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DOE/CS/32224-T1

ASHRAE HANDBOOK OF EXPERIENCES IN THE DESIGN AND INSTALLATION OF SOLAR HEATING AND COOLING SYSTEMS

By
Dan S. Ward
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July 1980

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NTIS-25
SP-25

Work Performed Under Contract No. AC01-76CS32224

American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.
New York, New York

U.S. Department of Energy

Solar Energy

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Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161.

Price: Printed Copy A11
Microfiche A01

DOE/CS/32224-T1
Distribution Category UC-59

AMERICAN SOCIETY OF HEATING, REFRIGERATING, AND
AIR-CONDITIONING ENGINEERS, INC.

HANDBOOK OF EXPERIENCES IN THE DESIGN AND INSTALLATION OF SOLAR HEATING AND COOLING SYSTEMS

JULY 1980

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EXECUTIVE SUMMARY

This handbook consolidates a large body of information on the experiences encountered in the design, installation, operation, testing, and maintenance of solar heating and cooling systems. A substantial portion of the information is derived from the U.S. Department of Energy's National Solar Data Network and other government-supported solar heating and cooling system projects. While these federally-funded projects may not portray the experiences of privately funded and commercial state-of-the-art solar systems, they do constitute systems which have sufficient data to allow for definitive conclusions to be made on the system's performance and operation. In addition, problems of durability and reliability experienced in the federally funded programs are common to a wide variety of solar systems, and vary significantly from the National Demonstration Program's experience only in degree and not in the specifics of different types of problems.

This handbook details a large array of problems encountered, including design errors, installation mistakes, cases of inadequate durability of materials and unacceptable reliability of components, and wide variations in the performance and operation of different solar systems. It should NOT be inferred, however, that this cataloging of problems in solar system design and installation implies that solar heating and cooling systems are not technically or economically feasible. In reality, the reverse is true; solar heating and cooling systems can be economically and technically feasible. Many well-designed and properly installed systems have provided significant energy savings and demonstrate the practical, technical and economic justification for the use of solar energy in reducing the use of non-renewable energy resources.

The theme of this handbook might very well be: It works, if you do it right. Based on experiences of operating solar heating and cooling systems, it can be concluded that substantial savings in non-renewable energy resources can be achieved with the use of solar energy. Among the well-designed and properly installed systems evaluated in this handbook, the average energy savings were:

Type Solar System	Savings in Non-Renewable Energy Resources (million Btu/year per square foot of collector area)
Domestic hot water	0.22
Passive space heating	0.10 - 0.29 *
Active space heating	0.19
Active space cooling	0.03
Potential active cooling	0.11 **

The majority of solar heating and cooling systems discussed in detail in this handbook did not achieve the energy savings per square foot of collector given above. The major purpose of this handbook is, therefore, to present the reasons for the reduced performance of many systems and to provide a compendium of data which details the problems encountered by operating solar systems. The emphasis on problems should not be construed to imply that solar systems typically encounter more problems than do conventional HVAC systems. Rather, a discussion of problems may be as instructive and useful as a detailed report on successful systems.

* Based on certain assumptions (see Section 3.4.1)

** Based on certain assumed system modifications and subsequent improvements (see Section 5.4.9)

Some of the significant conclusions from the development of this handbook are:

- Systems that followed long-standing and recognized design and installation practices, that utilized proven system design, and that were properly controlled performed well.

- A substantial portion of the problems associated with operating solar systems were problems directly related to conventional HVAC (Heating, Ventilating and Air Conditioning) practices. These problems include:
  (1) Inadequate or non-existent specifications
  (2) Lack of application of good engineering practices
  (3) Failure to adhere to good HVAC procedures
  (4) Use of improper design tools or methods
  (5) Unacceptable cost reduction attempts
  (6) Lack of detailed design and/or planning
  (7) Improper or non-existent maintenance
  (8) Lack of availability of maintenance or operating manuals

- Solar-related problems that have reduced the performance of solar heating and cooling systems include:
  (1) Improper application of solar design methods
  (2) Improper selection and integration of system components (specifically the poor matching of components with load and with other components within the solar system)
  (3) Inappropriate or unacceptable components (e.g., collectors) that contained design flaws
  (4) Design and installation of innovative, unique, or "experimental" systems without adequate testing, control, and/or instrumentation to ensure proper operation
  (5) Improper installation procedures
  (6) Poor selection of operating modes
  (7) Insufficient analysis of system hydraulics

- The major factors which resulted in reduced or unacceptably low performance include:
  (1) Excessive thermal losses from the system to the interior and exterior of the conditioned space
  (2) Unacceptable and excessive electrical energy consumption in operating the solar system
  (3) Lack of proper controls in operating the solar system and specific solar components (e.g., the absorption chillers in solar space cooling systems)
  (4) Lack of adherence to architectural constraints

- Solar domestic hot water (DHW) systems in general performed well. With few exceptions, system thermal efficiencies (useful heating divided by solar energy daily input) of 22 to 33 percent were achieved.

- The importance of system design (single tank, double tank, drain-back, drain-down, etc.) and choice of heat transfer fluid (water, air, water glycol, silicone, etc.) were shown to be of major importance in the level of performance of DHW solar systems.

- Freeze protection design has been shown to be a critical factor in the choice of solar DHW system designs.

- Passive space heating systems generally performed at high levels and with minimal operating problems. However, many of these passive systems included active elements and thus could be considered hybrid systems.

- Temperature variations in passive space heating systems ranged from 5°F (2.8°C) for closely controlled environments (i.e., residences) to as much as 35°F (19°C) for warehouses.

- Movable day/night insulation (e.g., beadwalls, insulating curtains, louvers, etc.) were effectively used to reduce night time heat losses without interfering with daytime collection of solar radiation.

- Those greenhouses included in this evaluation provided only minor contributions to the heating of the conditioned space, but did function as effective temperature buffers between the conditioned space and the ambient.

- The economic feasibility of passive solar heating systems was demonstrated to depend upon the ability of the passive system to: (1) reduce the heat loss to the ambient (particularly through the collection surfaces) and (2) to minimize temperature fluctuations within the occupied space and to maintain comfort conditions. These objectives were met by:
  (1) Reducing night time heat losses with some form of day/night movable insulation or by the use of attached greenhouses as temperature buffers
  (2) Limiting temperature fluctuations with substantial thermal mass (which was not subject to excessive thermal losses to the ambient) and with the use of hybrid or active/passive system components to improve heat distribution

- Direct gain passive heating systems without intervening mass walls, greenhouses, or other temperature buffers caused considerable temperature variations for systems contributing more than ten to twenty percent of the space heating requirements.

- Passive designs did not always consider effects of moisture accumulation and the need for reasonable levels of natural or forced ventilation.

- On average, active space heating systems performed at unexpectedly low levels. However, several systems which had received careful attention to detail in their design, installation and operation performed well.

- Active space heating systems utilizing air-heating solar collectors with pebble-bed storage and water-heating solar collectors with water storage performed with equivalent efficiencies and savings in non-renewable energy resources. Neither of these two major types of solar heating systems was shown to operate at significantly higher performance levels.

- The majority of solar active cooling systems evaluated were net energy losers, i.e., they utilized more conventional energy (in the form of electricity to operate the solar system) than they supplied from the solar components. Alternatively two systems (one residential and the other commercial) performed well and achieved significant energy savings.

- Given the realistic potential improvements in two of the operating solar systems and the modified calculation of potential real-energy savings, it may be concluded that solar cooling systems can achieve savings in non-renewable energy resources of 0.11 million Btu per year per square foot of collector.

- In order for solar cooling systems to be technically and economically feasible, thermal heat losses and solar operating electrical energy consumption must be minimized, internal and system controls must be optimized for maximum performance, and component, system and load requirements must be properly integrated. Solar space cooling systems can achieve system overall efficiencies of 20 to 35 percent when careful attention to design is a prerequisite.

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GLOSSARY

active system	A solar system that uses pumps or fans to circulate a heat transfer fluid through solar collectors and to distribute heat to the building; the opposite of a passive system.
air-type collector or air collector	A solar collector that uses circulating air as the heat transfer fluid
ambient temperature	The temperature of the surroundings as measured by a dry-bulb thermometer.
antifreeze loop	A circuit, consisting of the solar collectors, a pump, and a heat exchanger through which an antifreeze solution is pumped.
aqueous solution	A mixture of a substance (such as ethylene glycol) with water.
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 345 East 47th Street, New York, New York 10017.
ASME	American Society of Mechanical Engineers, 345 East 47th Street, New York, New York 10017.
auxiliary system	A system that provides heating or cooling when solar energy alone is insufficient, a back-up system.
Btu	British thermal unit. The amount of heat required to raise one pound of water one degree Fahrenheit; the basic unit of heat in the English system of units.
cathodic protection	A method of corrosion protection in which a highly reactive metal bar is placed in the system liquid. To be effective the metal bar must be more reactive than the most reactive metal component in the system and must have a continuous electrical path to the most reactive metal component.
centrifugal pump	A type of pump in which a liquid is flung outward by the rotation of an impeller. See positive displacement pump.
coil-in-tank heat exchanger	A coil of tubing surmerged inside a tank. One heat transfer fluid is pumped through the tubing while the other flows over the outside of the tubing by natural convection.
collector	A device constructed to absorb solar energy and convert it to useful heat.
collector coolant or fluid	A heat transer fluid used in solar collectors
convection, forced	A means of transferring heat in which the heat transfer fluid is moved by external means such as a pump or fan.
convection, natural	A means of transferring heat in which the heat transfer fluid is moved by the buoyancy of its warmer parts.
cooling season	The time of year (usually June to September, but varying with climate) when air conditioning is desirable to maintain comfortable room temperatures.
daily storage temperature range	The difference between the warmest storage temperature attained in a day and the coolest storage temperature reached on the same day.
DHW	Potable domestic hot water.
differential thermostat	A device that uses a measured temperature difference (such as the temperature difference between the collectors and storage) to control a device (such as a pump or fan).

drain-back system	A method of protecting the solar collectors against freezing by draining the collector water into the storage tank whenever the collector pump shuts off.
drain-down system	A method of protecting the solar collectors against freezing by draining the collector water into storage by means of vents and valves, at near freezing conditions.
effectiveness of heat exchanger	The ratio of actual heat transferred in a heat exchanger to the maximum possible heat that could be transferred in a perfect heat exchanger.
expansion tank	A device used to limit the pressure increase caused by thermal expansion of the liquid in a sealed system. The expanding liquid compresses air in the expansion tank.
f-Chart	A method devised by the University of Wisconsin for calculating the performance of solar energy systems.
flow rate	The volume or mass of fluid that flows past a point in a pipe or duct per unit of time. In the English system the units of volumetric flow rate are typically gallons per minute or cubic feet per minute, and the units of mass flow rate are typically pounds per minute (ℓ/s or kg/s).
fluid	A substance that cannot retain its shape without an external container; a gas or a liquid.
forced convection	See convection, forced.
FRP	Fiberglass-reinforced-plastic.
head	The maximum distance a liquid can rise in a pipe. Head is used as a measure of pressure.
heat distribution	As used here, heat distribution refers to transport of heat from storage to the parts of a building where heat is required.
heat exchanger	A device for transferring heat from one fluid to another while preventing irreversible mixing of the two fluids.
heating season	The time of year (usually October to May, but varying with climate) when heating is required to maintain comfortable room temperatures.
heat of fusion	The amount of heat per unit mass that must be removed from a liquid to freeze it when the liquid is initially at its freezing temperature.
heat storage device	A device that absorbs heat and holds it until the heat is needed to warm a building or domestic hot water.
heat transfer coefficient	The amount of heat that can be transferred across a unit area of surface per unit of time per unit of temperature difference between one side of the surface and the other (Btu/hr·°F·ft^2), (W/m^2·°C).
heat transfer fluid	A liquid or gas used to transport heat from one location to another. Typical heat transfer fluids include air, water, and antifreeze solution.
HVAC	Heating, Ventilating and Air Conditioning.
hybrid system	A solar energy system that combines features of active and passive systems.
hydronic system	A heating system in which water is heated by solar energy or by a boiler and distributed to heat exchangers located at various points in the building. The heat exchangers in a hydronic system are typically radiators, baseboard convectors, fan coil units, floor panels, or ceiling panels.

insolation	The amount of solar energy incident on a unit of surface area per unit of time (Btu/hr·ft^2). (W/m^2). Notice the differences in spelling and meaning between insolation and insulation. Insolation is an acronym from incoming solar radiation.
insulation	A material used to restrict the flow of heat or electricity.
laminar flow	Fluid flow in which little mixing between fluid layers occurs. Laminar flow occurs at low Reynolds numbers. Compare with turbulent flow and transitional flow.
latent heat	The quantity of heat per unit mass required to change phase. Heat of fusion is an example of latent heat.
life-cycle cost analysis	A method of comparing the cost of a solar energy system with the cost of a conventional system by totaling the costs of each system over the lifetime of the solar system. Costs usually included are first cost, mortgage interest, fuel, electricity, repairs, tax rates, insurance, etc.
liquid-type collector or liquid collector	A solar collector that uses a circulating liquid as the heat transfer fluid.
maximum operating storage temperature	The highest temperature at which the storage system can operate. Maximum operating temperature may be determined by the maximum temperature that the collectors can attain, the temperature limitations of materials in the system, the boiling point of water, or the pressure limitation of a sealed system; whichever is less.
minimum operating storage temperaure	The lowest temperature at which useful heat can be extracted from storage.
natural convection	See convection, natural.
net positive suction head	The absolute head (pressure) available at the inlet to a pump, abbreviated NPSH. Pumps will be damaged by cavitation if the NPSH does not exceed the pump's requirement.
nonpotable fluid	A fluid which does not meet Public Health Service standards for drinking water or state or local standards for drinking water.
operating temperature range	The difference between maximum operating temperature and minimum operating temperature for a specified length of time. See daily temperature range.
parasitic losses or parasitic power	The power required to circulate heat transfer fluids and operate the controls of a solar system.
passive system	A solar system that does not use pumps or fans to circulate a heat transfer fluid through solar collectors or to distribute heat to the building; the opposite of an active system.
payback period	The length of time until the fuel savings of a solar system begin to exceed the difference in cost between a solar system and a conventional system. See life-cycle cost analysis.
phase change system	A type of thermal energy storage system in which heat is stored by melting a substance and released by freezing the substance.
plenum	A space at the inlet and outlet of a rock bed used to distribute the air uniformly to the rocks.
potable water	Water that meets federal, state, and local quality and safety standards for human consumption.
pressure gradient	A change of pressure per unit of length.

psi	Pounds per square inch; a unit of pressure. Unless otherwise specified, pressure is measured relative to atmospheric pressure.
resistance heating	A method of heating with electricity in which electricity passing through a resistor is converted directly to heat.
retrofit	As used here, retrofit means to install a solar energy system in an existing building or in a building not originally designed for using solar energy.
R-value	Resistance of insulation to heat conduction given in units of $°F \cdot ft^2 \cdot hr/Btu$ ($m^2 \cdot °C/W$).
sensible heat	Heat that, upon flowing into a storage medium, increases the temperature of the medium. The constant of proportionality between the flow of heat and the temperature increase is the heat capacity of the medium.
sensor	A device that measures pressure or temperature and relays the information to a controller.
shell-and-tube heat exchanger	A type of heat exchanger consisting of a bundle of tubes within an outer shell and with internal baffles to direct the fluid flow. One heat transfer liquid is pumped through the space between the tubes and the shell.
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association, 8224 Old Court House Road, Vienna, Virginia 22180.
solar house	A house that derives a substantial portion of its heat from the sun.
space heating	Heating a building to maintain a comfortable indoor temperature.
stagnation temperature	Temperature of collector absorber plate at equilibrium, no-flow condition.
storage medium	As used here, a storage medium is a substance that stores heat in a solar system.
storage system	The part of a solar system that includes a storage medium in a container with heat exchangers, pump, valves, and other components necessary to transfer heat into and out of the storage medium.
temperature stratification	Thermal stratification.
tempering valve	A valve that limits the temperature of water flowing from a domestic hot water tank by mixing it with cold water.
thermal stratification	Separation of hot and cool parts of the storage medium within the storage unit.
thermosyphoning	Motion of a fluid caused by buoyance of its warmer parts; natural convection.
thermosyphon system	A pumpless solar system in which buoyancy, acting on water heated by the collector, causes the water to rise into the storage tank. Thermosyphon systems are usually limited to domestic hot water systems in the tropics because the storage tank must be mounted above the collectors and there is no protection against freezing.
toxic fluid	A gas or liquid that is poisonous, irritating and/or suffocating, as classified in the Hazardous Substances Act, Code of Federal Regulation, Title 16, Part 1500.
wraparound heat exchanger	A tank that has fluid passages wrapped around it. The fluid passages are typically a tube soldered to the outside of the tank (a traced tank) or a metal panel with integral fluid passageways clamped around the tank.

CONVERSION OF UNITS

1. Multiply (MBtu/year·ft^2) by 11.4 to obtain GJ/year·m^2
2. Multiply (Btu/lb·°F) by 4.19 to obtain kJ/kG·°C
3. °C = (°F - 32)/1.8
4. Multiply Btu/(day·ft^2) by 11.4 to obtain kJ/m^2(day)
5. Multiply ft^2 by .0929 to obtain m^2
6. Multiply Btu by 1.055 to obtain kJ
7. Multiply ft^3 by .0283 to obtain m^3
8. Multiply gpm by .0631 to obtain ℓ/s
9. Multiply hp by .746 to obtain kW
10. Multiply ft^3/m by .472 to obtain ℓ/s
11. Multiply ton by 3.52 to obtain kW
12. Multiply \$/ft^2 by 10.7 to obtain \$/m^2

ACKNOWLEDGEMENTS

Numerous individuals and organizations have contributed substantially to the content and development of this handbook. Special thanks are extended to Dr. John C. Ward, who read and reread several proofs of the handbook and who made invaluable comments and suggestions. Thanks also to Ms. Kathi McKenna who typed, retyped, and edited the entire manuscript, and to Jeff Grebe for his review and calculations.

The ASHRAE Technical Evaluation Committee (TEC) contributed a great deal of expertise, experience, insight and professional effort in extensive reviews of the handbook at several stages of its development. The TEC members:

A.B. Newton, Chairman
Warren Spofford
William Fleming
Stan Mumma
and
Doug Jardine

were instrumental in ensuring the usefulness and completeness of the handbook. Their contributions are sincerely appreciated.

Acknowledgement must also be made to H. Jackson Hale of the Department of Energy, and F.H. Faust and Ignacio de Dios, Jr. of ASHRAE for their financial, administrative, and moral support.

Other major contributors include Carlo LaPorta and Boyce Greer for their contributions at the TEC review meetings, Joe Brown and Mike Myers of Marshall Space Flight Center for the case studies, the contributors to the First and Second Annual Solar Heating and Cooling Systems Operational Results Conference, and to all of the individual authors of the many publications, reports, and technical articles which were used as the basis of information for this handbook. All of these people are to be commended for their efforts.

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July 1980

1. INTRODUCTION

1.1 OBJECTIVES OF THE HANDBOOK

The principal objective of this handbook is to provide information on the performance and operation of solar heating and cooling systems so as to enable future designs to maximize the performance of solar systems and to eliminate repeated and costly errors. It is intended that these objectives be attained by providing a compendium of practical experiences gained in the design, construction, testing, and operating phases of residential and commercial solar heating and cooling projects. These projects include those in the U.S. Department of Energy's Solar Heating and Cooling Demonstration Program and numerous other projects sponsored by private and public interests. This experiences handbook, therefore, is intended as a means of dissemination of experience and information to engineers, architects, builders, and contractors.

This effort is intended as a culmination of prior work which attempts to update the solar heating and cooling experiences information base. This base will certainly not be the final word on solar heating and cooling applications, but can constitute the foundation on which experience in the solar field could be expanded.

1.2 SCOPE OF THE HANDBOOK

This handbook covers the performance and operational experience with solar heating and cooling systems. Applicable systems include: Active solar space heating systems, passive solar space heating systems, and active solar space cooling systems. Performance and operational data have been acquired from the U.S. Department of Energy (DOE), the National Solar Data Program, solar system project reports, solar conference proceedings, design review reports, cost study reports, progress reports, commercial sources, and various solar energy journals and publications. Topics included in this handbook are:

1) Technical and performance criteria
2) Durability and reliability of sytems and components
3) Solar system performance
4) Analysis of the performance of solar heating and cooling systems
5) An overview of prior experiences
6) Management plans and logistics and
7) Cost factors

The scope of durability and reliability problems will be limited to problems previously encountered in operating systems. Performance of solar systems is limited to quantitative values of major energy flows in the system (in and out of solar collector array, thermal storage, chiller, etc.) and the system thermal and total energy savings. Detailed performance of solar components is considered only in its impact on total system performance.

Another limitation in the scope of the handbook is the emphasis on solar-related system problems and not conventional HVAC problems (which may have been encountered in the solar system operation). The objective is to limit considerations to the special or important aspects of designing, installing, operating, and maintaining solar heating and cooling systems, and the resulting effects on system performance.

1.3 SOURCES AND TYPES OF INFORMATION

Sources and types of information include:

1) Commercial and residential solar system project reports and program reports - obtained from Marshall Space Flight Center, DOE Operations Offices (Chicago, San Francisco), U.S. Department of Housing and Urban Development, and numerous DOE sup-support contractors.
2) Solar Heating and Cooling Systems Operational Results Conferences (1978 and 1979)
3) Second Solar Heating and Cooling Commercial Demonstration Program Contractors' Review (1978)
4) Preliminary issue of Solar Heating and Cooling Project Experiences Handbook - prepared by DOE Project Management Centers in Chicago, San Francisco, NASA Marshall Space Flight Center, ASHRAE, and the University of Alabama at Huntsville.
5) Design Review, Cost Analysis, and Program Manager Reports on the commercial solar demonstration program - obtained from NASA Marshall Space Flight Center and Mueller Associates.
6) Reliability and maintenance data - obtained from Argonne National Laboratory and NASA MSFC.

7) Solar system and component standards - obtained from the National Bureau of Standards.
8) Numerous other design, installation, and/or operating handbooks, publications, and reports (which are detailed in Section 2.9).

Information included in the handbook consists primarily of general durability, reliability and design problems, and quantitative performance data of solar heating and cooling systems. In some cases, specific, detailed problems are discussed along with proposed solutions. The overall intent is to allow solar practitioners to visualize the previous problems encountered in solar heating and cooling systems, and to hopefully avoid these and similar problems in future installations.

1.4 IMPORTANCE AND APPLICABILITY OF HANDBOOK

1.4.1 Importance of Practical Experience

The basic theoretical understanding of the thermal processes involved in solar energy systems are reasonably well understood. Even the interactions between different components in an integrated solar system can be predicted with sufficient accuracy provided that detailed calculations consider all the factors of thermal and total energy flows. However, this requirement for detailed information on the interaction of different components can result in very complex calculations and may require the use of large, high speed computers.

The alternative is to utilize experience gained in the design, installation, and operation of systems, combined with limited calculations. This latter alternative is preferred with respect to solar heating and cooling systems for practical reasons. The current experiences do provide evidence of the practicality and feasibility of solar heating and cooling systems. This body of practical experience thus combines a proof-of-concept demonstration with a compendium of lessons learned. Awareness of these factors should lead to improved performance of future systems and substantially increase the commercial feasibility of solar systems.

1.4.2 Usage of Handbook

In using this handbook, several factors should be noted. These include the limited sources of good data, the limitations on conclusions that can be drawn from the data, the apparent emphasis on system problems versus system successes, and the comparison of different systems.

A substantial portion of the information and performance data presented in this handbook is derived from DOE's National Solar Data Program. This is due primarily to the ready availability of the data from these instrumented systems. Data from commercial and privately funded projects is substantially less available due to the high costs of instrumentation and data analysis.

In addition, the National Demonstration Program had a high percentage of unique and innovative solar designs. These systems, which were more experimental than demonstrational, had a higher failure rate than some of the later designs. It is extremely important that the reader recognize that numerous solar heating and cooling systems have operated with virtually no problems and at high performance levels. Emphasis on problems encountered in solar systems is necessary in order to alert solar designers and installers to potential problems and difficulties. Systems that have worked well, on the other hand, do not provide the same quantity of information as do systems that have encountered problems.

A significant portion of this handbook will compare the performance and operation of different solar systems. These comparisons will include residential and commercial solar heating and cooling systems, and will analyze the differences in performance and operation between:
1) Different solar designs and systems and
2) Solar versus conventional HVAC systems.

1.5 HANDBOOK FORMAT

In evaluating the experiences with solar heating and cooling systems, three critical factors are of importance. These factors include the ability of a solar system to:
1) Operate within design specifications
2) Operate without major difficulties over its design life
3) Provide energy savings of non-renewable energy resources.

From the designer's and installer's viewpoint, the requirements are to:

1) Make the system work,
2) Make the system last, and
3) Make the system efficient.

These requirements of a solar system are embodied in the handbook format, which organizes the experiences in solar heating and cooling systems into:

1) Durability, reliability, and design problems
2) Performance of solar heating and/or cooling systems
3) Analysis of systems performance
4) Performance evaluation of systems, and
5) Experiences overview

Durability, reliability, and design problems are presented in terms of individual subsystems and components. These subsystems include:

1) Solar collector subsystem
2) Heat transfer fluids
3) Thermal storage
4) Passive solar components and
5) Piping/ducting

In addition to discussions on subsystems, reliability/operational problems, case studies, and an annotated bibliography are discussed in some detail. Reliability/operational problems are further subdivided into problems associated with freezing, boiling, control system, pumps/fans, and valves/dampers.

It is noteworthy that while this handbook details many specific problems, it does not necessarily present solutions. In some cases the solution is obvious (i.e., DON'T DO THIS!). In other cases a specific solution may or may not be suggested (other than choose an alternative design). Limitations in the scope of the handbook narrow its function to an identification of experienced (and in some cases potential) problems. The annotated bibliography (section 2.9) provides additional reading and some solutions to many of the design, installation, and operational problems identified herein.

The section on Performance of Solar Heating and Cooling Systems includes a criterion for evaluating the design and performance of a solar system and a catalogue of the actual measured and calculated performance of DHW, active and passive space heating, and space cooling systems. The performance criterion is based on solar collector performance, solar system thermal performance, and overall energy savings by the solar system.

The performance is analyzed with respect to the factors that contributed to successful systems and, alternatively, resulted in reduced performance of some systems. These are detailed in sections on:

1) Selection and integration of components
2) Collector array performance
3) Problem related variations in collector performance
4) Thermal losses
5) Solar system operation electrical energy requirements
6) Solar controls-caused variations in system performance, and
7) Architectural constraints on system performance

The following section consists of general comments and conclusions on the performance evaluation of DHW, passive and active space heating, and active space cooling systems.

Appendices include:

A. Management and Logistics
B. Case Studies
C. Mathematical Formalism of Parameters
D. Economics
F. References

2. DURABILITY, RELIABILITY AND DESIGN PROBLEMS

2.1 INTRODUCTION

In order to make a realistic and practical assessment of the overall performance of operating systems, it is essential that the question of durability and reliability of the components constituting the solar system be given some priority of consideration. If a solar heating or cooling system has inherent problems of material and component failures or unreliable controls, then the question of how efficiently the system performs is no more than academic. From a design and installation viewpoint, there are three critical factors which, in order of priority, are:

1) Make the system work
2) Make the system last
3) Make the system efficient

The first of these factors is essentially a question of reliability or operating the system in such a manner so as to meet design specifications and to ensure savings in non-renewable energy sources by the solar system. This is primarily a question of providing components which perform in accordance with design specifications, and integrating these components into an operationally-feasible system, and providing controls which operate the system in the desired manner.

The second factor is the durability of the system, i.e., the question of whether or not the system will perform over its design lifetime. Reliability ensures that the system does in fact operate according to design; durability ensures that the system will continue to operate during its design life without material or component failures.

After a particular system design is proven to operate reliably and without degradation, then the designer should be concerned with improving the efficiency of the system. This is not to suggest that efficiency is not important; it is a necessary but not sufficient condition. For example, a naval antiaircraft gun which is capable of firing 120 rounds per minute is of questionable benefit if it tends to jam after the third or fourth round.

Because of the importance of durability and reliability in solar heating and cooling systems, there has developed in the solar industry an extensive compilation of publications dealing with the operational results of systems and, in particular, the question of problems encountered and failures observed. These publications include detailed reports on the lessons learned in attempting to make different systems work and to keep them working. This handbook will not attempt to cover the gamut of durability and reliability concerns of solar systems, but will limit the discussion to an identification of specific difficulties and the overall assessment of problem areas. Specifics on any given problem can be further investigated by the reader, by referencing one of the other publications listed in Section 2.9 and Appendix E.

It should also be noted that many of the problems of durability and reliability experienced by the solar industry are in reality problems associated with conventional HVAC (heating, ventilating, and air conditioning) practices. Adherence to proper HVAC practices and methods would undoubtedly have avoided many of the failures experienced in solar heating and cooling systems. The reader is encouraged to consult the ASHRAE Handbook of Fundamentals [1] and other relevant publications for recommended HVAC practices and methods.

In considering the problems encountered at solar installations within the area of durability and reliability, it is convenient to consider the problems from a component viewpoint. In this way we can describe the specifics of actual problems encountered and attempt to point out considerations which are necessary in designing trouble-free solar installations.

2.2 SOLAR COLLECTOR SUBSYSTEM

The principal problem areas encountered in the design, installation, and operation of solar collectors and solar collector arrays include those problems due to:

1) Weathering
2) Corrosion
3) Differential thermal expansion
4) Thermal deformation at stagnation conditions
5) Thermal shock
6) Degradation of collector fittings and subcomponents
7) Collector materials degradation, and
8) Electrical power failures (i.e., electrical power to operate the solar system)

Sparkes and Raman [2,3] considered collector subsystem related problems in some detail and have classified the collector problems as:

I. Design related problems (components and subsystems)
   A. Materials-related problems
   B. Flow-related problems
II. Installation, start-up, and maintenance problems

Much of the information presented in this section is based on the results reported by Sparkes and Raman [2,3] and by other authors and investigators [4,5].

2.2.1 Design Related Problems

2.2.1.1 Materials Related Problems

2.2.1.1.1 Solar Collector Covers. Materials used for solar collector covers include window and tempered glass and a wide variety of plastics. With glass cover plates, the main problem encountered has been breakage caused by mechanical and thermal stresses. Thermal stresses in the glass due to large temperature differences between different points of the glass can lead to early failure (i.e., breakage). These temperature differences are sometimes caused by partial shading (particularly of the second or lower cover), and can be as large as 50°F (10°C). These thermal stresses, combined with mechanical defects in the form of hairline scratches or tiny chips in the glass cover edges, will cause breakage of the glass cover. The inner, or second glass cover, is subjected to the most stress and consequently will be the first to go. Single cover collectors generate less stress on the glass and consequently have a much lower failure rate.

Thermal and mechanical stresses can also occur due to the differential thermal expansion between a cover and the collector module frame. In general it is necessary to leave a minimum of one-fourth inch (0.64 cm) space around the entire edge of a glass cover (or structurally-integral plastic cover) in order to ensure that the greater thermal expansion of the cover (over the metal collector frame) does not cause severe cramping of the cover. If cramping is present and the cover is glass, the glass will break. If the cover is plastic, it will buckle, and wind loading will generally accomplish the remainder of the destruction of the plastic cover. Several instances of inadequate space for differential thermal expansion have resulted in glass breakage [6].

In addition, glass must be adequately supported in order to prevent scratching of the glass (and thus early breakage) due to the differential thermal expansion. Figure 1 [6] illustrates some of the possibilities for the attachment of glass to a metal framework. In figure 1(a), the metal scratches a "groove" in the glass, and in figure 1(b), a chip is caused. In either case the glass would be destroyed (tempered or not). Figure 1(c)shows the situation where a butyl tape (or equivalent) prevents this damage. The butyl tape is the type used in automobile windshields.

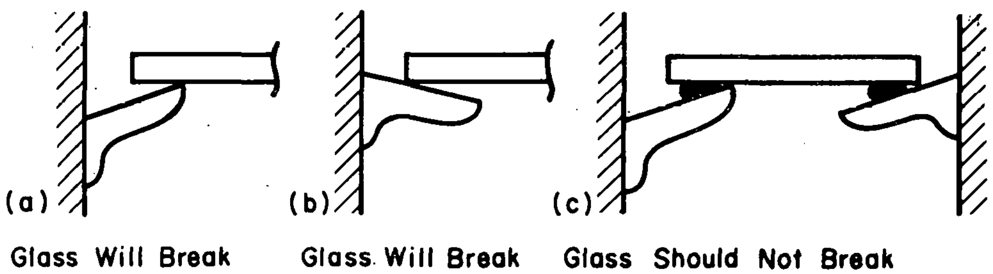

Figure 1. Schematic of Glass/Frame Attachments (Greatly Exaggerated)

Mechanical defects in the form of hairline scratches or cracks have also been found in tubular collectors. These defects have resulted in tube failures at high operating temperatures and pressures. Many collectors now utilize tempered glass for mechanical strength and for safety. A large percentage also use low-iron glass cover plates for maximum transmission of solar radiation.

The use of tempered glass has been recommended for solar collectors as a means of obtaining lower failure rates. However, the greater mechanical strength of the tempered glass is not the only reason for its use. In reality the use of factory-cut double strength window glass is quite sufficient provided that it is not chipped in any way during the installation phase. If the window glass is chipped, it will most likely break eventually. On the other hand, if the tempered glass is chipped by an installer, it will break immediately. From the viewpoint of an installer it is easier to replace the glass during the installation phase than it is later, after the job is completed. In addition, tempered glass breaks into a large number of relatively harmless bits of glass while ordinary window glass breaks into large, potentially harmful shards. Because of the potential danger of large, knife-like glass slivers from window glass sliding off the roof, tempered glass is preferred for safety reasons.

Plastic cover plates have special problems with regard to ultraviolet degradation and degradation from exposure to high temperatures. In addition, the thermal expansion and contraction of the plastic cover plates can make them sag (particularly those which are "stretched" over the collector and have effectively no structural strength). When this sagging occurs, it may cause severe overheating and melting of the plastic. In addition, even minimal sagging can result in a "fluttering" of the cover plate under windy conditions and eventual failure.

A clear distinction should be made between "plastic" covers and fiberglass-reinforced polyester (FRP), which is also a plastic. FRP has proven to be much more durable than other plastics. Several decades of experience in the commercial greenhouse industry have led to the conclusion that both glass and FRP have distinct advantages and disadvantages, but that both are suitable for use in modern greenhouse construction. Other plastics are generally not used in modern commercial greenhouses. FRP is usually less expensive than glass but may be less durable. Glass, on the other hand, must be very positively supported in order to prevent breakage (and thus usually requires a structurally sound collector module frame).

In many installations the covers of the solar collector modules act as a partial or full watertight roofing of a building. Problems have been encountered because of inadequate waterproofing and/or flashing of the solar collector array [3]. In addition, the collector cover plates and their integration with the rest of the collector module have not always provided watertight seals against weathering. It is essential to provide a leak-proof roof; only strict adherence to good roofing practices has allowed for the expected quality in leak-proofing of roofs. The addition of a solar collector array as the waterproofing surface must therefore be addressed with some care.

One final noteworthy point should be made. Hail damage of glass cover plates has been minimal [7]. In one case a severe hailstorm caused approximately $35 to $50 million damage to a community but only $800 for solar collector glass broken on approximately 11,000 square feet (1024 m^2) of collector.

2.2.1.1.2 Absorber Plates. Aluminum absorber plates, when connected to piping made of a different metal, e.g., copper, have resulted in galvanic corrosion which has shown up as pinhole leaks in the collectors. Steel absorber plates have been prone to both rust and corrosion. Rust has primarily been limited to the external surface of the steel plate when exposed to a leaky collector module; i.e., the leaky module allowed a significant amount of water between the cover plate and the absorber plate. This moisture build-up between the plates has been due to both major defects (causing an effectively non-watertight roof) and minor defects (which allow a slow but inevitable moisture build-up). The first problem can be effectively dealt with, and most good collectors do not suffer from this problem. The second problem has been approached by the use of desiccants within the collector. Experience has shown, however, that their useful life is only one or two years in many cases. It may be that some moisture accumulation within the collector will have to be accepted and rust and corrosion protection included as a necessary precaution.

Internal corrosion and rusting of steel absorber plates have sometimes been experienced in drain-down and drain-back systems where fresh air intakes provide a continuing source of oxygen. Nitrogen refill capabilities can reduce this problem substantially but such systems are expensive. Most systems with steel collectors, therefore, leave the system filled with antifreeze solutions which include corrosion inhibitors. Note, however, that with ethylene and propylene glycol solutions, stagnation temperatures (no-flow condition) and to a lesser degree boiling conditions within the collector can degrade the glycol and form corrosive organic acids.

Chemical corrosion of copper plates is rare but has occurred in some cases due to the chemicals in the flux used while soldering the piping to the absorber plate.

Differential thermal expansion between the absorber plate and the collector frame (even when different metals or materials are used, as they frequently are) is of comparatively little concern. This is due to the fact that the absorber plate for most collectors is normally thermally isolated from the frame. Thus there is generally an adequate space between the absorber plate and the frame in order to allow for differential thermal expansion (whether or not filled with insulation).

Some collectors have utilized the building's frame structure for solar collectors and in particular the rafters as the collectors' frame. In cases where the cover plates have been physically located between the rafters, breakage and/or buckling has usually occurred. In cases where the cover plates are just above the rafters and only the absorber plate and insulation are between the rafters, the collectors have held up quite well in most cases. It must be noted that a principal reason for failures has been the warping of the wood and the lack of precise dimensions between rafters.

2.2.1.1.3 Absorber Plate Coating. For some absorber plate surfaces, changes in the surface color, cracking, peeling, pitting, and outgassing of black paint coatings have been observed. Some of this degradation has been shown to result from a lack of quality control by the manufacturer. This conclusion was reached due to the differences in efficiency of different collector modules of the same model.

Black chrome has been found to give the most durable and efficient selective surface [2], and it has the advantage that it can be prepared so as to provide uniform quality. While the cost of surfaces depends on the treatment (whether selective or non-selective) and the individual manufacturer's prices, it appears that the use of a good black chrome surface is now justified. Not only has the black chrome proven to be quite durable, but the use of a selective surface has the advantage that, in general, the double cover flat-plate collector with a non-selective surface may be substituted for by a single cover selective surface collector. Thus the relatively poorer durability of a second or lower cover can be effectively replaced with the higher durability of the black chrome selective surface.

A common problem in collectors has been "outgassing". This can oocur from the binder in a black paint absorber coating, from sealants used in the collector, from the binder material used in fiberglass insulation located behind the absorber plate, or when wood is a collector component. When subjected to high temperatures (generally in excess of 160°F, 70°C), some of these materials or binders form vapors which move through the collector, and condense on the relatively cooler inner surface of the cover plate. This has resulted, in some cases, in a drastically lowered cover plate transmissivity. This problem has apparently been significantly reduced in the more recent collector designs, but is not yet completely eliminated. Lower temperature collectors have had considerably fewer outgassing problems. For example, one installation has operated for seven years at temperatures generally below 120°F (50°C) without any measurable problems [8].

2.2.1.1.4 Collector Heat Transfer Fluids. The most common fluids used as the collector heat transfer medium are air, water, and water/glycol solutions. Problems with air-heating collectors are primarily leakage problems which may increase with time.

Water and ethylene glycol solutions constitute one of the most common heat transfer liquids used in solar collectors. When water is used without glycol, corrosion inhibitors are added with the subsequent need for regular monitoring of the pH. Lack of such regular monitoring has been one of the causes of corrosion in many systems.

Ethylene glycol has caused several problems. When exposed to high temperatures (greater than about 270°F, 130°C), the glycol breaks up, forming organic acids. These acids can corrode the collector material and the sealants. Ethylene glycol is also toxic so that double separation between the glycol solution and the potable water supply is required. Propylene glycol solution is often considered non-toxic enough to waive the double separation requirement, but at high temperatures, propylene glycol also breaks up and forms corrosive organic acids. Because of this, it may also require double separation, particularly because of code restrictions and the fact that a relatively non-toxic antifreeze might be replaced with the more commonly available ethylene glycol. Single separation from potable water supplies could then be hazardous.

Most heat transfer fluids currently being used in solar heating and cooling systems (other than water or air), will degrade with time or under the extreme temperatures that may exist at stagnation. In addition to the possibility of corrosion, fluid properties may change, resulting in freezing and boiling problems (see Sections 2.7.1 and 2.7.2). Entrapped air occurrences, acidity levels, corrosion inhibitors, and liquid degradation must be periodically checked. Possibilities for chemical and/or galvanic reactions should be resolved in the design phase.

In addition, the roof and/or other building materials may be susceptible to chemicals in an antifreeze solution, oil, or other "exotic" heat transfer liquids. Spillage of heat transfer liquids may present hazards due to toxicity, fire potential, and potable water contamination, in addition to effects of the loss of liquid itself. In general, the main problems encountered with the heat transfer liquids have been corrosion, degradation of the liquid itself, and a loss of efficiency.

The use of more "exotic" liquids such as silicone liquids, oils, diethyl phthalate, et al, have sometimes resulted in interesting problems. In general, no other liquids have shown exceptional promise in terms of durability, reliability, requirements for pumping power, safety, and performance; and which would suggest the immediate replacement of water or water/glycol solutions.

The durability of air as a heat transfer fluid also seems assured.

There is also the question of sizing pumps and expansion tanks, selecting flow rates, type of valves, etc. according to the liquid properties. In many systems, sufficient attention has not been paid to the effect of the heat transfer liquid on system design and sizing.

The low specific heat of some liquids will lower the collection efficiency or increase the pumping energy used. However, some of these liquids have advantages such as chemical stability. More testing and experience is needed to decide which liquids are the best for different kinds of solar systems (see Heat Transfer Fluids).

2.2.1.1.5 Internal Piping Material in Collectors. Material problems relating to the collector piping materials include the ones noted above for the absorber plate materials, as well as (a) melting or breaking up of the bonding of the collector piping to the absorber plates in some collectors at stagnation temperatures, and (b) leaks in the joints in the collector internal piping arising from thermal expansion and contraction.

2.2.1.1.6 Collector Fittings. Differential thermal expansion between the collector modules and other solar components and structure have sometimes led to leakage from fittings. This is particularly crucial between the collector modules and the manifolding which ties the modules into a single collector array. There, the fittings have undergone severe stress from the differential movement of the manifold and the collector modules. This has required the replacement of automobile-type rubber hoses in numerous installations after only one or two years of service.

On long collector arrays, for example, pipe expansion and contraction has resulted in over an inch of travel on the supply and return headers. Rigid pipes connected to the collector have, in some cases, depended upon the absorber being able to move. However binding or lack of space for absorber movement has been a problem; therefore offsets in the rigid connector should be considered to help isolate the collector's absorber plate from the header pipe expansion.

The use of flexible hose has provided for expansion compensation, but only when installed properly. In cases where the hose was a short straight connection between the collector and manifold, expansion cycling has gradually either worked the hose loose or stressed the hose to failure. A shoulder on the end of the pipe connection will prevent the problem of working the hose loose, but will not protect against failure.

A variety of hose clamps to attach flexible hose to the collectors and headers has been used. The screw type connectors have occasionally sliced the hose when over-tightening occurred and smooth lines available for use with screw clamps were not used. Screw clamps have also been subjected to too much torque for the specific hose material to be used. This has resulted in the hose suffering a "set" over time and thus requiring a yearly maintenance procedure to check the clamp tightness.

Spring clamps have been successfully used in avoiding hose damage, but have occasionally allowed for movement working the hose loose (discussed above). The advantage of the spring clamp is that it automatically compensates for any hose "setting" which might occur. Crimp clamps have been used in only a few instances but may suggest a promising alternative.

One precaution, which has not always been taken, is to ensure that the mating pipe of the header has been deburred before installing the hose. There have been several cases where small slits have been cut into the inside surface of the flexible hoses when deburring was not done, resulting in leaky connectors.

Sealants within a collector have suffered degradation from ultraviolet radiation or contact with hot glycol solutions. Rubber connectors have been cauterized to the point of brittleness by high temperatures in some collectors. Corrosion has been caused by the flux used while soldering the connections in a collector. This latter problem appears to arise from the zinc chloride in the flux which, in the presence of water, can react with copper and some absorber plate coatings.

Some collectors have used wood as the frame material, which can cause problems. After having been subjected to high temperatures on a daily basis, the wood can become very dry so that the ignition point is low enough to constitute a fire hazard. In addition, the wood can become warped, causing more leaks than is permissible.

2.2.1.1.7 Tracking Mechanisms. At the time of preparation of this handbook, written information on operating experiences with tracking collectors was not available. However, the reader is encouraged to reference the following report for comprehensive design, controls, fluids, patents, manufacturers information on tracking and concentrating collectors: "A Survey of Tracking Mechanisms and Rotary Joints for Cooland Piping", EG&G Engineering. A report prepared for the U.S. Department of Energy under contract DE-AC04-79-AL10748, August 1979 [9]. Note that de-tracking, either by movement of the collectors or by natural de-tracking if the tracking mechanism stalls, usually prevents high stagnation temperatures in tracking arrays.

2.2.1.2 Flow-Related Problems

2.2.1.2.1 Thermal Problems at Stagnation Conditions. Stagnation conditions can result in temperatures of 300°F to 500°F (150°C to 260°C) for flat-plate collectors and potentially as high as 700°F (370°C) for evacuated tube and 1000°F (550°C) for concentrating collectors. Such temperatures can severely affect the durability of some solar collectors.

The problems at stagnation conditions have included outgassing (see above), thermal shock, damage from violent boiling of the collector liquid (see Section 2.7.2), and severe degradation of materials. In several cases the internal piping material in the solar collector modules exposed to stagnation temperatures have resulted in the melting or breaking up of the bonding of the collector piping to the absorber plates, and/or leaks in the joints of the collector internal piping arising from thermal expansion and contraction.

All installations at some time experience stagnation temperatures. It is therefore essential that the collector design provide for these eventualities. For example, the effect of an electrical power failure to the system or collector pump/blower is to cause stagnation conditions and its attendent severe stresses on the collector materials and components. In essence, therefore, it is unrealistic to attempt to provide a protection means such as collector covers to prevent stagnation conditions and the accompanying stresses on the solar collector array. It is far more desirable to design and install the solar collector array properly to begin with so that it can withstand prolonged stagnation conditions.

2.2.1.2.2 Thermal Shock. In some cases solar collectors have been damaged by thermal shock when the collectors were initially filled. On a sunny day the empty (dry) solar collector can easily reach high stagnation temperatures (350 to 1000°F, 180°C to 550°C). In this situation, the fluid entering the dry collectors on initial start-up is significantly cooler than the absorber plate. This has resulted in some cases in broken glazings and absorber warpage and, in a few cases, led to the exploding of evacuated tubes. This situation can be avoided by ensuring that the collector system is filled during the early morning (during non-freezing conditions) before the sun has a chance to heat the collectors. During the summer, this might require filling as early as 6:00 am.

Thermal shock can also be an on-going operational concern with drain-down and drain-back systems because they are emptied and filled on a more routine basis. High limit temperature sensors may be required for installation on the collector absorber which will prevent activation of the collector pump whenever the collector temperature is too high for safe starting.

2.2.1.2.3 Drainage/Venting. Some collector designs tend to trap air bubbles within the collector, which in turn traps liquid. At stagnation this liquid can boil and rupture the collector tube. In winter the trapped liquid can freeze. A collector must be selected so that the pipes in it are pitched to ensure proper draining and venting and for proper operation in the case of a thermosyphon system.

Absence of venting can cause moisture to condense inside some collectors. Vent holes or weep holes are provided to allow the condensed moisture to escape and release the pressure. Proper placement of the vent holes is necessary in order to prevent blockage by snow, dust, mud dobbers, or debris which can lead to excessive pressure build-up inside the collector. This depends on both design and installation actors.

A related problem is that control valves have sometimes been located so as to isolate venting and/or relief valves. The subsequent non-draining and/or pressure build-up have caused damage in some installations.

Excessive pressure build-up in evacuated tubular collectors have caused tubular collectors to explode. This has been due in part to the U-shaped flow pattern in most evacuated tubes, which results in air nd/or vapor locks during operations. Boil off pressures in evacuated tube collectors may exceed 300 psi (2000 KPa) because of the small diameter tubes being in series in these collector modules. While many of these problems have been reduced, they have not been completely eliminated.

2.2.2 Installation, Start-Up and Maintenance Problems

The installation or mounting of a solar collector array onto or djacent to a building must consider the building's waterproofing requirements, the penetrations for mounting, piping and electrical connections, any damage during installation, possible drain problems necessitated by the collector mounts, and the ability to shed snow. The solar collector array may be mounted as an integral part of the roof, as a portion of the roof, independent but attached to the roof, or as a separate structure. In all cases the mounting materials must be resistant to weathering and degradation and should not pose an aesthetically displeasing appearance.

A variety of problems have been encountered in the mounting of collectors and in the collector/structure interface in general. For example, mounting collectors so that they directly touch the roof increases the possibility of moisture accumulation under the collector, leading to rotting of the wood under the collector. A clearance should be left between the collector and the roof or the base of the collector should be flashed onto the roof.

In several systems with collectors mounted directly on the roof, the flashing at the base of the collector was not installed properly. For direct mounting of the collectors on the roof, it is best to mount them on roofing paper and then flash and seal them. There have been instances of rain leaking through collectors installed as weather seals. Checking the flashing and the gaskets is needed in order to correct this.

In many cases, the roof tilt is not the same as the desired collector tilt, either due to architectural constraints or because the project was a retrofit installation. When the collectors are mounted at a steeper angle than the roof, adequate attention has not always been paid to static loading, wind loading, and aesthetics in designing the support structure. Wind loading, for example, has been a serious problem in some cases, where collectors were mounted at an angle to the roof. In the more extreme case, collectors have blown off their supports because of inadequate bracing. In one installation, an auxiliary reflecting panel was not installed firmly and was blown off.

Snow loading of the collectors in winter (and the weight of the liquid in the collectors) have not always been included in the static load on the roof when estimating the required roof strength. Giving the collectors a sufficiently large tilt has aided in snow removal from the collectors. Larger tilts, however, should be made with attention to the aesthetics of mounting the collectors and to optimum collector tilt. Structural cross members and horizontal flashings between collector modules do not generally impede snow slide off. It is highly recommended that space be allowed for the snow to slide off the collector completely and to avoid snow build up at the bottom of the collector.

A problem that relates to both the design and installation of the collectors is the potential blocking of vents in the collectors. Vents or weep holes are needed in some collectors in order to prevent moisture condensation inside the collector. In some installations, snow, dust, or dirt have blocked the vent holes, leading to fogging and also pressure build-up within the collector.

Collectors designed with external headers require considerably more work in installation than those with internal headers. In addition, factory installed joints have resulted in fewer leakage problems than field installed joints (as in collectors with external headers) [3]. Also seals and gaskets caused problems due to differential thermal expansion in the internal and external piping. This latter problem is only one of the installation problems which have arisen in connecting collectors. In many cases the inlet and outlet tubes are not located conveniently on the collector. In some cases the attachment of the piping from the absorber plate has broken off the frame, making it virtually impossible to tighten any fitting connection to the collector module. In most cases, however, the major problem is the need for additional space in order to properly connect two or more collector modules or the collector module to the headers. This results in a larger area which must be protected from snow, ice, dust and dirt, etc. In cases where the plumbing connections were directed downwards, problems arose between the roof penetrations and the roof structure (rafters, joists, etc.) In general, installations should avoid multiple roof penetrations. Conversely, with proper connections, externally manifolded headers can often offer greater installation flexibility.

Ground-mounted collectors have not always been installed with their lowest point at least one or two feet (0.3 to 0.6 m) above the ground so that accumulation of snow has resulted in covering part of the collector. In other cases, shading by trees and other buildings has reduced the effectiveness of the solar collector array for portions of the year.

Sufficient attention has not always been paid to the aesthetics while mounting the collectors. The use of dormers for mounting collectors at a steeper tilt than the roof can enhance the aesthetics, provide better resistance to wind loads [1] and reduce heat losses from the collectors [3]. The use of dormers can also avoid snow and debris accumulation.

Some collectors have adhesive paper covering the top cover plate for protection. Removing the paper in freezing weather has created problems due to the paper becoming brittle when it is very cold. Another problem is that at high temperatures the glue can spread on the cover plate in a thin layer and can be very difficult to remove.

Air locks have occurred in the collector loop because of improper fill operations, lack of proper air vents, and poor piping design. Some systems lack a proper fill mechanism which make it difficult to fill the system after draining it for maintenance. The collector loop liquid should be tested as per manufacturer's specifications (about twice a year) and its pH (acidic concentration, hydrogen ion concentration) measured. This is necessary for corrosion control and for checking whether the liquid needs to be replaced. Lack of maintenance and occasional monitoring are some of the factors leading to early corrosion in many collector systems.

In addition to the above, there have been problems relating to the collector support structures where (a) the roof trusses had the wrong pitch for solar collector mounting in some of the projects. In some projects, it was possible to remedy this by replacing them with trusses of the proper pitch and (b) in another project, the roof structure was unable to accommodate the original collector configuration. The solution was to redesign the system, eliminate use of some of the collectors, and develop a new array support design.

Other collector mounting problems have included: The mounting brackets for the collector were not usable in some systems, in others they were not approved by the city engineer; the clamps for the collector would not hold or fasten because it was impossible to attach a flat washer to a round surface; in a site-built collector system, the plexiglass cover sheets sagged between the roof rafters; leakage of the fittings to the collector manifold and in the collector joints occurred, requiring resoldering, recaulking, and reinstallation of protective covers for the piping. Also, collector outlet nipples were not installed perpendicular to the piping, causing a fitting problem and leading to additional work to make the proper connections. A complaint often made by some installers of collector systems is regarding the difficulty of working on a steep pitched roof. Apart from the increased cost of labor, the workmanship can suffer because of this difficulty. In many installations, provision is not made for easy access to the collectors, which may be required for later maintenance work. A working area is desirable and should have a durable surface. Local building codes should be consulted in this regard.

2.3 HEAT TRANSFER FLUIDS

Collector heat transfer fluids have been discussed earlier. Because questions of durability and reliability of the collector heat transfer fluid are important, the reader should refer to both sections in considering the selection of a heat transfer fluid. It is of course common for the system fluid used in the load loops to be different from the collector fluid. Much of the information below is taken from reference [10].

2.3.1 Air as a Heat Transfer Fluid

The majority of solar heating and cooling systems utilize air, water, and water/glycol mixtures. Air has been used for both space and DHW heating applications. There have been instances where air-heating DHW systems have experienced freezing problems in the air-to-water heat exchanger (see Section 2.7.1) and suffered from excessive air leakage in ducts and collectors (see Section 4.5.3.4), resulting in reduced performance (see Section 5.1).

Air system designers should also be aware of possible problems associated with the quality of air. Dust accumulation can lead to large pressure drops in the collector and storage circuits and even transfer this dust to the living space. In maintaining proper HVAC practices and in avoiding costly cleaning, filters should be provided at appropriate points in the ductwork. Deposition of dust and other impurities in rock storage in the presence of moisture may, under certain operating conditions, lead to bacteria growth. Some experiences have shown, however, that a storage temperature of 140°F (60°C) is sufficient to eliminate any algae growth. The concern of some people about nuclear radiation problems from rocks in the storage bed has been found to be unsubstantiated; there have been no reports of radiation problems to date [11].

It should be noted that dust and smog related health problems, if any, are not the results of solar systems. Natural infiltration of air into a house can lead to the same problems irrespective of the presence of solar equipment.

In comparing air and liquid as heat transfer fluids, air has the distinct advantage of not leading to freezing, boiling or corrosion problems. These same problems are of major concern in liquid-heating systems. The primary disadvantages of air systems are reflected in comparing performance with liquid systems (see Section 5.3.6). The relatively low heat capacity of air, together with air leakage, may sometimes lead to serious deteriorations in performance. Furthermore, air systems do not easily lend themselves to solar cooling applications.

2.3.2. Water as a Heat Transfer Fluid

From a performance viewpoint, water is the most thermally efficient heat transfer fluid used. Water provides high efficiencies for use in solar collectors, heat exchangers, and other components. On the other hand, from a durability and reliability viewpoint, water can freeze, boil, lead to corrosion, and cause some scaling of heat exchanger surfaces.

Water in the presence of dissimilar metals can cause galvanic corrosion. The experienced installer should be aware of the local pH and mineral hardness problems because of the effects of hard water on conventional hot water systems. If water quality conditions are extreme, installation of a water softener to protect the solar DHW system should be considered. Softening will help to reduce scaling but it increases the potential for corrosion. Metals in water precipitate out at 160°F (70°C). Furthermore, softening adds sodium, a highly conductive metal and such precipitation generally leads to scaling in the lower velocity collector tubes. Whatever the local water conditions, a qualified water treatment engineer should be consulted to prescribe a treatment to make problem water safer for plumbing materials.

Freezing is one of the major concerns in using water as the heat transfer fluid. Two types of designs that do not utilize low freezing liquids in the collector loop are:

Drain-down systems - The water in the collector is drained whenever near-freezing conditions of the collector loop piping are reached (e.g., temperatures of 35-40°F, 2-5°C) and typically utilize automatic control valves which open, allowing the water to drain into storage

Drain-back systems - The water in the collector is drained whenever the collector pump shuts off (automatic control valves are not required). In this case the piping circuit has to be carefully designed.

Both types of systems require care in designing in order to ensure that they drain completely and automatically (see Section 2.7.1). In addition, thermal shock may occur when refilling the collector loop (see Section 2.2.1.2.2).

2.3.3. Water/Glycol Solutions

Care must be taken that the proper glycol/water mixture and heat exchanger have been specified. Ethylene glycol/water mixtures are toxic and require a double-walled heat exchanger for DHW systems or whenever there is a possibility of mixing with the potable water system. Foodgrade propylene glycol (U.S.P.)/water mixtures -- when certified non-toxic -- may be used with a single-walled heat exchanger if no toxic dyes or inhibitors have been added to the mixture. However, care should be exercised in not replacing with toxic glycols. In this regard, designers should consult local codes! The effect of boiling on glycol is discussed in the subsection on boiling problems.

Water/glycol solutions should be at least 25 percent glycol in order to prevent freezing in most parts of the United States. Freeze protection down to 10°F (-12°C) below the historic low of the region is recommended. A 50/50 solution is good down to about -32°F (-35°C) and maximum freeze protection is achieved with a 40/60 water/glycol mixture. Designers/installers are urged to refer to manufacturer's recommendations, since antifreeze solutions designed specifically for automobiles do not all possess the same properties.

Lower concentrations of glycol (20 to 25 percent) in water at low temperatures may cause the crystallization of the water but not the glycol, leading to a slush. In this case pipe will not burst, since the volume change will be compensated for in the expansion tank, however the higher concentrations of glycol are recommended to ensure adequate freeze protection.

Glycol solutions should not be used with zinc galvanized plumbing because the required corrosion inhibitors react with zinc. Glycols may damage certain materials such as the butyl rubber membranes in certain types of expansion tanks. If water/glycol mixtures are exposed to air through an air vent or a vacuum breaker at high temperatures, acids will form. If these conditions occur, the pH, inhibitor strength, and solution concentration of the water/glycol must be checked and the solution replaced if necessary. Periodic checks and replacement will be required in any case. In order to take advantage of antifreeze corrosion inhibitors, it is necessary to utilize a minimum concentration of glycol of about 30 percent.

Glycol solutions can leak through joints where water would not. Good seals and/or tape should be used. Glycols should be dyed with non-toxic food coloring dye (if not bought that way) to help identify leaks. Make-up supply, in case of leaks, should not be added automatically from the city water supply as this will reduce the glycol concentration. DEPENDING ON LOCAL CODES, water/glycol solution should be drained into dry wells or waste drains and not sanitary or storm sewers.

2.3.4 Other Heat Transfer Fluids

Each heat transfer fluid has differing properties, such as viscosity, specific heat, freezing, boiling, and flash points that will determine the size and design of many components (see Table 1).

Table 1. Properties of Heat Transfer Fluids

Medium	Specific Gravity	[1]Viscosity Centipoise	[2]Heat Capacity (Btu/lb,°F)	[3]Freezing Point °F
Water	1.00	0.5 to 0.9	1.0	+32
50 wt. % Water-Ethylene glycol	1.05	1.2 to 4.4	0.83	-33
50 wt. % Water-Propylene glycol	1.02	1.4 to 7.0	0.85	-28
Paraffinic oils	0.82	12 to 30	0.51	+15
Aromatic oils	0.85	0.6 to 0.8	0.45	-100
Silicon oils	0.94	10 to 20	0.38	-120

[1] Because viscosity is sensitive to temperature, values are given for a temperature range of approximately 80 to 140°F (26 to 60°C)

[2] Multiply [Btu/lb,°F] by 4.19 to get [KJ/kg,°C]

[3] °C = (°F - 32)/1.8

2.3.4.1 Paraffinic Mineral Oils

Paraffinic or mineral oils are petroleum based heat transfer fluids. Their useful temperature range between freezing and boiling is greater than that of water and they are electrically non-conducting. These oils have a higher viscosity than water and may require a larger pump. Because they will break down into tar-like materials under prolonged exposure to heat, periodic replacement is necessary.

Paraffin oils are considered toxic and usually require a double-walled heat exchanger for DHW or for connections to potable water (see building codes and/or the HUD Minimum Property Standards). The flash point of paraffinic oils may be subject to code restrictions.

2.3.4.2 Silicone Liquids

Silicone heat transfer liquids are quite inert and will cause neither galvanic corrosion nor degradation of roofing materials. They also have a very high flash point. These fluids have high viscosities and low specific heats compared to water; therefore large pumps and a flow rate of about 2.5 times that used for water are typical in order to remove the same heat at the same rate (in the same temperature range). Lower flow rates may use less electrical power, but only at reduced efficiency. Silicones are incompatible with expansion tanks fitted with neoprene or butyl rubber diaphragms.

Silicone liquids will leak readily through piping flaws and pump seals that would otherwise retain water. Even sweat-soldered joints in copper pipe will leak if not properly soldered. A manufacturer-recommended pipe sealant should be used at all threaded joints. Check all manufacturer supplied connections for proper sealant.

Avoid using silicone tubing or silicone sealants in the system. Sealless (canned) pumps or pumps with magnetic drives can be used. Some assurance should be provided that the pump is operating when the motor is turned on. Use non-acidic flux when soldering to prevent contamination of the neutral heat transfer fluid.

2.3.4.3 Aromatic Oils

Aromatic oils have lower viscosities than paraffins; this allows smaller pumps to be used. They also have lower flash points, which make them less safe to use.

Aromatics will dissolve roofing tar and most elastomer seals. Viton seals should be used in pumps whenever paraffinic or aromatic hydrocarbon oils are used.

2.3.4.4 Water/Glycerine Solutions

A 40/60 solution of water/glycerine (glycerol) is non-toxic and is sometimes used without a heat exchanger with double separation for DHW or with connections to the potable water supply. Water/glycerine solutions have higher viscosities than water/glycol solutions and, therefore, may require a larger pump. Also, glycerine solutions are subject to biological contamination and may become corrosive if overheated.

2.3.5 Corrosion

The combination of dissimilar metals and heat transfer fluids that conduct electricity will lead to some galvanic corrosion where the more chemically active metals are attacked. It can be avoided by:

1) Using a non-conductive heat transfer fluid such as silicone or hydrocarbon oil.
2) Using one metal throughout the whole system. If the metal is copper and the transfer fluid is water, there is no need to add corrosion inhibitors unless there are water softeners in an open system.
3) Using an air-to-water system with no dissimilar metals on the water side.

Water/glycol antifreeze mixtures require an added inhibitor because the glycol breakdown products include acids. Aluminum in the piping system will also require an inhibitor. Most commercially available heat transfer liquids are sold with inhibitors already added, but some will require the installer to formulate the proper mixture. Most of the common inhibitors carried in solution are sacrificial (the inhibitor is attacked rather than the plumbing) and, therefore, require the installer or the owner to follow a regular maintenance schedule to replace the transfer fluid or to update the inhibitor.

The system must be flus.... out completely prior to filling in ....er to remove solder flux, metal filings, etc. Direct connections between dissimilar metals must be avoided. The use of insulating washers (plastic, rubber) or silicone hoses between dissimilar metals will reduce galvanic reactions at that point but will not eliminate it if the liquid is electrically conductive. Note that softening increases conductivity.

In solar collector systems, as in any system involving circulating liquids, it is not sufficient to use a dielectric fitting to separate dissimilar metals from direct contact. Copper ions can be carried by the fluid and deposited on another metal, causing pitting. Although complex systems of corrosion protection are available for mixed metal systems, THE BEST SOLUTION IS SIMPLY NOT TO MIX METALS.

Thoroughly flushing out a system before filling with a liquid helps to prevent galvanic corrosion. Filings of one metal lodged in an absorber plate or heat exchanger coil of a dissimilar metal can cause galvanic corrosion.

In flushing out a water/glycol system it is best to use water only. This is because the anti-leak inhibitors in the glycol may tend to plug any filters used in the piping system.

2.3.6 Pressure Tests

Pressure tests at 1.5 times the maximum working pressure are an essential factor in providing leak-proof systems. The pressure tests should include the piping and collectors and should use air instead of expensive liquids. However, it is wise to conduct a final pressure test with the liquid to be used in the system. The manufacturer should be consulted before testing.

2.3.7 Heat Transfer Fluid Checklist

It is recommended that the designer be aware of at least the following heat transfer properties before selection. These can be obtained from manufacturers.

2.3.7.1 Design Properties

1. Normal collector operating temperature range, including start-up
2. Stagnation temperature
3. Maximum vapor pressure of fluids
4. Acceptable kinematic viscosities at the start-up temperature and at the design operating temperature
5. Maximum pumping power required per unit of power transferred
6. Expected half-cycle (years)
7. Melting point, pour point, boiling point
8. Heat of vaporization, coefficient of thermal expansion, surface tension
9. Thermal degradation temperature
10. Maximum temperature recommended for long-term use
11. Specific heat, thermal conductivity, viscosity, density, vapor pressures - at several temperatures in the operating range

2.3.7.2 Handling Properties

1. Fire resistance, flash point, fire point, autoignition temperature, oxygen index (percent oxygen)
2. Physical appearance
3. Compatibility with metals, plastics, elastomers and other construction materials at 70°F (20°C) and at maximum use temperature
4. Chemical sensitivity of the fluid to the following substances: Water, inorganic bases, trace quantities of strong acids, chloride ions, soldering and welding fluxes, oxygen
5. Solvents with which the fluid is immiscible
6. Physiological effects
7. Biodegradability characteristics
8. Recommended fire extinguishing agents for the fluid

2.3.7.3 Other Information

1. Current price per gallon - 5 gallon cans, 55 gallon cans, tank truck lots
2. Other

2.3.8. Experiences with Heat Transfer Fluids

In the next few pages are presented the responses to a questionnaire directed to designers and manufacturers of solar energy collectors and collection systems. This survey was done by Monsanto Research Corporation and the results are extracted from the following publication: "Superior Heat Transfer Fluids for Solar Heating and Cooling Applications", September 1979, prepared by Monsanto Research Corporation for the U.S. Department of Energy under contract number EM-78-C-04-5356 [12].

Table 2. Manufacturers of Flat-Plate Collectors and the Heat Transfer Fluids they Use or Recommend [12]

Mfr. org. no.	Activity: Collectors	Activity: Systems	Collector surface area[a] (ft^2)	Collector surface area[a] (m^2)	Fluids used[b]
C-3	✓	✓			Freon® Glycol-water solution Silicone Water
C-6	✓	✓	45,000	4,181	Dowfrost® Heat transfer fluid ST-92
C-9	✓	✓	15,000	1,394	Ethylene glycol-water solution Glycol Propylene glycol-water solution Water
C-18	✓	✓	1,200	111	R-114
C-19	✓	✓	5,000	465	Water
C-26	✓	✓	22,000	2,044	Water
C-29	✓		5,000	465	Sun-Temp
C-30	✓		125,000	11,613	Dowfrost® Solargard G
C-36	✓		2,000,000	185,806	Water
C-38	✓	✓	600	56	Propylene glycol NUTEK Silicone
C-39	✓	✓			Ethylene glycol Glycerine
C-40	✓		7,200	669	Water
C-48	✓		50,000	4,645	Prestone II -water solution
C-63		✓	5,000	465	Distilled water, containing 2% Virco Pet 31 corrosion inhibitor
C-75	✓	✓	2,500	232	Sun-Temp
C-80			75,000	6,968	Water
C-94	✓	✓	75,000	6,968	Prestone II
C-97	✓	✓			Water
C-108	✓	✓	10,000	929	Suntherm HTF-100
C-115	✓		2,000	186	
C-122	✓	✓	450,000	41,806	
C-123	✓	✓	45,000	4,181	UCAR Food Freeze
C-124	✓	✓	100,000	9,290	Water
C-130	✓		90,000	8,361	Glycol-water solution
C-137		✓	8,000	743	Water
C-146	✓		3,000	279	Propylene glycol
C-153	✓		300	28	Prestone II
C-165	✓		5,000	465	Thermia C
C-166		✓	1,000	93	Glycol-water solution
C-167		✓	300	28	Prestone II
C-169		✓			Glycol-water solution
C-172	✓	✓	200	19	Water
C-196	✓	✓	30,000	2,787	Water
C-197	✓	✓	7,000	650	Low-viscosity heat transfer oils

(continued)

TABLE 2 (continued)

Mfr. org. no.	Activity		Collector surface area[a]		Fluids used[b]
	Collectors	Systems	(ft^2)	(m^2)	
C-199	✓	✓	7,000	650	Sun-Temp
C-200	✓				Therminol® 44
C-208	✓	✓	100,000	9,290	Dowfrost®-water solution (50/50)
C-215	✓	✓	2,000	186	Water
C-218	✓	✓	50,000	4,645	Water (potable)
C-219	✓		25,000	2,323	Dowfrost®
					Dowtherm® SR-1
C-222		✓	5,000	465	Experimental fluid for extreme temperature range
					Experimental fluid for mid-temperature range
C-224	✓	✓	600,000	55,742	Glycol-water solution
					Water
C-230	✓		600	56	Ethylene glycol
C-236	✓		320	30	Water
C-254	✓	✓	2,200	204	Potable water
C-257	✓	✓	25,000	2,323	Ethylene glycol
					Propylene glycol
					Water
C-258	✓	✓	4,000	372	Dow-Corning Q2-1132
C-259	✓		512	48	
C-261	✓		5,000	465	Silicone fluid SF-96(500)
C-263	✓	✓	660,000	61,316	Water
C-264	✓		2,200	204	Sun-Temp
C-280	✓	✓	75,000	6,968	Dowtherm® A
					Dowtherm® J
					Glycols
C-285	✓	✓			Sun-Temp
C-294	✓	✓	3,000	279	Water (with and without inhibitor for algae)
C-295	✓				
C-301	✓	✓	25,000	2,323	Water
C-302	✓	✓	10,000	929	Water
C-313	✓	✓	8,000	743	Water
C-319		✓	1,700	156	Propylene glycol
					Water
C-324	✓		4,000	372	Solar Winter-Bar
C-326		✓	15,780	1,466	Ethylene glycol
					Water
C-331	✓	✓	8,000	743	
C-337	✓	✓	14,000	1,301	Silicone
C-354	✓		3,000	279	Diala® AX
					H-30 Solar Collector Fluid
					Sun-Temp
					Syltherm™ 444

(continued)

TABLE 2 (continued)

Mfr. org. no.	Activity		Collector surface area[a]		Fluids used[b]
	Collectors	Systems	(ft^2)	(m^2)	
C-356	✓		1,000	93	Mineral oil
C-367	✓	✓	10,000	929	Glycol
					Proprietary fluid
C-380	✓	✓	200	19	Syltherm™ 444
					Water
C-395		✓	10,000	929	Water
C-398	✓	✓			Sunsol 60
C-405	✓		6,000	557	
C-409	✓		4,620	429	Sun-Temp
C-427	✓				Automotive transmission oil
					Water
C-432		✓	1,000	93	Dow-Corning Q2-1132
C-435	✓	✓	4,000	372	Dowfrost® (with 1.75% dipotassium phosphate)
					Propylene glycol
C-439		✓	12,000	1,115	Propylene glycol
					Sunsol 60
					Water
C-441	✓	✓	8,000	743	Water (with inhibitor)
C-450	✓	✓			Syltherm™ 444
C-463	✓	✓	50,000	4,645	H-30 solar Collector Fluid
					Dowfrost®
					Dowtherm® SR-1
					Drewgard 100 (additive)
					Sun-Temp
C-465	✓	✓	20,000	1,858	Dowfrost®
					Dowtherm®
					Sun-Temp
C-469		✓	1,000	93	Brayco® 888
					Sunsol 60
					Sun-Temp
					Syltherm™ 444
C-477	✓	✓	500	46	Propylene glycol (with pH indicator and amine-type oxygen-scavenging inhibitor)
C-489	✓	✓	3,500	325	Sun-Temp
					Therminol® fluids
C-492	✓	✓	30	3	
C-493	✓		640	59	Water (potable)
C-499	✓		1,000	93	Dowtherm®
					Propylene glycol
					Solargard
C-501	✓		200	19	Water (distilled)
C-503	✓		7,200	669	Water
C-523	✓	✓	2,500	232	Propylene glycol
C-567	✓	✓	2,400	223	
C-568	✓	✓	9,820	912	Prestone II
					Water
C-575	✓		60,000	5,574	Glycol-water solution
					Syltherm™ 444
					Water
C-576	✓	✓	6,000	557	Water
C-578		✓	5,500	511	Water (with Nalco 8334 inhibitor)
C-580		✓	9,000	836	Water

[a]Approximate annual production rate in 1977.

[b]The manufacturers of most fluids listed in this table are identified in Table 13.

Table 3. Flat-Plate Collectors; Problems Encountered with Selected Fluids [12]

Mfr. org. no.	Fluids used	Problems Yes	Problems No	Problems encountered with the selected fluids
C-3	Ethylene glycol-water solution Freon® Silicone Water		✓	
C-6	Dowfrost® Heat transfer fluid ST-92		✓	
C-9	Ethylene glycol Ethylene glycol-water solution Propylene glycol-water solution Water	✓		Degradation in stagnating system.
C-18	R-114		✓	R-11 and R-12 are alternate suitable fluid, depending upon the application.
C-19	Water		✓	
C-26	Water	✓		Formation of mineral deposits from tap water. Occasional growth of algae in captive water.
C-29	Sun-Temp	✓		Fluid attacks rubber seals; special seals are required.
C-30	Dowfrost® Solargard G	✓		Fluids have degraded under stagnation conditions. Originally alkaline solution have become acidic (pH = 4.6 to 5.1).
C-36	Water		✓	
C-38	NUTEK Propylene glycol Silicone	✓		Degradation of NUTEK. Mild toxicity of propylene glycol. High viscosity of silicone, necessitating the use of pumps of higher power requirements.
C-39	Ethylene glycol Glycerine		✓	
C-40	Water		✓	
C-43	Therminol® 66 Silicone	✓		The fluids place an upper temperature limit (316°C, 600°F) on the collector operation. The operating temperature limit of the collector itself is 538°C (1,000°F). The fluid must also be pumpable at temperatures down to -8°C (10°F). We found Therminol® 66 to perform closest to our requirements.
C-48	Ethylene glycol-water solution		✓	
C-63	Distilled water, containing 2% Virco Pet 31 corrosion inhibitor		✓	
C-75	Sun-Temp		✓	
C-94	Prestone II		✓	
C-97	Water		✓	
C-108	Suntherm HTF-100		✓	
C-123	UCAR Food Freeze		✓	
C-124	Water		✓	
C-130	Glycol-water solution	✓		A minor problem with internal corrosion of carbon steel absorbers when the inhibitor was not added initially.
C-137	Water		✓	
C-146	Propylene glycol		✓	
C-153	Prestone II		✓	
C-166	Glycol-water solution		✓	
C-167	Prestone II		✓	
C-169	Glycol-water solution	✓		Corrosion.
C-172	Water		✓	
C-196	Water		✓	
C-197	Low-viscosity heat transfer oils	✓		Destruction of roofing materials and staining. Leakage through threaded plumbing fittings. Increase of viscosity at low temperatures.
C-199	Sun-Temp		✓	
C-200	Therminol® 44	✓		The fluid leaked through the seal of the circulating pump. Since no nitrogen blanket was used, the fluid also decomposed.
C-208	Dowfrost®-water solution (50/50)		✓	
C-215	Water		✓	
C-218	Water (domestic)		✓	
C-219	Dowfrost® Dowtherm® SR		✓	Leakage, caused by low surface tension, presents a minor problem.
C-222	Proprietary fluids	✓		Testing laboratories have been unable to certify due to lack of proper instrumentation.
C-224	Glycol-water solution Water	✓		
C-230	Ethylene glycol		✓	
C-257	Ethylene glycol with inhibitor Propylene glycol with inhibitor Water (potable)	✓		Cost of the glycols. High power requirement for the pumping of glycols.
C-258	Dow Corning Q2-1132		✓	
C-259	Water		✓	
C-261	Silicone fluid SF-96(500)	✓		Silicone fluid leaks through the pump.

(continued)

TABLE 3 (continued)

Mfr. org. no.	Fluids used	Problems Yes	Problems No	Problems encountered with the selected fluids
C-263	Water		✓	
C-264	Sun-Temp		✓	
C-277	Water		✓	
C-280	Dowtherm® A Dowtherm® J Glycols	✓		Oils degrade rubber gaskets. Degradation of glycols causes corrosion. Low heat transfer efficiency of these fluids.
C-285	Sun-Temp		✓	
C-294	Water (with and without inhibitor for algae)		✓	
C-301	Water	✓		Deposits from very hard water have caused poppet valve seats to leak and check valve pivots to seize.
C-302	Water		✓	
C-313	Water		✓	Lime scale buildup due to poorly maintained systems.
C-316	Sun 21		✓	
C-319	Propylene glycol Water		✓	
C-324	Solar Winter-Bar		✓	
C-326	Ethylene glycol Water		✓	
C-337	Silicone	✓		Low heat transfer efficiency.
C-354	Diala® AX H-30 Solar Collector Fluid Sun-Temp Syltherm™ 444			Degradation of rubber and leakage.
C-356	Mineral oil		✓	
C-367	Glycol Proprietary fluid		✓	
C-380	Syltherm™ 444 Water		✓	
C-395	Water		✓	
C-398	Sunsol 60		✓	
C-409	Sun-Temp	✓		Bleeding of asphalt when leakage occurs on an asphalt tile roof.
C-427	Automotive transmission oil		✓	
C-432	Dow-Corning Q2-1132		✓	
C-435	Dowfrost® (with 1.75% dipotassium phosphate)		✓	
C-439	Propylene glycol Sunsol 60 Water	✓		Economic/performance problems. When glycols are used instead of water, additional expenses are incurred because of additional circulator and heat exchanger requirements.
C-441	Water (with inhibitor)		✓	
C-450	Syltherm™ 444	✓		Leakage with poor joints. Air venting on startup.
C-463	H-30 Solar Collector Fluid Dowfrost® Dowtherm® SR-1 Drewgard 100 (additive) Sun-Temp	✓		The relatively high vapor pressure of glycol solutions allows boiling to occur under some conditions, increasing the probability of airlock formation and requiring the use of large expansion tanks to prevent buildup of excessive pressures. Gaseous products are generated from glycol solutions through degradation and electrolytic activity. Hydrogen is one of the constituents. The generation of gaseous products increases system pressure and creates safety hazards. Corrosion appears to be a minor problem with glycol solutions. A major objection to the "oily" fluid is their poor heat transfer efficiency. Additionally, they are generally not compatible with the nonmetallic seat and seal materials commonly used in the pumps, valves and expansion tanks of hydraulic systems. The use of compatible materials for these components entails significant additional cost. The "oily" fluids also dissolve some components from common building materials (i.e., asphalt shingles) and cleaning is difficult if leakage occurs.
C-465	Dowfrost® Dowtherm® Sun-Temp		✓	

(continued)

TABLE 3 (continued)

Mfr. org. no.	Fluids used	Problems Yes	Problems No	Problems encountered with the selected fluids
C-469	Brayco® 888 Sunsol 60 Sun-Temp Syltherm™ 444	✓		Sunsol 60 needs to be replaced every 3 years. Compositional information regarding this fluid could not be obtained from the manufacturer. The silicone fluid is very viscous.
C-477	Propylene glycol (with pH indicator and amine-type oxygen-scavanging inhibitor)		✓	
C-493	Water (potable)		✓	
C-499	Dowtherm® Propylene glycol Solargard	✓		Glycols have short service life. Dowtherm is expensive for the clients.
C-501	Water (distilled)	✓		Corrosion problems were encountered when pool water was used for heat transfer.
C-503	Water		✓	
C-523	Propylene glycol		✓	
C-575	Glycol-water solution Syltherm™ 444 Water		✓	
C-576	Water		✓	
C-578	Water (with Nalco 8334 inhibitor)		✓	
C-580	Water		✓	

Table 4. Flat-Plate Collectors; Physical Performance Requirements for Fluids [12]

Mfr. org. no.	Operating temperature range (°C)	Operating temperature range (°F)	Stagnation temperature (°C)	Stagnation temperature (°F)	Maximum vapor pressure (atm)	Maximum vapor pressure (psi)	Viscosity Startup	Viscosity Operation	Pumping power[a]	Half-life (years)
C-3	≲71	≲160	204 to 260	400 to 500						10
C-6	≲77	≲170	154	310						
C-9			149 to 204	300 to 400	27	400				20
C-26	2 to 82	36 to 180	177	350	1.0[b]	15			0.005 to 0.029	
C-29	-1 to 93	30 to 200	149	300	0.7	10				20
C-30	16 to 107	60 to 225	204	400			8 cp	2 cp		2.5
C-36	29	85	85	185						
C-38	-1 to 82	30 to 180	177	350						5 to 10
C-39			82	180						10
C-40	-18 to 82	0 to 180	177	350						
C-48	≤82	≤180	135, 177	275,[c] 350[d]						2
C-63	≤157	≤250	154	310						
C-75	-29 to 82	-20 to 180	204	400	2.0	30				
C-94	-29 to 260	-20 to 500	≤482	≤900	51	750				10
C-97	16 to 90	60 to 195			10.2	150				
C-108	38 to 71	100 to 160	138	280						
C-115	≤116	≤240	260	500						
C-122	16 to 82	60 to 180	135	275	8.5	125			0.05	25
C-123	16 to 104	60 to 220	135	275						
C-124	27 to 93	80 to 200	149	300						
C-130			204	400						
C-146	27 to 77	80 to 170	121	250						2.5
C-153	21 to 82	70 to 180	204	400	4.1	60				
C-165	-23 to 93	-10 to 200	116	240						
C-166	1 to 93	33 to 200								>20
C-169	-29 to 66	-20 to 150			4.0	59			0.03	20
C-172	4 to 66	40 to 150	149	300						
C-196	16 to 93	60 to 200	149 to 204	300 to 400[e]	1.0	15	<1 cs		<0.02	
C-197	≤71	≤160	≤177	≤350			–[f]		0.026	
C-199	-23 to 149	-10 to 300	∿149	∿300						20
C-208	≤93	≤200	232	450						20
C-215	4 to 82	40 to 180	177 to 204	350 to 400						7
C-218	4 to 82	40 to 180							0.17	
C-219	-18 to 82	0 to 180	≤204	≤400	5.9 8.6	87[g] 127[h]				
C-222	30 to 104	90 to 220	≤246	≤475	3.4	50				20
C-224	27 to 107	80 to 225	204	400						
C-230	60 to 116	140 to 240	116	240	1.7	25				10
C-254	≤88	≤190	≤177	≤350						
C-257	27 to 66	80 to 150	163	325						
C-258	-18 to 149	0 to 300								
C-261	-29 to 82	-20 to 180	177	350						
C-263	50 to 100	122 to 212	160	320						
C-277	-46 to 93	-50 to 200	177	350						10
C-280	-37 to 104	-35 to 220	≤196	≤385					0.025	10
C-294	-38 to 88	-37 to 190	≤204	≤400	8.5	125			0.05	25
C-295	≤82	≤180	121 to 163	250 to 325						

(continued)

TABLE 4 (continued)

Mfr. org. no.	Operating temperature range (°C)	(°F)	Stagnation temperature (°C)	(°F)	Maximum vapor pressure (atm)	(psi)	Viscosity Startup	Operation	Pumping power[a]	Half-life (years)
C-301	-29 to 99	-20 to 210	≤204	≤400						
C-302	10 to 71	50 to 160	121	250					0.0372	
C-313	24 to 66	75 to 150	177	350						
C-319	10 to 88	50 to 190	204	400						
C-324	30 to 60	90 to 140	204	400						25
C-326	-18 to 121	0 to 250	121, 149	250, 300						
C-331	16 to 93	60 to 200	185	365						10
C-337	21 to 110	70 to 230	288	550	100	1,470	50	10		20
C-380	-34 to 193	-30 to 380	193	380						
C-395	-18 to 177	0 to 350	177	350	2.4	35				
C-398	≤60	≤140	149	300						
C-409	≤71	≤160	149	300	5.4	80				>20
C-427	4 to 104	40 to 220	177	350	24	350				
C-432	27 to 82	80 to 180	149	300	8.2	120				5
C-435	≤100	≤212	150 to 180	302 to 356	2.4	35		2.5 - 3 cp		4
C-439	10 to 62	50 to 180	177 to 204	350 to 400	2.0	30				>10
C-441	16 to 82	60 to 180								
C-450	≤71	≤160	204	400						10
C-463	-18 to 104	0 to 220	191	375	0.07	1[i]	20 cp[j]	1 to 5 cp[k]		
C-465	-29 to 149	-20 to 300	204	400			50 cp	1 cp	0.021	2
C-469	-40 to 93	-40 to 200	177	350	4.1	60	100 cs[l]	3 cs[m]	0.013	40
C-477	32 to 49	90 to 120	88	190		65			0.04	20
C-489	≤93	≤200	204	400	4.4	100		3 cp[m]		10
C-492	1 to 102	33 to 215	149	300						15
C-493	21 to 99	70 to 210	204	400						
C-499	-18 to 116	0 to 240	135 to 232	275 to 450	2.0	30	8 cs[n]	0.8 cs[o]		5
C-501	100 to ~150	212 to ~302	>150	>302	0.7	10				>20
C-503	-18 to 82	0 to 180	177	350						
C-523	-29 to 100	-20 to 212	>135	>275						
C-568	-29 to 82	-20 to 180	83	182						25
C-575	32 to 82	90 to 180	204	400						
C-576	27 to 82	80 to 180	177 to 204	350 to 400						10
C-578	2 to 116	35 to 240	260 to 316	500 to 600			1 - 6 cp			5
C-500	≤121	≤250	121	250						

[a] The required power is expressed as a fraction of power delivered by the solar collector.

[b] At 100°C (212°F).

[c] Single-glazed collector.

[d] Double-glazed collector.

[e] Dependent upon the absorbance of the coating.

[f] Rapid flow at startup temperature is not essential.

[g] At 38°C (100°F).

[h] At 149°C (300°F); the maximum value.

[i] Desired vapor pressure.

[j] At 21°C (70°F).

[k] At 54°C (130°F).

[l] At -40°C (-40°F).

[m] At 93°C (200°F).

[n] At 27°C (80°F).

[o] At 99°C (210°F).

Table 5. Flat-Plate Collectors; Fire Resistance Requirements for Fluids [12]

Mfr. org. no.	Maximum hot surface exposure temperature (°C)	(°F)	Fire resistance specifications associated with the specific collectors	Codes and regulations
C-3	204 to 260	400 to 500	Proposed specification of flash point 56°C (100°F) above the temperature of the hottest surface which the fluid may contact.	
C-6	204 to 316	400 to 600		
C-9	204	400		HUD minimum property standards.
C-26	177	350		
C-29	93	200	Not combustible at stagnation temperature.	
C-30	204	400		None encountered to date.
C-38	82 to 93	180 to 200	Must be essentially nonflammable.	
C-48	177	350		
C-63	154	310	We intend to use nonflammable fluids.	
C-94	482	900		
C-108			Flash point 140°C (300°F).	
C-123	163	325		HUD minimum property standards.
C-124	149	300	Noncombustible.	
C-146			Should not ignite upon impingement onto the surface of an oil-burning furnace.	
C-166	204	400	Systems sold nationwide. Fluids must meet the codes of all states.	
C-169	150	302		
C-172	149	300	Nonburning	
C-196			Nonflammable.	
C-197	177	350	Intend to use nonflammable fluids.	
C-199	149	300		
C-208	232	450		
C-215	204	400		
C-218	171	340	Stagnation temperature.	
C-219	82	180	204°C (400°F) flash point. Leak during normal operation.	
C-222	116	240	Fluid should be nonflammable.	
C-254			Since water is used, there are no fire resistance specifications.	
C-257	163	325		
C-263	71	160		
C-280				Must meet HUD minimum property standards.
C-301	204	400	Leak onto collector surface during stagnation.	
C-302	93	200		
C-319	204	400		

(continued)

TABLE 5 (continued)

Mfr. org. no.	Maximum hot surface exposure temperature (°C)	(°F)	Fire resistance specifications associated with the specific collectors	Codes and regulations
C-326	149	300	To date, 121°C (250°F).	
C-337	288	550		
C-380	190	380	Absorber plate temperature.	
C-395	177	350	Must be totally fireproof.	
C-427	85	185		
C-432	149	300	Nonflammable.	
C-435	150	302	Fluid should be nonflammable.	
C-439	177 to 204	350 to 400	Should be fire-resistant in collector at stagnation.	
C-441	154	310	Nonflammable.	
C-450	400	752		
C-465	204	400	Nonflammable.	
C-469	177	350	Flash point above 232°C (450°F).	HUD minimum property standards.
C-489	204	400	Absorber plate under stagnation conditions.	
C-492	121	250		City of Los Angeles code.
C-499	232	450	Ignition properties of fluids must comply with HUD Standard S-515-8.2.2.	HUD Minimum Property Standards of 1977.
C-501			Must be nonburning, nontoxic and noncorrosive. If a leak would occur, the fluid would impinge onto a surface at a temperature above 150°C (302°F).	Los Angeles County Solar Code, 1978, by Building and Safety Divisons, Department of County Engineer.
C-523				NFPA.
C-568	54	130		Nonflammable fluids or protection from flames to satisfy building codes.
C-575	204	400		
C-576			Completely fire-resistant.	
C-578			Flash point above 260°C (500°F).	

Table 6. Flat-Plate Collectors; Compatibility Requirements for Fluids [12]

Mfr. org. no.	At max. op. temp. (°C)	At max. op. temp. (°F)	Metals: At 21°C (70°F)	Metals: At max. op. temp.	Plastics: At 21°C (70°F)	Plastics: At max. op. temp.	Elastomers: At 21°C (70°F)	Elastomers: At max. op. temp.	Other materials: At 21°C (70°F)	Other materials: At max. op. temp.
C-3			Aluminum	Aluminum						
			Copper	Copper						
C-26	99	210	Copper	Copper						
C-29							Neoprene	Neoprene		
C-30	204	400	Copper	Copper			Silicone rubber	Silicone rubber		
C-39			302 SS		MMA	MMA	SBR		60/40 solder	60/40 solder
			6061 aluminum		HDPE	HDPE	Buna-N			
			Silver		PVC		Viton			
			Copper		Nylon		RTV silicone			
					Epoxy					
C-63			Aluminum	Aluminum	Nylon	Nylon	Silicone rubber	Silicone rubber		
C-94			Copper	Copper						
			Stainless steel	Stainless steel						
			Low carbon steel	Low carbon steel						
C-97			Copper	Copper						
C-108	71	160	Aluminum	Aluminum	PVC	PVC	Neoprene	Neoprene		
			Copper	Copper	CPVC	CPVC	Viton	Viton		
			Brass	Brass	ABS	ABS				
C-123			Copper		Teflon		Buna-N			
			Iron				Butyl			
							Silicone			
C-146			Copper	Copper						
C-153			Copper				Polybutadiene			
C-172			Copper		Polyester					
C-196			Copper	Copper			EPDM	EPDM		
			Brass	Brass			Silicone	Silicone		
			Bronze	Bronze						
			Stainless steel	Stainless steel						
C-199			Copper	Galvanized fittings					Lead solder	
									Silver solder	
C-208	104	220	Copper	Copper			Silicone	Silicone		
			Iron	Iron						
			Stainless steel	Stainless steel						
C-215	204	400	Copper	Copper	PVC					
					CPVC					
C-218			Copper	Copper						
C-219	82	180	Aluminum	Aluminum	CPVC					
			Brass	Brass						
			Copper	Copper						
			Cast steel	Cast steel						
C-222			Copper	Copper						
			Aluminum	Aluminum						
			Ferrous metals	Ferrous metals						
C-230	116	240	Copper	Copper						
C-236			Copper	Copper						
C-257			Brass	Brass	CPVC	CPVC			Lead-tin solders	Lead-tin solders
			Copper	Copper	PVC				Silver-tin solders	Silver-tin solders
									Pump seals	Pump seals
C-261	177	350	Copper	Copper						
C-280	104	220	Copper	Copper			EPDM	EPDM		
			Steel	Steel			Polybutadiene	Polybutadiene		
C-294			Copper							
C-295			Copper				Viton			
C-301			Brass	Brass	Teflon	Teflon	Buna-N	Buna-N	Alumina seals	
			Copper	Copper						
			Stainless steel	Stainless steel						
C-302	93	200	Brass	Brass						
			Bronze	Bronze						
			Copper	Copper						
C-326			Copper	Copper	PVC	PVC				

(continued)

TABLE 6 (continued)

Mfr. org. no.	At max. op. temp. (°C)	At max. op. temp. (°F)	Metals At 21°C (70°F)	Metals At max. op. temp.	Plastics At 21°C (70°F)	Plastics At max. op. temp.	Elastomers At 21°C (70°F)	Elastomers At max. op. temp.	Other materials At 21°C (70°F)	Other materials At max. op. temp.
C-331			Brass							
			Copper							
			Galvanized metal							
C-337	110	230	Aluminum	Aluminum			Fluorosilicone	Fluorosilicone	Solder (95/5)	Solder (95/5)
			Copper	Copper			Viton	Viton		
C-395			Brass	Brass					Fiberglass	Fiberglass
			Copper	Copper						
			Stainless steel	Stainless steel						
C-432	149	300	Aluminum	Aluminum					Polystyrene,	
			Copper	Copper					polyurethane	
			Galvanized steel	Galvanized steel					and urea-for-	
									maldehyde	
									foams	
C-439	82	180	Castiron steel	Castiron steel						
			Copper	Copper						
			Stainless steel	Stainless steel						
C-441	82	180	Aluminum	Aluminum	CPVC		Neoprene	Neoprene		
			Copper	Copper	Polypropylene		Silicone	Silicone		
C-450			Aluminum	Aluminum						
			Copper	Copper						
C-463	191	375	Brass	Brass			Buna-N	Buna-N	Solders	Solders
			Copper	Copper			Butyl	Butyl		
			Steel	Steel			Ethylene-	Ethylene-		
			Zinc	Zinc			propylene	propylene		
							Neoprene	Neoprene		
							Silicone	Silicone		
C-465	149	300	Copper	Copper	Polyethylene	Polyethylene	Buna-N	Buna-N	Polyisocyanate	
			Steel	Steel	Teflon	Teflon	Neoprene		Roofing	
									materials	
									Silicone	
									sealants	
C-469			Aluminum	Aluminum			Silicone	Silicone	Pump gaskets	Pump gaskets
			Copper	Copper						
			Steel	Steel						
C-477			Copper				Neoprene			
			Steel							
C-486			Aluminum	Aluminum			EPDM	EPDM	Roofing	
			Bronze	Bronze			Neoprene	Neoprene	materials	
			Copper	Copper			Silicone	Silicone		
			Steel	Steel						
C-492	149	300	Copper							
C-499			Aluminum	Aluminum			Neoprene	Neoprene	Cork	Cork
			Copper	Copper						
			Steel	Steel						
C-523			Copper	Copper						
C-568	82	180	Aluminum	Aluminum	Polypropylene	Polypropylene				
			Copper	Copper						
C-575			Copper	Copper						
			Iron pipe	Iron pipe						
C-578			Aluminum	Aluminum			Silicone	Silicone	Roofing	
			Copper	Copper					materials	
			Steel	Steel						
C-580									Wood	

Tabl(Flat-Plate Collectors; Physiolc_ al Safety, Biodegradability, and Other Requirements [12]

Mfr. org. no.	Physiological safety requirements	Biodegradability requirements	Maximum acceptable price per gallon ($)	Other requirements
C-6			6.00	
C-9	Nontoxic.		0.20	
C-29	Nontoxic.		3.50	
C-38			5.00	Should have viscosity similar to that of water. The operating temperature range should extend from -23°C to 182°C (-10°F to 360°F). Should have a life-time of 5 years. Should be available in test quantities.
C-63			5.00 to 8.00	
C-75			3.00	
C-94				Price should be competitive with that for ethylene glycol-based fluids.
C-108			10.00	
C-122	Nontoxic.	Biodegradability required.	0.50	Should be miscible with water. Cost should be lower than that of ethylene glycol.
C-123		None.	3.00	
C-146			3.00	
C-153			0.75	
C-165			2.00	
C-169	None.	None.		
C-172			0.01	
C-196	Nontoxic.	Completely biodegradable.	0.01	
C-197			2.00	
C-199			5.00	For domestic hot water systems.
			0.25 to 0.50	For space heating systems.
C-208	Toxicity lower than that of propylene glycol.			Availability in 5-gallon containers for domestic hot water systems.
C-215	Nontoxic.		0.10	
C-218		None		
C-219	Low acute oral toxicity.		5.00	High specific heat (0.85 to 1.0 cal/g). Low coefficient of thermal expansion (5.4%/100°C). High surface tension (60 dynes/cm). Low viscosity (2-4 cp at 38°C). High boiling point (177°C to 204°C). Low freezing point (-26°C to -7°C). Noncorrosive to building and plumbing materials (i.e., roof shingles, CPVC, copper, aluminum). Low rate of degradation. The fluid should neither cause nor facilitate electrolytic corrosion.
C-222	Nontoxic.		6.00 to 8.00	Low dielectric constant.
C-230	Nontoxic upon skin contact.	None.	2.50	
C-257	Very low toxicity		2.00	Compatibility with roofing materials.
C-261			3.00	
C-280	Nontoxic according to FDA and Los Angeles County Health Department guidelines.	None as yet.		
C-302	Nontoxic.	Preferably biodegradable.	0.15	
C-324			3.60	
C-326	Installation personnel must be able to handle the fluid.			
C-337	Nontoxic.	None.	12.50	
C-380			50.00	
C-435	Nontoxic.			
C-439	Nontoxic upon contact with skin.		10.00[a] 0.75[b]	
C-441			3.00[a] 0.50[b]	
C-463			5.00	
C-465	Nontoxic.		0.10	
C-469	Nontoxic, noncarcinogenic.	Biodegradability required.	1.00	
C-489			6.00	Water-soluble. Should not leak readily through threaded connections.
C-492	None.	None.		
C-499	Nontoxic.	Must be biodegradable.	5.00	
C-501	Must meet OSHA requirements.	Should be biodegradable.		Fluid should be compatible with use in heat exchange systems for potable water.
C-523	Should be usable in the proximity of potable water.			
C-568				Availability on site in less than 30 days after placing an order. Materials should qualify for air freight shipment. Should be readily washable with water.
C-575			6.00	
C-578			3.00	Specific heat approx. 0.5 cal/g°C

[a]For fluids used in the collector-heat exchanger array.

[b]For fluids that are also used for energy storage.

Table 8. Manufacturers of Concentrating Collectors and the Heat Transfer Fluids they use and Recommend [12]

Mfr. org. no.	Activity: Collectors	Activity: Systems	Collector surface area (ft^2)	Collector surface area (m^2)	Fluids used	Remarks
C-2	✓	✓	9,600	892	Caloria® HT-43	
					Dowtherm® A	
					Syltherm™ 800	
					Therminol® VP-1	
					Water (potable)	
					Water (treated boiler feed water)	
					Water-ethylene glycol solution containing up to 50 vol-% ethylene glycol	
C-43	✓		2,800	260	Silicone	
					Therminol® 66	
C-54	✓		45,000	4,181	Caloria® HT-43	
					Freon® 113	
					Prestone II	
					Therminol® 66	
C-80	✓		100,000	9,290	Water	Evacuated tubes
C-154	✓	✓		35	Therminol® 66	
					Water	
C-167		✓	5,700	530	Prestone II	Evacuated tubes
C-226	✓		19	2	Ethylene glycol-water solution (50/50 vol-%)	
C-231		✓			Caloria® HT-43	
					Therminol® 66	
C-246		✓	480	45	Syltherm™ 444	
C-298	✓	✓	∿35,000	∿3,252	Ethylene glycol-water solution	
					Propylene glycol	
					Therminol® 55	
C-316		✓			Sunoco Heat Transfer Oil 21	
C-337	✓	✓	14,000	1,301	Silicone	
C-340	✓	✓	12,000	1,115	H-30 Solar collector fluid	
					Sun-Temp collector fluid	
C-367	✓	✓	3,000	279	Ethylene glycol	
					A proprietary fluid	
C-380	✓	✓	100	9	Syltherm™ 444	
C-489	✓	✓	3,500		Sun-Temp collector fluid	
					Therminols®	
C-493	✓	✓	750	325	Propylene glycol	
					Water	
C-495	✓		50,000	4,645		
C-573	✓	✓			Caloria® HT-43	
					Sunoco Heat Transfer Oil 21	
					Therminol® 66	
C-581	✓	✓	70,000	6,503	Syltherm™ 800 (Dow-Corning X2-1162)	

Table 9. Concentrating Collectors; Problems Encountered with the Selected Fluids [12]

Mfr. org. no.	Fluids used	Problems Yes	Problems No	Problems encountered with the selected fluids
C-2	Caloria® HT 43 Dowtherm® A Syltherm™ 800 Therminol® VPI Water (potable) Water (treated boiler feed water)	✓		In one of the programs (150 kw) anticipating some degradation of the Caloria® HT-43 heat transfer fluid. Planning to replace volatilized material with fresh makeup oil. When the fluid properties degrade substantially, will drain and refill the entire system with fresh oil. Would like to find a reasonably priced thermally stable substitute that would alleviate the need for fluid refilling.
C-43	Silicone Therminol 66	✓		The fluids used limit the collector temperature to 316°C (600°F) whereas its design capability is 538°C (1000°F). The fluid also has to be pumpable at ambient temperatures down to -12°C (10°F). Found Therminol® 66 to meet all requirements of this application closest.
C-54	Caloria® HT-43 Freon® 113 Prestone II Therminol® 66	✓		Typical problems associated with hydrocarbon working fluids. Toxicity considerations of working fluids in contact with potable water systems.
C-80	Water			The recommended operating fluid for this firm's evacuated tube type collector is water. The manufacturer does not use nor recommend any other heat transfer fluids.
C-154	Therminol® 66 Water		✓	
C-167	Prestone II		✓	
C-226	Ethylene glycol-water solution (50/50 vol-%)	✓		Corrosion.
C-246	Syltherm™ 444			None of the present heat transfer fluids meets all the design criteria. The silicone oils require high pumping power.
C-298	Ethylene glycol-water solution Propylene glycol Therminol® 55		✓	No major problems. Double-walled separation is required between the toxic heat transfer fluid (ethylene glycol) and domestic water supply. This requirement lowers the heat transfer efficiency of the heat exchanger and increases its cost.
C-337	Silicone	✓		Low heat transfer efficiency.
C-340	H-30 Solar Collector Fluid Sun-Temp Collector Fluid	✓		Need fluids with better high-temperature capabilities, at a lower price.
C-380	Syltherm™ 444		✓	
C-493	Propylene glycol Water		✓	

Table 10. Concentrating Collectors; Physical Performance Requirements for Fluids [12]

Mfr. org. no.	Operating temperature range (°C)	Operating temperature range (°F)	Stagnation temperature (°C)	Stagnation temperature (°F)	Maximum vapor pressure (atm)	Maximum vapor pressure (psi)	Viscosity Startup	Viscosity Operation	Pumping power[a]	Half-life (years)
C-2	60 to 316	140 to 600	>538	>1,000	23.8	350	1,000cs[b]	0.7 cs[c]		>10[d]
C-39	191 to 316	375 to 600[e]								
	577 to 843	1,250 to 1,550[f]								
C-43	≤316	≤600	>538	>1,000	0.6[g]	8.8[g]				
C-54	66 to 260	150 to 500								
C-154	≤288	≤550		–[h]						
C-167	≤149	≤300	370	700						1
C-226	21 to 88	70 to 190	110 to 149	230 to 300						
C-231	-1 to 327	30 to 620	649	1,200			3,000 cs	0.3 to 0.6 cs	0.02	
C-246	-29 to 93	-20 to 200	177	350	6.8	100			0.06	10
C-298	38 to 121	100 to 250	249	480	8.5	125	–[i]	–[i]		
	60 to 177	140 to 350	416	780					0.01	5
C-316	-12 to 302	10 to 575	316	600						7
C-337	21 to 110	70 to 230	288	550	0.13	1.9 –[j]	50 cp	10 cp		20
C-340	-18 to 427	0 to 800								10 to 20
C-380	-34 to 193	-30 to 380								
C-489	66 to 93	150 to 200	204	400	6.8	100	3 cp[k]			10
C-493	≤177	≤350	177	350						>5
C-495	≤104	≤220	149	300						
C-573	593 to 704	1,100 to 1,300	1,093	2,000	0.14 to 0.20	2 to 3	50 to 70 cp[l]	2 to 5 cp[m]	0.01 to 0.02	10 to 15
C-581	260 to 399	500 to 750	<427	<800	<1	<14.7	26 cs[n]	0.48 cs[o]		>10

[a] The required power is expressed as a fraction of power delivered by the solar collector.

[b] At -29°C (-20°F).

[c] At 316°C (600°F).

[d] Ten percent low-boiling volatiles in 10 years.

[e] Line-focusing collector.

[f] Point-focusing collector.

[g] At 316°C (600°F).

[h] Typical of parabolic trough collector.

[i] Prefer to have viscosities up to 60% of the values of water.

[j] Desirable to have very low vapor pressure.

[k] At 93°C (200°F).

[l] At 21°C (70°F).

[m] At 316°C to 538°C (600° to 1000°F).

[n] At -19°C (-3°F).

[o] At 316°C (600°F).

Table 11. Concentrating Collectors; Fire Resistance Requirements for Fluids [12]

Mfr. org. no.	Maximum hot surface exposure temperature (°C)	(°F)	Fire resistance specifications associated with the specific collectors	Codes and regulations
C-2	316	600	Approximate maximum temperature of receiver surface.	
C-43	~316	~600	The temperature is hottest on the surface of the pipe at the collector outlet.	
C-54	288	550	Under operating conditions.	
	649	1,200	In failure mode.	
C-167	149	300		None.
C-200	427	800	Internal leak onto a ~427°C (~800°F) surface in an evacuated tube. External leak onto a 177°C (350°F) manifold during normal operation, and onto a 371°C (700°F) manifold during stagnation.	
C-226				Should meet ASHRAE and HUD minimum standards for heat transfer fluids.[a]
C-231			If a receiver should reach its stagnation temperature of 649°C (1,200°F), it would leak fluid that would touch its hot surface. The autoignition temperature must exceed the normal operating temperature. The fire point must exceed 149°C (300°F).	
C-246	204	400	Would not use a fluid having a flash point below 315°C (600°F).	
C-298	~482	~900		
C-316	343	650		NFPA, Class I.
C-337	288	550		
C-340	371 to 427	700 to 800	Prefer "noncombustible" fluid.	
C-380	193	380	Absorber plate.	
C-489	204	400	Absorber plate under stagnant conditions.	
C-493	177	350	Flash point above 232°C (450°F).	HUD regulations.
C-495	149	300		
C-573	649	1,200	Autoignition temperature above the operating temperature.	
C-581			Autoignition temperature above the operating temperature. The leaking fluid would generally encounter surface temperatures lower than its own temperature under operating conditions, except for the surface of the backup heater.	NFPA national fire codes.

[a]The fluid should be labeled at the fill point of the closed loop with a record of its properties.

Table 12. Concentrating Collectors; Compatibility Requirements for Fluids [12]

Mfg. org. no.	At max. op. temp. (°C)	At max. op. temp. (°F)	Metals: At 21°C (70°F)	Metals: At max. op. temp.	Plastics: At 21°C (70°F)	Plastics: At max. op. temp.	Elastomers: At 21°C (70°F)	Elastomers: At max. op. temp.	Other materials: At 21°C (70°F)	Other materials: At max. op. temp.
C-2	316	600	Carbon steel	Carbon steel	None	None	None		Pipe fittings	Pipe fittings
C-39	316	600	302 SS		MMA	None	SBR		Basalt	Basalt
			6061 Aluminum		HDPE		Buna-N		Granite	Granite
			Silver		PVC		Viton		Rocks	Rocks
			Copper		Nylon		RTV Silicone		60/40 solder	60/40 solder
					Epoxy/amides					
C-43	538	1,000	Carbon steel	Carbon steel						
			Stainless steel	Stainless steel						
C-54	<316	<600		Steel						
				Copper						
C-167	127	260	Copper	Copper				Elastomeric gasket		
			Stainless steel	Stainless steel						
C-200	177-204	350-400	Copper							
			Iron pipe							
C-226	316	600	Carbon steel	Carbon steel	None	None			Pipe fittings	Pipe fittings
C-231			Copper	Copper	Teflon		Viton		Carbon	
			Mild steel	Mild steel						
			Zirconium	Zirconium						
C-246	177	350	Copper	Copper			Ethylene-propylene copolymer			
			Stainless steel	Stainless steel						
C-298	177	350	Copper	Copper			Viton	Viton		
			Steel	Steel						
C-316	204	400	Copper	Copper	CPVC					
					PVC					
C-337	110	230	Aluminum	Aluminum			Silicone	Silicone	95/5 solder	95/5 solder
			Copper	Copper			Viton	Viton		
			Steel	Steel						
C-489			Aluminum	Aluminum			EPDM	EPDM	Typical roofing materials	
			Bronze	Bronze			Neoprene	Neoprene		
			Copper	Copper			Silicone	Silicone		
			Steel	Steel						
C-493			Copper				EPDM			
							Viton			
C-573			Alloy steel	Alloy steel					Iron ore	Iron ore
			Low carbon steel	Low carbon steel					Rocks	Rocks
			Stainless steel	Stainless steel						
C-551	399	750	Carbon steel	Carbon steel					Iron ore	Iron ore
			Stainless steel	Stainless steel						

Table 13. Concentrating Collectors; Physiological Safety, Biodegradability and Other Requirements [12]

Mfr. org. no.	Physiological safety requirements	Biodegradability requirements	Maximum acceptable price per gallon ($)	Other requirements
C-2	Should be nontoxic.	Depend on local codes.	1.00	Usable to 399°C (750°F).
C-43	Should be nontoxic upon inhalation and contact with skin.		5.00	
C-54	Should be nontoxic.	Should be biodegradable.		
C-167	None.	None.		
C-200			5.00	
C-226				Nontoxic dyes should be incorporated into ethylene glycol-based fluids.
C-231	None.	None.	5.00	
C-246	Should be nontoxic and safe in potable water.		10.00	
C-298			10.00	
C-316			0.80	
C-337	Should be nontoxic.	None.	12.50	
C-380			50.00	
C-489			6.00	Fluid should not leak through threaded pipe connections.
C-493	Need nontoxic fluid for single-wall heat exchangers. Documentation of toxicity test results for transmittal to local code officials.		3.00	
C-573	Liquid and vapor, and reaction products with air and water should be nontoxic at handling and operating temperatures.		1.00 to 2.00[a] 2.00 to 3.00[b] 3.00 to 4.00[c]	
C-581	Nontoxic.		20.00	The degradation products should not deposit on heat transfer surfaces. Specific heat greater than 0.49 cal/g°C at 316°C (600°F). Density greater than 0.67 g/cm³ at 316°C (600°F). Thermal conductivity greater than 0.12 watts/meter °K [0.067 But/hr ft °F at 316°C (600°F)].

[a]For a fluid that has a useful operating temperature range from 21°C (70°F) to 316°C (600°F).

[b]For a fluid that has a useful operating temperature range from 21°C (70°F) to 427°C (800°F).

[c]For a fluid that has a useful operating temperature range from 21°C (70°F) to 538°C (1,000°F).

Table 14. Information on Commercially Available and Developmental Heat Transfer Fluids [12]

Company	Trade name of fluid	Chemical type or composition	Price per gallon ($)		
			5 gal	55 gal	Tank truck
Bray Oil Company	Brayco® 888	Synthetic hydrocarbon			
	Brayco® 888 HF	Synthetic hydrocarbon			
Dow Chemical USA	Dowfrost®	Propylene glycol with inhibitor	–[a]	–[a]	–[a]
	Dowtherm® A	Eutectic mixture of biphenyl and diphenyl oxide	18.00	8.20	7.00[b]
	Dowtherm® G	Mixture of di- and triaryl ethers	21.00	9.60	8.20[b]
	Dowtherm® J	Alkylated aromatic hydrocarbons	14.50	6.30	5.30[b]
	Dowtherm® LF	Mixture of diphenyl oxide and methylated biphenyl	19.30	8.50	7.60[c]
	Dowtherm® SR-1	Ethylene glycol with inhibitor	–[a]	–[a]	–[a]
Dow Corning Corporation	Syltherm™ 444	Poly(dimethylsiloxane)	23.40	20.10	
	Request X2-1162	Poly(dimethylsiloxane)			
Du Pont Company	Freon® 11	Fluorocarbon	7.40	6.40	4.60
	Freon® 114	Fluorocarbon	11.20	10.30	8.80
	Freon® TA	Mixture of fluorocarbon and aliphatic ketone	10.40	8.80	7.10
Ethyl Corporation	ESH-4	Hydrogenated polyalphaolefin	5.90	5.50	5.00
	ESH-5	Hydrogenated polyalphaolefin	5.90	5.50	5.00
	ESH-6	Hydrogenated polyalphaolefin	5.90	5.50	5.00
Exxon Company	Caloria® HT-43	Petroleum hydrocarbons and additives	Not available	1.34 - 1.82[d]	.99[e]
General Electric Company	SF-96 (20)	Poly(dimethylsiloxane)			
A. Margolis and Sons Corp.	Silogram Heat Transfer 43	Paraffinic type oil with oxidation inhibitor			
Mark Enterprises	H-30 Solar Collector Fluid	Synthetic polymeric hydrocarbon	7.00	5.60	4.40
Mobil Oil Corporation	Mobiltherm 600	Paraffinic base hydrocarbon	2.59	2.04	1.80[f]
	Mobiltherm 603	Mixture of aromatic and paraffinic hydrocarbons	2.19	1.64	1.40[f]
Monsanto Industrial Chemicals Co.	Therminol® 44	Modified ester	18.00	7.50	7.10
	Therminol® 55	Synthetic hydrocarbon mixture	16.15	2.45	1.85
	Therminol® 60	Polyaromatic compounds	18.00	6.60	6.20
	Therminol® 66	Modified terphenyl	19.29	8.19	7.79
	Therminol® 88	Mixed terphenyls	~7.50[g]	~6.70[h]	~6.10[i]
	Therminol® VPI	Eutectic mixture of biphenyl and diphenyl oxide[j]	18.20	7.32	6.97
	MCS-1958	Halogenated aromatic		~10 to 20[k]	~10 to 20[k]
	MCS-1980	Mixed terphenyls and higher polyphenyls		~9.30[k]	~9.30[k]
	MCS-2046	Mixed terphenyls and higher polyphenyls		~8.30[k]	~8.30[k]

(continued)

TABLE 14 (continued)

Company	Trade name of fluid	Chemical type or composition	Price per gallon ($) 5 gal	55 gal	Tank truck
Nuclear Technology Corporation/ NPD Energy Systems, Inc.	Sunsafe™ 100	Ethylene glycol formulation with inhibitors (to -18°C)			
	Sunsafe™ 130	Ethylene glycol formulation with inhibitors (to -34°C)			
	Sunsafe™ 200	Propylene glycol formulation with inhibitors (to -18°C)			
	Sunsafe™ 230	Propylene glycol formulation with inhibitors (to -34°C)			
PPG Industries	Zerex®	Ethylene glycol with inhibitors	–[a]	–[a]	–[a]
Practical Solar Heat, Inc.	Practical Solar Fluid	Propylene glycol with inhibitor			
Resource Technology Corporation	Sun-Temp Collector Fluid		6.42	4.50	
Shell Oil Company	Diala® AX	Refined mineral oil and oxidation inhibitor	–[a]	–[a]	–[a]
	Thermia® Oil C	Refined mineral oil and oxidation inhibitor	–[a]	–[a]	–[a]
Stauffer Chemical Company	Stauffer 3664A	Polyol ester-based fluid			
Sun Oil Company	Sunoco Heat Transfer Oil 21	Paraffinic type petroleum oil	–[a]	–[a]	–[a]
	Sunoco Heat Transfer Oil 25	Paraffinic type petroleum oil	–[a]	–[a]	–[a]
Sunworks	Sunsol 60				
Texaco	Texatherm	Refined paraffinic oil from petroleum stocks	–[a]	–[a]	–[a]
Union Carbide Corporation	Prestone II	Ethylene glycol with inhibitor	3.50	3.40	–[a]
Uniroyal Chemical	Uniroyal PAO-13C	Synthetic saturated polyalphaolefin		7.75	7.00

[a]Prices provided by local distributors.

[b]For 4,000-gal lots.

[c]For 4,706-gal lots.

[d]Price range for delivered fluid.

[e]F.O.B. Baytown, Texas.

[f]For 5,000-gal minimum order. Price applies to the East Coast region.

[g]10- to 14-lb bags.

[h]40- to 229-lb bags.

[i]300-lb bags.

[j]26.5 wt-% biphenyl and 73.5 wt-% diphenyl oxide.

[k]Estimated price for commerical quantities.

2.4 THERMAL STORAGE

2.4.1 Introduction

Problems in thermal storage units have been characterized by degradation of insulation, thermal expansion problems, fluid leakage and loss, internal flow blockage, deterioration of storage walls, and in some pebble-bed storage units, settling of the pebbles with resultant unwanted flow channeling or short circuiting.

Both water tank and pebble-bed storage have been used extensively in solar installations and constitute the two major storage methods. Both these storage forms require substantial space. It is important to make good use of space and therefore storage may be located in a basement, garage, or crawlspace or by burying the storage installation. Maintenance and liquid thermal expansion considerations must be included in any design/installation. For example, some allowance must be made for the expansion of the large volume of liquid in a storage tank over an operating temperature range of 35°F to over 210°F (2°C to 99°C). In some cases the expansion tank has not been sized properly to account for this expansion.

Due in part to the thermal expansion and to the pressure of the water in the tank, leakage and fracture of the tank wall has been a problem at some installations. Concrete tanks in particular are susceptible to this durability problem. In one system, a four-inch thick foam tank with metal strap reinforcing showed signs of sagging (bulging out) under its own weight and developed a leak. A set of plastic cool storage tanks demonstrated similar behavior until they were fully and continuously supported on the bottom half of the horizontal cylindrical mount. There have also been steel and fiberglass tanks in unpressurized systems that have cracked under the pressure of the weight of water. The linings of concrete tanks have not proven satisfactory in leak proofing, especially at temperatures higher than 180°F (82°C).

In drain-down systems, where the collector liquid drains back into the storage tank, the tank sizing should allow space for the liquid draining back. Some earlier designs did not adequately allow for this.

A problem occurring in pebble-bed storage is a lack of cleaning the pebbles or rock prior to installation. In these cases the rock had a great deal of fine material and dust in it, which created problems by adding dust and fine particles to the air in the building. Rock with dust and impurities can also affect the pressure drop across the storage and the heat transfer properties. Even though this problem can be taken care of by cleaning the rock before installing it in a storage bin, there have been some cases where the rock bins have failed to charge fully because of partial or complete flow blockage. In some cases the flow blockage was due to the use of the pebble-bed storage bin as a trash receptacle during the building construction phase.

In other pebble-bed storage systems, there is evidence of wood warping, wall settling, etc. Such walls and the lid of a pebble-bed unit must be designed so as to be durable under large temperature changes. Warping may not be a serious concern in a pebble-bed unit with vertical air flow, however numerous air systems have been used with horizontal air flow through the rock storage. Because of the tendency of rocks to settle over time, air gaps above the rocks have occurred. These air gaps allow air to take the path of least resistance and pass over the top of the rocks, thus short-circuiting the storage. This effect can be reduced by allowing the rocks to be tamped down and settle for a few weeks before the top of the rock box is installed and also by installing baffles from the top of the box into the rock and perpendicular to the air flow.

In a few cases flow channeling has inadvertently been designed into the rock storage. This occurrs when the inlet and outlet plenums do not allow flow through the entire cross-section of the pebble-bed. Only when the plenums are open to the entire cross-section of the pebble-bed can the full volume of rock be active in the energy storage process.

2.4.2 Types of Thermal Storage

2.4.2.1 Sensible heat storage (most common)

1. Water storage (see Table 15 for advantages and disadvantages of tank types)
   a. Steel tanks
   b. Concrete tanks
   c. Wood containers

d. Fiberglass tanks (limited to 140°F/60°C temperature - refer to manufacturer's literature). Polypropylene tanks are capable of withstanding higher temperatures than polyethylene. However, there is little experience with using these materials in solar systems.

2. Rock box (or pebble-bed)
3. Other design types
   a. Storage for drain-down systems
   b. Rock/oil storage
   c. Direct contact liquid-liquid heat exchanger/storage
   d. Cool storage
   e. Off-peak storage
   f. Multiple temperature storage using:
      (i) Multiple tanks
      (ii) Stratification methods and/or devices
      (iii) Multiple materials (e.g., latent heat units)

2.4.2.2 Latent heat storage

1. Encapsulated phase change
   a. Air
   b. Liquid

2.4.2.3 In-ground storage

1. Seasonal storage - ground water aquifers
2. Other
   Due to limited experiences and a lack of availability of sufficient data for in-ground solar storage it will not be discussed in this handbook

Table 15. Advantages and Disadvantages of Tank Types

Steel Tank	Fiberglass Tank	Concrete Tank	Wood Tank with Liner
ADVANTAGES			
Can be designed to withstand pressure	Factory-insulated tanks are available	Cost is moderate	Cost is low
Much field experience is available	Considerable field experience is available	May be cast in place or precast	Indoor installation is easy
Connections to plumbare easy to make	Some tanks are designed specifically for solar		
Some tanks are designed specifically for solar storage	Fiberglass does not corrode or rust		
DISADVANTAGES			
Complete tanks are difficult to install indoors	Maximum temperature is limited, even with special resins	Careful design is required to avoid cracks, leaks, excessive cost	Maximum temperature is limited
Are subject to rust and corrosion	Fiberglass tanks are relatively expensive	Concrete tanks must not be pressurized	Must not be pressurized
Steel tanks are relatively expensive	Complete tanks are difficult to install indoors	Connections to plumbing are difficult to make leaktight	Not suitable for underground installation
	Must not be pressurized		

2.4.3 Characteristics of Thermal Storage

Any evaluation of thermal storage units must consider the following:

1. The operating temperature range
2. The heat capacity per unit volume
3. The characteristics of the container
4. The method of supplying heat to (charging) and extracting heat from (discharging) the storage unit

5. The pumping energy requirements (e.g., electrical) for charging and discharging the unit
6. The heat loss characteristics of the unit, insulation requirements
7. Other characteristics of the storage medium
8. Cost
9. Temperature stratification/distribution
10. Input/output temperatures (collection may add heat but decrease the maximum temperatures, particularly in pebble-beds).

2.4.4 Specification Design Problems

2.4.4.1 Sensible heat storage - Water Storage

Some common problems experienced with water tanks or containers are outlined below:

2.4.4.1.1 Thermal expansion. Allowance has to be made for the expansion of the large volume of liquid in the storage tank over a temperature range of 35°F to 210°F (2°C to 99°C) in most cases. The expansion tank or storage tank in drain-back systems has to be sized appropriately to allow for this. One of the thermal design problems is not allowing sufficiently for thermal expansion.

2.4.4.1.2 Leaks. Cracking and leaking of storage tanks has been a problem in several solar installations [2]. The primary reason is that the storage tank design did not allow for the large pressure exerted by the water. There have also been steel and sometimes fiberglass tanks in unpressurized systems that have cracked under the pressure of the weight of water.

2.4.4.1.3 Corrosion. Corrosion of steel tanks has been a problem at some installations. Tanks with appropriate storage lining material for corrosion protection should be selected which can withstand the temperature and pressure conditions that will be experienced.

2.4.4.1.4 Stratification. Stratification in liquid and air storage tanks is desirable. High velocity discharges into liquid tanks can destroy this stratification [13,14]. Storage tank sensors should be located to take advantage of this stratification [14].

2.4.4.1.5 Evaporation. Vapor loss from the unsealed and unpressurized storage tanks can lead to undesirable condensation on a cold wall of a storage room [15]. This evaporation may be minimized by sealing of the tank lid, being careful not to restrict the overflow line. This is particularly applicable to concrete tanks and wood containers. An unsealed tank can also be a safety hazard [16]. Storage tanks should not vent excess humidity/condensation to the rest of the house. In addition, air from across the top of a water tank should not be allowed to reach the conditioned space (i.e., don't make space conditions equivalent to that of an indoor pool).

2.4.4.1.6 Flow channeling. Flow channeling is undesirable if it occurs across the storage tank within one flow loop. For example, in some systems the inlet and outlet to the building heating loop were installed on the same end of the storage tank. The inlet and outlet from storage should be installed at locations which will tend to improve flow distribution. A submerged horizontal discharge pipe with many holes along its length will help greatly in assuring that flow channeling will be reduced within a given loop.

2.4.4.1.7 Concrete tank experiences. Unlike steel, concrete is porous. If the tank is supplied in two halves, the joint should be well sealed and it is preferable that the two halves have grouted-in steel ties to prevent any leakage due to movement. In addition, the coating should be able to span minor cracks that often tend to develop. Therefore cementatious and epoxy waterproofings would tend to be less desirable because they cannot span cracks. Liquid applied lastomers or plastic liners are capable of spanning minor cracks and might be considered if the temperature and liquid water additives are compatible. Also insulate the bottom of the concrete tanks. In some cases it was found that designers have integrated the concrete storage container into the general structure of the building and used the foundation walls as sides of the storage tank [17]. This approach, though less expensive, has two related drawbacks. One is that there is a tendency to draw off and dissipate the heat through conduction within the wall which wastes heat in the tank and, secondly, it can add to the summer cooling load by radiating heat into an occupied space.

2.4.4.2 Sensible Heat Storage - Rock Box

The most common storage type for air systems is the rock box. Some of the experiences associated with rock storage are outlined below.

2.4.4.2.1 Heat losses. Heat losses from rock storage boxes are a serious problem in some installations. Inadequate sealing around the perimeter has often led to air leakage into and out of the rock boxes, leading to serious degradation in performance. Adding flashing around the perimeter of the rock bin and proper sealing, especially around the lid, during installation can help reduce these heat losses.

2.4.4.2.2 Uncleaned rock. A problem that has arisen with rock storage is the use of rock that has not been properly cleaned or rock with a great deal of fine material in it. This can create problems by adding dust and fine particles to the air. Rock with fine dust or particles can also affect pressure drop and heat transfer properties. It is recommended that the rocks be cleaned before installing [18].

2.4.4.2.3 Health factors. Bacteria, radon gas, odors, etc. can arise in connection with rock bed storage units (see Section 2.3).

2.4.4.2.4 Stratification and flow channeling. Stratification is especially important in rock box storage. Furthermore, charging and discharging of storage in a rock bed should be done in reverse directions to enhance system efficiency making proper use of stratification. There are basically two charging/discharging designs for rock boxes, horizontal bed and vertical bed. Rock beds designed for horizontal air flow do not always achieve good stratification and, if a space is left at the top during construction, the air flow by-passes the rocks [13]. This is caused by rocks settling over time and creating an air space above the rocks [4]. If the rocks are tamped down and allowed to settle for a few weeks before the top of the rock box is installed, the air gap will be minimized. If an air gap does occur, air will take the path of least resistance and pass over the top of the rocks, essentially by-passing storage. To be certain this will not occur, baffles should be installed from the top of the box into the rocks perpendicular to the air flow.

In a few cases flow channeling has inadvertently been designed into the rock box. This occurs when the inlet and outlet plenums are not large enough and do not allow flow through the entire cross-section of the rock box thus leading to a part of storage being inactive and therefore less efficient.

In vertical flow boxes, there have also been cases of channeling of flow [18]. This can be avoided by designing entrances to the box as large as possible with well-designed (not abrupt) transition pieces leading to a low pressure drop plenum. Vertical boxes with cinder blocks have resulted in too large a pressure drop in the plenum [18]. Plenums are a critical design feature of rock boxes and must be carefully considered.

2.4.4.2.5 Box deformation/buckling. Improper design of the box, leading to bulging under its own weight, can lead to failure of wood supports over time [2,18). Bulging can also lead to leaks if not properly sealed and can be a source of heat losses. In some systems the rock box shows wood warping or wall settling. The walls and lid of a rock box must be designed to be durable under large cyclical temperature stresses.

2.4.4.3 Sensible Heat Storage - Other design types

2.4.4.3.1 Storage for drain-down/drain-back systems. In drain-down or drain-back systems in which the collector fluid drains back into the storage tank, the tank sizing should allow room for the fluid draining back. Some earlier designs did not allow adequately for this fluid return [2]. Using collectors with steel absorber plates with drain-down design could lead to corrosion problems due to the cyclic exposure of the plate to air and water.

2.4.4.3.2 Rock/oil storage. A novel storage medium has been employed in one case where the solar system consists of trough type concentrating collectors using Therminol 44 as the energy transport fluid [19]. Solar energy provides a portion of the heating, domestic hot water, and cooling (using Rankine cycle power systems) requirements of the building. Solar storage consists of 60% (by volume) rocks and 40% (by volume) Caloria HT 43. The use of rocks for storage was primarily to cut down on use of the more expensive Caloria HT 43.

One of the problems encountered during the early stage of operation was the difficulty experienced in attempting to provide clean rocks for the storage tank due to residues and moisture that accompanied the small rocks. The investigators recommended incorporating an additional rock washing following placement within the tank [19]. Following this wash, the rocks should be dried as much as possible and the tank must be heated slowly to allow venting of the moisture without a rapid pressure increase.

2.4.4.3.3 Direct contact liquid-liquid heat exchanger/storage. This type of storage is of both sensible heat [20] and latent heat [21]. Due to their limited use and due to lack of sufficient data, they will not be covered in this handbook. However, references have been provided for further reading.

2.4.4.3.4 Cool storage. The use of cool storage has been made at a few solar cooling installations. Its use is dependent on:

1. Whether cycling of the chiller will lead to significant degradation of the coefficient of performance during periods of low cooling loads. This has been a problem where the heat capacity of the chiller has been large [2]
2. Space requirements
3. Design requirements
4. Cost

The advantages of cool storage are presently under investigation and its use has been made primarily where system design requirements justified its use [14]. There is some evidence that, with proper control of the chiller, cool storage is not necessary.

2.4.4.3.5 Off-peak storage. Unless low off-peak electric rates are available for an electric boiler, it is generally undesirable to use an auxiliary heat source to heat the storage tank [4]. There is usually no technical benefit in storing auxiliary energy since its availability at most times is assured. And there is a penalty associated with energy storage due to stand-by heat losses. However, waste heat from other sources may be useful.

However, there can be economic advantages for off-peak storage if the local utility provides differential rates for the off-peak and peak electricity usage. These differential rates should be at least 2¢/kW-hr [22]. In this case electrical resistance heating provides thermal heat to a ceramic or brick thermal storage unit (instead of heating the solar storage unit with electricity). This may however, result in high demand charges and may require a substantially increased size of service (i.e., large transformer). In any case, it will require the cooperation of the electrical service company.

2.4.4.3.6 Multiple temperature storage. Problems in achieving temperature stratification in water or liquid tanks include the possibility of increased electrical pumping energy in using multiple tanks (large pressure drops may result) or multiple latent heat units. Alternatively, a stratified vertical storage unit has now been successfully operated in a solar heating and cooling system [22].

2.4.4.3.7 Latent heat storage. An outstanding feature of latent heat storage is the compactness of the storage unit compared with sensible heat storage units. The volume of Phase Change Material (PCM) required to store a given amount of heat is less than the volume of sensible heat storage material required to store the same amount of heat. This allows much greater flexibility in choosing a location for the storage unit. Further, since the unit is small, much less insulation is required to maintain reasonable thermal losses.

Thermal stratification does not occur in phase change storage systems because their temperatures remain nearly constant throughout the change/discharge cycle. If the melting point is chosen so that the storage unit provides heat at slightly above the minimum temperature required by the system, then the output from the collector need be only a few degrees warmer than the minimum temperature regardless of whether the storage unit is charged or discharged. By contrast, a sensible heat storage system typically operates at 40 to 60°F (22 to 33°C) above its minimum operating temperature when it is fully charged. Thus a collector coupled to a phase change storage system can operate at a lower, more efficient average temperature than a collector coupled to a sensible heat storage system.

Due to a lack of availability of information with operating experiences of PCM's, it will not be further discussed here.

2.4.5 Specific Location and Installation Problems

After the storage type has been selected and properly sized, the next decision to be made is the best location. This involves:

1. Space considerations
2. Insulation, waterproofing and freezing protection (if needed)
3. Functional design application (cooling, heating)
4. Aesthetics
5. Structural considerations
6. Safety considerations

The storage choices are:

1. Above grade - inside the building
2. Above grade - outside the building
3. Below grade storage
4. Combination inside/outside storage

2.4.5.1 <u>Above grade - Inside the building</u>

2.4.5.1.1 <u>Advantages</u>

1. An interior installation is relatively easier and less costly to insulate than the other alternatives
2. Does not require external waterproofing
3. Heat losses from storage are utilized in heating the space in winter.
4. Leaks are easily detected
5. Access for repairs is relatively easy

2.4.5.1.2 <u>Disadvantages</u>

1. If installed within the conditioned space and used to store heat during the cooling season, it could add significantly to the cooling load [23]. Also, an improperly insulated storage could lead to overheating of the space during the heating season.
2. Living space is occupied by the storage
3. Uninsulated tank supports could be a source of heat loss. A thermal break between the supports or between the supports and the underlying slab should be used.
4. If boiling of the storage water is a possibility, it should be vented to the outside. This has often not been done.
5. Some concrete and rock boxes have been installed without appropriate lids. These could be safety hazards.
6. Leaks in a tank could pose several problems, including damaging the building interior
7. Steel or FRP tanks are difficult to install in an existing building

2.4.5.2 <u>Above grade - Outside the building</u>

2.4.5.2.1 <u>Advantages</u>

1. Saves valuable building space
2. Heat losses from storage do not add to the cooling load (when applicable). Overheating of space due to uncontrolled heat losses are not a problem since losses are to the ambient air.

2.4.5.2.2 <u>Disadvantages</u>

1. Greater degree of detail required in insulating. The insulation should be impervious to water in case a leak should develop in the waterproofing. If the tank is supported by insulation, the insulation should be capable of supporting the filled tank without crushing. If the tank sits on supports, it is important to isolate or insulate the support in order to minimize losses.
2. Adequate waterproofing required. Rain and snow require close attention to waterproofing. If waterproofing fails, the insulation effectiveness can be severely reduced. All penetrations for valves, supports, piping, sensors, etc. should be adequately waterproofed.
3. Freezing can be a problem. The storage tank and piping need to be sufficiently insulated or other means provided to ensure freezing does not occur, particularly in climates where severe cold spells are experienced.
4. Lack of aesthetic considerations could lead to an unsightly extension of the building.
5. Uncontrolled storage and piping heat losses could be significant and are not useful in heating the house.
6. Leaks and boiling are not easily detected, therefore tank sights for monitoring the water level will be required.

2.4.5.3 Below grade storage

2.4.5.3.1 Advantages

1. They combine the building space saving feature of exterior location and eliminate the aesthetic problem.
2. Elimination of the possibilities of conditioned space overheating as in the above grade storage locations.

2.4.5.3.2 Disadvantages

1. Cost of excavating, weatherproofing, insulating, and supports can be significant, particularly in periodic flood plain areas.
2. Access for repairs or maintenance could be a problem.
3. Ground water and moist soil could lead to large heat losses and add to the possibility of corrosion and degradation of storage wall materials. If possible, buried tanks should not be located below the water table where conductive losses and chances of degradation of materials can be increased. Below grade storage also requires special hold down structures and waterproofing details. One investigation [17] has noted that:

> "The first and foremost concern that should be considered prior to deciding on a buried location is elevation of the water table. If it is above the height of the bottom of the proposed tank, there are several problems which require a special hold down structure and waterproofing. It is best to avoid this condition. However, if there is no alternative, you must design the support so that the tank is held up when full and down when empty. This is usually accomplished with a large concrete footing with tank saddles and steel straps to hold down the tank. It is most important that the supporting members do not break the waterproof integrity of the tank. One way of doing this is to reinsulate and waterproof the tank prior to installation and to install insulation capable of withstanding the point loads imposed by the supports and steel straps and to strengthen the waterproofing in these areas. Remember that just as the supports can crush the insulation and tear the waterproofing under compression, the steel straps can do equal damage if the tank tends to float.
>
> "If there is no water table problem, then there probably is no need for any concrete support, especially if the soil has good drainage. A common approach to installation is to prefoam the tank with urethane and then wrap it with nylon fabric and a bitumastic material and set it in a sand bed. If done carefully, this seems to work well. Another approach is to set the tank in granular insulation and place a plastic sheet over the top. If the surrounding ground is porous, this approach can work. However, if it is not, water will tend to back up into the insulation and extract your hard won solar energy.
>
> "Access to buried tanks is another area that must be thought out carefully. If access is required, then many of the same freeze problems encountered with exterior above ground installations are encountered here and must be solved. We have found a comparatively large number of systems that did not include water level indicators -- much to the operator's chagrin. Again, if access is provided and if the exiting pipes, etc. are bunched, the access area can be minimized. Please provide drainage so that water will not be trapped.
>
> "When locating your buried tank, especially in retrofits, be sure to locate it sufficiently far from the building so as not to cause undermining of the foundations. A good rule of thumb is to locate the tank at least three feet (one meter) away from the face of the building for each foot (0.3 m) below the footing you excavate -- but check with your structural engineer for your particular site."

2.4.5.3.3 Some guidelines for underground storage tanks [17]. For tanks installed underground, anchorage must be provided to prevent buoyant uplift when the tank is empty. The tank should be anchored to a concrete pad at least 6 inches (15 cm) thick and weighing at least as much as the water the tank can hold. The concrete should be covered with a layer of fine pea gravel, sand, or number 8 crushed stone at least 6 inches (15 cm) deep and spread evenly over the concrete to separate it from the tank. Fiberglass or steel hold-down straps should be anchored one foot (0.3 m) beyond the sides of the tank. The hold-down straps should pass over the top of the tank and should be tightened with turnbuckles to give a snug fit. Use at least a 5 to 1 safety factor when you calculate the strength of the hold-down straps and turnbuckles.

Backfill with pea gravel, sand, or number 8 crushed rock at least 2 inches (5 cm) all around the tank. The remainder of the backfill may be clean tamped earth or sand to a depth of 24 to 36 inches (60 to 90 cm) above the tank. Provide concrete pads for nozzles and manholes extending to grade.

In areas with a high water table, the tank insulation must be impervious to water or the tank must be installed in a vault provided with a sump pump.

It should be noted that both basement-like concrete shells with concrete slab roofs and ground-coupled heat pumps have worked well in the demonstration program [24].

2.4.5.4 Combination inside/outside storage or garage

The combination storage incorporates the advantages of both the inside storage and outside storage. The best location is in a corner of the building [25]. In the summer the room containing storage is open to the outside on two sides and completely insulated from the rest of the building. In the winter the room containing storage is open to the interior of the building on two sides and is completely insulated from the outside. Advantages include the fact that the heat losses will be partially utilized by the building in winter and significantly reduced in the summer. Easier maintenance on the storage tank and smaller problems if leakage occurs are some other advantages.

2.4.6 Expansion Tanks

Incorrect sizing of expansion tanks has been a frequent cause of trouble in solar systems. A method of determining expansion tank size is given in the ASHRAE Handbook and Product Directory, 1976 Systems, Chapter 15, "Basic Water System Design". Since the volumetric expansion of antifreeze solutions is greater than the volumetric expansion of water, systems using antifreeze require a larger expansion tank than do systems using water. For these systems, the method of calculating expansion tank size given in the ASHRAE Handbook and Product Directory must be modified as follows [26]:

- From the distributor or manufacturer of the fluid, obtain data on how the fluid's density changes with changes in temperature.
- Multiply the volume of fluid in the system by the fluid's density at the lowest temperature that you expect and divide the result by the fluid's density at the highest temperature that you expect. The result will be the total expansion of the fluid in the system (Part E in Equation 7 of the ASHRAE handbook mentioned above). All other parts of the ASHRAE method of sizing expansion tanks can be used without modification.

Two types of expansion tanks are available. One is a simple tank with an air space; the other uses a flexible diaphragm to separate the water from the air in the tank, thus preventing the water from absorbing the air. Both are effective, but the diaphragmless tank requires periodic replacement of the air absorbed by the water in the tank.

2.4.7 Common Problems Experiences with Storage Systems

Table 16 presents some examples of problems actually experienced with solar energy storage systems, what caused the problems, and how they were dealt with [27].

Table 16. Storage System Problems Encountered [27]

Problem	Description	Resolution
Stratification in liquid tank not accomplished	High velocity input prevented stratification and reduced efficiency	Diffusers were installed to minimize
High thermal loss in buried tank	Water getting into insulation of buried tank increased heat loss	Provide ground drainage, provide waterproof insulation or locate above ground
Heat loss	High heat loss at night was thought to be caused by heat escaping through the tank insulation because of high groundwater. Further investigation showed a faulty thermocouple that allowed pump to run all night, rejecting heat to atmosphere.	Thermocouple was replaced and replacement also failed. It was then found that the thermocouples used were not suitable for the temperatures experienced. They were replaced with high temperature thermocouples.
Heat loss	Groundwater around tank caused high heat loss	Additional insulation and stones were placed under and around the tank to improve drainage.
By-pass of rock bed	A rock bed designed for horizontal flow had an air space at the top which permitted the air to by-pass the rocks.	Redesign to use vertical flow through the rock bed. (Horizontal flow also reduces the desired stratification effects.)
Leakage	Leakage existed at joints of fiberglass tank after tank was assembled on-site from two halves.	Carefully assemble following the manufacturer's recommendations and using the recommended sealing materials.
Leakage	Fiberglass tanks leaked through wicking action in some fiberglass threads that extended through the tank.	Seal all exposed fiberglass threads
Sewer gas in the house	A sewer drain was installed under a rock bed to remove any water. The heat in the bin evaporated the water in the drain trap, letting sewer gas into the house.	Changed drain to a location outside rock bed
Heat loss through insulation	Buried concrete tank leaked water through the tar seal, soaking the insulation, increasing heat loss.	Changed to above ground storage tank
Heat loss	Heat loss from DHW tanks exceeded manufacturer's specifications. Investigation showed the added solar piping and instrumentation provided an increased heat leak path.	Adequately insulate all exposed piping and instrumentation connected to the storage tank
Oversized storage tank	Tank was too large for collector area and tank temperature never exceeded 135°F (57°F)	Replaced tank with one that provided 2 gallons (7.6 liters) of storage for each square foot of collector
Contaminated heat exchangers	Heat exchangers supposedly of refrigeration quality were contaminated with machine oil and metal filings.	Units were returned to vendor for cleaning
Heat transfer losses	Heat transfer from collector loop through the heat exchanger into the storage tank was not as good as assumed.	A parallel heat exchanger was added. (This was considered less expensive than replacing with a more desirable larger single heat exchanger.)
Corrosion	Investigation indicated that the corrosive condition of the ground itself might create problems with underground storage tank.	Installed cathodic protection for the tank (sacrifical magnesium anodes) and coated tank with a rubberized vapor barrier.

Table 16. Storage System Problems Encountered (continued)

Problem	Description	Resolution
Heat loss	Underground tank insulation was damaged by lack of proper support in rocky soil. Maintaining watertight insulation on underground storage tanks is difficult. Water in insulation increased heat transfer.	Check waterproofing prior to installation, provide proper support, install carefully, patch any bad spots in insulation, and backfill carefully.
Incorrect inlet and outlet	Flow from collector to tank entered at bottom of tank. Flow back to collector was also from bottom of tank, causing short circuit in flow path and eliminating benefit of stratification.	Flow from collector to tank should enter at top where water is hottest. Flow back to collector should be from bottom of tank (on opposite end of inlet if no distribution manifold is used.
Materials	Material planned for inside coating of storage tank melted at 180°F (82°C)	Changed to a compound stable at 250°F (145°C)
Saturation of insulation	An open-cell foam was applied to the tank. This acted as a sponge, collected water, and increased heat loss	Use closed-cell foam
Too many tank penetrations	The fiberglass storage tank had all feed and return pipes for the solar collector loop, house loop and domestic hot water loop through the tank below the water level. This resulted in leaks that were difficult to seal.	Two of the three loops were pressurized with positive pressure to the pump suction. Only suction line to the unpressurized loop needed to be below the water level to provide positive suction to the pump. All others could be brought into the tank above the waterline and even the one suction line tank penetration could be above the water level if a foot valve was added.

2.5 PASSIVE SOLAR COMPONENTS

2.5.1 Introduction

Solar passive systems are characterized by reliance on natural convection and radiation and by heat collection and storage devices that are typically integrated with the building structure. Passive space heating is defined here as the direct or indirect collection of incident solar radiation for space and/or DHW heating purposes by means not requiring the forced circulation of a heat transfer fluid, either for solar collection or for delivery of solar heat to the various parts of the heating load (with the exception of an air distribution system blower). In brief, solar passive systems accomplish heat transfer by natural means and do not require forced and/or mechanical movement of the heat transfer medium.

Solar passive systems are defined so as not to include energy conservation features. Energy conservation includes those design features which are intended to reduce a heating and/or cooling load, solar passive features are intended to increase the amount of available heat in order to meet the heating and/or cooling load. Therefore, this definition of passive solar heating excludes:

1. The effective insulation of walls, ceilings and roof
2. The provision of tight construction
3. Building entries with double doors
4. Walls without windows on the north side of the building
5. Windbreaks around the house
6. Partial burying of the structure in the ground (berms)
7. Double or triple glazing
8. Wood windows and doors

All of these measures are useful in reducing heat loss from the building but they have nothing to do with solar heating. They are just as desirable in an electrically or fossil fuel heated house as in a solar heated house since they reduce the cost of space conditioning. One or more of these features should included in a passive solar home ju s they should be included in any energy-efficient building. In general, energy conservation comes first; then solar.

In effect, this definition limits passive solar heating to the use of transparent surfaces, primarily on the south wall or roof of the building which transmit solar radiation to the interior and in which there is generally sufficient thermal mass for storage of a portion of the absorbed solar radiation for reducing temperature fluctuations and night time utilization. The use of greenhouses attached to the wall of buildings is also considered another form of passive solar heating. However, greenhouses may or may not provide direct solar heating of the living space. In most cases, greenhouses serve as moderate temperature zones that partially insulate the south walls from excessive losses as well as being used to furnish partially warmed air to the living space by venting or by circulation.

The attractiveness of passive solar heating is due to the absence of solar mechanical equipment and also to the utilization of additional conventional building components such as glass, concrete walls and floors, etc. for solar heat recovery. In reality, most passive heating systems do require the use of equipment for air distribution such as fans, temperature and humidity control sensors, mechanisms to operate night insulation, and for excess heat disposal, and others. Without this equipment, passive heating would not be able to provide comfort conditions to building occupants while substantially reducing the cost of space conditioning by conventional sources.

2.5.2 General Problem Areas and Design Suggestions

A major portion of this section is based on a study by the Franklin Research Center on experiences with passive design [28]. Other problems and design suggestions have been derived from other sources [29,30,31, 32, 33].

2.5.2.1 Insufficient contribution of useful heat from the solar "system" to the living spaces

This problem typically occurs as a result of a small undersized collector area or inadequate storage mass coupled with a relatively large collector area. In some cases, houses have windows placed evenly on the north, south, east and west facades, ignoring the greater heating benefits of south-facing windows. Solar collection area must be carefully sized and placed in the building with adequate control measures to prevent overheating in summer or excessive heat loss in winter.

Solar storage must also be carefully sized and placed in the building. Storage materials must be chosen on the basis of adequate heat capacity and potential distribution capability. In many frame houses, water or masonry elements are not included and thus do not provide solar storage for night time or extended periods of solar heating. When storage is added, it is often undersized and not adjacent or directly coupled to the collector area. Rock storage beds are often included with little understanding of how one efficiently charges and discharges this kind of storage or the temperatures needed for effective distribution. All storage materials (especially materials such as sand) must be evaluated for their ability to absorb, conduct, hold, and emit solar heat gain. Granular materials such as rock and sand exhibit very little grain-to-grain heat conduction. Storage must be charged and discharged by air flow.

A basic understanding of the collector aperture to storage mass relationship is necessary to ensure adequately sized and located collector and storage -- balanced for the optimum solar contribution [see, e.g., 34,35,36].

2.5.2.2 Inadequate or inefficient distribution of collected heat

This problem is directly related to the storage mass-to-living space relationship. The distribution of solar heat needs to be logically conceived and properly executed based on engineering principles. When radiation distribution to the living space is used, the storage mass must be adjacent to the occupants and rooms needing heat, not the less used spaces such as closets and stairwells.

When convective distribution is used, the logical flow of hot and cold air must be understood. Stratification must be anticipated and handled appropriately. Drawing arrows to indicate heat flow does not guarantee heated air flow throughout the house. A key to success is placing storage in the right position for distribution. This has often not been done.

The passive solar heat tribution system should provide a s___ e and direct link between collection and storage as well as between storage and the living space.

2.5.2.3 Poor use of controls

In several cases users have been unable to regulate heat flow. This can be a serious problem. The success of passive solar systems depends upon the controls which speed, slow, or stop the flow of heat from coming in or going out of the house as required. These controls -- registers, backdraft dampers, movable insulation, exhaust vents, etc. -- are often not stock items and need to be carefully sized, placed, and detailed to ensure the proper operation of passive solar homes.

2.5.3 Specific Problem Areas and Design Suggestions

2.5.3.1 Direct Solar Gain - Collection

1. Do not oversize collector or glazing areas. This could lead to excessive temperatures and uncomfortable conditions during the day. Avoid direct uncontrolled or excessive heating of occupants. Every attempt should be made to diffuse and redirect sunlight to the storage mass around the room or consider direct gain systems for rooms without daytime occupancy so that overheating and glare will not be a problem to the occupant.
2. Summer shading devices should be considered in order to prevent overheating from the solar collection/storage systems. Proper orientation and proper tilt are the first two steps to attain both effective winter collection and summer shading. Operable rather than fixed louver shading devices should be considered to allow solar collection throughout the heating season and, at the same time, provide for control during spring and fall for best comfort conditions. Adequately sized overhangs, exhaust vents, operable windows, and deciduous tree planting may help in reducing overheating of the building. Clerestories may be used to bring natural light and heat to the north rooms.

2.5.3.2 Direct Solar Gain - Storage

1. Do not undersize solar storage. If storage is inadequate, the solar gain which potentially could be stored could cause uncomfortable overheating or may have to be exhausted or vented.
2. Try to provide storage mass in proper relation to all collector areas. Often second floors lack storage mass although they have significant potential for collecting energy.
3. As much of the storage mass as possible should be located where it will be exposed to direct sunlight. This eliminates or reduces use of blowers to transfer collected heat to storage.
4. Avoid use of loose rocks or sand as direct gain storage for incoming sunshine. A limited amount of heat absorption of the top layer can be relied on, but the conductive distribution of heat down through these materials (charging) is very poor. Rock beds charged with hot air from the top of a direct gain living space should only be used for secondary storage (for prevention of overheating). Water and solid masonry storage, if considered, should preferably be used in the living spaces for direct radiant and convective distribution. Direct gain storage should not be covered with materials such as carpet, linoleum, or fabrics which prevent solar absorption and heat radiation.

2.5.3.3 Direct Solar Gain - Distribution

1. Avoid heating direct gain solar storage away from occupied spaces. Provide for solar heat distribution throughout the house, especially in areas which do not receive direct sunlight. Small fans can be used to circulate solar heated air to remote space or existing mechanical distribution systems can be integrated for distribution of passively gained solar heat.
2. Do not expose storage mass, such as floor slabs and vertical walls, to the outside without insulation to prevent heat loss. The solar storage mass will radiate most easily to the coldest side unless prevented by a thermal break.
3. Avoid leaving large glass areas designed for direct solar gain exposed at night. Much of the stored heat in the house will flow out through the glazed area to the cool ambient air. Although double and triple glazing will limit this flow, movable insulation over the glass is an additional barrier to heat loss through

large glass area and is almost always mandatory for achieving energy savings.

2.5.3.4 Indirect Solar Gain - Collection

Include summer shading and/or summer exhaust vents to prevent excessive heat storage or degradation of collector glazings in overheated periods.

2.5.3.5 Indirect Solar Gain - Storage

Do not size the massive walls heated by direct solar (Trombe walls) by making the wall too thick for effective radiant distribution to the house at night. While some Trombe walls can be 12 to 18 inches (30 to 45 cm) thick, walls thicker than 12 inches (30 cm) may radiate stored heat many hours later than desired [23]. Therefore thermal lag times of walls should be considered.

2.5.3.6 Indirect Solar Gain - Distribution

Avoid insulating a storage wall from the space it radiantly heats. Caution must be taken not to block radiant heat transfer with closets, bookshelves, and finished wall materials. In many climates, heat loss to the outside should be prevented with movable insulation over the glass or between the glass and the storage wall.

Avoid heating by convective distribution alone without adequate heat transfer surface and thermal isolation of the Trombe wall at night. Rough masonry surfaces and smaller water storage containers will provide better convective heat transfer. Movable insulation over the glass to prevent excessive heat loss to the outside is necessary if night time heating is to be provided by Trombe wall convection alone.

Some form of controls (vents and backdraft dampers) are required for effective convective Trombe wall operation. Their size, location, and functions must be clearly understood. In both convective and radiant Trombe wall heating systems, adequate distribution should be provided throughout the house.

2.5.3.7 Solarium - Isolated Solar Gain

2.5.3.7.1 Collection. Prevent excessive summer overheating of the solarium space. Consider vertical glazed areas for shading in the summer or provide vents and movable shading devices and possibly ventilating or exhaust fans.

2.5.3.7.2 Storage. A decision should be made whether the solarium must be thermally regulated for the plants or materials within or if it can be allowed to fluctuate for maximum solar collection and storage. In order to keep "occupied" solariums comfortable, solar storage should be provided within to prevent overheating and movable insulation must be added to all glazed areas to prevent excessive heat loss. Balance the storage mass in the solarium with the temperature needed and place all other solar storage within or adjacent to the living space. In "unoccupied" solariums, temperatures may be allowed to fluctuate considerably, with care taken only to prevent heat loss from the solar storage to ensure that the solar heat collected is distributed to the house and not to the colder ambient air.

If rock or pebble-bed storage is contemplated, proper care should be taken in locating, charging, and discharging of storage to maintain comfort conditions for occupants. This has often been designed incorrectly, leading to non-optimum performance. In addition, evaluate storage materials for their heat absorption, conduction, storage capacity, and emission capabilities. Do not overestimate the capabilities of loose earth, loose gravel, or sand as heat storage materials.

2.5.3.7.3 Distribution. Care should be taken in locating the solarium space. To simplify distribution, the collector/storage arrangements should be adjacent to the living spaces which need heat. Passive designs should specify expected temperature swings. Based on one study, occupants may expect temperature swings as little as 5°F (3°C) but more often 15-18°F (8-10°C) [31].

2.5.4 Conclusions

Because most passive solar designs have to be integrated with some form of auxiliary (mechanical, wood stove/fireplace) or active solar heating, sufficient care should be taken to ensure a proper interface. Designs should be made to optimally balance percentage of solar contributions for winter heating against the cost-effectiveness, complexity, and summer cooling requirements.

The challenge confronting passive solar energy design is one of storing and controlling heat to maintain suitable comfort standards within a building. Moderately effective controls, designed to deal with other passive solar problems include movable shades to control sunlight, movable insulation panels to reduce night time heat losses, integration of storage, controls and vents to reduce daytime overheating by means of natural convection.

Passive solar energy concepts can be applied to the design of residential and small commercial buildings. However for large buildings, particularly those with large ventilation requirements, passive systems are more difficult to implement.

The incorporation of passive solar heating techniques has been perceived by many people to involve substantially lower first costs than other methods. This is not always true because passive techniques require a high degree of detailed design and the use of equipment and materials in maintaining proper temperature control of the building.

2.6 PIPING/DUCTING [2,3]

2.6.1 General Problems

Problem areas in piping and ducting are generally limited to leakage problems. These may be caused by deformation of piping or ducts, freezing problems (see Section 2.7.1) water hammer, and improper drain-down. Thermal expansion problems in extended runs of solar system piping can usually be resolved with standard expansion loops or expansion joints. Unfortunately, multi-directional movement can present a more serious problem.

For piping connections, hoses can be damaged by exposure to ultraviolet radiation, stagnation temperatures, and liquid chemical reactions at elevated temperatures (see Section 2.2.1.1.6). Silicone hoses have resulted in long-term creep when held on by clamps and have caused eventual leakage. Metal flexible pipe appears to be one of the few long-term solutions.

Other problems include pipe sizing and layout. In some systems improper sizing has resulted in excessive pressure drops and/or in prevention of complete drain-down. Air locks have occurred in the collector loop because of an improper fill operation and lack of proper air vents. And some systems have lacked a proper fill mechanism, making it difficult to fill the system after draining it for maintenance.

There have been some cases where collector arrays and associated piping were not pressure tested prior to insulating, waterproofing, and burying of the pipes and storage tank. Leaks were subsequently discovered at system start-up. Insulation had to be removed from buried pipe to repair the system. Obviously, installation procedures should call for a pressure test of the system prior to insulating and burying the piping and storage tanks.

Proper provision should be made for draining or discharging of fluid from relief valves in the layout of the piping; these valves should be located such that any hot fluid spilling over will not present a hazard. Special provisions should be made for draining liquids other than water. In many systems these questions have not been attended to.

In many projects, a substantial amount of scale and dirt has been found in the system, left over from construction. It is necessary to flush and clean the system in order to prevent clogging and degradation of the pumps and valves.

Material problems in piping have been encountered in the use of galvanized piping when the temperature of the liquid is higher than about 130°F (55°C). Galvanized piping is not recommended for use in solar systems. Similarly, plastic pipes have ruptured under high pressure and temperature. PVC (poly-vinyl-chloride) pipes should not be used above 140°F (60°C) while CPVC pipes may be used only up to about 185°F (85°C). At higher temperatures, they become soft and are susceptible to rupture. In addition they can also result in curving and bending of the pipes and cracking at bends and joints. This can also create air pockets and water pockets, which can create problems in draining and venting.

Another area where leaks have been of some concern is in air collector systems. The performance of these systems has often been significantly degraded due to leaks in duct seams, damper shafts, collectors, and pebble-bed box joints. Duct seams should be caulked (usually with silicone sealant) and attention paid to tight quality installation procedures to minimize air (and heat) losses. Sloppy workmanship in duct installation has also been noticed at some installations, resulting in ducts of the wrong size, leaky ducts, damaged ducts, etc., which can lead to deterioration in performance of the system.

Absence of air filters in air distribution systems has led to dust problems in the air. Proper maintenance requires regular checking and annual replacement of the air filters and seasonal adjustment of the dampers. When this is not done, the system pressure drop is increased, resulting in increased energy being used by the blowers. Detailed procedures for calculating duct pressure drops and losses can be found in the ASHRAE 1977 Handbook of Fundamentals [1], "Air Duct Design Methods".

2.6.2 Seals

The temperatures, pressure, chemistry, and hardness of the solar system working fluids require that careful attention be given to the seals used in pumps, valves and fittings. Seals used in water heating collectors have failed due to high temperature, chemical attack, and/or ultraviolet radiation. In addition, sealed joints used in the collector loop have shown degradation due to weathering (see Section 2.2.1.1.6).

2.7 RELIABILITY/OPERATIONAL PROBLEMS

Reliability/operational problems include difficulties encountered in the operation of the solar system, which in general caused an interruption of energy supplied by the solar system and resulted in unpredictable system operation. The problems experienced included freezing or boiling of the collector heat transfer liquid, control system malfunctions and failures, and failures of other components in the solar system, such as pumps and fans.

2.7.1 Freezing Problems

The problem with freezing in solar systems has been considered in some detail by Chopra and Wolosewicz [37]. According to these investigators, a review of 47 operating solar systems (all part of the National Solar Demonstration Program) indicate an occurrence of freezing in approximately 30 percent of these sites. The data available from these sites indicated that water/glycol systems have provided more reliable freeze protection than water systems as long as an adequate glycol concentration was initially installed and then maintained.

In general, Chopra noted that the freezing problem in water systems was more complex and could in general be attributed to one or more of several problems including freeze detection sensor location, power failures, manifold slope, leaking control valves, frozen or improperly located vent valves, and a direct connection of the city water main to the thermal storage tank. It was clear that more stringent component reliability requirements were needed for a water system than for a water/glycol system.

Chopra also noted that the freezing potential of air collector systems with a domestic hot water preheat option was directly affected by the air damper leakage rate and that, with proper damper selection, installation and inspection as well as proper maintenace of the louver seals, this freeze problem could be avoided. Specific problems and/or recommendations are shown below.

2.7.1.1 Air Systems with Domestic Hot Water Heating

Freezing occurs when the air damper to the hot water heat exchanger leaks and causes the heat exchanger to rupture. The usual protection method for this component consists of properly specified, installed, and maintained leak-tight air dampers. It has been found that freezing can occur even when the leakage rate through the air damper system is only 15 percent of the total air flow rate.

2.7.1.2 Water/Glycol Systems

Freezing may occur due to an improper glycol concentration and/or in the event of development of a leak in the system, causing the make-up system to add water only to the collector loop. Glycol/water systems require frequent checks of the concentration of the antifreeze in order to ensure freeze protection and subsequent avoidance of damage. Automatic glycol make-up systems have been used successfully in some locations, however several investigators warn against their use due to the possibilities of malfunction. Another problem associated with glycol/water systems that will have an impact on system performance and durability, and could affect solar system freezing, is the degradation of the glycol over time.

2.7.1.3 Water Drain-Down Systems

Figure 2 [37] lists many of the experimental and possible freeze-related problems. Because drain-down systems require the water from the collector array and any exposed piping to drain before freezing can occur, these systems can operate reliably only if:

1. The air vents or the nitrogen purge valves open (May require a "heater strip" for vent)
2. The circulating pump must shut off, and
3. The inlet and outlet manifolds must be properly sloped (1/16 inch per foot; 5 mm/m) to ensure that the system can drain.

Vent Valve Problems	Control Valve Problems	Inlet/Outlet Manifold Problems	Other Problems
Vent frosted shut	Valves not bubble tight, holding tank fills with water and collectors cannot drain	Improper slope to lines (DB)	System back pressure lost and city water floods the collectors (DB)
Vent orientation incorrect	Control valves leak and holding tank level too low to trip switches	Manifold sags due to improper support (DB)	Pinhole leaks draw in cold air and cause remaining moisture to freeze (DB)
Vent location incorrect	Back check valves leak and city water pressures fills the collectors	Manifold bent when stepped on by workman (DB)	After system inoperative and on refilling, cool storage water froze in manifolds (DB)
Internal mechanism damaged by operating conditions	Combination valve and flow meter frosted closed	Improper use of expansion joints (DB)	Freeze detection problems
	Balancing valve orifice frosted closed	Collector hoses had low spot and trapped water	
	Valves improperly located in drain lines		
	Valve resistance too high to allow water to drain		

Figure 2. Freeze-Related Problems for Drain-Down and Drain-Back Systems [37]

2.7.1.4 Water Drain-Back Systems

Drain-back systems are inherently simpler systems than drain-down systems; drain-back designs do not require control valves. Consequently valve and vent problems do not occur. On the other hand, many problems occurring in drain-down systems also occur in drain-back systems. These common problems are denoted by "(DB)" in Figure 2.

2.7.1.5 Water, Circulating Water Systems

Freezing may occur in this case due to freeze protection sensor problems and/or failure of the circulating pump to run because of power failures. With a gravity drain-down system, the pump may not run and, therefore, the system cannot drain properly (see Figure 3) [37].

A circulating water system is a form of freeze protection technique most appropriate for areas where only brief freezing periods occur, generally in the warmer sections of the United States. Because this technique requires the circulation of warm water from the thermal energy storage tank through the collector array, it assumes uninterrupted supply of power to drive the freeze protection pump on demand. Power failures are especially prevalent under freezing rain conditions due to the ice-break of power lines.

Freeze Detection Sensor Problem	Circulating Pump Problem	Gravity Drain-Down Problem
Sensor fails to operate	Pump fails to run because of power problems	Pump does not turn on and system cannot drain
Improper sensor calibration		
Sensor located in close proximity to warm drain line		

Figure 3. Freeze-Related Problems for Circulating Water Systems [37]

2.7.1.6 General Component Guidelines

From the information reviewed on the operating solar systems, freezing problems are found to be primarily attributed to malfunctions and improper design use of the following components besides the other factors outlined earlier:

1. Vent valves
2. Control valves
3. Inlet and outlet manifold design
4. Air dampers

2.7.1.6.1 Vent valves. Automatic vent valves must be specified for all air assisted drain-down systems [3]. The specifications of this component must take into consideration the following:

- i. Maximum operating conditions (temperature and pressure)
- ii. Materials that can withstand these conditions
- iii. The proper venting rate
- iv. The effect of frost formation on the valve mechanism (consequences of vent failing in closed and open positions)
- v. The location of the valve (ideally at every turn down in the piping)

One of the methods of preventing vent valve frosting is to wrap the valve with electrical heat tape. The success of this technique, however, is dependent on the reliability of the local power supply.

(c) The specifications for manual vent valves used in glycol/water systems or circulating water systems are not as stringent as those used in drain-down systems. These manual vents are basically small globe or needle valves.

2.7.1.6.2 Control valves. Solar systems are generally built from off-the-shelf valves used in residential plumbing systems. These valves are adequate for their intended applications but the decision to use drain-down systems with or without a nitrogen purge requires the valves to be leak tight for both water and air. Obviously, drain-back systems do not need valves.

The valves that are normally selected for solar system applications have soft seats that seal the supply and the return lines against water and vapor when the system is not operating. The specified seat materials must be compatible with the solar system working fluids.

Three-way valves have been used in several solar installations and leakage problems have been reported [37]. In normal industrial applications, some leakage can be tolerated without causing serious operational problems. However, solar systems that are designed to drain-down require leak tight valves. Until leak proof three-way valves are made available and identified, it is usually better to use two single function valves.

2.7.1.6.3 Inlet and outlet manifold design. The major problems associated with solar system manifolds arise either because of improper support or improperly sloped lines. If the proper location for the manifold supports is not specified, the manifolds will deflect under their own weight and this deflection could eliminate any designed-in slope.

Further, these lines must be designed for thermal movements. If expansion joints are used to absorb the thermal effects, the manifolds must be guided and anchors must be provided. If these conditions are not met, the expansion joints can be deformed permanently, thereby preventing the manifolds from draining. Some expansion joints (e.g., bellow, bulbous type, etc.) may not be drainable. In "non-drain-down" systems, these joints may freeze during maintenance or may suffer mechanical damage after freezing.

Depending on the solar system configuration and on the available space, expansion joints can be replaced by properly designed U-bends. These U-bends must be horizontal to facilitate draining.

2.7.1.6.4 Air dampers. The possibility of freezing in solar air-heating systems can be reduced by using double damper systems, butterfly type valves (commonly used in isolation systems), or by use of better dampers. These higher quality dampers have self-inflating edges that are forced to mate with the fixed portions of the damper.

The designer must also consider the seals, bearings, and linkages when motorized dampers are selected. Because these components are installed in the hotter sections of air systems, seal deterioration could lead to freezing.

2.7.1.7 General Design Suggestions to Avoid Freezing Problems

Most freezing problems have been identified to occur due to:

1. Lack of attention to engineering details
2. Lack of knowledge of the specific requirements of solar systems. For example in conventional heating and ventilating systems, a valve does not usually require special sealing characteristics. Consequently a valve manufacturer's claim that his valves have a "tight closure" does not necessarily imply that they are air tight or bubble tight.

Chopra and Wolosewicz made several important conclusions. Although water systems are attractive because of the thermal and physical properties of water, the system designer must be aware of potential freezing problems. The design of a freeze-proof water solar system and the selection of its components requires careful evaluation of at least the following:

1. The freeze detection sensor must be reliable and must be properly located so warm convection currents from the water storage tank do not affect it. Not a concern in drain-back systems.
2. All joints must be properly designed and must be made so that pinhole leaks cannot develop. Such leaks can cause frost blockage that leads to more extensive freeze damage.
3. The manifold design must consider deflection due to the weight of a water-filled manifold and the effect of thermal expansion. A slope of at least 1/4 inch per foot (21 mm/m) after the manifold deflections are accounted for is recommended by Chopra [37]. If properly and frequently supported (i.e., no sags, assured slope, constant slope), 1/16 inch per foot (5.3 mm/m) is adequate [24].
4. Avoid installation of manifolds which "encourage" being stepped on by installation/maintenance workers.
5. The vent valves or the nitrogen purge valves must be specified to withstand stagnation conditions and must be freeze protected. Not a concern in drain-back systems.
6. The collector loop holding tank or the expansion tank must be properly located with respect to the system pumps and must be placed in a warm place.
7. The system pumps must be properly specified, must be located so that cavitation effects do not occur, and must maintain the proper system water level. Pumps should be located as low as possible in the piping system.
8. Water level indicating switches at the top of the collector array or evidence of water returned down the down pipe should be interlocked with a timer to stop the collector loop fill pump if the solar system does not fill in a specified time. Not a concern in drain-back systems.
9. The control valves for drain-down systems must be properly located with respect to head loss considerations, be air or water tight, and fail in a manner that assures system drainage. Not a concern in drain-back systems.
10. In the event of a power failure (and power failures will occur for as long as three days), all control valves in a drain-down system should be powered shut and fail open so that the solar system will be drained. Not a concern in drain-back systems.

11. Don't use globe valves. Ensure stems of ball or gate valves are in horizontal position for draining. Not a concern in drain-back systems.
12. The water storage tank should not be connected directly to the city water mains. Water/glycol systems should provide freeze protection as long as the proper glycol concentration is initially installed and maintained and any glycol make-up is performed manually and the proper glycol concentration is maintained. One of the problems associated with water/glycol solutions is the possibility of degradation of the glycol over time. Not a concern in drain-back systems.

The freezing potential of an air collector system with a domestic hot water option is directly affected by the air damper leakage rate. With proper selection, installation, inspection, and maintenance of the louver seals, as well as the addition of back-up dampers, by-pass loops, or the use of butterfly type valves, this freezing problem can be avoided.

It appears that the component reliability requirements are greater for water systems than for water/glycol or air systems. As the systems become more complex, the reliability of each component must be an important design consideration. However, a reliable component incorrectly used can still fail. The solar system designer must, therefore, be aware of the special requirements for these systems and be sure that the important engineering details are carefully worked out and that the installed system closely resembles the system that was designed.

2.7.2 Boiling Problems

There have been occurrences of boiling problems in liquid-heating collectors for the following reasons:

1. Interruptions of electrical power to the collector or heat exchanger pumps
2. Mechanical malfunction of pumps
3. Inadequate collector fluid flow rates
4. Inadequate thermal capacity in storage in relation to the collector area
5. Insufficient load and/or lack of provision of heat rejector mechanism
6. Incomplete draining of water in a drain-down or drain-back system
7. Evolution of non-condensibles which block flow in a portion of the collector array

Because of the nature of some of the factors (e.g., power failure) which could precipitate boiling, it is to be expected that over the life of the system some of these problems will occur Therefore measures should be adopted to prevent damage to the system. Some of the design features that should be incorporated in the system are:

1. Means to provide pressure relief for the boiling liquid
2. Appropriate location for boiling liquid to be discharged or condensed
3. Means to replace the liquid in proper concentrations
4. Consideration for the effects of liquid degradation

Specific problems have occurred in pressure relief valves due to obstructions or manual valves being placed between the collector and the pressure relief valve. Some relief valves have been set at too high a pressure, therefore increasing the possibility of damage [38].

Furthermore, the discharge location of the boiling liquid should be properly selected so that it does not pose a danger to people or hardware in the vicinity. This is particularly true when the collector fluid is other than water due to the toxicity and high temperature characteristics of some fluids. Boiling fluids have also resulted in a reduction of the integrity of watertight roofs by discharging to the underside of flashing. Some collector designs tend to trap air bubbles within the collector which in turn traps liquid. At stagnation this liquid can boil and possibly rupture collector tubes. Without proper drainage of fluid, this problem can lead to winter freezing too. In addition, this loss of liquid by boiling, leading to entrapped air, increases the probability of corrosion. This is particularly true of glycol/water systems which require approximately 30 percent concentration for corrosion protection and a range of 20 to 60 percent for freeze protection. Therefore a proper make-up system should be designed to ensure appropriate concentrations.

Ethylene glycol degrades at temperatures higher than 270°F (130°C) by breaking up and forming organic acids which may result in corrosion of the collector materials and sealants. Ethylene glycol is also toxic, which requires double separation at the heat exchanger if utilized for DHW or connected in any way to the potable water supply. Most propylene glycols are considered non-toxic enough to waive the double separation requirements but its use should be prevented because of the possibility of replacement with a toxic glycol. However, at high temperatures, propylene glycol also breaks up and forms corrosive organic acids and thus it may also require double separation [3].

Furthermore, the effects of boiling fluid on rubber hose connections between collector modules and collector array manifolds and collector mateials should be considered with respect to degradation.

Partial shading of collectors has also sometimes led to freezing/ boiling problems.

2.7.3 Control Subsystem Problems

Controls are a major factor to consider in the successful operation and reliability of a solar system. An unsatisfactory control system or improper logic can lead to serious degradation in performance or even complete failure of the solar system to perform [38].

Many of the larger systems are over-engineered and have complicated control systems and several modes of operation. The simpler systems have been found to be more reliable [3]. Failure of the more complicated control systems have often resulted in freeze-ups and overheating. Furthermore, the time and effort in determining a malfunction could be substantial.

Improper selection of the set point temperatures to activate pumps has often resulted in excessive cycling. This condition often occurs during periods of low insolation and could result in premature failure of the pump [3]. Allowing for an adequate deadband can remedy this problem.

Control system components have sometimes been defective. Some examples of this in actual systems include:

1. Defective temperature sensors
2. Failure of differential controller because of faulty circuitry and component failure

Proper location of sensors has very often been overlooked. Extraneous heat flow effects, stratification (especially in rock boxes [39]), thermosyphoning [11], and other dynamic effects can often lead to a deceptive signal being received by the controller. This can result in cycling and a non-optimum operation.

A control system not performing as designed can lead to a serious degradation in operating efficiencies. Example: Collector temperature sensors provide more accurate control when located on the absorber plate rather than in the return pipe above the collectors [4]. This is due to the fact that during start-up there is no flow and therefore the sensor mounted in the return pipe has to rely on natural convection and conduction for a very indirect measurement of the collector temperature.

Some of the other factors to be considered in selecting temperature sensors are ease and integrity of mounting, corrosion protection [4], heat and electrical capacity of sensor [11], and quality of sensors (accuracy, precision, linearity, repeatability, response time, drift, etc.).

Control logic should consider failure contingencies so that the occupants are not subjected to extreme changes in comfort conditions as a result of failure. In addition, the possibility of damage to hardware should be considered. Electronic monitoring of control system malfunctions has often proven to be useful in alerting occupants of the need for corrective measures.

2.7.4 Pump/Fan Problems

A problem often found in solar projects is the incorrect sizing and selection of pumps. Parasitic power consumption can be sufficient to negate the advantages of solar energy collection. A pump should be sized so that it operates at a high pumping efficiency. Furthermore, incorrect sizing of pumps has led to flow balance problems. Higher collector temperatures and lower collector efficiency also result if the pump is undersized for the collector array [3,12].

Other problems relating to pumps in a solar system have been the following:

1. Pump cavitation
2. Pumps burning out because of the rupturing of defective piping or because of faulty controls
3. Pumps with expansion tanks on their suction side have been subject to wear out from cavitation problems [3,40]. The expansion tank should be on suction side, but needs sufficient liquid head to prevent cavitation.

Improper design has led to vapor locks/air locks and the need for manual priming of pumps. Pumps should be mounted at the lowest possible point in the system.

Erosion may occur with fluid velocities of over three feet per second (0.90 m/sec). This results in the need for streamlining at the heat exchanger (within 1.5 diameters of inlet). Note however that too low a velocity may result in scaling.

The analysis of static pressure for each mode of operation is often not done in sizing fans. Furthermore, all sources of pressure drop should be considered in sizing. Also leakage of air has often not been considered in sizing, leading to inappropriate flow rates and degradation in performance. In addition the fans are often not properly sized to handle the excessive pressure drops imposed by duct turns, collectors, coils in the air stream, backdraft dampers, heat exchangers, and the pebble-bed storage. In one case, two blowers placed in series caused flow pulsations and poorly controlled flow rates. Blowers have in several cases operated noisily because the air ducts were sized smaller than required by good HVAC design.

Pump and/or fan damage has in general been minimal and limited to pump damage from backflow conditions or from chemical attack of the heat transfer liquid. Blowers have performed well, particularly when routine maintenance, such as oiling bearings, tightening or replacing belts, checking pulley tightness, and cleaning fan blades have been performed. Pump seals have also been damaged due to chemical attack by fluids at elevated temperatures.

2.7.5 Valves/Dampers

Problems with valves and dampers have sometimes seriously affected the reliability and performance of solar systems. However, in terms of durability, valves and dampers have performed well. In some cases, however, leaky dampers have resulted in freezing of air heating DHW systems (see Section 2.7.1). Regular maintenance of dampers is essential to keep air leakage to a minimum.

2.8 CASE STUDIES

Appendix B includes several examples of specific problems encountered and corrective action taken.

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2.9.1 Collector Subsystem

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3. PERFORMANCE OF SOLAR HEATING AND COOLING SYSTEMS

3.1 INTRODUCTION

In order to maximize savings in alternative fuels, it is essential to identify solar systems that have performed efficiently, and to distinguish between levels of performance in different solar installations. For the purposes of this handbook, a measure of the performance of a solar heating and cooling system, is defined in terms of:

1) System and subsystem efficiencies, and

2) System and subsystem energy outputs.

The principal objective is to document the relevant results, and to present an overall assessment of the performace of solar heating and cooling systems. Only those installations which have sufficient data, information, and documentation on which to base an objective opinion on a system's performance, have been considered in this assessment. In addition, certain installations of solar systems have been considered in greater detail than others, in order to determine cause and effect in system performance, i.e. the effect of design variations in increasing or decreasing the overall performance of a solar system.

Solar systems that performed well, achieved substantial savings in nonrenewable energy resources. In brief, these energy savings amounted to:

Type Solar System	Savings in Nonrenewable Energy Resources (million Btu/year·ft^2 of collector)**
Domestic Hot Water	0.22
Passive Space Heating	0.29 *
Active Space Heating	0.19
Active Space Cooling	0.03
Potential Active Cooling	0.11 @

This chapter is divided into the following sections:

- Criteria for Design and Performance Analysis
- Performance of Domestic Hot Water (DHW) Systems
- Performance of Passive Heating Systems
- Performance of Active Space Heating Systems
- Performance of Active Space Cooling Systems.

3.2 CRITERIA FOR DESIGN AND PERFORMANCE ANALYSIS

The criteria for design and performance analysis, presented here, is based on the following requirements:

1) The establishment of an unambiguous set of parameters, which allow for an objective and practical evaluation of solar system performance.

2) Parameters which will provide a measure of the quality of design, installation, and operational performance of a solar system and its major components.

3) Identification of the energy inputs/outputs of the solar system and its major components.

4) Clearly defined efficiencies, which allow for direct comparisons to be made between distinct installations and system designs.

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*Based on certain assumptions (see section on Performance Evaluation of Passive Systems).

@Based on certain assumed system modifications and subsequent improvements (see section on "Comparison with Conventional Cooling Systems").

**Multiply (million Btu/year·ft^2) by 11.4 to get (GJ/year·m^2).

3.2.1 Collector Parameters

Q_c is defined as the average[1] daily useful solar energy collected by the solar collector array. Units are Btu/day (kJ/day).

A is defined as the gross area of the solar collector array. The gross area of an array encompasses the entire area within the outer perimeter of the array, including space allowed for interconnecting individual collector modules. Units are ft^2 (m^2).

I is defined as the average daily total integrated value of the solar radiation incident upon the tilted surface of the solar collector array, per unit area of the collector. Units are Btu/day ft^2 (kJ/day m^2).

η_c is the average daily solar collector array efficiency, (dimensionless), and is defined as the fraction of the incident solar energy, collected by the collector array, i.e.:

$$\eta_c = \frac{Q_c}{A \quad I} \tag{1}$$

3.2.2 Thermal System Parameters

C is defined as the Coefficient of Performance (COP) of the heating or cooling unit (e.g. the C.O.P. of an absorption chiller), i.e. the heating or cooling of the conditioned space accomplished by the unit, divided by the thermal input to the unit (dimensionless).

β is defined as the system heat loss factor, i.e. the factor used to account for the degree of non-usefulness of the solar system's thermal losses to the interior of the conditioned space (dimensionless).

Heating Season. β equals the fraction of the solar system's thermal losses to the conditioned space, which is not useful, i.e. constitutes overheating of the space. In general those heat losses are not useful, whenever the heating demand of the space is less than the internal heat generation of the space (exclusive of the solar system's internal heat generation). During cold periods and for systems with low heat losses, most of the heat lost from a solar heating system to the conditioned space may be considered useful even with small, temporary temperature increases, because of the fact that much of the excess energy is stored in the thermal mass of the conditioned space's structure. In this case $\beta \simeq 0$. Nevertheless it is desirable to minimize uncontrolled heat losses in order to avoid significant overheating of the conditioned space.

Cooling Season. β accounts for the heat lost to the space and thus its non-availability to be used in operating a cooling unit, plus the solar heat required to operate a cooling unit (at some C.O.P.) in order to remove the additional cooling load caused by the solar heat losses to the space. Thus

$$\beta = 1 + 1/C$$

where C is the coefficient of performance of the cooling unit when operated with solar energy.

Q_I is defined as the average daily solar system heat losses to the interior of the conditioned space. Units are Btu/day (kJ/day).

Q_E is defined as the average daily solar system heat losses to the exterior of the conditioned space. Units are Btu/day (kJ/day).

Q_u is defined for space and DHW heating systems, as the average daily solar system heat delivered to the heating unit. Units are Btu/day (kJ/day).

[1]Averages for all parameters are, for the purposes of this handbook, taken over monthly time periods.

Q_u is defined for space cooling systems, as the average daily solar system heat delivered to the cooling unit, in excess of that required to remove the heat losses of the solar system from the conditioned space. Units are Btu/day (kJ/day).

These heat quantities within the thermal system can be related by:

$$Q_u = Q_c - Q_I \beta - Q_E \tag{2}$$

η_T is the average daily solar heating and cooling system thermal efficiency (dimensionless); and is defined as the average daily total useful heating and/or cooling of the conditioned space, divided by the solar radiation on the collectors; i.e.:

$$\eta_T = \frac{Q_u C}{A I} \tag{3}$$

3.2.3 Solar System Electrical Parameters

E is defined as the average daily electrical energy required to operate the solar system (e.g. pumps, controls, etc). Units are Btu(elec)/day (kJ/day).

S is the ratio of useful solar heating and/or cooling to the electrical solar-operating energy used, and is defined as the average daily useful heating and/or cooling by the solar system, divided by the average daily electrical energy used to operate the solar system (dimensionless).

This ratio, S, is defined by:

$$S = \frac{Q_u C}{E} \tag{4}$$

S_c is the ratio of controlled useful solar heating, to the electrical/solar-operating energy used, and is defined as the average daily useful and controlled heating by the solar system, divided by the average daily electrical energy used to operate the solar system (dimensionless).

In the definition of S_c, the heat losses to the space are not considered useful in any manner, so that $\beta = 1$. S_c is therefore defined as:

$$S_c = [Q_c - Q_I - Q_E] \frac{C}{E} \tag{5}$$

$S_c = S$ for solar cooling systems, by definition.

3.2.4 Total Energy System Parameters

Q_s is the average daily savings in nonrenewable energy resources by the solar heating and cooling system. Units are Btu/day (kJ/day). Q_s is defined by:

$$Q_s = \frac{Q_u C}{\eta_A} - \frac{E}{\eta_E} \tag{6}$$

where

$Q_u C$ = Solar useful heating and/or cooling, Btu/day (kJ/day)

E = Electrical solar-operating energy, Btu(elec)/day (kJ(elec)/day)

η_E = Overall efficiency of fuel-generated electrical power generation, distribution, and transmission (dimensionless).

η_A = Conversion efficiency of the auxiliary or conventional heating and/or cooling unit.

Depending on the type of conventional heating and/or cooling unit chosen, η_A may be quantified as:

$\eta_A = \eta_E$ for electric resistance heating.

$\eta_A = \eta_F$ for a fuel-fired furnace (e.g. natural gas, fuel oil, etc.) (η_F = furnace efficiency, dimensionless).

$\eta_A = C_c\eta_E$ for a heat pump or conventional air-conditioning vapor-compression unit, where

C_c = Coefficient of Performance of the heat pump or conventional air-conditioning, vapor-compression unit (dimensionless).

η_S is the average daily solar heating and cooling system overall efficiency (dimensionless); and is defined as the average daily total energy savings by the solar system, divided by the sum of the average daily total solar radiation on the collector plus the average daily electrical solar-operating energy (converted to its fuel energy equivalent). η_S is given in equation form as:

$$\eta_S = \frac{Q_s}{A\ I + E/\eta_E} \tag{7}$$

3.2.5 Example Calculations

3.2.5.1 Typical Efficiency Values of Conventional Equipment

The overall efficiency of fuel-generated electrical power generation, distribution, and transmission, η_E, is approximately 0.26 [41]. For Hydroelectric plants, $\eta_E \simeq 0.87$ (prime denotes hydroelectric).

For conventional DHW heating units, the conversion efficiency for natural gas heaters varies over a considerable range but typically averages 0.55 [24], i.e. η_c (DHW) = 0.55. For electric hot water heaters, η_c (DHW) $\simeq \eta_E$.

For conventional space heating, natural gas and fuel oil furnaces have typical efficiencies of 0.60 and 0.55, respectively. (These figures are national averages [42].) Therefore η_F (natural gas) $\simeq 0.60$, and η_F (fuel oil) $\simeq 0.55$. However, it should be noted that furnace efficiencies may easily range from 20 to 80 percent, depending upon the age and maintenance history of the unit.

Heat pump C.O.P.'s vary from 1 to 4 (or higher) for heating purposes. On a seasonal basis $C_c \simeq 1.5$ to 2.5. Thus

$\eta_A = C_c\ \eta_E \simeq 0.4$ to 0.65.

For space cooling, the seasonal coefficient of performance for a heat pump and/or a conventional air-conditioning vapor-compression unit is 1.8 to 2.5. Therefore $\eta_A = C_c\ \eta_E \simeq$ 0.47 to 0.65. Gas-fired absorption chillers have C.O.P.'s ranging from 0.65 for residential-sized units to greater than 0.9 for larger units.

For the purposes of this handbook, we will assume:

η_E = 0.26		Electrical power generation, distribution and transmission, overall efficiency
η_A (DHW Systems)	= 0.55	Conversion efficiency for natural gas DHW heaters
η_A (Space Heating)	= 0.60	Conversion efficiency for natural gas or fuel oil furnaces
η_A (Space Cooling)	= 0.65	Seasonal vapor-compression C.O.P., times η_E = 0.26.

3.2.5.2 Example System

Assume that an example space cooling system has the following measured values:

I = 1,586 Btu/day·ft^2 (18,080 kJ/day·m^2)

A = 631 ft^2 (58.6 m^2)

Q_c = 339,300 Btu/day (357,960 kJ/day)

Q_E = 50,900 Btu/day (53,700 kJ/day)

Q_I = 21,100 Btu/day (22,260 kJ/day)

C = 0.527

E = 16,800 Btu(elec)/day (17,725 kJ(elec)/day).

The other major parameters may then be calculated. For example

3.2.5.2.1 Collector Efficiency:

$$\eta_c = \frac{Q_c}{A\ I} = \frac{339{,}300 \text{ Btu/day}}{(631 \text{ ft}^2)(1{,}586 \text{ Btu/day}\cdot\text{ft}^2)} = .339$$

$$\boxed{\eta_c = 33.9 \text{ percent}}$$

3.2.5.2.2 System Thermal Efficiency:

β = (space cooling) = 1 + 1/C = 1 + 1/.527 = 2.90

$$Q_u = Q_c - Q_I\beta - Q_E$$

$$Q_u = 339{,}300 \text{ Btu/day} - (21{,}100 \text{ Btu/day})(2.90) - 50{,}900 \text{ Btu/day}$$

$$Q_u = 227{,}210 \text{ Btu/day } (239{,}700 \text{ kJ/day})$$

$$\eta_T = \frac{Q_u\ C}{A\ I} = \frac{(227{,}210 \text{ Btu/day})(0.527)}{(631 \text{ ft}^2)(1{,}586 \text{ Btu/day}\cdot\text{ft}^2)} = .1196$$

$$\boxed{\eta_T = 12.0 \text{ percent}}$$

3.2.5.2.3 System Energy Savings:

$$S = \frac{Q_u\ C}{E} = \frac{(227{,}210 \text{ Btu/day})(0.527)}{16{,}800 \text{ Btu(elec)/day}} = 7.1$$

$$Q_S = \frac{Q_u C}{\eta_A} - \frac{E}{\eta_E} = \frac{(227{,}210\ \text{Btu/day})(0.527)}{0.65} - \frac{16{,}800\ \text{Btu(elec)/day}}{0.26\ \text{(Btu(elec)/Btu)}}$$

$$\boxed{Q_S = 119{,}600\ \text{Btu/day}\ (126{,}200\ \text{kJ/day})}$$

3.2.5.2.4 System Overall Efficiency:

$$\eta_S = \frac{Q_s}{A\ I + E/\eta_E} = \frac{119{,}600\ \text{Btu/day}}{(631\ \text{ft}^2)(1{,}586\ \text{Btu/day}\cdot\text{ft}^2) + \left(\frac{16{,}800\ \text{Btu(elec)/day}}{0.26\text{(Btu(elec)/Btu)}}\right)}$$

$$\eta_S = 0.1123$$

$$\boxed{\eta_S = 11.2\ \text{percent}}$$

3.3 PERFORMANCE OF DOMESTIC HOT WATER (DHW) SYSTEMS

The performance of a wide variety of solar DHW systems is given in Tables 17 through 22. Table 23 provides a more specific identification of each of these systems. Tables 18 and 20 provide for direct comparisons of different designs under similar solar and load conditions.

In developing these tables, several assumptions are made, including:

C = 1.0, i.e. the coefficient of performance of converting solar heat into useful DHW heat, is unity,

β = 1.0, i.e. all of the heat losses from the solar system are nonuseful in meeting the DHW load, and

$\eta_E = 0.26$

$\eta_A = 0.55$

In addition, the interior and exterior heat losses from the solar system, Q_I and Q_E respectively, are combined to yield the total heat losses from the solar system, Q_L, i.e.

$$Q_L = Q_E + Q_I\ \beta \rightarrow Q_E + Q_I$$

Finally, the solar useful heating/electricity solar-operating energy used ratio, S, is assumed equal to S_c, and is therefore not included as a separate column in Tables 17 through 22, for DHW systems.

3.3.1 Performance Evaluation

Evaluation of the performance of solar DHW systems is detailed in the section on "Analysis of Systems Performance"; subsection: 1) "Overall Evaluation of Performance," 2) "Problems in Design and Sizing," 3) "Major Factors in Reduced Performance," and 4) "Performance Evaluation of Solar DHW systems."

In general the performance of solar DHW systems was excellent. With few exceptions, system thermal efficiencies of 17 to 42 percent were achieved. Several systems (identified in Tables 19 and 20 as systems 17, 20, 23, and 27) had system overall efficiencies of 35.1 to 55.7 percent, and corresponding average energy savings (of these four systems) of approximately 13.4 million Btu/year·system (14.1 GJ/year·system), or 0.22 million Btu/year·ft^2 (2.5 GJ/year·m^2) of collector.

Table 17. DHW Systems Annual Average Performance Parameters.

System Identification Number (see Table 23)	1	2	3	4	5	6	7[1]	8[1]	9[1]
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft²)[2]	1528	(IA = 1.43 x 10^6 Btu/day)[4]	1590	1568	1349	(IA = 3.28 x 10^6 Btu/day)[4]	(IA = 0.78 x 10^6 Btu/day)[4]	(IA = 0.33 x 10^6 Btu/day)[4]	(IA = 0.37 x 10^6 Btu/day)[4]
A - Gross area of collector array (ft²)[3]	76.6		520	6500	6254				
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[4]	33.2	397.7	359.2	2717.9	2894.8	1084.6	195.8	50.2	51.4
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	28.4	26.6	43.4	26.7	34.3	33.0	25.2	15.0	13.8
Q_L - Average daily solar system heat losses (1,000 Btu/day)[4]	2.6	58.1	119.2	430.0	95.6	624.8	119.3	25.0	51.3
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[4] ($=Q_c - Q_I\ \beta - Q_E$)	30.6	321.6	240.0	2287.9	2799.2	459.8	76.5	25.2	0.1
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	26.1	22.5	29.0	22.4	33.2	14.0	9.9	7.5	0.0

1. Systems 7, 8, and 9 are combined space and DHW heating systems, but include only data from DHW solitary operations.
2. Multiply [Btu/day·ft²] by 11.4 to obtain [kJ/day·m²].
3. Multiply [ft²] by 0.0929 to obtain [m²].
4. Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 18. Comparison of DHW System Performance [48].

Unit[2]	System	System Losses (%) Collector Losses	System Losses (%) Piping Losses	System Losses (%) Storage Losses	Thermal Efficiency (%)	System[3] Efficiency (%)	Solar Fraction (%)	Collection Area (ft^2)[4]
	Thermosyphon							
10	Single tank direct, liquid	69.5	0[1]	7.2	23.3	22.6	50.3	54
	Active							
11	Single tank direct, liquid	55.4	1.7	14.7	28.2	21.4	40.9	36
12	Double tank direct, liquid	59.5	1.8	20.9	17.8	12.5	39.8	54
13	Single tank indirect, liquid	70.2	2.0	5.7	22.1	19.6	48.6	54
14	Double tank indirect, liquid	65.9	2.8	14.7	17.1	14.6	37.9	54
15	Double tank indirect, air	78 1	3.3	12.1	6.6	3.1	21.7	80

[1] Accurate piping losses incalculable due to characteristic variable flow rate of thermosyphon system; piping losses included in collector losses for thermosyphon.

[2] See Table 23

[3] System Efficiency = $\frac{Q_u - E}{AI}$

[4] Multiply [ft^2] by 0.929 to obtain [m^2].

Table 19. DHW Systems Annual Average Performance Parameters.

System Identification Number (see Table 23)	16	17	18	19	20	21	22	23	23[2]
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[3]	1486	1486	2093	1360	1383	1132	1312	1339	
A - Gross area of collector array (ft^2)[4]	54	54	585	42.2	44	76.6	1782	105	
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[5]	32.5	23.9	433.3	-	17.2	23.2	457.9	51.5	
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	40.5	29.8	35.4	-	28.3	26.8	19.6	36.6	
Q_L - Average daily solar system heat losses to exterior (1,000 Btu/day)[5]	7.6	2.6	337.2[1]	-	2.6	3.6	123.6	16.9	
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[5] ($=Q_c - Q_I\ \beta - Q_E$)	24.9	21.3	96.1	14.5	14.6	19.6	334.3	34.6	
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	31.1	26.6	7.8	25.2	24.0	22.6	14.3	24.6	
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[5]	4.0	1.9	27.0	2.3	0.0	2.0	67.6	10.1	2.6
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u$C/E)	6.2	11.3	3.6	6.3	∞	9.8	4.9	3.4	13.3
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[5] ($=Q_u C/\eta_A - E/\eta_E$)	30.0	31.4	70.9	17.5	26.5	27.9	347.8	24.1	52.9
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	31.4	35.9	5.3	26.4	43.6	29.6	13.4	13.4	35.1

Underlined numbers denotes calculation by authors from available data.

1. All heat losses are to exterior of conditioned space.
2. System 23, where electrical energy from pumps is considered to be added as heat to circulating loop.
3. Multiply [Btu/day·ft^2] by 11.4 to get [kJ/day·m^2].
4. Multiply [ft^2] by 0.0929 to get [m^2].
5. Multiply [Btu/day] by 1.055 to get [kJ/day].

Table 20. Comparison of Different Solar DHW Systems [54].

Unit[2]	Fluid	Collection Area A (ft^2)[1]	Tank Arrangement	Heat Exchanger	Solar Heat Losses Q_L (kBtu/day)[3]	Solar Useful Heating Q_u (kBtu/day)[3]	Thermal Efficiency η_T (%)	Electrical Operating Energy E (kBtu/day)[3]	S_c	Energy Savings Q_s (kBtu/day)[3]	Overall Efficiency η_s (%)
24	Silicone	38.9	Double	Computer Flow	16.9	35.0	35.8	7.6	4.6	34.4	22.6
25	Silicone	57.4	Single	Internal Coil	16.9	17.7	8.0	3.3	5.4	19.5	5.3
26	Drain-Back	49.4	Single	Wrap Around	12.2	31.1	19.6	2.2	14.1	48.1	26.8
27	Drain-Back	34.5	Double	Internal Coil	24.7	24.0	42.4	2.0	12.0	35.9	55.7
28	Glycol	62.5	Double	Wrap Around	16.5	22.9	11.0	2.4	9.5	32.4	12.8
29	Glycol	42.3	Double	Counter Flow	15.5	18.7	6.2	5.4	3.5	13.2	1.7

ALL MARKETED SYSTEMS

#25 and #29 have flow restrictors in order to obtain recommended flow rates.

Underlined numbers calculated by author from available data.

[1] Multiply $[ft^2]$ by .0929 to obtain $[m^2]$.

[2] See Table 23

[3] Multiply [kBtu/day] by 1.055 to obtain [mJ/day].

Table 21. DHW Systems January Monthly Average Performance Parameters.

System Identification Number (see Table 23)	16	17	19	20	21	22	23	23[1]
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[2]	970	970	929	1360	919	1122	1352	
A - Gross area of collector array (ft^2)[3]	54	54	42.2	44	76.6	1782	105	
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[4]	22.9	13.5	-	-	-	-	43.7	
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	43.7	25.7	-	-	-	-	30.8	
Q_L - Average daily solar system heat losses (1,000 Btu/day)[4]	1.6	0.6	-	-	-	-	10.7	
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[4] ($=Q_c - Q_I\ \beta - Q_E$)	21.3	12.9	7.7	18.0	19.0	200.0	33.0	
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	40.6	24.6	19.6	22.0	27.0	10.0	23.2	
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[4]	2.3	1.1	1.5				8.2	2.1
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_uC/E$)	9.1	11.6	5.1	∞	10.0	4.0	2.8	15.7
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[4] ($=Q_uC/\eta_A - E/\eta_E$)	29.9	19.2	8.2				28.5	51.9
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	48.8	33.9	18.2				16.4	34.6

Underlined numbers denotes calculation by authors from available data.

1. System 23, where electrical energy from pumps is considered to be added as heat to circulating loop.
2. Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].
3. Multiply [ft^2] by 0.0929 to obtain [m^2].
4. Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 22. DHW Systems July Monthly Average Performance Parameters:

System Identification Number (see Table 23)	16	17	18	19	23	23[1]
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[3]	1843	1843	1941	1500	1641	
A - Gross area of collector array (ft^2)[4]	54	54	585	42.2	105	
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[5]	39.5	34.2	347.1	-	68.4	
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	39.6	34.3	30.6	-	39.7	
Q_L - Average daily solar system heat losses (1,000 Btu/day)[5]	12.3	3.0	297.2[2]	-	26.6	
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[5] ($=Q_c - Q_I\,\beta - Q_E$)	27.2	31.2	49.9	18.9	41.8	
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	27.2	31.3	4.4	29.8	24.3	
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[5]	5.0	2.6	23.6	3.1	13.8	3.6
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_uC/E$)	5.4	12.2	2.1	6.1	3.0	11.6
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[5] ($=Q_uC/\eta_A - E/\eta_E$)	30.2	46.7	-4.2	22.4	22.9	62.2
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	25.4	42.6	-0.3	29.8	10.2	33.4

Underlined numbers denotes calculation by authors from available data.

1. System 23, where electrical energy from pumps is considered to be added as heat to circulating loop.
2. Thermal heat losses are to exterior only.
3. Multiply [$Btu/day \cdot ft^2$] by 11.4 to obtain [$kJ/day \cdot m^2$].
4. Multiply [ft^2] by 0.0929 to obtain [m^2].
5. Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 23. Identification of Systems Described in Tables 17 through 22.

ID	Reference(s)	Description	Months of Data
1	[43,44]	Direct, single tank, single-family	Feb-Jul
2	[43]	Direct, single tank, multi-family	Apr-Jun
3	[43,45]	Direct, double tank, multi-family	Mar-Aug
4	[46]	Direct, Industrial laundry	Nov-Dec
5	[47]	Indirect, double tank, restaurant	Jun-Aug
6	[43]	Indirect, multi-family, continuous circulation	May-Jul
7	[43]	Space and DHW heating, liquid	Jun-Jul
8	[43]	Space and DHW heating, liquid	Jul
9	[43]	Space and DHW heating, liquid	Jul
10	[48]	Thermosyphon, direct, single tank, liquid	Jul-Dec
11	[48]	Direct, single tank, liquid	Jul-Dce
12	[48]	Direct, double tank, liquid	Jul-Dec
13	[48]	Indirect, single tank, liquid	Jul-Dec
14	[48]	Indirect, double tank, liquid	Jul-Dec
15	[48]	Indirect, double tank, air	Jul-Dec
16	[49]	Direct, double tank	Jul-Jun
17	[49]	Indirect, single tank	Jul-Jun
18	[50]	Indirect, double tank	Jul-Oct
19	[51]	Direct, single tank	Jan-Jul
20	[52]	Passive, direct gain	1 year
21	[52]	Direct, single-family	1 year
22	[52]	Indirect, multi-family, continuous circulation	1 year
23	[53]	Indirect, double tank, Interstate Highway Visitor Center	Jan-Aug
24	[54]	Indirect, double tank, silicone	~3 weeks
25	[54]	Indirect, single tank, silicone	~3 weeks
26	[54]	Direct, single tank, water	~3 weeks
27	[54]	Direct, double tank, water	~3 weeks
28	[54]	Indirect, single tank, water/glycol	~3 weeks
29	[54]	Indirect, double tank, water/glycol	~3 weeks

3.4 PERFORMANCE OF PASSIVE SPACE HEATING SYSTEMS

3.4.1 Definitions/Assumptions

As discussed in the section on Durability and Reliability of Passive Solar Heating and Cooling, solar passive systems are characterized by reliance on natural convection and radiation and by heat collection and storage devices that are typically integrated with the building structure. Passive Space Heating is accordingly defined herein as the direct and/or indirect collection of incident solar radiation for space and/or DHW heating purposes, by means not requiring the forced circulation of a heat transfer fluid, either for solar collection or for delivery of solar heat to the various parts of the heating load (with the exception of air distribution blowers). In brief, solar passive systems accomplish heat transfer by natural means and do not require forced and/or mechanical movement of the heat transfer medium.

Solar passive systems are defined not to include energy conservation features. Energy conservation includes those design features which are intended to reduce a heating and/or cooling load; solar passive features are intended to increase the amount of available heat to meet the heating and/or cooling load. Typically passive systems are in addition to energy conservation features or designs.

The annual performance of a variety of solar passive space heating systems is given in Table 24. Tables 25, 26, and 27 give monthly values of three, specific, passive installations. Table 28 provides a more specific identification of each of these passive systems.

In developing Tables 24 through 27, several assumptions are made, including:

C = 1.0, i.e. the coefficient of performance of converting solar heat to useful space heating is unity.

$\eta_E = 0.26$

$\eta_A = 0.60$

S_c assumed to be zero (except for system 40, where $S_C = 5.1$)

Figure 4 (below) provides the total energy flows for a 176-day period for the system summarized in Table 25 [34].

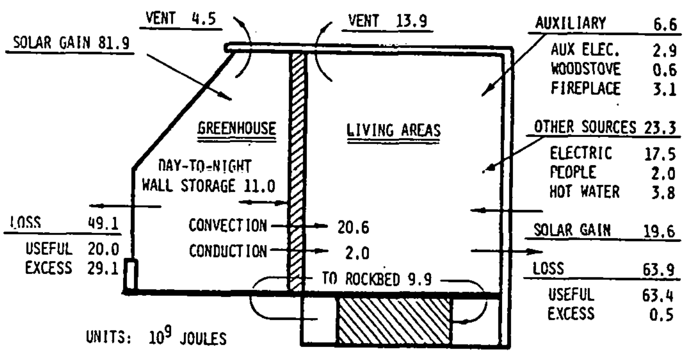

Figure 4. Total energy flows for the 176-day period from Nov. 1, 1978 through Apr. 24, 1979. [34]

3.4.2 Overall Evaluation

Passive systems performed with minimal operating problems and high efficiencies. System thermal efficiencies ranged from 23 to 44 percent, for most of the systems. System overall efficiencies were also excellent, ranging from 29 to 61 percent. On the other hand, temperature variations in the conditioned space ranged from 3 or 4°C to as much as 20°C, and in some cases, resulted in significant overheating. Two solar passive designs (Systems 32 and 37 in Table 20) which included day/night insulation on glazings, sufficient thermal mass to reduce temperature variations to within a few degrees celsius and realistic energy conservation features, achieved potential energy savings of 69.5 and 185.5 million Btu/year (73.3 and 195.1 GJ/year), or about 0.29 million Btu/year·ft^2 (3.3 GJ/year·m^2) of collecting surface area.

3.5 PERFORMANCE OF ACTIVE HEATING SYSTEMS

3.5.1 Introduction

The annual performance of a variety of solar active space heating systems is given in Tables 29, 30, and 31. Table 36 provides a more specific identification of each of these active heating systems. Tables 32 through 35 show monthly data on several selected systems. In developing Tables 29 through 35, it is assumed that C = 1, η_E = 0.26, η_A = 0.6 (see section on Criteria for Performance Analysis).

3.5.2 Overall Evaluation

On average, active heating systems performed at unexpectedly low levels. However, several systems which had received careful attention to details of design, installation, operation and maintenance performed quite well. One residential- and one commerical-sized active heating system had system thermal efficiencies of 30 and 32 percent, respectively. These two systems had overall efficiencies of 39 and 42 percent, respectively. The residential-sized system (identified as system 61 in Table 31) achieved an annual energy savings of 106 million Btu/year (112 GJ/year), or about .16 million Btu/year·ft^2 (1.8 GJ/year·m^2) of collector. The commercial-sized system (identified as system 62 in Table 31) had equivalent energy-saving values of 1438 million Btu/year (1520 GJ/year), and 0.19 million Btu/year·ft^2 (2.1 GJ/year·m^2) of collector. These factors may be interpreted to mean that well-designed, active space heating systems can provide these expected energy savings.

3.6 PERFORMANCE OF SPACE COOLING SYSTEMS

3.6.1 Introduction

The annual performance of a variety of solar space cooling systems is given in Tables 37 and 38. Tables 39 and 40 provide more detailed, monthly data on several selected systems. Table 41 provides a more specific identification of each of these solar cooling systems. In calculating the values in Tables 37 through 40, it was assumed that η_E = 0.26, η_A (cooling) = 0.65, and η_A (DHW) = 0.55.

3.6.2 Overall Evaluation

All but three of the systems reported on in Tables 37 and 38 had negative energy savings, and in some cases the solar cooling system used substantially more energy than a conventional system could be expected to use. Two systems (identified as systems 78B and 79 in Table 38), however, had significant energy savings. These systems (1 residential and 1 commercial) obtained system thermal efficiencies of 12-12.4 percent. Their system overall efficiencies were 11.2 and 5.1-5.3 percent, respectively. The residential-sized system (#78B) achieved an annual energy savings of about 16 million Btu/year (16.8 GJ/year), or approximately .03 million Btu/year·ft^2 (.34 GJ/year·m^2) of collector. The commercial system (system 79) had equivalent values of 130 million Btu/year (137 GJ/year), or about .02 million Btu/year·ft^2 (.22 GJ/year·m^2) of collector.

It should be noted that these efficiencies are much lower than those of well-designed and properly controlled cooling systems in commercial sizes.

Table 24. Passive Systems Annual Average Performance Parameters.

System Identification Number (See Table 28)	31	32	33	34	35	36	37	38	39	40
									Roof/Windows	
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1346	1636	1140	(IA=.15 x 10^6 Btu/day)[3]	1023	1127	1226	1667	1866/800	1644
A - Gross area of collector array (ft^2)[2]	$\underline{850}$	520	-		1440	$\underline{720}$	300	400	338/141	440[f]
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	-	494.8	-	73.2	$\underline{618.7}$		169.5	291.9	163.5/68.6	238.4
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	45[a]	58.2	43[a]	48.7	42[a]	46[a]	46.1	43.8	25.9/60.8	33.0
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	-	-	-	$\underline{8.3}$	-	-		$\underline{4.2}$	-	-
β - System heat loss factor (non-usefulness of heat losses to interior)	-	.01(Living Area) .59(Greenhouse)		0[g]			$\underline{0.26}$	0[g]		.16[g]
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\ \beta - Q_E$)	286.6	317.6		64.9	$\underline{383.0}$	315.0	$\underline{125.4}$	287.7	175.1	217.3
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	25[b]	37.3		43.2	26[b]	39[b]	$\underline{34.1}$	43.2	23.5	30.1
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	3.2[c]	5.5			42.0[d]	5.0[e]	4.8			23.2
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u$C/E)	$\underline{90}$	60.2			$\underline{9.2}$	$\underline{6.3}$	$\underline{26.1}$			9.6
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	$\underline{465.4}$	$\underline{508.2}$			$\underline{476.8}$	$\underline{505.8}$	$\underline{190.5}$			$\underline{272.9}$
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	$\underline{40.2}$	$\underline{58.3}$			$\underline{29.2}$	$\underline{60.9}$	$\underline{49.3}$			$\underline{33.6}$

Underlined numbers denotes calculation by authors from available data.

[a]"Solar Utilization Efficiency" = $\frac{\text{Solar Energy Used for Space Heating}}{\text{Total Incident Solar Energy}}$ (Energy used includes thermal losses through glazings.)

[b]"Savings Efficiency" (Includes heat losses through glazings.)

[c]Greenhouse fan

[d]Solar System Operating Energy

[e]"Beadwall" Operating Energy

[f]Does not include 1055 ft^2 reflector

[g]Assumed Values (estimated)

[1]Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·ft^2].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 25. Passive Systems Monthly Average Performance Parameters [34].

System Identification Number 32 MONTH/	NOV	DEC	JAN	FEB	MAR	APR	Season
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1378	1474	1478	2058	1807	1652	1636
Gross area of collector array (ft^2)[2]							850
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	409.3	552.8	563.4	604.5	473.7	346.0	494.8
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	57.2	72.2	73.4	56.5	50.4	40.3	58.2
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\ \beta - Q_E$)	228.9	392.5	443.1	390.4	259.5	168.8	317.6
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	32.0	51.2	57.7	36.5	27.7	19.7	37.3
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	5.3	5.3	4.2	7.4	5.3	5.3	5.5
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u$C/E)	43.7	78.2	100.6	51.2	52.6	31.3	60.2
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	361.1	633.8	722.3	622.2	412.1	260.9	508.2
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	30.3	49.8	56.8	35.0	26.5	18.3	36.0

Underlined numbers denotes calculation by authors from available data.

A_{Total} = 519.7 ft^2 (48.3 m^2)

$A_{greenhouse}$ = 344.3 ft^2 (32 m^2)

	β	Room Temperature[4] Low	High	Average
(living area)	0.01	60°F	75°F	68°F
(greenhouse)	0.59	45°F		64°F

[1]Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·ft^2].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

[4]°C = (°F-32)/1.8.

Table 26. Passive System Monthly Average Performance Parameters [35].

System Identification Number 39 / MONTH	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	Season
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[1] ROOF	2084	1373	1838	1660	2256	2025	1919	1732	1866
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[1] WINDOWS	1122	75	1765	1788	1174	0	0	0	800
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3] ROOF	68.6	183.6	138.2	96.0	237.4	270.1	175.1	126.6	163.5
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3] WINDOWS	128.7	16.9	107.6	142.4	84.4	0	14.8	24.3	68.6
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	9.9	39.5	22.2	17.1	31.1	39.4	27.0	21.6	25.9
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I \beta - Q_E$)	118.2	177.2	217.3	224.7	227.9	185.7	107.6	101.3	175.1
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	13.7	37.3	25.0	27.6	24.6	27.1	16.6	17.3	23.5

Notes:

$A_{roof\ aperture}$ = 338 ft^2 (31.3 m^2)

$A_{south\ window}$ = 141 ft^2 (13.1 m^2)

[1]Multiply [$Btu/day \cdot ft^2$] by 11.4 to obtain [$kJ/day \cdot ft^2$].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 27. Passive System Monthly Average Performance Parameters [36].

System Identification Number 40 / MONTH	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	Season
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[1]	1884	1430	1632	1442	1987	1752	1524	1359	1644
A - Gross area of collector array (ft^2)[2]									440
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	227.9	182.5	246.9	230.0	359.8	268.0	204.7	155.1	238.4
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	27.4	29.0	34.4	36.4	41.2	34.8	30.6	25.2	33.0
β - System heat loss factor (non-usefulness of heat losses to interior of space)	0.23	.07	.00	.00	.00	.00	.57	.67	.16
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\ \beta - Q_E$)	187.8	179.4	246.9	230.0	359.8	268.0	128.7	79.1	217.3
η_T - Average daily solar heating/cooling system thermal efficiency ($Q_u C/AI$)	22.7	28.5	34.4	36.4	41.2	34.8	19.2	12.9	30.1
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	14.8	26.4	21.1	19.0	32.7	28.5	20.0	17.9	23.2
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u C/E$)	13.0	6.7	11.6	12.2	11.0	9.5	6.2	4.4	9.6
S_c	3.9	4.9	6.3	6.0	6.9	4.8	3.4	2.3	5.1
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	256.1	197.5	330.3	310.3	473.9	337.1	137.6	63.0	272.9
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	31.6	29.3	45.0	47.7	51.5	41.6	20.0	10.3	36.5

Underlined numbers denotes calculation by authors from available data.

$A_{roof\ aperture} = 338\ ft^2\ (31.3\ m^2)$ *$\beta = 1$ when $Ta > 50°F\ (10°C)$[4]

$A_{south\ window} = 141\ ft^2\ (13.1\ m^2)$ $\beta = 0$ when $Ta < 50°F\ (10°C)$

Monthly average calculated by average daily values.

[1]Multiply [$Btu/day \cdot ft^2$] by 11.4 to obtain [$kJ/day \cdot ft^2$].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

[4]$°C = (°F-32)/1.8$.

Table 28. Identification of Solar Passive Heating Systems described in Table 24.

ID	Reference(s)	Description	Months of Data
31	[55]	Combination sunspace (greenhouse)/mass wall system, single family	Nov-Apr
32	[34]	Greenhouse/mass wall, single family	Nov-Apr
33	[55]	Direct Gain (south facing window wall and overhead sky light), single family, 2500 gal (9475ℓ) water-filled tubes, storage near windows/skylight and concrete slab floor	Feb-Apr
34	[43]	Direct Gain, Mass wall	Dec-May
35	[55, 56]	Direct Gain, warehouse	Dec-Apr
36	[55]	Combination Direct Gain/drum wall and bead wall movable insulation, single family	Nov-Apr
37	[57]	Earth covered, Direct Gain, Mass wall, Bead wall	Nov-May
38	[43]	Greenhouse, Mass wall, Remote rock bed	Mar-Apr
39	[35]	Clerestory windows across roof, hinged insulation panels, water plastic bags storage	Oct-May
40	[36]	Hybrid: Active air heating system 440 ft^2 (40.8 m^2) Passive: Adobe Mass Wall, Direct Gain Cooling--Night Evaporative Cooling	Oct-May

Table 29. Active Heating Systems Annual Average Performance Parameters.

System Identification Number	41	42	43	44	45	46	47	48	49	50
IA - Solar Radiation on tilted surface of solar collector times gross area of collector array (million Btu/day)[1]	.722	.884	.545	.757	.391	.625	.435	.323	.519	13.99
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[1]	212.7	169.8	101.5	138.8	84.4	181.1	38.6	61.1	71.1	3635..
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	29.5	19.2	18.6	18.3	21.6	29.0	8.9	18.9	13.7	26.0
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[1]	160.3 (142.6[a])	6.3	85.5 (55.2[a])	10.6	47.1	23.2	29.9[a]	52.6 (33.0[a])	16.2[a]	-
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[1]	-	107.0 (93.2[a])	1.4	97.7 (94.4[a])	24.2	140.7	-	-	-	-
β - System heat loss factor (non-usefulness of heat losses to interior of space)	-	0.0[b]	0.0[b]	1.0[b]	0.3	0.84	-	-	-	-
C - Heating or cooling unit Coefficient of Performance (dimensionless)	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[1] ($=Q_c - Q_I\ \beta - Q_E$)	52.4	163.5	14.6	30.5	30.1	40.0	8.7	8.5	54.9	1743
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	7.3	18.5	2.7	4.0	7.7	6.4	2.0	2.6	10.6	12.5

Underlined numbers denotes calculation by authors from available data.

[a]Thermal losses from storage.

[b]β assumed to be zero by reference publication.

[1]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 30. Active Heating Systems Annual Average Performance Parameters.

System Identification Number	51	52	53	54	55	56	57	57[f]	58	58[f]
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1403	982	1061	1210	1061	1216	1605		1108	
A - Gross area of collector array (ft^2)[2]	436	1932	2685	400	2496	512	335		675	
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	207.6	493.3	398.9	106.5	900.4	205.5	128.4		221.9	
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	33.9	26.0	14.0	22.0	34.0	33.0	23.9		29.6	
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	17.0	61.6[a]	15.4	-	64.3[a]	29.7[a]	5.7		78.7	
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]	163.1[a]	-	92.3[a]	63.2[a]	-	-	60.7[a] 6.9[c]		-	
β - System heat loss factor (non-usefulness of heat losses to interior of space)	.05[b]	-	1.0[b]	-	-	-	0.7		-	
C - Heating or cooling unit Coefficient of Performance (dimensionless)	1.0	1.0	1.0	1.0	1.0	1.0	1.0		1.0	
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\ \beta - Q_E$)	109.1	294.5	300.7	80.5	438.7	89.8	59.1[h] 21.1[d]		132.0[h] 11.2[d]	
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	17.8	15.5	10.6	16.6	16.6	14.4	14.9		19.1	
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	29.6	17.4	44.2	9.1	9.9	17.6	15.8[e,g]	23.3	18.5[e,g]	34.3
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u$C/E)	6.9	16.9	6.8	8.8	44.5	5.1	5.1	3.4	7.7	4.2
S_c	1.8	-	-	-	-	-	3.5	2.4	7.7	4.2
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	68.0	423.9	331.2	99.2	693.1	82.0	72.9	44.1	167.6	106.8
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	9.4	21.6	11.0	19.1	25.8	11.9	12.2	7.0	20.5	12.1

Underlined numbers denotes calculation by authors from available data.

[a]Thermal losses from storage.

[b]β assumed by reference publication.

[c]Heat losses from ducts/piping.

[d]Solar energy delivered to DHW load.

[e]Collector pump electrical operating energy.

[f]Distribution fan electrical operating energy.

[g]Solar heating pump electrical operating energy.

[h]Solar energy delivered to space heating load.

[1]Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 31. Active Heating Systems Annual Average Performance Parameters.

	'76-'77	'77-'78	'77	'78	'77-'78	'77-'78	'78-'79	'77-'78	'78-'79
System Identification Number	59	59	60	60	61	61	61	62	62
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1837	1765	1037	900	1499		1733	1697	1654
A - Gross area of collector array (ft^2)[2]	340	340	220	220	722		623	7705	7705
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	113.6	113.9	60.6	~57	363.2		356.9	4201.9	4226.5
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	18.2	19.0	26.6	~29	33.6		33.2	32.1	33.2
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	26.2	30.9	10.4 (4.8[a])	7.8 (3.1[a])	6.6[c] 13.8[a]		-	-	-
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]	42.5[a] 30.6[c]	58.5[a] 9.5[c]	-	-	28.4		-	618.0	564.9
β - System heat loss factor (non-usefulness of heat losses to interior of space)	0.0[b]	0.0[b]	-	-	0.0[b]		-	.13	.20
C - Heating or cooling unit Coefficient of Performance (dimensionless)	1.0	1.0	1.0	1.0	1.0		1.0	1.0	1.0
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I \beta - Q_E$)	73.1[h] 14.3[d]	68.0[h] 15.0[d]	50.2	49.2	311.5[h] 31.1[d]		259.2[h] 34.4[d]	4119.4	4114.7
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	14.0	13.8	22.0	24.8	31.7		27.2	31.5	32.3
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	17.7	13.3	6.4	5.0	19.4[e]	35.4[e,f]	13.6[e] 8.2[g]	263.5	253.1
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u$C/E)	4.9	6.2	7.8	9.8	17.7	9.7	13.5	15.6	16.2
S_c	0.8	1.1	7.8	9.8	16.2	8.4	13.5	13.6	14.5
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	77.6	87.1	59.0	62.8	496.7	435.2	405.5	5852.2	5884.3
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	11.2	13.4	23.3	28.9	42.9	35.7	34.9	41.5	42.9

Underlined numbers denotes calculation by authors from available data.

[a] Thermal losses from storage.

[b] β assumed to be zero by reference publication.

[c] Heat losses from ducts/piping.

[d] Solar energy delivered to DHW load.

[e] Collector pump electrical operating energy.

[f] Distribution fan electrical operating energy.

[g] Solar heating pump electrical operating energy.

[h] Solar energy delivered to space heating load.

[1] Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].

[2] Multiply [ft^2] by 0.0929 to obtain [m^2].

[3] Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 32. Active Heating (AIR) System Monthly Average Performance Parameters [64].

System Identification Number 61 / MONTH		OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	Season
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]		1824	1165	1283	1208	1242	1882	1781	1765	1499
A - Gross area of collector array (ft^2)[2]										722
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]		439.0	308.2	352.7	320.2	305.1	439.4	372.6	380.6	363.2
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)		33.3	36.6	38.1	36.7	34.0	32.3	29.0	29.9	33.6
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	g	22.7	23.0	3.5	3.5	4.2	29.7	6.1	28.3	6.6
	s	21.7	9.8	5.9	4.9	8.7	19.8	24.6	20.2	13.8
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]		49.2	21.0	29.1	4.4	10.3	33.2	45.2	50.3	28.4
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\ \beta - Q_E$)		304.8[h]	263.5[h]	292.9[h]	264.1[h]	292.2[h]	389.9[h]	354.1[h]	332.1[h]	311.5[h]
		89.9[d]	57.9[d]	50.4[d]	47.7[d]					31.3[d]
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)		30.0	38.2	37.1	35.7	32.6	28.7	27.5	26.1	31.7
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	c	21.1	19.3	21.7	17.1	16.7	20.3	19.6	19.4	19.4
	f	3.3	14.8	25.1	27.4	23.7	11.4	6.2	10.3	16.0
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_uC/E$)		16.2	9.4	7.3	7.0	7.2	12.3	13.7	11.2	9.7
S_c		14.2	8.8	6.7	6.9	7.0	11.3	12.0	9.5	8.4
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_uC/\eta_A - E/\eta_E$)		564.0	404.5	392.2	348.5	331.6	527.9	490.9	439.3	435.2
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)		37.6	41.6	35.5	33.4	31.5	35.7	35.4	31.6	35.7

Underlined numbers denotes calculation by authors from available data.

[c] Collection pump electricity
[d] DHW heating
[f] Fan distribution electricity
[g] Heat losses from ducting
[h] Space heating
[s] Heat losses from storage to ground
β (assumed) = 0

[1] Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].
[2] Multiply [ft^2] by 0.0929 to obtain [m^2].
[3] Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 33. Active Heating (Liquid, Drain Back) System Monthly Average Performance Parameters [62].

System Identification Number 58 / MONTH		OCT	NOV	DEC	JAN	FEB	MAR	APR	Season
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft²)[1]		1267	847	948	907	998	1268	1393	1108
A - Gross area of collector array (ft²)[2]									675
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]		132.1	158.9	231.8	213.1	350.7	279.4	197.1	221.9
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)		15.4	27.8	37.9	34.8	52.1	32.6	21.0	29.6
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]		46.3	43.4	57.9	27.0	115.2	135.3	130.0	78.7
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]		-	-	-	-	-	-	-	-
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I$ b $- Q_E$)	h	72.8	104.4	166.0	180.5	225.0	129.2	51.6	132.0
	d	13.0	11.1	7.9	5.5	10.5	14.9	15.5	11.2
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)		10.0	20.2	27.2	30.4	35.0	16.8	7.1	19.1
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	c	3.6	5.6	8.9	10.2	13.5	9.8	6.7	8.3
	g	1.8	7.1	14.9	27.8	17.8	6.2	1.4	10.2
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u$C/E)		15.9	9.1	7.3	5.6	7.5	9.0	8.3	7.7
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)		122.2	143.7	198.3	183.2	272.1	178.6	80.7	167.5
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)		13.9	23.2	27.1	24.8	34.3	19.5	8.3	20.5

Underlined numbers denotes calculation by authors from available data.

[h]Space heating
[c]Collector pump
[d]DHW heating
[g]Heating pump

[1]Multiply [Btu/day·ft²] by 11.4 to obtain [kJ/day·m²].
[2]Multiply [ft²] by 0.0929 to obtain [m²].
[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 34. Active Heating (Liquid) System Monthly Average Performance Parameters [58].

System Identification Number 50 / MONTH	OCT	NOV	DEC	JAN	FEB	MAR	APR	Season
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[1]	1290	801	741	1096	1357	1605	1665	1222
A - Gross area of collector array (ft^2)[2]								220
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	3516.1	1833.3	2419.4	3096.8	3750.0	4935.5	5900.0	3635.9
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	23.8	20.0	28.5	24.7	24.1	26.9	30.9	26.0
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I F - Q_E$)	612.9	500.0	2322.6	3612.9	3607.2	1080.0	466.7	1743.2
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	4.1	5.5	27.4	28.8	23.2	5.9	2.4	12.5

[1]Multiply [$Btu/day \cdot ft^2$] by 11.4 to obtain [$kJ/day \cdot m^2$].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 35. Active Heating (Liquid) System Monthly Average Performance Parameters [66].

System Identification Number 62 / MONTH	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	Season
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1851	1201	1446	1410	1829	1878	1880	1860	1654
A - Gross area of collector array (ft^2)									7705
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	4779.1	3029.4	3735.5	3380.1	4837.9	5035.1	4456.9	5029.4	4226.5
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	35.5	32.7	33.5	31.1	34.3	34.8	30.8	35.1	33.2
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	-	-	-	-	-	-	-	-	-
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]	405.7	296.7	490.0	668.2	728.0	585.8	624.6	922.3	564.9
β - System heat loss factor (non-usefulness of heat losses to interior of space)	.6	.1	0	0	0	0	.6	.4	.2
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I \beta - Q_E$)	4530.8	3000.0	3735.5	3380.1	4837.9	5035.1	4082.5	4634.1	4114.7
η_T - Average daily solar heating/cooling system thermal efficiency ($Q_u C/AI$)	31.8	32.4	33.5	31.3	34.3	34.8	28.2	32.3	32.3
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	288.2	206.5	219.9	200.0	269.2	291.0	274.9	309.0	253.1
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u C/E$)	15.7	14.5	17.0	16.9	18.0	17.3	14.8	15.0	16.2
S_c -	15.2	13.2	14.8	13.6	15.3	15.3	13.9	13.3	14.5
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	6442.9	4205.4	5380.1	4864.3	7027.8	7272.6	5746.9	6535.0	5884.4
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	41.9	41.9	44.9	41.8	46.5	46.7	37.0	42.1	42.9

Underlined numbers denotes calculation by authors from available data.

[1]Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 36. Identification of Solar Active Heating Systems described in Tables 29 through 35.

ID	Reference(s)	Description	Months of Data
41	[43]	Space and DHW (liquid)	Jan-Jul
42	[43]	Space and DHW (liquid)	Mar-Apr
43	[43]	Space and DHW (liquid)	Mar-Jul
44	[43]	Space and DHW (air)	Jan-Mar
45	[43]	Space and DHW (air)	Mar-Apr
46	[43]	Space and DHW (air)	Feb-Jul
47	[43]	Space and DHW (air)	Oct-Mar
48	[43]	Space and DHW (air)	Dec-Mar
49	[43]	Space and DHW (air)	Dec-Mar
50	[58]	Heating and Cooling System, office bldg. (7 different types of flat plate collector)	1 year
51	[59]	Space and DHW (air), single family	Mar-Apr
52	[60]	Space heating (air), Garage ($T_r \geq 50°F$; $10°C$) and office area ($T_r \simeq 68°F$; $20°C$)	Nov-Mar
53	[60]	Space heating (air), elementary school	Oct-Mar
54	[60]	Space heating (water), office/warehouse	Oct-Mar
55	[60]	Space heating (air), for gymnasium (school), DHW heating for locker room, hot air for grain drying	Oct-Mar
56	[60]	Space and DHW heating (water), single family, solar assisted heat pump, drain-down	Oct-Mar
57	[61]	Space and DHW heating (liquid), water/ glycol, single family	Mar-Apr
58	[62]	Space and DHW heating (liquid), drain back, single family	Oct-Apr
59	[63]	Space and DHW (air), single family (mobile home)	Oct-May
60	[64]	Space heating only (liquid), water/ glycol, office bldg.	Feb-Apr 77
60	[64]	Space heating only (liquid), office bldg.	Jan-Mar 78
61	[64, 65]	Space and DHW (air), single family, (used as office)	Oct-May/ Dec-May
62	[66]	Heating and Cooling System, Conference Center and Library	2 years

Table 37. Cooling Systems Annual Average Performance Parameters.

System Identification Number	71	'78 72	'79 72	73	74	75	76
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1621	1542	1681	1854	1969	1483	1652
A - Gross area of collector array (ft^2)[2]	714	3840	3840	3650	4950	11000	12660
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	287.7	877.5	774.7	1977.9	3756.4	2285.0	4629.3
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	24.9	14.8	12.0	29.2	38.5	14.0	26.7
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	139.8[a]	627.7	316.3[a]	155.6[a]	91.9	614.4 (608.7[a])	-
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]	-	-	4.6	37.9	183.8	307.2 (304.3[a])	-
C - Heating or cooling unit Coefficient of Performance (dimensionless)	0.60	0.46	.586	.366	.34	.423	.62
β - System heat loss factor (non-usefulness of heat losses to interior of space)	-	3.17	2.71	3.73	3.95	3.36	-
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\,\beta - Q_E$)	147.9	249.8	446.1 338.3[c]	1838.9 1680.9[c]	2936.0 2758.7[c]	638.4	2729.6
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	7.7	1.9	4.7	11.4	11.4	1.7	8.1
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	115.0	-	206.7	1931.5	2851.3	500.2	-
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_uC/E$) = S_c	0.8	-	1.5	0.4	0.4	0.5	-
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_uC/\eta_A - E/\eta_E$)	-305.8	-	-294.0	-6195.0	-9201.1	-1508.4	-
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	-19.1	-	-4.1	-43.6	-44.4	-8.3	-

Underlined numbers denotes calculation by authors from available data.

[a]Heat losses from storage

[c]Solar heat delivered to cooling unit only

[1]Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 38. Cooling Systems Annual Average Performance Parameters.

System Identification Number	77A	77B	78A	78B	'78 79	'79 79
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[1]	1877	1999	1621	1586	1881	1804
A - Gross area of collector array (ft^2)[2]	1923	1923	631	631	7705	7705
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	448.4	692.8	299.1	339.3	3033.2	2962.1
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	12.4	18.0	29.2	33.9	20.9	21.3
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	146.0 91.5[a]	87.3 62.8[a]	32.1	50.9	544.1	489.1
Q_I - Average daily solar system heat losses to interior (1,000 Btu/day)[3]	7.7	7.7	45.4	21.1	-	-
C - Heating or cooling unit Coefficient of Performance (dimensionless)	.72	.54	.605	.527	.69	.68
β - System heat loss factor (non-usefulness of heat losses to interior of space)	2.39	2.85	2.65	2.9	-	-
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I \beta - Q_E$)	284.0	583.6	146.8	227.1	2489.1	2473.9
η_T - Average daily solar heating/cooling system thermal efficiency ($Q_u C/AI$)	5.7	8.2	8.7	12.0	12.0	12.4
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	100.5[d]	106.4[d]	53.0	16.8	463.5	464.5
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u C/E$) $= S_c$	2.0	3.0	1.7	7.1	3.8	3.7
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	-72.0	75.6	-67.2	119.5	859.6	801.5
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	-1.8	1.8	-5.5	11.2	5.3	5.1

Underlined numbers denotes calculation by authors from available data.

[a]Heat losses from storage

[d]Does not include electrical energy to blowers

[1]Multiply [$Btu/day \cdot ft^2$] to 11.4 to obtain [$kJ/day \cdot m^2$].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 39. Cooling System (Residential) Monthly Average Performance Parameters.

System Identification Number 71 / MONTHLY	MAY	JUN	JUL	AUG	Season
I - Solar Radiation on tilted surface of solar collector (Btu/day·ft^2)[1]	1361	1581	1437	1699	1621
A - Gross area of collector array (ft^2)[2]					714
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	305.3	300.7	247.1	297.7	287.7
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	31.4	26.6	24.1	24.5	24.9
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3]	186.7	142.5	81.9	148.2	139.8
C - Heating or cooling unit Coefficient of Performance (dimensionless)	.68	.62	.57	.53	0.60
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I\,\beta - Q_E$)	118.6	158.2	165.2	149.5	147.9
η_T - Average daily solar heating/cooling system thermal efficiency (Q_uC/AI)	12.2	14.0	16.1	12.3	12.8
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	93	120	112	125	115
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_uC/E$)	0.9	0.8	0.8	0.6	0.8
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_uC/\eta_A - E/\eta_E$)	-233.6	-310.6	-285.9	-358.9	-305.8
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	-17.6	-19.5	-19.6	-21.2	-19.1

Underlined numbers denotes calculation by authors from available data.

[1]Multiply [Btu/day·ft^2] by 11.4 to obtain [kJ/day·m^2].

[2]Multiply [ft^2] by 0.0929 to obtain [m^2].

[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 40. Cooling System (Commercial) Monthly Average Performance Parameters.

System Identification Number 79 / MONTHLY	MAY	JUN	JUL	AUG	SEP	Season
I - Solar Radiation on tilted surface of solar collector ($Btu/day \cdot ft^2$)[1]	1911	1949	1827	1820	1900	1881
A - Gross area of collector array (ft^2)[2]						7705
Q_c - Average daily useful solar energy collected by array (1,000 Btu/day)[3]	2834.1	3085.3	3209.5	3371.6	2625.6	3033.2
η_c - Average daily solar collector array efficiency (%) (Q_c/AI)	19.2	20.5	22.8	24.0	17.9	20.9
Q_E - Average daily solar system heat losses to exterior (1,000 Btu/day)[3] *	535.5	535.5	507.1	523.2	626.5	544.1
C - Heating or cooling unit Coefficient of Performance (dimensionless)	.71	.69	.69	.70	.66	.69
Q_u - Average daily solar system heat delivered to heating/cooling unit (1,000 Btu/day)[3] ($=Q_c - Q_I \beta - Q_E$)	2298.6	2548.9	2702.4	2848.3	1999.1	2489.1
η_T - Average daily solar heating/cooling system thermal efficiency ($Q_u C/AI$)	11.3	11.9	13.4	14.4	9.2	12.0
E - Average daily electrical energy used to operate the solar system (1,000 Btu/day)[3]	421.8	458.8	510.0	510.0	413.3	463.5
S - Solar useful heating and/or cooling/electrical solar-operating energy used ratio ($=Q_u C/E$)	4.0	3.9	3.7	4.0	3.3	3.8
Q_s - Average daily savings in nonrenewable energy resources by solar system (1,000 Btu/day)[3] ($=Q_u C/\eta_A - E/\eta_E$)	888.5	941.1	907.2	1105.9	440.2	859.6
η_S - Average daily solar heating and cooling system overall efficiency ($=Q_s/(AI + E/\eta_E)$)	5.4	5.6	5.7	6.9	2.7	5.3

Underlined numbers denotes calculation by authors from available data.
*Heat losses in machinery space are vented to exterior.
[1]Multiply [$Btu/day \cdot ft^2$] by 11.4 to obtain [$kJ/day \cdot m^2$].
[2]Multiply [ft^2] by 0.0929 to obtain [m^2].
[3]Multiply [Btu/day] by 1.055 to obtain [kJ/day].

Table 41. Identification of Cooling Systems detailed in Tables 37 through 40.

ID	Reference(s)	Description	Months of Data
71	[67,68]	Space heating, cooling and DHW, single family (Data for space cooling only)	May-Aug
72	[69,70]	Solar cooling and DHW, office building Concentrators	Mar-Aug 78 Jun-Aug 79
73	[70]	Solar cooling, Space and DHW heating, Recreation and Health Center (data for cooling only)	Jun-Aug
74	[70]	Heating and Cooling, Elementary School Evacuated tube 2 100-ton absorption units No Storage	Jun-Aug
75	[70]	Heating and Cooling, Elementary School, (water) Reflector	Jun-Aug
76	[60]	Office building, Heating and Cooling, 174-ton Absorption chiller 7 different types flat plate collectors	1 year
77	[71]	Design A Single family, water/glycol B Single family, water/glycol	Jul-Aug (2 yrs)
78	[12,25]	Space heating and cooling and DHW (Data for space cooling only) Design A Design B	Jul-Aug Aug
79	[72]	Conf. Center and Library, Heating and Cooling	2 years

4. ANALYSIS OF SYSTEMS PERFORMANCE

4.1 INTRODUCTION

In evaluating the performance of solar systems discussed in the section, several factors are considered. These factors include:

1. Comparison of predicted and measured performance of specific designs and/or installations
2. Comparison of performance of alternative designs
3. Design, installation and operational features which affect the overall performance, and
4. Common errors and/or problems in design, installation, and operational procedures and/or methods.

In comparing the performance of solar systems the emphasis will be limited to two primary areas, i.e.:

1. Performance comparison of different solar system designs and/or installations and
2. Comparison of high performance solar heating and cooling systems and conventional HVAC alternatives.

4.2 DESIGN METHODS

Numerous methods are available for predicting solar system performance, including detailed computer simulations, hand-held computer methods, simplified procedures not requiring computers and rules of thumb. Tables 42 through 47 [73] provide information on some of the various design methods available. This information includes:

1. Applications of computer methods and
2. Characteristics of hand calculation methods.

Limitations on the usefulness of these methods, which affect the results for different methods to different degrees, include:

- Virtually all methods for active solar designs are based on the Hottel-Whillier-Bliss model of the solar collector and are therefore limited by the same assumptions of that model (see Section 4.5.2.1 and Appendix C).
- Testing results for collector characteristics are usually based on optimum noon time conditions.
- Only limited validation of some of the design methods by comparison with carefully-measured operating systems has been accomplished. Many of the simplified methods have received no validation with experimental results.
- Methods that deal with passive systems (both computer and hand calculations) are limited in number.
- There are very few hand calculation methods capable of analyzing solar cooling systems and they may not be capable of evaluating different types of solar cooling systems [73].
- Most of the calculation methods do not have the ability to account for all design variations (such as different collector characteristics, control system variations, modifications in system component integration, operating temperatures, etc.).
- Many of the models do not consider the electrical energy required for operating the solar system and the effects of this electrical energy usage on the total energy savings capability of the solar system. Alternatively, DOE-2, BLAST, SEE and TRACE do; although even these methods do not always provide comparisons with conventional systems electrical usage.
- The accuracy of all models is dependent upon the accuracy of input data. Specifically, the estimated heating and/or cooling load may be in error on the order of 15 percent (some calculation methods have been shown to be in error by overestimating heating loads by more than 100 percent). In addition, solar radiation data are typically in error by five percent and, in some cases, by as much as 15 percent.
- Many methods (particularly hand calculation methods) do not consider the effects of storage temperature variations.
- Numerous methods are highly empirical.

Table 42. General Characteristics of Computer Methods [73]

Program Name	Originator	Date	Cost	Development Status: Users Manual	Development Status: Program Manual	Development Status: Public Availability	Life Cycle Economics	Load Model: UA	Load Model: Internal Detailed	Load Model: External	Solar Emphasis: Primary	Solar Emphasis: Secondary	Solar Type: Active	Solar Type: Passive	Collector Fluid Type: Liquid	Collector Fluid Type: Air
BLAST	U.S. Army Construct. Engineering Lab	1977	$300	*	*	*	*		*			*	*	*	*	*
CBS	Los Alamos Sci. Lab	1979	$300	*	*	*		*		*	*		*		*	*
DEROB	University of Texas at Austin	1973							*		*			*		
DOE-1	Argonne Nat'l Lab	1977	$400	*	*	*	*		*			*	*	*	*	*
EMPSS	Arthur D. Little	1978		*	*	*	*		*	*	*		*		*	*
FCHART	Univ. of Wisconsin	1976	$100	*		*	*	*		*	*		*		*	*
FREHEAT	Colorado State Univ.	1979	$150		*	*		*	*					*		
HISPER	Marshall Space Flight Center	1977		*		*			*		*		*		*	*
LASL	Los Alamos Sci. Lab	1975	None			*		*			*		*		*	*
PASOLE	Los Alamos Sci. Lab	1977	$175						*		*			*		
RSVP	Booz, Allen, Hamilton	1977		*	*	*	*	*		*	*		*		*	*
SESOP	Lockheed	1975	$530	*		*	*	*		*	*		*		*	*
SHASP	Univ. of Maryland	1978	None			*		*			*		*			*
SHSEMOD	Jan F. Kreider	1973					*	*		*	*		*		*	*
SIMSHAC	Colorado State Univ.	1974							*	*	*		*		*	
SOLCOST	Martin Marietta	1976	$300	*		*	*	*	*	*	*		*		*	*
SOLHEAT	Natural Heating Systems	1978			*		*	*		*	*		*		*	
SOLOPT	Texas A&M Univ.	1977	None	*		*	*	*			*		*		*	
SOLPAS	Martin Marietta	1978							*		*			*		
SOLSYS	Sandia Laboratories	1975		*	*	*	*	*	*	*	*		*		*	
STOLAR	Colorado State Univ.	1977	None			*		*			*		*		*	
SUN	Berkeley Solar Group	1974						*			*		*		*	
SZOKO	S.V. Szokolay	1977			*	*		*			*		*		*	
SYRSOL	Syracuse Univ.	1976	None			*		*	*		*		*		*	
TRNSYS	Univ. of Wisconsin	1974	$200	*	*	*		*	*	*	*		*		*	*
UWENSOL	Univ. of Washington	1978	$200	*	*	*			*			*		*		
WATSUN	Univ. of Waterloo	1978	$170	*	*	*	*	*		*	*		*		*	*

Table 43. Passive System Capability Chart [73]

Program	Direct Gain	Trombe Wall	Water Wall	Roof Ponds	Thermic Diode[1]	Attached Sun Room	PCES Elements[2]	Window Management[3]	Heat Pipe	Natural Vent	Evap. Cooling	Thermo Syphon
BLAST	*											
DOE-1	*											
DEROB	*	*	*	*	*	*	*	*	*	*	*	*
FREHEAT	*	*	*	*		*			*	*		
PASOLE	*	*	*	*		*		*		*	*	
SOLHEAT	*											
SOLPAS	*	*	*	*		*		*		*	*	
SUN	*	*	*	*		*						
UWENSCL	*	*	*	*		*		*		*		

1. Thermic diode - This is a concept being developed by S. Buckley at M.I.T. Such a device allows heat flow in one direction but not in the reverse direction

2. PCES Elements - Phase change material which is encapsulated in suitable building materials

3. Window management - This applies to methods that increase insulation over windows by various processes. This includes beadwalls, movable insulation drapes. etc.

Table 44. Characteristics of Hand Methods [73]

No.	Author(s)	Description	Application Active				Application Passive		Collector Fluid		Life Cycle Economics		Typical Calculation Interval			With data supplied by	
			SH	DHW	COM	SC	TW	WW	LIQ	AIR	PRIM	SEC	DAY	MO	YR	USER	METH
1	S.A. Klein	F-Chart	*	*	*				*	*		*		*		*	
2	Balcomb and Hedstrom	Solar load ratio	*		*				*	*				*			*
3	Barley and Winn	Relative areas	*	*	*				*	*	*				*		*
4	G.F. Lameiro	GFL			*				*	*					*		*
5	S.A. Klein	$\bar{\phi}$-curves	*	*	*				*					*		*	
6	Klein and Beckman	$\bar{\phi}$-F-Chart	*	*	*	*			*					*		*	
7	USEC	Building code	*	*	*				*					*			*
8	P. Lunde	Performance curves	*	*	*				*					*	*	*	
9	Liu and Jordan	Utilizability factors	*	*	*				*	*				*		*	
10	D. Watsun	Appendix to book	*	*	*				*	*		*		*			*
11	Balcomb and McFarland	Passive	*	*			*	*						*	*		*
12	J.C. Ward	Minimum cost sizing	*	*	*				*	*	*				*	*	
13	Bell and Gossett	Design manual	*	*	*				*	*			*			*	*
14	D.S. Ward	Realistic sizing	*	*	*	*			*	*		*		*		*	
15	D. Hittle et al	CERL	*	*	*	*			*	*		*		*	*		*
16	Swanson and Boehm		*	*	*	*			*	*			*				
17	Kreider and Lameiro	G-Chart (tm)	*	*	*				*	*							
18	Kohler and Sullivan	TEANET	*				*	*					*			*	*
19	Haslett and Monaghan		*	*	*				*	*				*	*	*	

Table 45. SHAC Manual Design Methods Summary [74]

The following table describes solar heating and cooling manual design methods. This table does not give all of the design methods applicable to SHAC analysis, but it does contain the most currently used and best known methods. These methods do not require access to a computer although some (e.g., F-Chart) have been implemented on computers. They vary in degree of sophistication from the simple, almost rule-of-thumb type to methods requiring programmable calculators. Some of the latter type methods are available from the source indicated as prerecorded programs on magnetic cards.

Description	Author	Availability			Application				Col. Type		System Type		Tools Required				Basis of Method			Output		
		Cost ($)	Date	Reference/Source	Space Heat	Domestic Hot Water	Space Cooling	Economic Analysis	Liquid	Air	Active	Passive	Graphs/ Tables	4 Function Calculator	Scientific Calculator	Programmable Calculator	Detailed Simulation	F-Chart Correlation	Other Correlation	Solar Fraction	Optimum Area	Other
A Simplified Method for Calculating Solar Collector Array Size for Space Heating	J.D. Balcomb and J.C. Hedstrom		1976	Sharing the Sun: Solar Technology in the Seventies. Vol. 4, American Section, International Solar Energy Society, 1976, pp. 281-284.	•	•			•	•	•		•	•					•		•	
Passive Solar Design Handbook	J.D. Balcomb and Bruce Anderson		Early 1980	Will be available from NTIS, 5285 Port Royal Road Springfield, VA 22161	•							•	•	•					•	•		
Optimal Sizing of Solar Collectors by the Method of Relative Areas	C.D. Barley and C.B. Winn		1978	Solar Energy, Vol. 21, No. 4, 1978, pp. 279-289.	•	•		•	•	•	•		•		•			•			•	
MESH	Dr. John Clark		1978	Dr. John Clark Central Solar Energy Research Corp 1200 6th Street, Room 328 Detroit, MI 48226	•			•	•	•	•				•				•	•	•	
Predicting the Performance of Solar Energy Systems	U.S. Army Construction Engineering Research Lab		1977	Rept. No. AD-A035 608/9 ST (NTIS)	•	•	•	•	•	•	•		•	•					•	•	•	
Copper Brass Bronze Design Handbook—Solar Energy Systems	Copper Development Association	3	1978	Copper Development Association, Inc. 1011 High Ridge Road Stamford, CN 06905	•	•			•	•	•		•	•					•	•		
PEGFIX and PEGFLOAT	W. Glennie	75 both	1978	Princeton Energy Group 729 Alexander Road Princeton, NJ 08540	•							•				•	•					•
Solarcon Programs ST355 and ST365	R.W. Graeff	239 both 142 each 15 weather data	1977	Solarcon, Inc. 607 Church Street Ann Arbor, MI 48104	•	•		•	•	•	•					•		•		•		
Solarcon Program ST33	R.W. Graeff	138	1979	Solarcon, Inc.	•							•				•	•			•		
Solar Heating Systems Design Manual	ITT Corporation Fluid Handling Division	2.50	1977	Bulletin TESE-576, Rev. 1 ITT Training & Education Dept. Fluid Handling Division Morton Grove, IL 60053	•	•			•	•	•		•	•			•			•		
A General Design Method for Closed-Loop Solar Energy Systems	S.A. Klein and W.A. Beckman		1977	Proceedings of the 1977 Annual Meeting, Vol. 1 American Section, International Solar Energy Society, 1977, pp. 8.1-8.5.			•		•		•		•		•			•		•		
Solar Heating Design by the F-Chart Method	S.A. Klein, W.A. Beckman, and J.A. Duffie	10	1977	John Wiley and Sons, New York, N.Y., 1977 (Publisher)	•	•		•	•	•	•		•	•				•		•		
A Design Procedure for Solar Heating Systems	S.A. Klein, W.A. Beckman, and J.A. Duffie		1976	Solar Energy, Vol. 18, No. 2, 1976, pp. 113-127.	•	•			•				•		•				•	•		
TEANET	J.T. Kohler and P.W. Sullivan	95	1978	Total Environmental Action, Inc. Church Hill Harrisville, NH 04350	•							•				•	•					•

Table 45 (continued)

The GFL Method for Sizing Solar Energy Space and Water Heating Systems	G.F. Lameiro and P. Bendt		1978	Rept. No. SERI-30 Solar Energy Research Institute 1617 Cole Boulevard Golden, CO 80401	•				•	•	•		•		•			•		•		
A Design Handbook for Direct Heat Transfer Passive Solar Systems	R.M. Lebens	10	1978	Northeast Solar Energy Association P.O. Box 541, 22 High Street Brattleboro, VT 05301	•			•				•	•			•	•			•		
A Rational Procedure for Predicting the Long-Term Average Performance of Flat-Plate Solar Energy Collectors	B.Y.H. Liu and R.C. Jordan		1963	Solar Energy, Vol. 7, No. 2, 1963, pp. 53-70.	•	•			•	•	•		•		•		•			•		
Pacific Regional Solar Heating Handbook	Los Alamos Scientific Lab		1976	Rept. No. TID-27630 (NTIS)	•	•			•	•	•		•		•				•	•		
Prediction of the Monthly and Annual Performance of Solar Heating Systems	P.J. Lunde		1977	Solar Energy, Vol. 20, No. 3, 1977, pp. 283-287	•	•			•		•		•	•					•	•		
SCOTCH Program	R. McClintock	195 Thermal alone, 175; econ. anal. alone, 75	1977	SCOTCH Programs P.O. Box 430734 Miami, FL 33143	•	•		•	•	•	•					•		•		•	•	
Solar Heating of Buildings and Domestic Hot Water	Naval Facilities Engineering Com. E.J. Beck, Jr. and R.L. Field		1976	Rept. No. AD-A028 862/8ST (NTIS)	•	•		•	•	•	•		•		•			•		•	•	
PCTS	J. Schoenfelder	F-Chart Therm. 35; F-Chart Econ. 35	1978	Central States Research Corp. P.O. Box 2623 Iowa City, IA 52240	•	•		•	•	•	•					•		•		•	•	
Domestic Hot Water Manual Using Sunearth Solar Collector Systems	Sunearth Corp.	3	1976	Sunearth Solar Products Corp. Technical Services R.D. 1 P.O. Box 337 Green Lane, PA 18054		•			•		•		•					•		•		
An Averaging Technique for Predicting the Performance of a Solar Energy Collector System	G.H. Stickford		1976	Sharing the Sun, Vol. 4, 1976, pp. 295-315.	•	•			•	•	•	•	•		•		•			•		
Calculation of Long-Term Solar Collector Heating System Performance	S.R. Swenson and R.F. Boehm		1977	Solar Energy, Vol 19, No. 2, 1977, pp. 129-138.	•				•	•	•		•		•			•		•		
Minimum Cost Sizing of Solar Heating Systems	J.C. Ward		1976	Sharing the Sun, Vol. 4, 1976, pp. 336-348	•			•	•	•			•		•			•		•	•	
Designing and Building a Solar House; Your Place in the Sun	D. Watson	9	1977	Garden Way Publishing Charlotte, VT 05445	•	•		•	•	•	•		•	•			•			•		
SEEC I — Heat Load, Monthly Solar Fraction, Economics	C.B. Winn	125	1976	Solar Environmental Engineering Co., Inc 2524 East Vine Drive Fort Collins, CO 80522	•	•		•	•	•	•					•		•		•		
SEEC II — Collector Optimization, Annual Solar Fraction, Economics	C.B. Winn, D. Barley, G. Johnson, J. Leflar	95	1978	Solar Environmental Engineering Co., Inc	•	•	•	•	•	•	•					•		•			•	
SEEC III — SEEC II Plus Insulation Optimization	C.B. Winn, D. Barley, G. Johnson, J. Leflar	125	1978	Solar Environmental Engineering Co., Inc.	•	•		•	•	•	•					•		•			•	
SEEC VI — Passive Solar Heating	C.B. Winn, D. Barley, G. Johnson, J. Leflar	125	1978	Solar Environmental Engineering Co., Inc.	•			•				•				•			•	•		
Sunshine Power Programs for Modeling Solar Energy Components and Systems	G. Shramek	30-60	1977	Sunshine Power Co 1018 Lancer Drive San Jose, CA 95129	•	•		•	•	•	•	•				•	•	•	•	•		•
Mazria Design Patterns (Rule-of-Thumb) in The Passive Solar Energy Book	Edward Mazria	11	1979	Rodale Press, Emmaus, PA 18049	•				•	•		•	•				•			•	•	•

Table 46. SHAC Computer Methods Smmary [74]

The following summary table notes the most frequently used and currently available solar analysis computer methods. The information was obtained largely from a program author survey conducted by Arthur D. Little, Inc. for the Electric Power Research Institute and primarily reflects the opinions of each program author.

Most summary table categories are self-explanatory, however, the intended user category needs emphasis. Programs suitable for use by builders were limited to the interactive type program that interrogates the user by a question and answer methodology. Architects/engineers use mainly design-oriented computer programs and generally restrict their analysis to standard input/output options of the program. The research engineer generally has hands-on access to the program and is very familiar with both the operation and assumptions of the program and the details of the system being analyzed.

Program Name	Latest Version	Availability				Application							Intended Users			Computation Interval		Computer Versions Available	Economic Analysis	Sponsor
		Purchase ($)	Time Share	Special Arrangements	Comments	User Manual	Service Hot Water	Space Heating	Space Cooling	Process Heat	Active System	Passive System	Research Engineers	Architect/Engineers	Builders	Hour	Month			
ACCESS*	1978	10,000		●	No cost to EEI members	●	●	●	●	●	●		●	●		●		IBM	●	Edison Electric Institute (EEI)
BLAST*	1978	300	●		Training available	●	●	●	●		●		●	●		●		CDC	●	USAF, USA, GSA
DEROB	1979	Nom.				●		●	●			●	●	●		●		CDC		NSF, ERDA, DOE
DOE-2*	1979	400	●			●	●	●	●		●	●	●	●		●		CDC	●	LASL, DOE
EMPSS	1978	500		●	Consulting with ADL	●	●	●	●		●		●	●		●		IBM	●	EPRI
F-CHART	1978	100	●		Training available	●	●	●			●			●	●		●	CDC, IBM UNIVAC	●	DOE
FREHEAT	1979	150			Limited documentation			●				●	●			●		CDC	●	DOE
HISPER	1978	Avail. on request			Limited documentation		●	●	●		●		●			●		UNIVAC		NASA, MSFC
HUD-RSVP/2	1979	175	●		Based on F-CHART	●	●	●			●	Δ		●	●		●	CDC UNIVAC	●	HUD
SHASP	1978	Avail. on request				●	●	●	●		●		●			●		UNIVAC	●	DOE
SIMSHAC	1973	300					●	●			●		●			●		CDC	●	NSF
SOLAR-5	1979		●					●	●			●		●	●	●	●	CDC		UCLA, DOE
SOLCOST	1979	300	●			●	●	●			●			●	●		●	CDC, IBM UNIVAC	●	DOE
SOLOPT	1978	20				●	●	●			●		●				●	AMDAHL	●	Texas A&M Univ.
SOLTES	1979				Available Argonne Fall 1979	●				●	●		●			●		CDC		Sandia
SUNCAT	1979	Nom.			Limited documentation	●		●				●	●	●		●		Data General Eclipse	●	NCAT
SUNSYM[1]	1979		●	●		●	●	●	●		●			●		●		IBM	●	Sunworks Comp. Systems
SYRSOL	1978	Nom.			Avail. but not actively marketed		●	●	●			●	●	●	●	●		IBM	●	ERDA, NSF, DOE
TRACE SOLAR*	1979		●	●		●	●	●	●	●	●		●	●		●	●	IBM	●	The Trane Co.
TRNSYS	1979	200	●		Training required	●	●	●	●		●	Δ	●			●		CDC, IBM UNIVAC		DOE
TWO ZONE	1977			●		●		●	●			●	●			●		CDC	●	LBL
UWENSOL	1978	200				●		●				●	●			●		CDC		State of Wash.
WATSUN II	1978	200				●	●	●			●		●			●		IBM	●	Nat'l Research Center of Can.

*Programs are primarily developed for large-scale, multi-zone applications

Δ Being added

Table 47. HVAC Computer Programs Summary [74]

The following table lists programs intended primarily for building heating and cooling load analysis. Some provide for solar analysis, but in such cases it is secondary to the conventional energy analysis. The programs have been generally developed and maintained by heating, ventilating, and air conditioning (HVAC) consulting engineers for their own analysis use, however, most are available through special arrangements with the contact.

Program Name	Sponsor	Author	Contact	Original Release	Current Version	Application: Building Load	Application: HVAC System	Application: Active Solar	Application: Passive Solar	Availability: Source Cost ($)	Availability: Time Share	Availability: Special Arrangements
ECUBE III	American Gas Association (AGA)	Subcontractors	American Gas Association 1515 Wilson Blvd. Arlington, VA 22209 (203) 841-8400 David G. Wood		1979	●	●		●	No info	●	●
ENERGY 1	Gibson-Yackey-Trindade Assoc.	Robert Gibson	Gibson-Yackey-Trindade Associates 311 Fulton Ave. Sacramento, CA 95811 (916) 483-4369 Robert Gibson	1974	1974	●	●	●		Not avail.	N/A	●
EP	Energy Management Services (EMS)	EMS	Energy Management Services 0435 SW Iowa Portland, OR 97201 (503) 244-3613 Robert M. Helm	1976	1978	●	●	●	●	Not avail.	●	●
ESAS	Ross F. Meriwether & Associates, Inc.	Ross F. Meriwether	Ross F. Meriwether & Associates 1600 NE Loop 410 San Antonio, TX 78209 (512) 824-5302	1969	1978	●	●	●	●	Not avail.	N/A	●
ESP-1	Automatic Procedures for Engineering Consultants (APEC)	Stone and Webster	APEC, Inc. Executive Off. Grant-Doneau Tower Suite M-15 40th & Ludlow Streets Dayton, OH 45402 (513) 228-2602 Doris Wallace	1977	1978	●	●	●		6,000 no cost to APEC members	N/A	●
HACE	William Tao & Associates (WTA)	WTA/Computer Services, Inc.	WTA/Computer Services, Inc. 2357 59th Street St. Louis, MO 63110 (314) 644-1400 Richard Lampe	1970	1978	●	●	●	●	Not avail.	●	●
NECAP	NASA	NASA	NASA, Langley Research Center Mail Stop 453 Hampton, VA 23665 (804) 827-4641 Ron Jensen	1974	1975	●	●	●			N/A	N/A
SCOUT	Guard, Inc.	Guard, Inc.	Guard, Inc. 7440 N. Natchez Ave. Niles, IL 60648 (312) 647-9000 Robert Henninger	1976	1978	●	●		●	See contact	N/A	●
SEE	The Singer Co. (NSF Grant)	W.S. Fleming & Assoc., Inc., The Singer Co.	The Singer Co. Climate Control Division 62 Columbus St. Auburn, NY 13201 (315) 253-2771, X391 Philip Parkman	1975	1977	●	●	●	●	Not avail.	N/A	●
Westinghouse Programs	Westinghouse Electric Corp.	Westinghouse Electric Corp.	Westinghouse Electric Corp. Energy Systems Analysis 2040 Ardmore Blvd. Pittsburgh, PA 15221 (412) 256-3168	1964	1978	●	●	●	●	Not avail.	N/A	●

N/A = Not applicable

Nevertheless, comparisons of actual performance of operating solar systems with the various design methods can provide insights into:

1. The degree to which actual system performance meets the expectations of the design methods,
2. Potential improvements in performance with variations in design and operation,
3. Effects on performance due to changes in installation procedures, and
4. Limitations on performance of specific solar system designs.

4.3 OVERALL EVALUATION OF PERFORMANCE

In evaluating the performance of various solar heating and cooling systems, it must be emphasized that many of the installations which had sufficient data acquisition and information available were research and development or experimental projects. Many of the systems utilized unique and innovative design concepts and therefore do not accurately represent the performance of commercially available state-of-the-art systems. In addition, these "experimental" systems had the majority of system deficiencies and/or problems.

On the other hand, SYSTEMS THAT FOLLOWED PROPER DESIGN PRACTICES, USED PROVEN DESIGNS, AND WERE PROPERLY CONTROLLED PERFORMED WELL.

It should also be noted that systems that work well do not receive the same amount of discussion. Problems are discussed in order to avoid future difficulties. Correct procedures are reported only as alternatives to faulty procedures.

Because of the wide variations in performance of many systems, <u>average performance</u> values of numerous systems have <u>limited usefulness</u>. For example, it can be concluded that, on average, solar heating and cooling systems have <u>not</u> performed up to expectations [75]. On the other hand, numerous systems have performed well and within expected design limits.

A substantial portion of the problems associated with operating solar systems with low performance levels are problems that are directly related to conventional HVAC (Heating, Ventilating and Air Conditioning) practices. These include:

- o Inadequate or nonexistent specifications
- o Lack of application of good engineering practice
- o Failure to adhere to good HVAC practices
- o Improper tools, methods (e.g., short cuts) used to little or no advantage
- o Unacceptable cost savings attempts
- o Work attempted too quickly and/or with insufficient planning
- o Poor choice of materials due to lack of detailed design
- o Improper or nonexistent maintenance, and
- o Lack of availability of maintenance or operating manuals

The objective of this handbook, however, is not to consider those problems associated with conventional heating and cooling systems design, installation, and operation. Such information is readily available in other ASHRAE publications and from other sources. Rather, it is the problems specific to solar hardware, design, and installation that are addressed here.

The solar-related problems that have reduced the performance of solar heating and cooling systems include:

- o Improper design methods and practices
- o Improper selection and integration of system components (specifically the poor matching of components with load and with other components within the system)
- o Inappropriate or unacceptable components which contain serious design flaws
- o Design and installation of "experimental" systems without adequate control and/or instrumentation to ensure proper operation
- o Improper installation procedures and/or methods
- o Poor selection of operating modes
- o Insufficient analysis of system hydraulics
- o Inappropriate control strategy as related to solar

Incorporated within the problems listed above are several major factors which have resulted in reduced performance including:

1. Excessive thermal losses
2. Unacceptable electrical energy requirements for solar system operation
3. Poor choice of controls (equipment and methodology), and
4. Lack of adherence to architectural constraints.

4.4 PROBLEMS IN DESIGN AND SIZING

Design-related problems, based on experiences, include:

1. Rule of thumb sizing methods provide useful estimates of collector and component sizing but are generally inadequate [2]. The primary difficulty appears to stem from the fact that the rules of thumb are only applicable to a very limited number of designs and applications of solar systems. In addition some rules of thumb were based on inadequate experimental and data information bases.
2. Space heating load estimates are subject to uncertainties of 15 to 25 percent (and as high as twice the actual load). One source of uncertainty is the use of the degree day method in which the empirical correction factor is subject to som error [2]
3. Incorrect estimates of domestic hot water (DHW) loads are common. These uncertainties occur when the design loads are based on a rough average, such as 15 to 20 gallons (57 to 76 liters) per person per day [2].
4. Sizing methods are commonly misapplied due to the mismatching of load profiles. For example a sizing method for DHW for residential use is not applicable for sizing a commercial application because of the substantial difference of the DHW load demand profile over 24 hours.
5. Errors in design calculations arise from uncertain data, especially weather and solar radiation data [2]. Direct measurements of these data may not be availabl for the site in question, so that data may have to be taken from a nearby or similar site. Differences in local topography can result in a sizable differenc in the weather data and differences in elevation, cloud cover (including time of day of occurrence), atmospheric haze, etc., can give rise to differences of 25 to 30 percent in the estimate of incident solar energy [2].
6. Collector efficiency and system efficiency based on collector and component parameters may be in error due to:
   a. Errors in component test data
   b. Use of net instead of gross collector area (collector characteristics are usually based on gross area)
   c. Errors in estimated flow rates and/or temperatures, which in turn may reduce collector and system performance
   d. Errors incurred when collector modules are used in series configurations without appropriate corrections being made (for a given flow rate, series configurations reduce performance)
   e. Errors due to use of instantaneous or steady-state parameters that are not corrected for daily usage.
7. Incorrect use of such common methods as F-Chart and other similar sizing methods The F-Chart method is sometimes being used outside its range of validity [2]. The underlying assumptions of sizing methods should be checked before using the method for a particular installation.
8. Sizing of heat exchangers, pipes, ducts, fans, and pumps is generally not addressed in solar design methods. Conventional procedures should nevertheless be followed closely.
9. There has been a general lack of consideration of the solar system operating electrical energy requirements. Such lack of consideration of electrical energy consumption of pumps, fans, controls, etc. has led to system designs which are net energy losers, i.e., the electrical solar-operating energy usage (in terms of fossil fuel consumption) has exceeded the thermal solar energy gained.
10. Insulation for pipes, ducts, components, etc. has generally been inadequate, installed improperly, and, in some cases, nonexistent. Non-insulation of pipes, for example, can sometimes reduce the amount of solar energy delivered to the load by half [14]. Note, however, that in general smaller pipe sizes (approximately 1/2 inch, 1.27 cm) with short runs (less than two feet, 0.61 m) should not be insulated.
11. Selection of collector type (liquid or air heating, flat-plate, evacuated tube, or concentrating/tracking) has been done without apparent consideration of costs efficiency characteristics, overall system integration, and end use. Operating parameters (such as temperatures, flow rates, solar radiation availability, diffuse versus beam radiation, efficiencies, etc.) should be considered with the selected collector type AND with the solar system requirements. It is important to note that flat-plate, evacuated tube, and concentrating collectors have distinct performance advantages AND disadvantages in different design applications (see Section 4.5.2).
12. Many solar design methods for systems with heat pumps and/or absorption chillers and hybrid passive/active components tend to be inadequate and difficult to

utilize [2]. It is essential that complete design methods which consider all factors be used in order to obtain realistic projections on future system performance.

13. Difficulties in collector flow distribution, proper filling of liquid-heating collector arrays, and air leakage in air-heating collector arrays have been encountered in many systems [76]. This is a critical area (see Section 4.5.3.2).
14. System designs have apparently been conducted backwards in many cases. Rather than selecting a collector and then designing a system which will "fit" a particular building or application (as has been done in many cases), it is essential to consider the building/application requirements first, then select an appropriate system type and, finally, select components.

4.5 MAJOR FACTORS IN REDUCED PERFORMANCE

4.5.1 Selection and Integration of Components

Numerous systems have been designed without proper matching of the system to the load requirements. In addition, specific components (such as collector types and thermal storage units) have been selected without due consideration of load requirements. Inevitably the design has violated the KISS principle (Keep It Simple, Stupid).

For example, selection of DHW system designs have not always been based on the specific requirements of a particular site's climate and heating requirements. Different design types of DHW systems include:

1. Direct[1] heating, combination collector/storage (non-pumped, passive)
2. Direct heating, thermosyphon (non-pumped)
3. Direct heating, pumped
4. Indirect[2] heating, non-pumped
5. Indirect heating, non-freezing, liquid, pumped
6. Indirect heating, air, pumped (and/or blown)

The direct heating, combination collector/storage unit is potentially the simplest of the designs and is in keeping with the design principle, that the simplest way to obtain hot water is to heat the water directly. Disadvantages of the system are the potential for freezing and the architectural and structural constraints on installation of this type of system into existing (and in some cases) new buildings.

The direct heating thermosyphon has been shown to achieve the highest performance of system types 2, 3, 5 and 6 [49]. However, constraints on the thermosyphon include potential for freezing and the requirements for placing the hot water storage at a higher elevation than the collector. It follows that thermosyphon systems are more suited for particular climates than others. Indirect air-heating DHW systems can be considered competitive in non-freezing climates, but may have an even greater potential in freezing situations. The use of air-heating DHW systems in non-freezing regions may nevertheless be competitive, based on tentative results of DHW systems for two high schools in New Mexico [77].

Use of attached greenhouses to provide additional space heating for a building cannot always be considered cost-effective unless aesthetic or other advantages can be gained by the addition of the greenhouse. In addition, greenhouses without night insulation have limited ability to provide energy to other spaces.

Installation or incorporation of passive heating features without:

1. Substantial thermal capacity of walls, floors, ceilings, and/or other storage form or
2. Acceptability of larger interior temperature variations

cannot be recommended. Buildings with allowable interior temperature fluctuations (e.g., warehouses, etc.) and/or with large thermal mass constituents (e.g., concrete floors, walls, etc.) can utilize passive designs to best advantage.

Because the intent of an active <u>space</u> heating system is to heat space, i.e., air, water-heating systems might be considered less than appropriate for active residential space heating systems. This is based on the fact that durability and reliability factors of freezing, boiling and corrosion can be expected to potentially reduce the long-term effectiveness of water-heating active space heating systems.

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[1]Direct heating systems have no heat exchangers

[2]Indirect implies the use of a heat exchanger between the collector fluid and the storage fluid.

For example, spring, fall, and summer boiling problems of water (or water/glycol mixtures, etc.) may be a serious problem, particularly with active combined space and DHW heating systems [4]. In some cases solar collectors have been damaged by thermal shock when the collectors were filled with liquid on sunny days (when the absorber plate temperatures were in the range of 400 to as high as 800°F, 204°C to 427°C). Thus boiling and thermal shock problems can be associated with both drain-down, drain-back, continuously filled, and indirect liquid heating systems. While these design problems are not specifically liquid system problems, they are important considerations. Alternatively, excessive damper and duct leakage may effectively nullify any potential gains for an air-heating system.

Poor integration of components within a system has also been observed. Use of hot water storage with air-heating collectors has the disadvantage of severely reducing performance by eliminating the stratification of temperature, easily obtainable in pebble-bed (rock) storage units.

The most frequent example of poor integration of components to particular applications is the use of "high performance collectors" such as tracking concentrators (operating temperatures of 200 to 500°F, 93 to 260°C) for lithium bromide absorption cooling. Lithium bromide absorption chillers require solar operating temperatures of only 110 to 220°F (43 to 104°C) and typically 160°F (70°C) with proper chiller controls! In addition, ambient temperatures associated with cooling requirements for residences are typically 80 to 100°F (25 to 35°C). Thus the difference between collector operating and ambient temperatures is equivalent to space heating requirements (i.e., 70 to 80°F, 40 to 45°C). Flat-plate collectors capable of meeting space heating loads can therefore achieve greater energy collection in space cooling applications (with proper tilt of the collector array), because of the greater solar radiation intensity during the cooling season [78]. The use of concentrating collectors for space cooling is therefore not required and, in fact, may only be a better choice for space cooling equipment requiring substantially higher input temperature requirements (e.g., ammonia-water absorption chillers, Rankine cycle cooling, etc.). Such systems, of course, sacrifice most of the diffuse radiation.

An "experimental" system as used in this handbook is a system design which does not have a proven record of performance. Such a proven record requires that the system shall have been operational over a reasonable period of time and the system design's detailed performance measured by extensive instrumentation. Innovative, unique, and clever system designs may have a high failure probability [78]. The quality of "improved" components/designs is always a significant factor in the ultimate performance of a system.

4.5.2 Collector Array Performance

4.5.2.1 Collector Efficiency

The Hottel-Whillier-Bliss (HWB) model has been the standard tool for determining collector efficiency for four decades, and has been shown to closely represent the steady-state performance of some collector arrays under uncontrolled field conditions [79]. The HWB model may be represented in equation form [80,81] by:

$$\eta_c = F_R(\tau\alpha) - F_R U_L[(T_i - T_a)/I_I]$$

where:

- F_R is the solar collector heat removal factor, dimensionless
- $(\tau\alpha)$ is the effective product of the cover transmittance and the absorber plate absorptance (taking into account the internal and multiple reflections), dimensionless
- U_L is the solar collector overall heat loss coefficient, Btu/hr·ft^2·°F (W/m^2·°C)
- T_i is the inlet fluid temperature to the solar collector array, °F (°C)
- T_a is the ambient (outdoor) temperature, °F (°C)
- I_I is the instantaneous solar radiation intensity on the plane of the collector, Btu/hr·ft^2 (W/m^2)

The usual method of collector evaluation is to test under steady-state, controlled conditions (e.g., ASHRAE 93-77) and measure the energy gains from the collector by use of the equation:

$$Q_c = \dot{m}C_p\ (T_o - T_i)$$

where:

$\dot{m}$ is the transport fluid mass flow rate, lb/hr (kg/s)

C_p is the transport fluid specific heat at constant pressure, Btu/lb•°F (kJ/kg•°C)

T_o is the outlet fluid temperature from the solar collector array, °F (°C)

The collector efficiency, η_c, is then given as the actual energy collected, Q_c, divided by the product of the incident solar energy, I_T, and the gross area of the collector, A (the net area of the collector's absorbing surface (any definition of "net" area) should not be used!). Experimentally, an efficiency curve of the type depicted in Figure 5 is obtained from which applicable values of $F_R(\tau\alpha)$ and $F_R U_L$ are readily calculated.

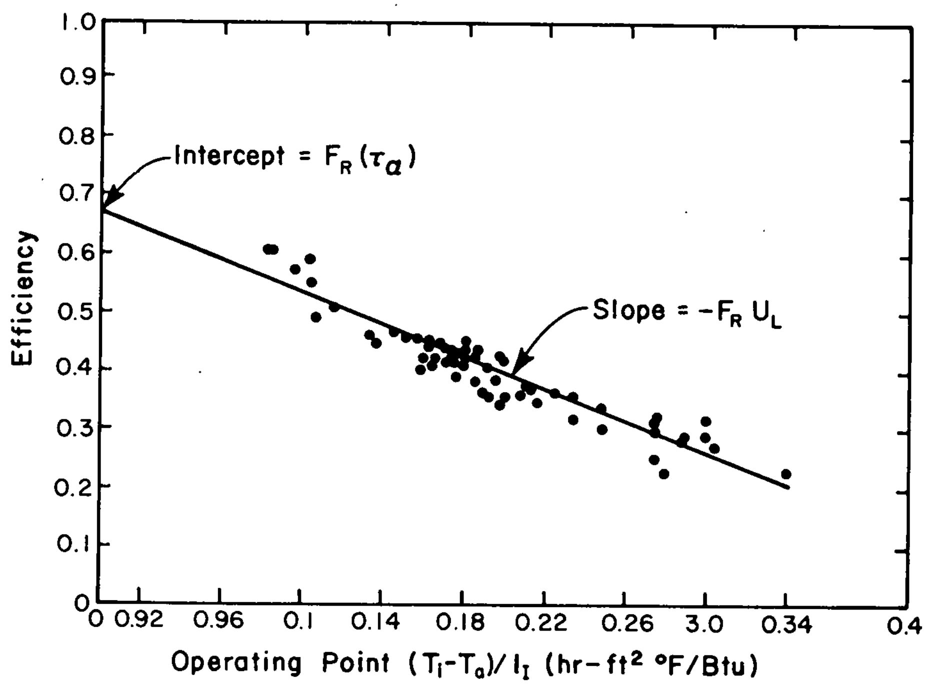


Figure 5. Typical Steady-State Collector Efficiency Curve

A curve derived in this manner is typically provided by solar collector manufacturers and subsequently used by the system designer to predict collector array performance. However collector arrays in the field are sometimes composed of multiple solar panels in series configurations (primarily residential applications; commercial installations are typically in array banks) and, in addition, seldom operate in a steady-state condition, i.e., they are exposed to a variety of dynamic factors such as clouds, wind, diurnal variations in sunlight and shade, etc.

Nevertheless, the instantaneous efficiency curve derived from experimental tests in many cases compares adequately to a curve drawn through carefully selected "quasi-steady-state" points obtained from operating systems [79]. Thus the experimental curve may often be reliably used in design methods provided that corrections are made for the use of the collector in an array rather than as a stand alone panel.

It must also be noted that collector arrays include headers and piping which are not included in the experimental single collector module tests. Losses from these sources are a subtraction against array performance despite the fact that good manifold design and insulation can minimize the effects and can be accounted for in the expectations of collector performance. (In a system design the losses in headers/piping associated with arrays must be separately calculated and subtracted from the array.) The fact that these heat losses must be accounted for does not constitute a problem, as treating pipe losses separately is a standard HVAC practice. It is noteworthy, however, that a particular building may not be appropriate for solar if the piping run is too long.

Of potentially greater importance is the level of agreement between the steady-state derived curve and the actual performance of collector arrays under dynamic conditions.

4.5.2.2 Comparisons of Actual and Predicted Collector Performance

Comparisons of actual and predicted collector energy gaines (i.e., values of Q_c) are shown in Table 48 [81]. In evaluating this table it is worthwhile to consider McCumber's comments.

Table 48. Comparison of Actual and Predicted Energy Gains [81]

System	Predicted Monthly Q_c (1)	Actual Monthly Q_c (1)	Percent Deviation from Predicted (%)
A-frame	1.56	0.83	-46.5
Alpha Construction	6.43	4.69	-27.1
Aratex	168.97	121.95	-27.8
Facilities Development	15.17	10.46	-31.1
Florida Gas	9.68	9.23	- 4.6
Hogate's Restaurant	95.52	86.08	- 9.9
Reedy Creek	54.71	25.08	-54.2

(1) Million Btu/month (multiply by 1.055 to obtain GJ/month)

The A-frame solar energy installation is designed to supply the total domestic hot water requirements for a family of five, but the load is in reality provided by only two people. The effect of this diminished load is to cause a large percentage of the operating points to be located to the right on the efficiency curve. In addition, the A-frame site is located in a windy area (Hawaii) which in turn leads to higher collector heat losses. The percent error in actual energy gain from prediction is consistent from day to day, indicating that the operating conditions did not deviate significantly. The mean daily error in solar collected useful energy is a negative 23,370 Btu/day (24,650 kJ/day). This, however, cannot be considered a typical system.

The Alpha Construction Company solar energy installation is an air-heating system, designed to supply space and DHW heating to a single family dwelling. The mean daily error, i.e., overprediction of solar collected useful heat, Q_c, is 56,199 Btu/day (59,290 kJ/day).

The Aratex solar energy installation is designed to preheat water for a large industrial laundry. The system showed a consistent daily deviation from the predicted value of -27.8 percent.

The Facilities Development solar energy installation provided domestic hot water heating to a 31 unit condominium. The monthly deviation between actual and predicted energy gain was -31.1 percent. The average daily average in Q_c was a reduction of 152,032 Btu/day (160,390 kJ/day).

The Florida Gas sola energy system is designed to provide heating, cooling, and domestic hot water to a 1548 square foot (144 square meter) single family dwelling. Table 48 indicates a monthly error of -4.6 percent, i.e., the actual energy gain, Q_c, was 4.6 percent less than the predicted value for the month of August. Examination of the daily values indicates deviations between +125 percent and -18 percent. The majority of the positive deviations occurred on days when every collection was very low, which implies overcast conditions and a high ratio of diffuse to beam energy. McCumber [81] has concluded from this example that the linear HWB model is appropriate for this application of this particular collector.

The Hogate's Restaurant solar energy installation provides hot water heating for kitchen use. The tabular data given in Table 48 shows the monthl energy gain to be 9.9 percent lower than predicted. The daily errors are not consistent, with some positive errors showing up on days of low energy gain. When energy gain was high, the errors are negative and of approximately the same magnitude. This implies a better conversion of diffuse sunlight than that of beam radiation. The mean daily deviation between actual energy gain and predicted energy gain was a reduction of 304,612 Btu/day (321,365 kJ/day).

The Reedy Creek utilities solar energy installation employs a concentrating collector, consisting of reflective parabolic troughs and linear receivers (absorbers). The receivers are mounted on level arms which rotate to maintain the receiver in the focal region of the parabolas. This installation provides hot water to the generator of an absorption chiller at temperatures in excess of 170°F (77°C). Table 48 shows the monthly energy gain to be 54.2 percent less than predicted. The original design had determined that diffuse radiation was assumed to be concentrated on the receiver -- an erroneous assumption.

"Experience has shown that concentrating collectors, as a class, have fewer steady-state operating conditions than flat-plate collectors. This is thought to be due to the added requirement of maintaining the receiver in focus against all the environmental disturbances. A higher percentage of transient points leads to a greater deviation from the expected energy gain. The daily error percentages are consistent with the monthly error percentage" [81].

> The general flaw in McCumber's analysis is the invalid application of the prediction method. For example, the manifold/header and exterior collector loop piping losses were not accounted for by the IBM (McCumber) analysis. Thus the collector arrays do not perform as predicted, primarily because the losses in headers and piping associated with solar collection are virtually ignored. If these losses are accounted for in each of the installations, then the collector array performance can be reasonable accurate.

McCumber has subsequently quantified the dynamic effects of field operating conditions on the energy gain of a collector array [82]. This was done by deriving instantaneous efficiency curves from field data by techniques described in reference [79] and using the field-derived curve (instead of the single panel laboratory-derived curve) in the energy gain comparison. McCumber then concluded that, in general, dynamic effects result in errors on the order of five percent. This roughly corresponds to more precise estimates of reductions in collector modules in series flow configurations (based on theoretical models) by Oonk, et al [83].

4.5.2.3 Causes of Deviations from Efficiency Curves

Figure 6 shows performance comparisons of several systems. (The "histograms" refer to the percentage of time that the collector array operated under specified conditions of $[(T_i-T_a)/I]$.) Reductions in the efficiency axis intercept (i.e., when $(T_i-T_a/I_I = 0)$ may be caused by reduced values of:

1. F_R, the collector heat removal factor. F_R is a direct function of flow rate and thermal conductivity of the bond between the absorber plate and the fluid transport tubes (ducts) and an inverse function of the heat loss coefficient.
2. τ, cover transmissivity. This may be due to opaque substances on the outer surface of the glazing (dust, debris, etc.) or condensation and/or outgassing residue on the inner surface of the glazing, and
3. α, plate absorpvitiy. This may be low due to the deterioration of the absorber coating.

Reduction in the operating point intercept (i.e., when $\eta_c = 0.0$) is not caused by variations in F_R, but can be due to:

1. Decreases in τ and/or α, and/or
2. Increases in U_L, the collector heat loss coefficient. U_L is strongly affected by wind and ambient temperatures and is also subject to increase as installation and perimeter insulation decreases in effectiveness.
3. Array not totally filled (i.e., some collectors inoperative).

4.5.2.4 Comparison of Array Performance to Single-Panel Prediction

McCumber [81] has analyzed 50 collector installations. Of these, four were in close agreement (5%) with the single panel projection, 12 were substantially better (actually collected more) than the single panel prediction, and 34 were substantially worse (actually collected less) than the single panel prediction. Table 49 summarizes these findings [82].

4.5.2.5 Variations of Instantaneous Efficiency Curve with Time

Figure 7 presents the monthly instantaneous efficiency curves for a single site. Notice that the efficiency axis intercept varies only slightly but the operating point axis intercept varies from 0.6 to 0.72 $hr \cdot ft^2 \cdot °F/Btu$ Variations in the operating point axis intercept

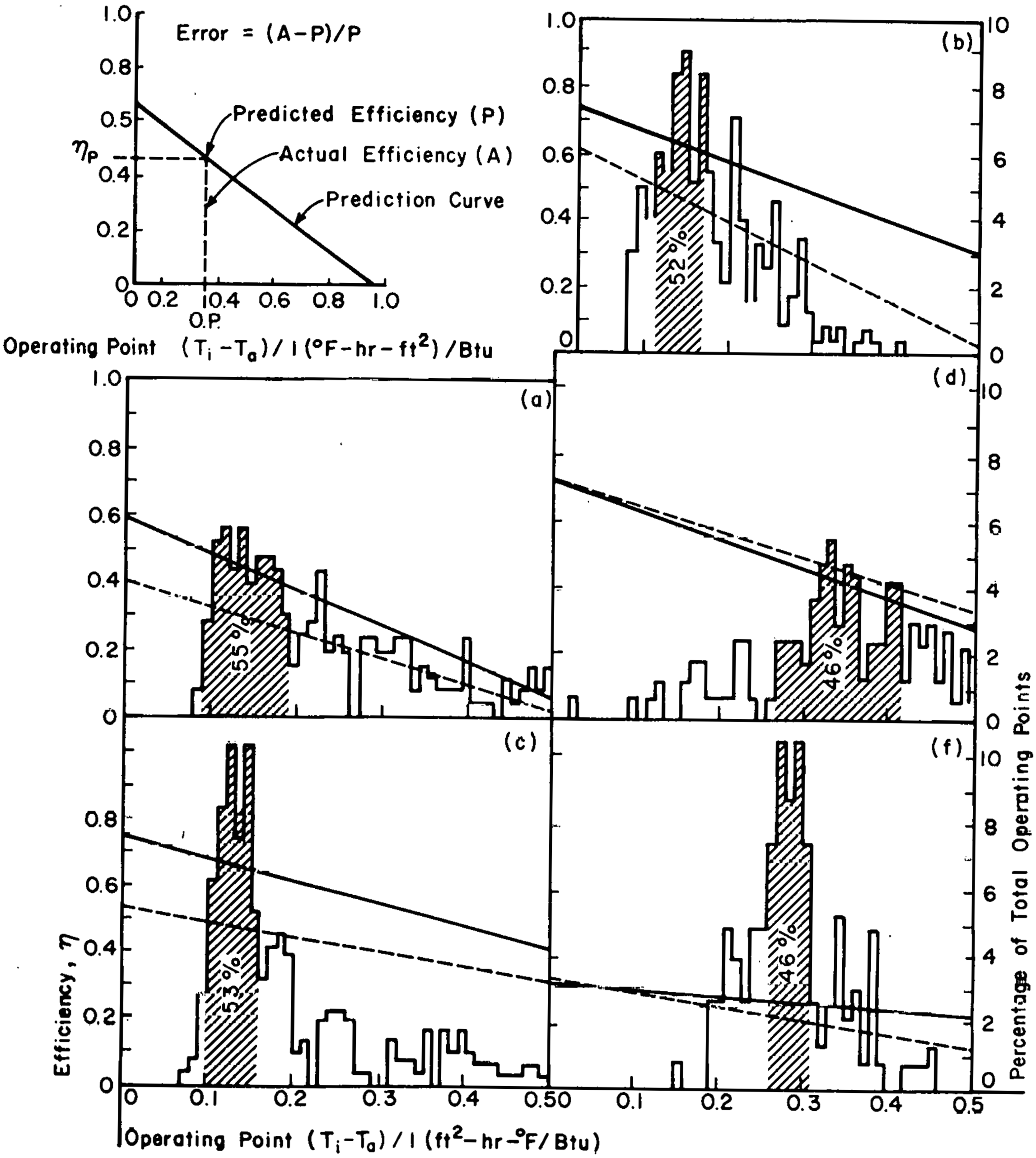


Figure 6. Collector Array Performance Comparison and Operating Point Histrograms (Lab Curve —; Field Curve --)

Table 49. Comparison of Array Performance to Single Panel Prediction [82]

			Within 5%	Better	Worse
Liquid	Single glazed	non-selective	0	6	2
		selective	1	0	3
	Double glazed	non-selective	1	3	7
		selective	0	0	7
	Evacuated tube		1	0	0
	Concentrator		0	0	2
Air	Single glazed	non-selective	0	0	6
		selective	0	0	1
	Double glazed	non-selective	1	3	5
		selective	0	0	0

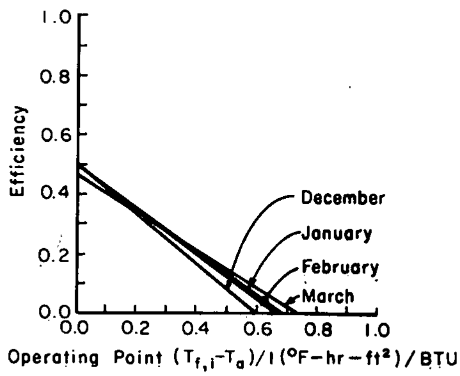


Figure 7. Variations of Efficiency Curves by Month [82]

appear to be due to changes in U_L brought about by changes in the external ambient temperature without corresponding changes in the operating temperature, i.e., T_i.

4.5.2.6 Relative Performance of Collector Types

Figure 8 presents the relative performance of representative collector types. Figure 8(a) for example, is the HWB curve for a single glazed non-selective absorber collector array plotted over a histogram of the collector operating points for an example month. Figures 8(b) through 8(j) depict other generic collector configurations. The notes on the curves are the energy acquired by the collector array normalized to the number of square feet of collector area. The significant aspects of any collector array are: (1) energy acquired, (2) the cost of its acquisition, and (3) the delivery temperature.

The curves in Figure 8 indirectly indicate the delivery temperature, higher delivery temperatures being implied by a concentration of operating points to the right of the graph or low values of solar insolation. Relative comparisons among types must consider the design operating point. Thus a single glazed non-selective absorber collector, which has a high efficiency intercept and low operating point intercept, will not operate efficiently at the inlet temperatures (as reflected by the operating range) required for absorption chiller operation (temperatures in excess of 165°F, 75°C).

4.5.3 Problem-Related Variations in Collector Performance

4.5.3.1 Introduction

Significant reductions in collector array performance have also been due to system problems. While dynamic effects may reduce performance of field installed collectors by five

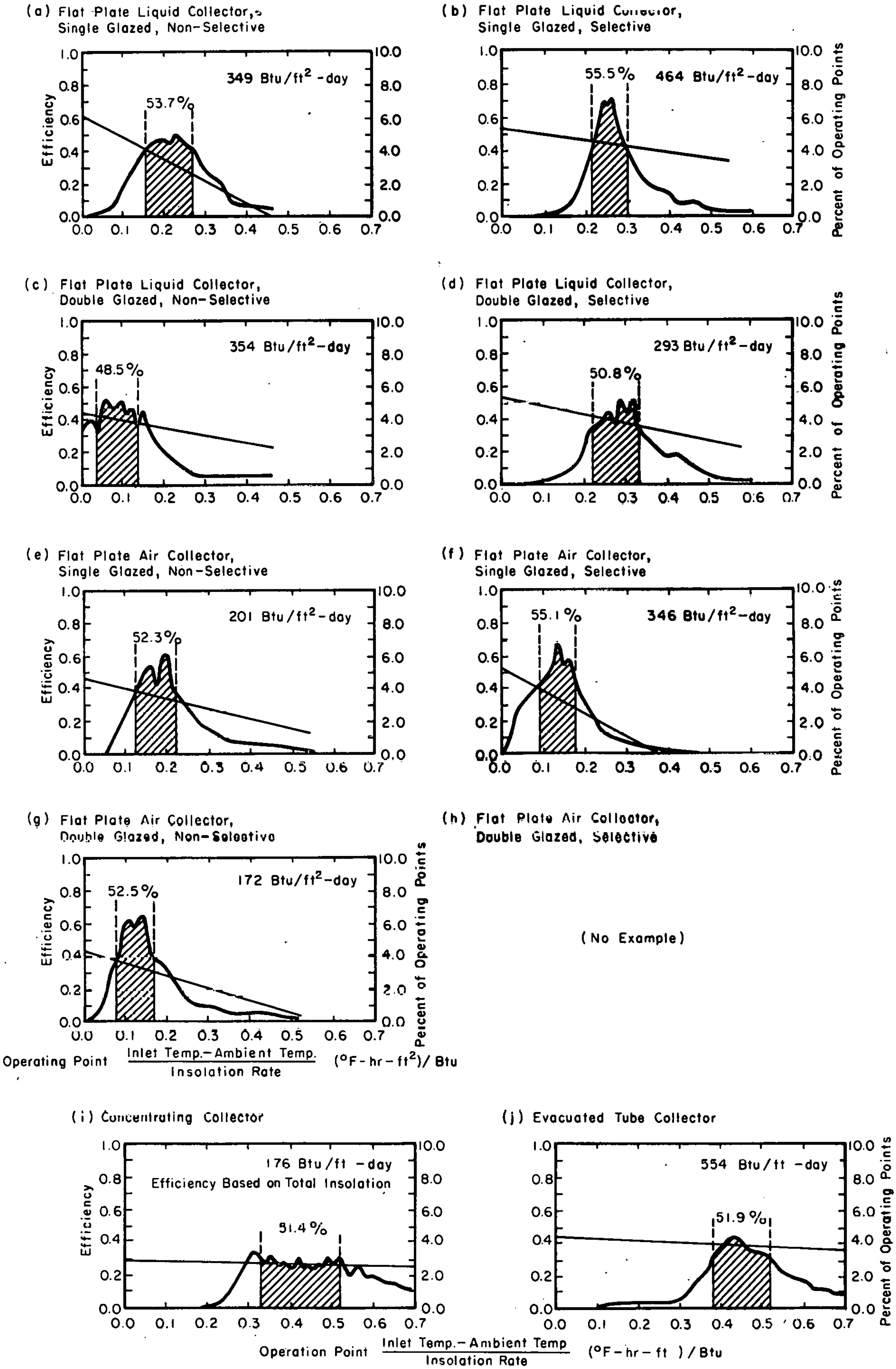


Figure 8. Comparative Performance at Representative Collector Designs [82]

percent from that of single panel collector expectations, system design, installation, and operation problems or errors may cause much greater degradations of performance.

Several of the major problems in collector array performance which have been observed in the field are:

1. Inadequate flow distribution, including a lack of adequate filling of the collector array with heat transfer liquids,
2. Unanticipated thermal storage temperature stratification (or insufficiently accounted for),
3. Unacceptable rates of heat transfer fluid leakage
4. Degradation of collector characteristics due to lack of design, installation and operational considerations, and
5. Prediction methodology does not represent field application, i.e., prediction assumptions are not met in real life.

4.5.3.2 Inadequate Flow Distribution

4.5.3.2.1 Flows within arrays. Collector arrays composed of more than four collector modules are normally connected in a parallel flow configuration. Flow through an inlet manifold is distributed to a set of single modules (or series-connected pair or triplets of modules -- see Figure 9), with the intent of each module or module pair to receive an equal fraction of the total collector flow. For example if the collector flow rate for the array shown schematically in Figure 9 is 8 gpm (0.5 l/sec), each of the module pairs should receive a flow rate of 1 gpm (.06 l/sec) or one-eighth of the total flow.

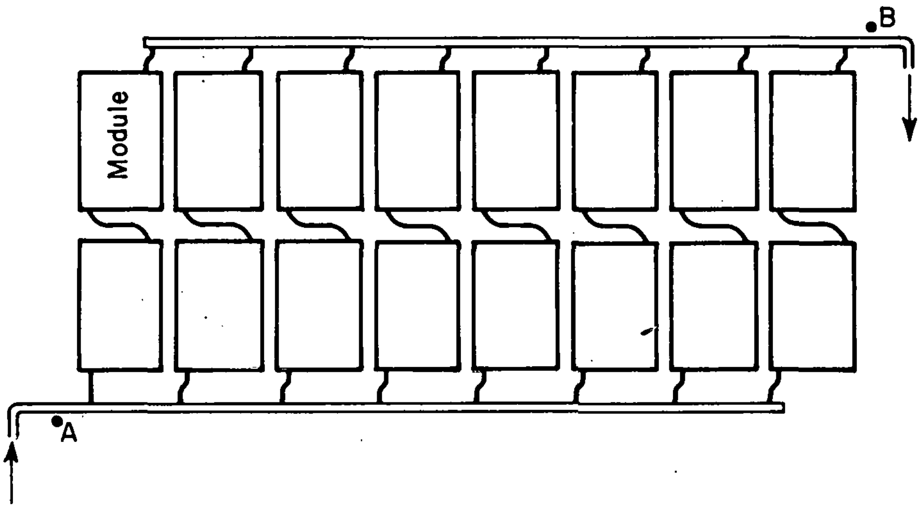


Figure 9. Schematic of Collector Array Flow Configuration Through Modules

The performance of a solar collector array is dependent upon the absorber area of the collector. However, if in a collector array several collector modules are inoperative (i.e., no useful heating is being accomplished), then the effective area is considerably reduced from the absorber area [76]. If, for example, the collector array shown in Figure 9 had the third and fourth modules on the top row inoperative, then the third and fourth modules at the bottom are also inoperative and thus the area of the array which is collecting useful heat is reduced by one-fourth of the gross area. Such reductions could partially account for the differences in observed collector array efficiencies and the predicted efficiencies of the collector modules for some of the systems discussed by McCumber [81] (see Table 48).

A collector module within an array could become inoperative if the flow of the heat transfer fluid through the module was in any way interrupted or restricted. This modular no-flow condition may be experienced in a liquid-heating collector by: Flow constriction of the collector tubes by debris, a combination of glycol leak protection and foreign matter combining to plug one or more tubes, damage to the collector absorber causing crimping and/or closure of a tube, and by air (or steam) pockets which would prevent liquid flow (particularly in open flow, i.e., trickle type, collectors).

Prevention of plugging of collectors with debris and foreign matter is easily accomplished by proper use of filters when flushing the collector array initially and during normal operations.

Damage to the collector absorber (such as during shipping or installation) can be avoided by proper attention to procedures. The elimination of air pockets, however, is a design function and depends upon the ability of the system to completely fill, and keep filled, the collector array. This is a question therefore of proper hydraulic design.

4.5.3.2.2 Hydraulics - Interrupted Flow. Several instances of inadequate filling of the collector array have been observed with subsequent reduction in the collector's useful heat output. These instances have been due, in general, to two specific design features. The first case is shown in Figure 10 and involves the characteristics of the pump being used to fill the collector and the total static head of the collector loop (including the collector array itself). In Figure 10, a pump curve is shown as a plot of pressure head against flow rate. The family of curves indicates the pressure head versus flow rate condition for the various numbers of collector modules receiving flow (assuming the number of modules in Figure 9). If the total static head, H, is greater than h_{max} (as shown in Figure 10), flow will not be achieved in all of the collector modules.

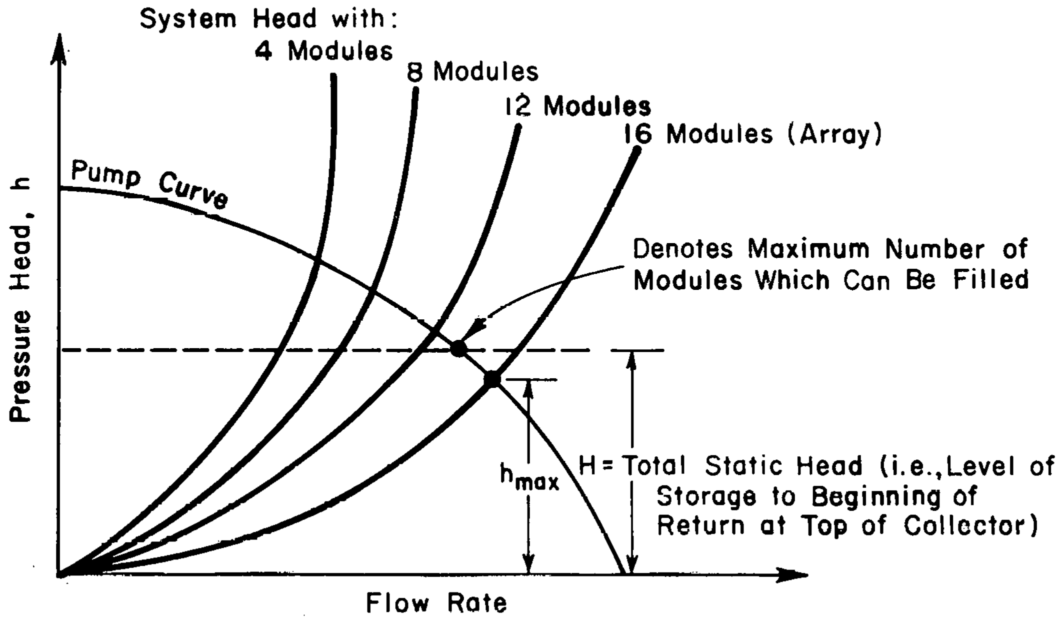


Figure 10. Pressure Head Versus Flow Rate for a Collector Pump and Collector Array

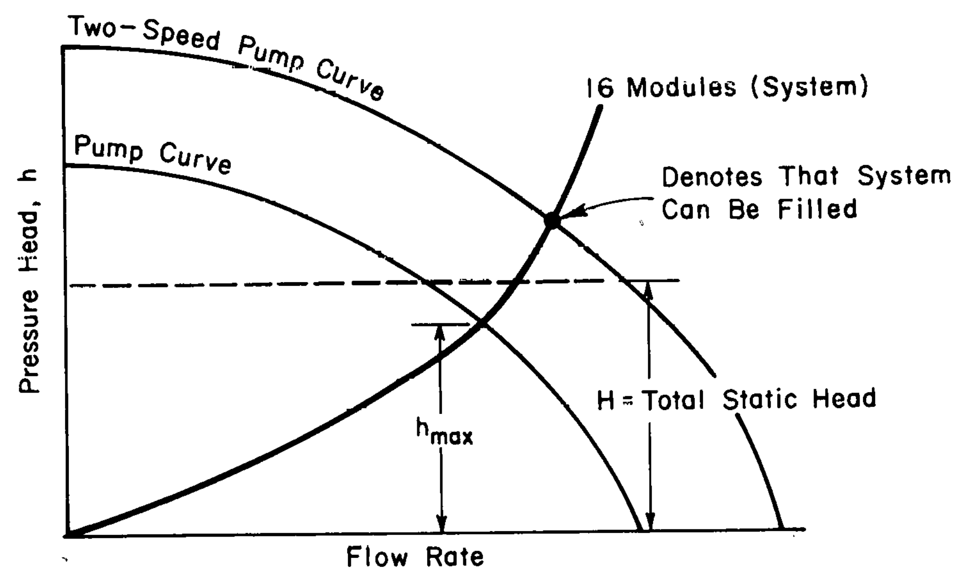


Figure 11. Pressure Head Versus Flow Rate for a Two-Speed Pump and Collector Array

A two speed pump could be added in some cases for the initial filling of the collector array in order to avoid this problem (see Figure 11). The two speed pump would also help to avoid the potential problem of filling a very cold collector array by moving the water through the array fast enough to prevent freezing upon the initial filling of the day for a drain-down or drain-back system.

However, this will not necessarily eliminate a second design problem of maintaining a completely filled collector. For example, Figure 12 shows the relationship between the frictional pressure drop, Δp_m, across the collector module (or series-connected modules) and the pressure static head, h_A, caused by the difference in height between the inlet and outlet of the collector module and/or array. If the collector module pressure drop, Δp_m, is less than the pressure head, h_A, and one module is momentarily not quite filled, the liquid will take the path of least resistance and by-pass that module. Inevitably, because of air pockets incurred in filling operations, from dissolved air in the collector liquid, or from possible steam generation, modules will have their flow occasionally interrupted. If $\Delta p_m < h_A$, then the module flow interruption cannot be corrected and some collector modules will become inoperative. The inoperative modules will normally be located in the center of the array (see Figure 12b).

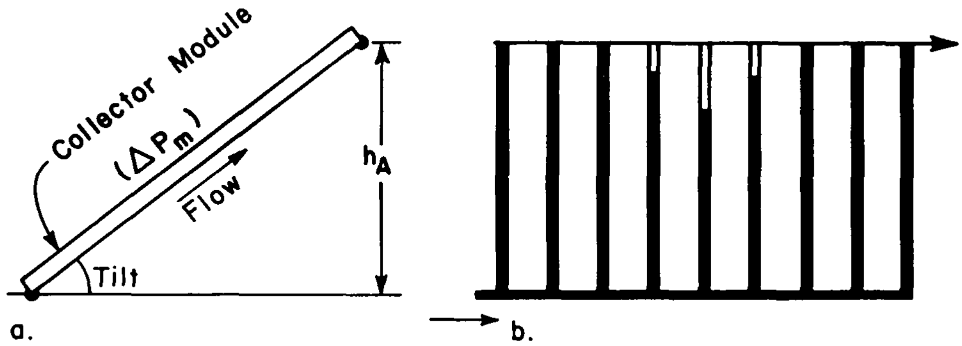


Figure 12. Comparision of Collector Module Pressure Drop and Pressure Head

4.5.3.2.3 Hydraulics - Flow Distribution. A lack of equal flow distribution through all collector modules in an array (e.g., in Figure 9) can also degrade the performance of a collector array. Several cases of poor flow distribution have been observed [84,85] and are again due to inappropriate pressure drops in various portions of the collector loop. Inadequate flow distribution is a potential problem in air-heating as well as in liquid-heating collectors.

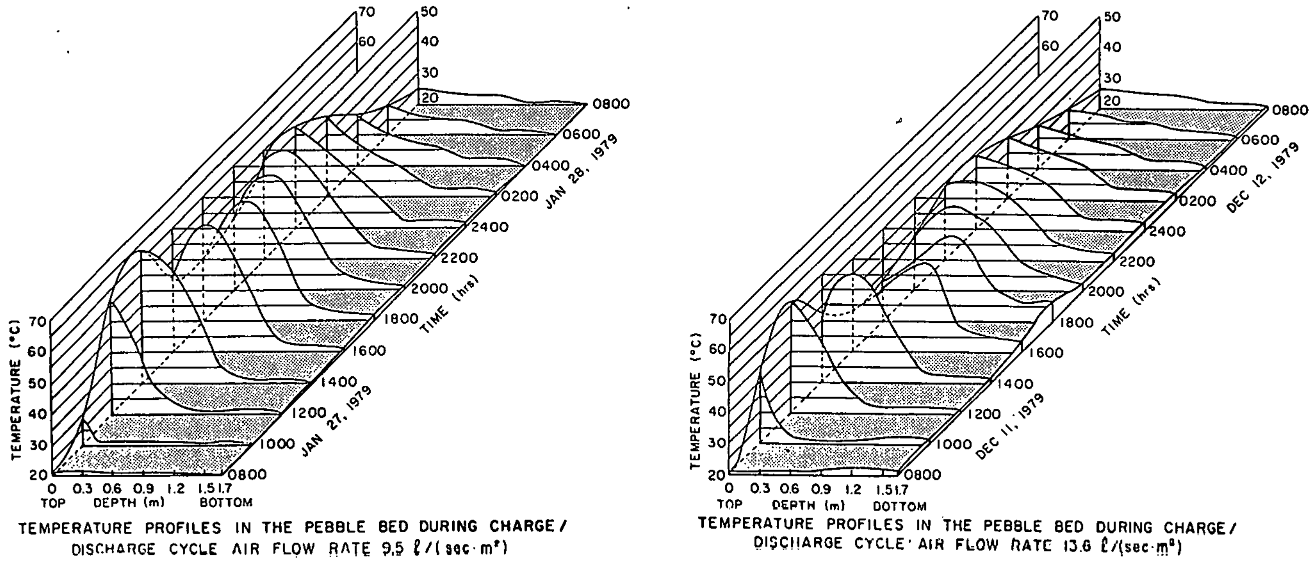
For the case of "diagonal flow" across a collector array (as shown in Figure 9 where the collector fluid enters at one corner, labeled A, and exits at the opposite corner, labeled B), the condition for equal flow distribution [86] is that the pressure drop in the modules (or series-connected modules), must be greater than 90 percent of the total pressure drop from point A to point B, Δp_{AB}. This is easily achieved by ensuring that the inlet and outlet manifolds are large enough to ensure a minimal pressure drop (and less than 10 percent of the total collector array pressure drop).

4.5.3.2.4 Recommendations. If we combine the flow distribution requirement with the requirements for proper filling of the collector, we can summarize three critically important hydraulic design features for the collector array heat transfer fluid characteristics:

1. $\Delta p_m > 0.9\ \Delta p_{AB}$ (see Figure 9)
2. $\Delta p_m \geq h_A$ (see Figure 12)(liquids only)
3. $h_{max} > H$ (see Figure 10)(liquids only)

4.5.3.3 Thermal Storage Temperature Stratification

Thermal storage temperature stratification is the variation of temperature within the thermal storage unit along the path of heat transfer fluid flow. Temperature stratification is most prevalent in pebble-bed storage units used with air-heating collector systems. Figure 13 shows a set of temperature profiles in a pebble-bed storage.



(a) (b)

Figure 13. Pebble-Bed Storage Temperature Stratification Profiles with Respect to Time of Day [67]

The advantage of temperature stratification is that the temperature of the heat transfer fluid from storage to the inlet of the collector array is a minimum. This results in the maximum efficiency for the collector. In air-heating systems this inlet temperature is typically 70°F (20°C), even with collector outlet temperatures of 150°F (65°C). Temperature stratification in hot water thermal storage units is normally more difficult to obtain because of the mixing of hot and cold water by forced convective currents. Numerous techniques have been developed in order to obtain temperature stratification in water storage tanks. Temperature differences of 25°F (15°C) have been achieved in some cases with a resulting four percent increase in collector efficiency [87].

Some projects have reported temperature stratification in water storage tanks with a resulting improvement in system performance. Such temperature stratification has been due in many cases to "short circuiting" of the storage liquid through a tank. In one case a reduced flow rate in the collector loop produced a lower mixing of hot and cold water and, when combined with the location and positioning of the tank connections (see Figure 14), caused a three percent increase in collector efficiency [12].

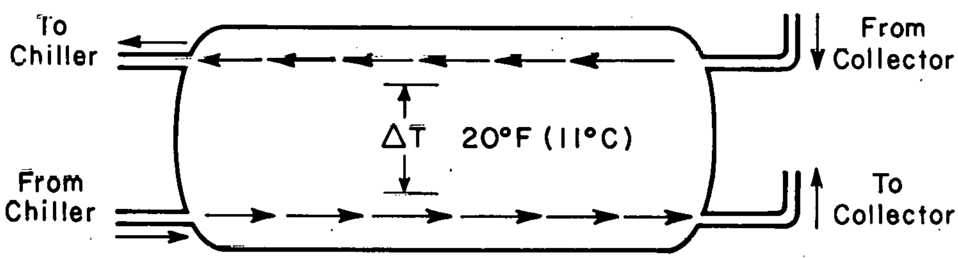


Figure 14. "Short Circuiting" of Liquid Through a Hot Water Storage Tank Resulting in a 22°F (12°C) (approximate) Temperature Stratification

The important factor is that the reduced collector flow rate, which resulted in slightly improved collector performance, used less electrical power to run the collector pump. Such factors must be included in the detailed design of the system if it is to perform up to expectations.

4.5.3.4 Heat Transfer Fluid Leakage

Heat transfer fluid leakage in liquid systems is unacceptable. Accordingly, the effect of liquid leakage on performance has not been considered.

In air-heating systems, air leaks into the collector loop may not significantly affect the overall system performance if the leaks displace the building's air infiltration (and/or exchange) which would otherwise occur during collector operation. Close [88] notes that where leakage does replace natural infiltration, collector leaks will result in better performance over collectors which don't leak. Jones, et al [89] notes that a leak of 10 percent would not significantly affect the performance of the system but leaks greater than 10 percent would be detrimental to the system performance. In addition, air leaks can bring dirt and other impurities into an otherwise closed system, thus interfering with performance of collectors, heat exchangers, storage materials, etc.

If the air leakage does not displace natural ventilation, the role of air leaks into a solar system is to decrease the efficiency of the system over the case where no leaks occur [89]. Table 50 provides a theoretical estimate of the effects of air leakage on collector performance where it is assumed that:

1. Leaks do not displace natural ventilation and
2. Leakage of the collector itself is not considered (because such leaks would already be realized in the collector module testing)

In Table 50, $F_R(\tau\alpha)$ and F_RU_L are the resulting collector parameters and f is the fraction of the load carried by solar.

Table 50. Effects of Air Leakage on Collector Efficiency [89]

	$F_R(\tau\alpha)$	F_RU_L	η_c*	f
No leakage	0.52	0.85	30.8%	74.2
Leaks into collector inlet duct	0.52	1.05	25.8%	70.0
Leaks equally divided between inlet and outlet ducts	0.51	1.10	23.5%	67.7
Leaks into collector outlet duct	0.50	1.14	21.5%	65.5

$*(T_i-T_a/I)= 50°F/(200\ \text{Btu/hr}\cdot\text{ft}^2) = 0.25\ \text{hr}\cdot\text{ft}^2\text{/Btu}$

Shingleton, et al [90] have analyzed the effects of air leakage on performance by considering measured leakage at instrumented sites where air flow measurements were conducted. All of the seven systems considered and where air flow surveys were accomplished, exhibited external air leakage in the collector array and the pebble-bed storage container as well as internal leakage through control, backdraft, and shut-off dampers. Some of the systems also exhibited leakage along duct seams and localized leakage at duct-to-component joints. Air leakage and blower flow rates were found to vary from one operational mode to another in response to the varying system pressure drop. Some leak locations were found to infiltrate air in one mode and exfiltrate air in another.

One of the systems surveyed is the basis for Shingleton, et al's [90] analysis and is described in Table 51. (Figure 15 shows a schematic of the system.) The results of the analysis are shown in Figure 16.

Shingleton, et al [90] has concluded that a system with various external and damper air leaks and an annual solar fraction of 40 percent (i.e., f = 40%), can realize significant savings by eliminating air leaks. In this example:

- o The elimination of all air leaks results in a 19 percent reduction in the seasonal auxiliary energy use.
- o Installation of low leakage dampers (one percent leakage) results in a six percent reduction in the seasonal auxiliary energy use [90].

The effects of the collector flow rate and installation procedures on air leakage (and ultimately on collector performance) has been observed by Karaki, et al [67]. Table 52 provides data on this system where in the west array all of the collector modules (16 in number) were lifted to the roof first. The cherry picker, which was used to lift the collectors, was

Table 51. Component Air Leaks in Basic Solar Air-Heating Systems [90]

Air Leak Rate
(Percent of Design Flow Rate)*

Leak Location	Operating Mode: Storage to house	Operating Mode: Collector to house	Operating Mode: Collector to storage
Collectors	- 8	-42	- 1
Storage	-17	- 1	+11
Solar blower	-17	-19	-11
Damper D2	5	3	14
Damper BD2	2	--	--
Damper MD1A	1	--	--
Damper MD2A	--	--	--
Damper MD2B	--	4	3

*Design flow rate = 1092 cfm (2325 l/s)

Notes: Leak sign convention for external leaks:
positive = leak out (exfiltration)
negative = leak in (infiltration)

"The air leakage in this system was measured after extensive efforts by the HUD support contractor to reduce air leaks where practical" [90].

Note Figure 15.

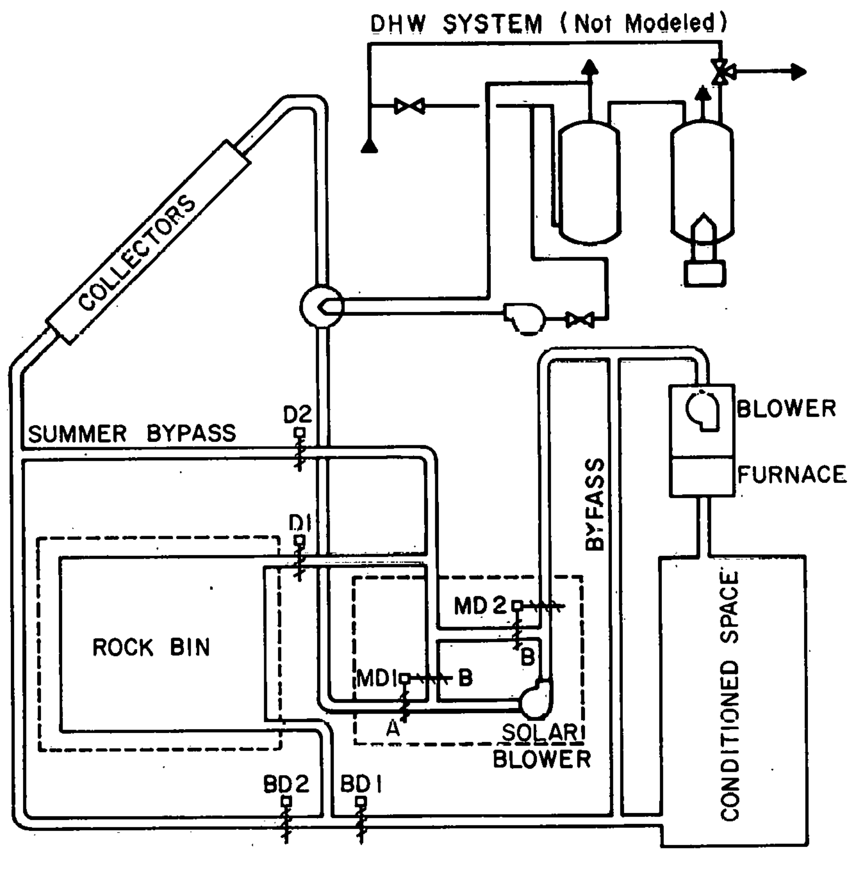


Figure 15. Solar Air-Heating System Schematic [90]

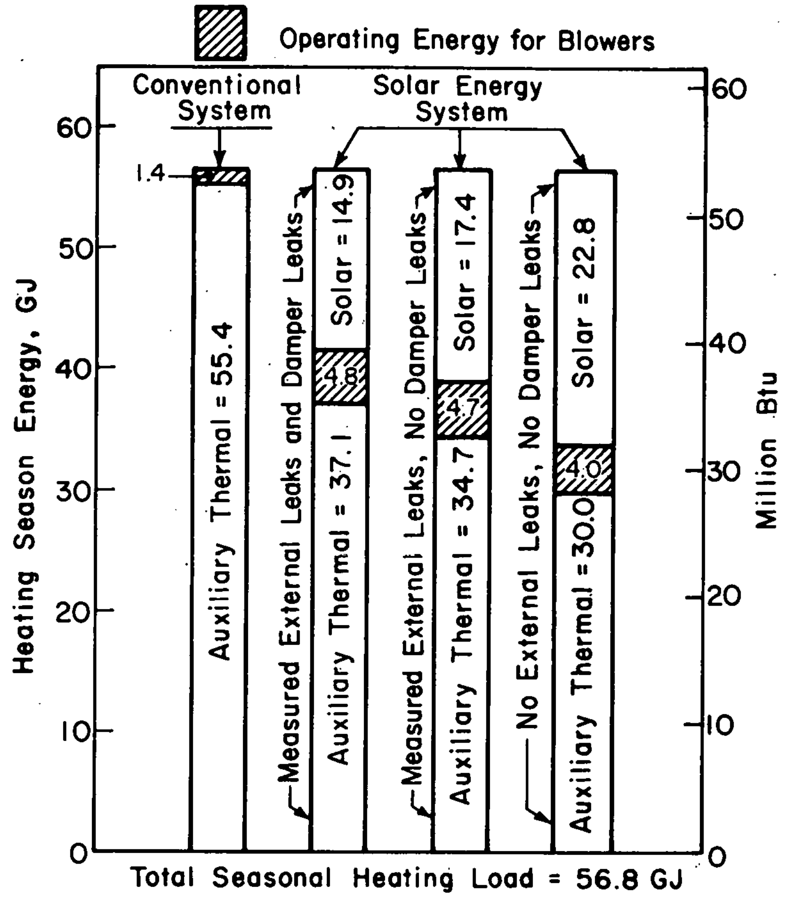


Figure 16. Computer Simulation Results for a Solar Air-Heating System in Madison, Wisconsin With and Without Air Leaks [90]

Table 52. Effects of Collector Air Flow Rate and Installation Procedure on Air Flow Leakage

	West Array	East Array
Flow rate = 6.5 cfm/sq. ft (1.31 l/s sq. m) *		
Leakage during storage charging	5%	13%
Leakage during direct space heating	9%	22%
Flow rate = 4.9 cfm/sq. ft (0.96 l/s sq. m) @		
Leakage during storage charging	5%	6%
Leakage during direct space heating	7%	18%

* Varied from 5.2 to 7.8 cfm/ft^2 (1.02 to 1.53 l/s per sq. m)
@ A reduction in flow rate of 27%

released and then left the job site. The modules were then moved into position by the installers and fastened to the roof. For the east array, collector modules were lifted to the roof as needed and fastened into position. The first method minimizes use time of the mechanical lifting equipment but the second minimizes overall installation time. Including pre-installation preparation time, cleaning the outer glazing, etc., the 16 collector modules in the west array were lifted to the roof in 2.5 hours (thus limiting the use of the cherry picker) and the array was completed in 10 additional hours. The east array was totally installed in 9.5 hours [67].

Karaki has concluded that "Leakage of air into collector arrays does not adversely affect an air-heating system." But this is contingent upon leakage acting as a preheat of infiltration air and that this has not been at the expense of increasing building infiltration. Air leakages of 5 and 20 percent existed in the west and east arrays, respectively. Leakages across closed dampers are significant, particularly when a damper is installed in a bi-directional duct and the reverse side of the damper is subjected to increased pressure.

Jones [89] has shown that, for a 15 percent leak equally divided between inlet and outlet ducts, the solar fraction is reduced from 74 to 68 percent when the air leakage is not a replacement for natural ventilation. Thus while the west array is probably acceptable in terms of leakage (5 to 9 percent), the east array has 6 to 10 percent more leakage (depending upon flow rate). This results in the west array providing 10 to 15 percent more useful heat, Q_c, (assuming that air infiltration into the collectors does not replace natural infiltration), than the east array at the higher flow rate and 5 to 10 percent more useful heat at the lower flow rate. Under the worst conditions the west array useful heat collection may exceed the east array heat collection by 25,000 to 30,000 Btu per day (26,400 to 32,700 kJ/day) (or about 17 percent more useful heat) [67].

4.5.3.5 Effects of Collector Degradation on Performance

The effects on performance due to the observed degradation of collector characteristics include performance reductions due to:

1. Degradation of plastic covers, including fiberglass-reinforced-polyester (FRP)
2. Scaling build-up in collector modules and piping, and
3. Inability of the collector module and/or components to withstand stagnation temperatures

All plastic covers should be expected to degrade over time due to ultraviolet radiation and effects of weathering (see Section 2.2.1.1.1). Performance degradation usually occurs by reductions in the transmissivity of the plastic. For example, FRP plastic covers with a transmissivity of 0.95 (i.e., 95 percent of the incident solar radiation is transmitted through the cover) will inevitably degrade to about 0.90 within one or two years. However, there is little or no further performance degradation after the transmissivity has been reduced to about 0.90. Thus the long range performance level of the FRP cover is about equivalent to that of ordinary glass covers.

Several systems which used steel tubes in the solar collector modules and which were vented on a regular basis have encountered problems with scaling build up in the collector module tubes [91]. This scaling has interfered with flow distribution with a resulting reduction in performance (see Section 4.5.3.2).

A more substantial problem occurs whenever the stagnation (equilibrium no-flow) temperature exceeds the rated maximum temperature of the collector. In one case [76] the stagnation temperature (≈450°F, 230°C) was considerably higher than the manufacture's rated or guaranteed temperature limit of the collector (300°F, 150°C). The result was that the solder used in the collector melted at ≈350°F (175°C) and the bonding between tube and absorber plate failed. The end result was that 55 to 60 percent of the collectors in the system became inoperative [76].

4.5.4 Thermal Losses

4.5.4.1 Introduction

Systems that perform well have always considered the effects of system heat losses in the design phase. Overlooking and/or underestimating the system and component heat losses in other installations has been a major factor in the inability of these solar heating and cooling systems to achieve the expected and/or predicted level of performance [78]. System components that incurred substantial and significant heat losses, which have in turn produced lower levels of performance, include:

1. Thermal storage units
2. Piping and/or ducting
3. Heat exchangers, pumps and blowers
4. Valves (relief, vent, shut off, etc.) and
5. Collector manifolds and collector module interconnections

4.5.4.2 Exterior/Interior Heat Losses

In considering the effect of heat losses on system performance, it is necessary to distinguish between exterior and interior heat losses. Exterior heat losses generally do not contribute toward reducing the heating load or increasing the cooling load and consequently can be accounted for by a simple reduction in the available solar useful heat. In this case the remaining solar heat available to the solar heating and cooling system is the collected useful heat, Q_C, less the exterior heat losses, Q_E.

Heat losses to garages, attics or other normally unheated spaces may, during the heating season, provide either useful heating of that space (essentially a slight improvement in temperature) or help reduce heat losses from the conditioned space by providing a partially heated "buffer" between the conditioned space and the ambient. While such heat losses may be marginally useful in some cases and thus may be a system/building design consideration, such losses will be considered in this handbook as exterior heat losses and therefore will not be considered useful in contributing to meeting the heating load. (This assumption is usually more applicable to residential installations.). Such heat losses during the cooling season would, of course, add to the cooling load.

Interior heat losses are those system heat losses which serve to heat the conditioned space. However, it is important to recognize that heat losses from solar system components that are physically located within the conditioned space include, in general, both interior and exterior heat losses. For example, heat losses from a thermal storage unit can occur:

1. Through the walls of the insulated unit
2. Through the structural base supporting the thermal storage unit (e.g., a horizontal tank with supporting saddles)
3. Via the piping connections to the unit
4. Through thermosyphoning of the heat transfer fluid through the collector (or storage side of the collector/storage heat exchanger) loop, load loop, DHW loop(s), etc.

Exterior located components which are adjacent to the conditioned space may also have interior, as well as exterior, heat losses.

For example, Figure 17 shows an installation where the thermal storage hot water tank with foam insulation (R-30) is located on a concrete slab in a building with concrete walls. The horizontal tank is supported by a steel saddle with four steel (pipe) legs. There are eight piping connections to the tank (to and from collector, to and from heating/cooling units, to and from DHW preheat tank, a drain connection, and a vent valve). The tank is also contained within an equipment room with wall R-11 insulation between the room and the conditioned space and R-19 insulation in the equipment room ceiling. The piping to the storage tank has R-6 piping insulation.

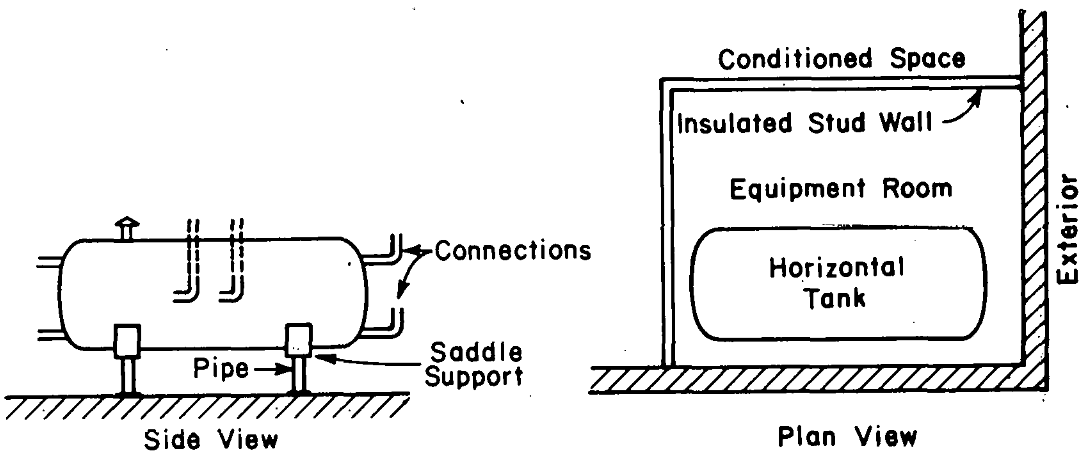


Figure 17. Installation and Insulation of a Thermal Storage Tank [92]

While the tank supports have a very small cross-sectional area, large heat conductivity of steel (relative to R-30 foam insulation) more than compensates for the small heat conducting cross-sectional area of the structural legs. The piping connections have an effective resistance to heat conduction of about R-6 (the value of the piping insulation) and, when combined with natural thermosyphoning, may reduce the effective R-value of the thermal storage by 30 to 60 percent [93]. Clearly the insulation of tank supports, the reduction in the number of connections, and the installation of check valves to prevent thermosyphoning will reduce these losses, but not eliminate them (see below).

Finally the heat losses to the equipment room do not provide useful heating of the conditioned space as such (but normally do provide overheating of the equipment room). Only that portion of the heat generation in the equipment room which reaches the conditioned space through the ceiling or stud wall can be considered interior heat losses. Those heat losses through the exterior basement walls and concrete slab (to ground) are exterior losses. Table 53 provides a summary of those heat loss mechanisms as a percentage of the total heat losses from the thermal storage unit.

Table 53. Heat Loss Components from Thermal Storage [92]

Method	Heat Loss (Btu/hr)*	Percentage of Total
1. Storage tank direct to ground (via support structure)	600	23
2. Storage tank to exterior, direct	100	4
3. Storage tank to equipment room		
a. Through foam insulation	400	15
b. Piping connections (conduction)	600	23
c. Thermosyphoning in load loops	900	35
d. Total to equipment room	1900	73
4. Total thermal storage heat loss	2600	100
1. Equipment room to interior		
a. Through ceiling	60	3
b. Stud wall	160	9
2. Equipment room to exterior		
a. Through concrete walls	1370	72
b. To ground via concrete slab	310	16
Total loss to equipment room	1900	100
Summary:		
1. Heat loss to interior of space	220	9
2. Heat loss to exterior of space	2380	91

*Multiply Btu/hr by 1.055 to obtain kJ/hr

It is important to realize that air-heating collector systems have similar problems. The effective thermosyphoning of cold freezing air in the collector loop has in many cases caused the freezing of DHW coils located within the conditioned space [94].

4.5.4.3 Usefulness/Nonusefulness of Interior Heat Losses

In numerous evaluations of operating solar systems, the heat losses from solar components (such as a thermal storage unit) located within the conditioned space have been considered as useful heat delivered to the space heating load. In effect the assumption has been that $\beta=0.0$. Such an assumption is only true for certain portions of the year. For example, Table 53 provides an estimate of the effects of overheating of a residential-sized solar system on a monthly basis. It is clear from Table 54 that the assumption that all heat losses to the interior of the space are useful is, in general, incorrect. In addition, overheating during the months of October, February, March, and April could represent an even greater problem.

Table 54. Solar System Heat Losses/Overheating of Conditioned Space [95]

	T>23°C	T>24°C	T>25°C	T>26°C
β (November)	.64	.28	.07	.01
β (December)	.46	.22	.09	.02
β (January)	.59	.29	.11	.02

β = Number of hours above T/Number of hours in month
Temperature setting = 72°F (22°C)

During the cooling season, system heat losses to the interior of the conditioned space not only reduce the available solar energy for operation of a cooling unit, they also increase the cooling load of the building. The reduction in the useful solar energy space cooling purposes thus includes the direct loss of available heat to the interior, Q_I, plus the solar heat required to operate a cooling unit in order to remove this heat at some coefficient of performance, C, i.e., Q_I/C. Thus:

$$Q_u = Q_c - Q_E - Q_I - (Q_I/C)$$

where:

Q_u = Useful heat delivered to the cooling unit

Q_c = Useful collected heat delivered from the solar collector array

Q_E = Exterior heat losses

Q_I = Interior heat losses and

C = Coefficient of performance of the cooling unit (See Section 3.2 and Appendix C)

or

$$Q_u = Q_c - Q_E - Q_I \beta$$

where:

$\beta = 1 + (1/C)$

Because of the factor, β, the interior heat losses have two to three times the deleterious effect of the exterior heat losses.

Table 55 provides clear evidence of the devastating effect of thermal losses to the interior of the building. Relocation of the thermal storage to the exterior and/or the improved insulation of the storage has been predicted to yield:

- o Percent increase in Q_uC (improved insulation of interior storage) = 215%
- o Percent increase in Q_uC (relocation of storage to exterior) = 47%
- o Percent increase in Q_uC (improved insulation and relocation of storage) = 393%

Table 55. Effect of Thermal Storage Heat Losses on Cooling Performance (residential 3-ton unit) [96]

	Sept	July 1-16	July 17-31	August
1. I - solar insolation (1000 Btu/day)*	1014.0	993.3	913.1	888.6
2. Heat delivered to storage (1000 Btu/day)*	325.2	325.5	301.0	300.0
3. Collector efficiency (line 2/line 1)	.321	.328	.330	.338
4. Thermal storage loss to interior (1000 Btu/day)*	99.2	105.3	65.2	68.4
5. C - COP of chiller	.377	.432	.495	.450
6. $\beta = 1+(1/C)$	3.65	3.31	3.02	3.22
7. Solar heat to chiller (1000 Btu/day)*	187.9	161.3	197.5	197.8
8. Q_u - Useful heat to chiller (1000 Btu/day)*	-75.0	-81.9	65.8	90.8
9. Q_uC (1000 Btu of cooling/day)* (line 8 x line 5)	-28.3	-35.4	32.6	40.9
10. Q_u (if thermal losses to exterior), (1000 Btu/day)*	187.9	161.3	197.5	197.8
11. Q_uC (if thermal losses to exterior), (1000 Btu/day)*	70.8	56.7	97.8	89.0

* Multiply Btu/day by 1.055 to obtain kJ/day

4.5.4.4 Thermal Storage Heat Losses

Heat losses from the thermal storage units of solar heating and cooling systems require careful attention in both air and liquid systems. A review of the chapter on the Performance of Solar Heating and/or Cooling Systems provides evidence that only 14 out of 53 systems with available data had storage heat losses of less than 10 percent of the useful collected heat.

In DHW heating systems the average percentage heat loss from storage (storage loss divided by collected heat, Q_c) was about 20 percent. Only four systems had less than 10 percent losses (averaging about six percent). Three combination space and DHW heating systems when operating in the summer had 10, 61 and 97 percent losses, respectively! Evaluation of identical data by different researchers [48,49,54] indicate that the method of accounting for heat losses in DHW systems is critical to the evaluation of relative performance levels of different types of systems. For example, for double tank direct and single tank direct DHW heating systems, one investigator obtained thermal storage efficiencies, η_T, of 31.1 and 26.8 percent, respectively, while another researcher obtained contradictory values of 17.8 and 22.1 percent, respectively. The only basic difference was in accounting for the amount of useful heat loss.

Passive systems showed storage losses averaging slightly more than 21 percent although the data are limited. Percentage losses were 9 to 36 percent within six systems.

Active heating systems incurred storage losses averaging 27 percent and ranging from 4 to 67 percent. These variations included the effects of location and type of system (residential or commercial). However, here there are distinct differences in different systems. Five of the best systems averaged only 6.5 percent while the remainder averaged 39 percent. A single subdivision [14] had an average loss of 28 percent and one group of federally funded projects averaged over 60 percent.

In average heating systems, of course, some of the heat loss from storage would be useful. However, in one example [14], the <u>noticeable</u> effect was overheating of the conditioned space from a comfort viewpoint. In another case the solar system was effectively losing five million Btu per month (5.3 GJ/month) [59]!

In cooling systems, storage heat losses averaged about 25 percent of useful heat collected. However six of the systems averaged less than 15 percent while four systems exceeded 40 percent.

Heat losses from thermal storage units cannot be eliminated but are capable of being reduced to 10 to 15 percent of Q_c on small systems and less than 5 percent on large systems. Some systems have done this. This requires careful design of the storage unit including tank or pebble-bed insulation, check valves/dampers to avoid thermosyphoning, pipe or duct insulation, low conductivity support structures, and well-considered location of the storage unit with respect to the building. In regard to location, several noteworthy factors should be considered:

1. Double tank DHW systems have 1.5 to 2.5 times the heat losses of single tank DHW systems
2. Passive water walls (because of natural convective currents within the water) will normally have greater heat losses than solid mass walls
3. All passive mass walls lose a significant portion (25%) of their heat through the foundation support to the ground
4. Two buried storage tanks had heat losses of 36 and 47 percent of the useful collected heat, Q_c. This resulted in a system with collector efficiencies of 24 and 30 percent and system thermal efficiencies of only 15 and 19 percent. However, one cooling system with buried storage had storage losses of 20 percent, which was corrected in following years to less than 10 percent.
5. Cooling system thermal storage heat losses to the conditioned space (other than a mechanical equipment room) have in many cases eliminated the technical feasibility of solar absorption cooling (i.e., resulted in negative overall system efficiencies). Table 56 provides a comparison of interior and exterior storage effects on system performance.

Note in Table 56 that the relocation of the thermal storage unit from the interior of the conditioned space to the exterior of the condiitioned space has the effect of:

1. A 7 percent decrease in total heat losses
2. A 58 percent increase in useful heat to chiller and
3. A 112 percent increase in total energy savings

In one case, excessive storage losses resulted in the solar cooling system being unused for a complete cooling season. Table 57 provides some of the performance and design information on this system.

Table 56. Absorption Cooling Performance with Respect to Location of Thermal Storage [76]

		Interior Storage	Exterior Storage
IA	Solar radiation on tilted surface of solar collector times the gross area of collector array (1000 Btu/day) *	1138.4	1113.6
Q_c	Average daily useful solar energy collected by array (1000 Btu/day) *	332.9	377.6
η_c	Average daily solar collector array efficiency (%) (= Q_c/IA)	29.2	33.9
Q_E	Average daily solar system heat losses to exterior (1000 Btu/day) *	35.7	56.7
Q_I	Average daily solar system heat losses to interior (1000 Btu/day) *	50.5	23.5
C	Heating or cooling unit coefficient of performance (dimensionless)	.605	.605
β	System heat loss factor (nonusefulness of heat losses to interior of space)	2.65	2.70
Q_u	Average daily solar system heat delivered to heating/cooling unit (1000 Btu/day) * (= $Q_c-Q_I\beta-Q_E$)	163.4	257.4
η_T	Average daily solar heating/cooling system <u>thermal</u> efficiency (Q_uC/IA)	8.7	14.0
E	Average daily electrical energy used to operate the solar system (1000 Btu/day)	19.3	19.3
Q_s	Average daily savings in nonrenewable energy resources by solar system (1000 Btu/day) * (= $Q_uC/\eta_a-E/\eta_e$)	77.9	165.3
η_S	Average daily solar heating and cooling system <u>overall</u> efficiency (= $Q_s/(IA+E/\eta_E)$)	6.4	13.9

* Multiply Btu/day by 1.055 to obtain MJ/day

Table 57. Effects of Heat Loss on Performance [97]

	Total Solar to Load	DHW	Heat	Cool	I	Q_c	η_c
March	3183.5	2256.1	927.4	--	1830	7629.4	25.9
April	4234.7	4082.7	151.7	--	1523	4783.0	19.5
May	2893.2	2739.0	154.2	--	1595	4565.5	17.8
June	3312.7	2889.7	423.0	--	1826	4607.0	15.7
July	2910.0	2910.0	--	--	1825	2733.9	9.9
Season	3300.7	2968.8	331.8	--	1720	4865.9	17.6

"Storage temperature was too low to operate the absorption chiller during the present season due to the high level of collection maintenance activity" [97].

1608 tracking concentrating collectors
16,080 square feet of floor space
Continuous circulation, space heating and cooling and DHW
352 ton chiller
2 each - 20,000 gallon storage, R-20 to R-25 observed (usually only one tank in operation in order to maintain higher $T_{storage}$
Energy units are 1000 Btu/day (multiply by 1.055 to obtain MJ/day)

4.5.4.5 Piping/Ducting Heat Losses

Ducting heat losses are caused primarily by air leakage from ducts (see Section 4.5.3.4). However, severe reductions in performance can also be experienced with uninsulated ducts. Table 58 [98] shows the devastating effect of the use of uninsulated ducts (in lieu of one inch insulation in or around ducts). In view of the relative high performance of ducts with only one inch of insulation, there is simply no justification for the use of uninsulated ducting; although ducts whose heat loss or gain is to the conditioned space may not require insulation.

Table 58. Ducting and Piping Heat Losses [98]

	Heat Losses (Btu/day) *	Percent of Collected Heat
Uninsulated piping	≈44,000	18%
2 inch insulated piping	≈14,000	6%
Filled piping heat capacity losses	≈44,000	18%
Uninsulated ducting	312,000	94%
1 inch insulated ducting	49,000	15%

Approximate figures; depend on length of piping, useful heat collection, etc.
*Multiply by 1.055 to obtain kJ/day

Losses from piping can reduce the performance in two distinct ways. During the operation of the collector, heat losses from the piping can account for 5 to 20 percent of the useful heat gain of the collectors. In one case of uninsulated piping, the heat lost between collectors and storage accounted for 39 percent of the useful heat collection [14]. Insulation of the piping subsequently reduced this loss to about 13 percent of the useful heat collection. The corresponding thermal system efficiencies for the uninsulated and insulated piping were 11.4 and 24.3 percent, respectively (see Table 59).

Table 59. Energy Losses as a Percentage of Solar Energy Collected [16]

	As Installed		Predicted when insulated *	
	(10^6 Btu/day)@	(%)	(10^6 Btu/day)	(%)
Solar energy collected	5.92	100	5.92	100
Solar energy lost between collector and storage	2.30	39	0.77	13
Solar energy lost from storage	1.67	28	1.67	28
Solar energy lost between storage and load	0.34	6	0.11	2
Solar energy delivered to the space heating load as controlled energy	1.09	18	2.85	48
Solar energy delivered to the DHW load	0.48	8	0.48	8
Solar energy not accounted for	0.04	1	0.04	1
Appropriate η_T	11.4%		24.3%	

* Insulation value of R-6
IA = 490,000 Btu/day
Q_c = 210,700 Btu/day
Q_u = 56,100 Btu/day
Multiply Btu/day by 1.055 to obtain kJ/day

Heat losses from piping can also adversely affect the performance of the solar system when the collector is not operating. These losses are due to the heat capacity of the liquid in the collector loop (and, to a lesser degree, the other pipes as well). These losses occur when the collector pump shuts down and the liquid is allowed to cool overnight or for extended periods of time during the day. Because the piping is typically not warmed by the sun (as in the case of the collector array), the heat lost is irreversible without the commencement of collector operations to reheat the liquid. Experience has shown that about 75 percent of the reheating of the liquid left in the piping comes from the thermal storage unit. Consequently this heat loss is significant. Ward [98] has estimated these "heat capacity" overnight losses at 40,000 to 50,000 Btu/day (42,200 to 52,800 kJ/day). Table 58 provides an indication of the potential magnitude of these losses for active space heating and cooling systems. It is noteworthy that drain-back systems do not experience such losses except to the extent that the piping itself is cooled.

4.5.4.6 Component Heat Losses

Individual components of the solar system that are not insulated can result in significant heat losses. In one case [107] where piping was already insulated (but not the heat exchangers, valves and pumps), an overall storage heat loss coefficient of 55 Btu/hr•°F (105 kJ/hr•°C) was measured with the storage tank vent open and no fluid flows in or out of the tank. When the heat exchangers were insulated (but most of the valves, sensor fittings and unions were not), the loss coefficient was reduced to 42 Btu/hr•°F (80 kJ/hr•°C). This resulted in a reduction in heat losses of 1300 Btu/hr (1372 kJ/hr) for a storage tank of 216 square feet (20.1 square meters) surface area and a temperature difference between the operating liquid and the ambient of 100°F (56°C).

It is particularly important to wrap relief valves, vent valves, shut off valves, etc. individually when the valves are exposed to the ambient. Without such precautions, some installations have experienced freezing at the location of the valves before other portions of the collector loop because of the higher heat losses.

4.5.4.7 Collector Manifolds and Intermodular Connections

Losses from the collector manifolds are usually considered as part of the piping losses. However, intermodule connections within a collector array are usually not included in the calculation of piping heat losses. Oonk [83] has shown that heat losses from intermodular connections can account from one to five percent of the heat losses of the collector array and therefore cannot be ignored in the design phase.

4.5.5 Solar System Operation Electrical Energy Requirements

The solar system electrical solar operating energy requirements is the second of two major problems which have been experienced in reduced performance of some of the solar heating and cooling systems (the first major problem is excessive thermal losses [78].

Table 60 provides a ranking of DHW systems by electrical solar operating energy per unit area of collector. In reviewing Table 60, several conclusions regarding the electrical usage by DHW systems can be made:

1. The ratio of controlled solar energy utilized to the electrical operating energy use, S_c, greater than 9.0 or 10.0 can be readily obtained with well-designed systems.
2. Electrical usage per unit area, E/A, of less than six percent is possible with well-designed systems.
3. Double tank systems are capable of higher collector efficiencies than single tanks but, in most cases, at the expense of greater electrical usage. (This is due in most cases to longer operating hours with double tank systems.)
4. On average, single tank systems have lower electrical usage and subsequent greater overall system efficiency than double tanks. However, system #27, a drain-down double tank system, outperformed all other systems. In addition, system #18 had storage losses of 78 percent. If these could be reduced to about 20 percent (as in system #16), the system overall efficiency would have been increased to 47.4 percent, considerably better than the single tank systems. (In general, double tank systems have higher overall efficiencies than single tank systems under lower solar insolation conditions, i.e., winter, but have lower efficiencies in summer.)
5. Silicone liquid systems generally performed poorly. In one case (#23), the excessive pumping energy is due to the use of silicone liquid and the use of a second pump to achieve the desired flow rate [53].

Table 60. Electrical Solar Operating Energy per Unit Area of Collector

E/A (Btu/ft^2·day)*	S_c	η_T (%)	η_s (%)	System Identification and Description	
0.0	∞	24.0	43.6	20	Direct thermosyphon
26.1	9.8	22.6	29.6	21	Direct, single family
35.0	11.3	26.6	35.9	17	Indirect, single tank
38.0	9.5	11.0	12.8	28	Indirect, double tank, water/glycol
38.1	4.8	14.3	13.4	22	Indirect, continuous circulation, multifamily
45.0	14.1	19.6	26.8	26	Direct, single tank, drain-back
46.2	3.6	7.8	5.3	18	Indirect, double tank
54.5	6.3	25.2	26.4	19	Direct, single tank
57.0	5.4	8.0	5.3	25	Indirect, single tank, silicone
58.0	12.0	42.4	55.7	27	Direct, double tank, drain-back
74.1	6.2	31.1	31.4	16	Direct double tank
96.2	3.4	24.6	13.4	23	Indirect, double tank, silicone
128.0	3.5	6.2	1.7	29	Indirect, double tank, water/glycol
195.0	4.6	35.8	22.6	24	Indirect, double tank, silicone

* Multiply [Btu/ft^2·day] by 11.4 to obtain [kJ/m^2·day]
See Section 3.2 for definitions of parameters

Surprisingly, six out of ten passive heating systems utilized electrical energy to operate fans and/or other solar system components. The ratio of solar energy utilized to the electrical used, S, ranged from a low of 9.2 for fan energy in a warehouse and 9.6 for a hybrid (combination active/passive system)to 26.0 for a beadwall and as high as 60 to 90 for fan distribution energy requirements. Electrical energy requirements in passive systems are generally to improve certain aspects of the system's operation, and cannot always be considered system requirements as much as component requirements. For example, in one case, the electrical energy is used only to operate the beadwall system in an earth-covered direct gain mass wall. The beadwall system has been added in an attempt to reduce night time losses. The energy requirements of the beadwall subsystem (about 4800 Btu(elec) per day) is therefore associated only with the energy savings at night and not necessarily with the energy gains during the day. Thus the value of S of 26 is not necessarily representative of the passive system. If the difference in night time heat losses with and without the beadwall is calculated, we obtain an energy savings (due to the beadwall) of about 5400 Btu/hr (5700 kJ/hr). If we consider the average night of ten hours and add two hours for cloudy periods, then the daily energy savings from the beadwall are about 65,000 Btu/day (68,600 kJ/day). The value of S for the beadwall is thus 65,000 Btu/day divided by 4800 Btu/day for a figure of 13.5. It is noteworthy that the energy savings attributed to the beadwall is about one-half of the useful energy delivered to load!

Active space heating systems use on the average approximately half the electrical energy per unit area of collector (31 Btu(elec)/ft^2 of collector, 350 kJ/m^2) as do the DHW heating systems (68 Btu(elec)/ft^2).) From Table 61, we see that the ratio of controlled solar energy utilized to the electrical operating energy used, S_c, range from less than one to over 16. Six systems have S_c values greater than 13!

The amount of electrical energy required per unit energy delivered to load (E/Q_u) averaged 12.1 percent. In the worst case the electrical energy reduced the useful energy savings by two-thirds. Four systems had E/Q_u values of 6.5 percent or less and provided the three highest overall efficiency systems. Two of these systems had overall system efficiencies of 42.9 percent.

In comparing air and water systems, we note that for this of systems:

System	E/Q_u	η_s	Range of η_s
Air	13.1%	20.6%	9.4 to 42.9%
Water*	13.6%	22.7%	11.0 to 42.9%

*(and water/glycol)

Air-heating collectors can achieve higher efficiencies when operated with higher mass flow rates, but only with increased electrical energy usage (blower power). As a lower limit on flow rate, a given depth (i.e., length of flow path) in a pebble-bed storage must be sufficient

Table 61. Electrical Usage and Efficiency of Active Space Heating Systems

System Number and Fluid		E/Q_u (%)	E/A (Btu/ft^2)*	S_c	η_s	Remarks
51	Air	27.1	68	1.8	9.4	
52	Air	5.9	9	16.9	21.6	
53	Air	14.7	16	6.6	11.0	
54	Water	11.3	23	4.8	19.1	
56	Water	20.0	34	5.1	11.9	
57	Glycol/water	19.7	47	2.4	12.2	
58	Drain-back	12.9	28	7.7	20.5	S_c = 4.2 with fan energy
59	Air	20.3	52	0.8	11.2	
59	Air	16.0	39	1.1	13.4	Second year
60	Glycol/water	12.7	29	7.8	23.3	
60	Glycol/water	10.2	23	9.8	28.9	Second year
61	Air	5.7	27	16.2	42.9	S_c = 8.4 with fan energy
61	Air	4.6	22	13.5	34.9	Second year
62	Water	6.2	33	14.5	42.9	
	Conventional HVAC			∿ 2.5		

*Multiply Btu/ft^2 by 11.4 to obtain kJ/m^2
See Section 3.2 for definitions of parameters

in order to enable effective heat transfer to the pebbles and to ensure adequate flow distribution in the rock box. Thus the air flow rate in air systems is critical. The effects of different air flow rates is described in part by Karaki [67] and is summarized below:

Flow rate	η_c	E**	Q_u*	Q_s*	η_s
13.1 l/s m^2	38%	18.64	324.8	469.6	45.7
@ 9.6 l/s m^2	35%	13.18	320.0	482.6	46.8

Notes: *(1000 Btu/day) (multiply by 1.055 to obtain MJ/day)
** Collection blower only
@ 27% reduction in flow rate

Cooling systems on the average were net energy losers because of excessive electrical energy usage and poor control methodologies. But, while five systems had a combined net loss of 17.5 million Btu/day (18.4 GJ/day) (about 850 Btu/day·ft^2 of collector; 9700 kJ/day·m^2 of collector), three systems had positive energy savings, totaling over one million Btu/day (1.055 GJ/day) (about 100 Btu/day·ft^2 of collector; 1140 kJ/day·m^2). System efficiencies, η_s, for the three energy saving systems ranged from 1.8 to 11.2 percent (although in two cases the initial designs had $\eta_s < 0$) and S_c ranged from 3 to 7.

4.5.6 Solar Control-Caused Variations in System Performance

An essential aspect of the reliability and efficient operation of a solar heating and cooling system is the control subsystem. Control of active solar systems (and, to some degree, passive systems) is a major factor in the overall performance of the system. Controls must be utilized in such a manner as to take best advantage of components and their integration into the solar system. When appropriate overall control strategies are combined with reliable control subsystem hardware, significant energy savings and/or reductions in component sizes are possible [99].

Control-caused variations in solar system performance can be categorized into two distinct groups:

- o Control strategy or methodology
- o Control sensors and hardware components

4.5.6.1 Control Strategies

Control strategies are particularly important because they establish the method by which the solar system is to be controlled. The control strategy includes the determination of when pumps, blowers, valves, and/or dampers are actuated, the setting of differential and absolute temperature set points, and the design of component protection circuits (e.g., freeze or boil protection circuits). The control of when pumps and blowers and other components are actuated is extremely important, particularly in solar cooling systems [100]. Established and proven procedures must be followed in order to maximize performance (see, for example, reference 101).

For example, "solar HVAC systems can often be designed for a given performance to require no more than 70 percent of the collector area required by the frequently used empirical design methods" [102]. Newton [102] has proposed dual storage units, one with 25 percent of total volume for high temperature operation, and one with 75 percent of total volume for lower temperatures. This design is expected to increase the average collector efficiency to the extent that 30 percent of the collector area may be eliminated. This performance improvement is based on the fact that any solar system which can make effective use of low temperature heat can obtain much more heat from a given collector array than a system which must have higher temperature heat. The critical factor is that, in both heating and cooling applications, there are only a few hours during the year wherein maximum water temperatures are needed in energizing the load handling equipment [101].

Differential temperature set points can have critical effects on the performance of a solar system. For example, Ward [103] has reported that, for the case of the temperature differential between the collector array and storage of 5°F ($\sim$ 3°C) greater than the specified design setting, the system thermal efficiency would be reduced from 39.5 percent to approximately 31.1 percent. Note that the "error" of 5°F may be due to an incorrect setting by a designer and/or installer or may be due to the inaccuracies of the control sensors in the collector array and/or storage unit.

Failures of protection circuits (such as freeze and/or boil protection circuits) have resulted in absorption chiller freeze-ups (i.e., solidification of the lithium bromide/water solution) due to the condensing water temperature being too low [104], collector heat transfer liquid boil-off and subsequent damage to the collector array [105], and numerous other difficulties in the operation of a solar system.

4.5.6.2 Control Sensors and Hardware Components

Bartlett [106] has evaluated numerous systems in which the control system has caused abnormal or anomalous operation of the solar energy system. Based on this evaluation of projects in the National Solar Demonstration Program, Bartlett classified the control problems into three groups:

1. Control sensor problems
2. Problems with controllers and
3. Problems with control actuating devices

Control sensors have been improperly placed in the system (and thus do not reflect the actual operating condition of the system), have failed in use, and have provided inaccurate information; all resulting in extraneous or non-existent energy flows. Controller problems have included malfunctions of equipment as well as incompatibilities with solar system designs. Control actuating device problems include component failure and improper operation (e.g., valves or dampers do not fully close). Leaky dampers, for example, experienced leaks of 12 to 40 percent of full flow [8].

4.5.6.2.1 Control Sensors. Control sensors must meet four major requirements:

1. Proper location (i.e., placed in a position so that the designed temperature (or other variable) is measured directly),
2. Provide accurate measurement (reliable readings over a wide range of conditions)
3. Resist failure (durable over extended periods of time and during worst-case conditions; such as sub-zero and/or stagnation temperatures), and
4. Protected against extraneous flows

A major difficulty with many commercially available liquid-heating solar collectors is that no allowance has been made in the collector module design for the insertion of a control sensor. This inadequacy also results in the inability of installers/maintenance personnel to quickly insert temporary data sensors into different collector modules in order to check for proper flow distribution. Alternatively, solar collectors designed for drain-down, drain-back or trickle type collector systems must have a means of attaching a control sensor to an absorber plate. In this latter case, the control sensor being attached to the absorber plate will not accurately reflect the temperature of the fluid. This difference in absorber and fluid temperature must be accounted for in the control methodology and setting of differential (and/or absolute) temperature set points.

Control sensors must be chosen so as to provide reliable and accurate readings and at the same time act as a durable component. Sensors with variable output for similar conditions will result in non-optimum control and subsequent reduced performance. Again, the possible inaccuracy of 5°F (2.7°C) may result in a reduction of system thermal efficiency from 39.5 to 31.1 percent.

Control sensors must be able to withstand stagnation (equilibrium, no flow condition) temperatures. This requirement is particularly noteworthy in evacuated tube collectors because of the much higher stagnation temperatures. Repeated failures of the collector array control sensor in an evacuated tube has been observed [105].

A fourth control sensor problem has been the effect of extraneous heat flows on a temperature-actuated control sensor. The extraneous heat flow causes either heating or cooling in the immediate vicinity of the temperature-actuated control sensor. Extraneous heat flows have resulted from solar radiation, wind and ambient temperature. They have also been associated with a storage tank, a boiler, or a hot or cold pipe located near the temperature control sensor. In some cases the incorrect signals have been caused by thermal gradients between two points along a pipe or duct under normal no-flow conditions. These thermal gradients have resulted in either heat transfer along the pipe or duct or free convection cells being set up within the piping loop where fluid is transported by buoyant thermal forces from the cooler to the hotter region.

There are two types of abnormal system operation caused by invalid inputs to the system controller resulting from extraneous heat flow. The first is degraded system performance as typified by the case where the collector inlet temperature is artificially high. Under this condition the solar energy collected over an extrended period may be significantly reduced because the collector circulation pump is not always operated when useful energy could be collected. This condition, even when significant, might go unnoticed.

Because of these experiences, all temperature control sensors for solar energy systems should be reviewed with respect to their location and susceptability to extraneous heat flows. Special attention should be given to their insulation. If these sensors are properly insulated, their susceptability to the influence of ambient temperature fluctuations will be greatly reduced. When heat flows are internal to the heat transfer fluid, such as those associated with thermosyphon cells, their influence can be reduced by proper placement of the sensors or by use of check valves or backdraft dampers. In addition, the designer or installer of a solar energy system should ensure that all control sensors are not faulty, inaccurate or mislocated [106].

4.5.6.2.2 Controller Problems. A controller problem is characterized by either the controller not functioning as designed or the controller functioning as designed in a solar energy system with which the design is incompatible. In many solar energy systems, the pump which circulates the heat transfer fluid through the collector is repeatedly cycled on and off. This abnormally high rate of on/off operation degrades system performance and could cause premature failure of the pump. This condition most often occurs during periods of low solar radiation such as early morning or late afternoon when the collection of solar energy is being initiated or terminated. The cause has been found most often to be in either the collector controller or the control sensors, and sometimes in the system dynamics.

The typical collector controller turns the circulation pump on or off based on the difference in temperature between the collector outlet and the bottom of storage. If the pump is off and this temperature difference rises above some preset value, ΔT_{on}, the circulation pump will be turned on. If the pump is on and this temperature difference falls below some preset value, ΔT_{off}, the circulation pump will be turned off. The temperature difference required to turn the pump on must be significantly greater than that required to turn it off. Otherwise frequent cycling of the pump will result. For example, when the pump is turned on, energy will be removed from the collector and the outlet fluid temperature will decrease. If ΔT_{off} is too close to ΔT_{on}, and the incident solar energy does not increase sufficiently, the pump will quickly be turned off. With no flow through the collector, the fluid outlet temperature will increase, and the pump will then be turned back on, starting another cycle. This will continue until the incident energy increases or decreases enough to keep the pump either on or off. The ratio of ΔT_{on} to ΔT_{off} that will prevent this system instability is determined by the characteristics of the collector and the environmental conditions. This ratio has been found to be typically between 5 and 7 for liquid collectors and between 1.5 and 3 for those using air. Setting the ratio too low will result in the described cycling and setting it too high will result in inefficient operation by not maximizing the amount of solar energy collected. For maximum efficiency, this ratio should be set during system operation in the field to a value slightly above that required to prevent anomalous cycling [106].

System dynamics in both start-up and no flow conditions have also been observed to cause unstable operation of the collector fluid circulation pump. If, during start-up time in the morning, the line from the storage tank to the collector contains a significant quantity of fluid which has reached ambient temperature during the night, then the system might cycle unnecessarily because of the cooler fluid passing through the collector ahead of the warmer fluid from storage. The cooler fluid will quickly lower the collector outlet temperature below that at the bottom of storage, thus turning the pump off. This problem has been eliminated in some solar energy systems by using a time delay in the controller to allow passage of the cooler fluid through the collector without the pump being turned off. During no flow conditions at night, it is also possible for enough thermosyphoning of the fluid between the cooler collector and the warmer storage to heat the temperature control sensor at the collector enough to turn the pump on. This condition has also resulted in instability since once flow is initiated, the sensor will be cooled quickly and the pump turned off. This condition has been avoided in many systems by the use of check valves to prevent thermosyphoning.

Anomalies have also been observed which were not caused by problems with any single controller but were caused by an incompatibility between two or more controllers. An example of this problem was a space cooling system which consisted of an absorption chiller in series with two vapor compression chillers. All chillers were serviced by a common chilled water pump which was controlled independently of the absorption chiller. After the absorption chiller had been operating normally for some time, the chilled water circulation pump was turned off but the independent control system for the absorption chiller did not turn the chiller off. The chiller froze the stagnant water in the evaporator, rupturing the tube bundle.

Other systems have been observed in which the incomptability was between the collector controller and the load controller. In several systems using air as the transport fluid, the collector controller allowed energy to be collected and stored late in the afternoon at a temperature lower than that which the load controller had defined as useful. The energy at a lower temperature was stored at the top of the rock bed, thus forcing the higher temperature energy, collected in the middle of the day, to the middle of storage. At night, when a demand for space heating existed, the temperature at the top of the rock bed was less than the useful temperature for heating, thus the heating demand was met completely with the auxiliary unit even though the air temperature at the middle of storage was above the minimum required for heating [106].

4.5.6.2.3 Control Actuating Device Problems. A control actuating device problem occurs when a pump, fan, valve, or motorized damper does not operate according to the controller Obvious examples of this class of problem are when the actuating device is broken or malfunction ing and cannot respond. Another example where the potential for degrading the performance is no as obvious is the case where a three-way valve or an air control damper does not fully close. This results in a flow path which is not properly terminated.

Of the nine systems having air/water heat exchangers for heating hot water, three systems experienced freeze damage due to leaky dampers. The performance impact was studied by modeling a standard air system and determining the effects of leakage through each damper on an individual basis. In most cases, the performance impact was insignificant. Either the energy lost due to damper leaks was only a few percent, or when the energy lost was more than a few percent, it was not actually lost to the system but was diverted to some beneficial use other than that intended. For example, if the system was in the collector to space heating mode, any energy leaking into the rock bed was not really lost from the system but could be used later in the storage to space heating mode.

However, in some cases, performance was degraded significantly when a damper leaked during a particular mode of operation. These dampers should be considered critical because of their potential for degrading the performance of the system.

One damper found to be critical was the damper whose function is to stop the flow of air through the collector when the system was in the storage to space heating mode. This mode typically occurs at night when the ambient temperature is well below the temperature of the conditioned space. If air is allowed to leak through the collectors during this time, the collector will reject energy to the colder environment. It was found that the energy rejected by the collector ranged from about 19 percent of the energy removed from storage (i.e., 81 percent of the energy removed from storage went into the conditioned space and 19 percent was rejected to the environment) for a 10 percent leak through the collector.

Another damper found to be critical to the system performance was the damper which prevents flow through the conditioned space in the summer collector to hot water mode. The effect of such a leak is to increase the cooling load because of the hot air being forced into the conditioned space. The significance of this additional cooling load is highly dependent on the normal cooling load, but this damper should also be considered critical and receive special attention.

It was found that some leaks were occurring because backdraft dampers were installed backwards. In addition, some were not properly closing because they had been damaged either in shipment or during installation. Also it was discovered that some were not properly adjusted. Some leaks were caused by motorized dampers which had not been properly wired. The appropriate damper was not being used in many cases. Standard dampers were being used where one with a low leak rate was required, manual dampers were being used where a backdraft damper was required, and dampers with a low temperature adhesive were being used in place of one utilizing high temperature adhesive. Failure of damper seals was also found to be the cause of excessive leakage. These failures were due to incorrect installation of seals during assembly, the breakdown of seal adhesive after installation, or seal deterioration due to high temperatures. The system designer should verify that the dampers being used do not have the problems which have been found to cause excessive leakage; especially the critical dampers discussed previously [106]. Table 62 provides a summary of control problems as reported by Kent and Winn [107].

Table 62. Summary of Control Problems [107]

Number/Type of Problems	Potential	Minor	Major
A. General System	2	4	9
1. Poor performance		2	5
2. Freezing problem	2		1
3. Leakage		1	2
4. Noise		1	1
B. Collectors	2	8	9
1. Leakage in manifolds	1	3	3
2. Leakage within collector	1	2	1
3. Slope		2	
4. Broken or damaged covers due to:			
a. Winter			1
b. Unknown causes			1
5. Insulation degradation			1
6. Serviceability			1
7. Poor design			1
8. Sealing		1	
C. Piping or Ducting	1	14	3
1. Leakage		5	1
2. Poorly insulated		5	
3. Sized improperly		1	1
4. Noise		1	
5. Freezing	1		
6. Installation quality		1	
7. Excessive heat loss			1
8. Aesthetics		1	
D. Valving or Dampers	1	3	5
1. Omission or mislocation		2	3
2. Leakage		1	1
3. Sticking	1		1
E. Storage	1	3	2
1. Leakage		3	
2. Excessive pressure drop			1
3. Excessive heating loss/freezing			1
4. Overheating/boiling	1		
F. Pump or Blower		6	4
1. Motor burnout		3	
2. Noise		1	1
3. Sized improperly			2
4. Leakage		1	
5. Serviceability		1	
G. Controller		3	5
1. Sensor mislocation		1	1
2. Inadequate room temperature control			1
3. Component failure		2	1
4. Sensor missing			1
5. Improper application			1

4.5.7 Architectural Effects on System Performance

System performance for particular installations may be affected by a building or structure's architecture. Architectural constraints on system design and subsequent performance may result from:

1. The imposed effect of aesthetics,
2. Shading of all or portions of the collector array,
3. Structural features affecting the location of solar system components or use, and
4. The length and size of piping/ducting runs resulting from the integration of the solar system with the building.

Aesthetics, for example, may require that the solar collector array be placed at a non-optimum tilt and/or orientation. Aesthetics may also eliminate certain component selection options (e.g., evacuated tube or concentrating collectors, cooling towers, exterior storage units, etc.). Aesthetics therefore represents a potential constraint on solar design.

Alternatively, the solar system design may result in improved or degraded aesthetics. Numerous examples are available where the architectural use of the solar collector array on the building provided for a pleasing visual effect (see for examples Figures 18 and 19). Conversely, a collector array covering the south-facing roof of a north-facing residence may indirectly detract from the aesthetics of the building by requiring the normal plumbing-type roof projections to be on the front of the building.

Shading of a collector array may be caused by other buildings/structures, trees, and components of the building in which the solar system is located. In residential applications, fireplace chimneys can cause a significant shading of a portion of the collector array. Alternatively, trees are usually not a shading problem for roof mounted collectors (except for tall, close-in trees). Shading of a collector array by other man-made structures may have

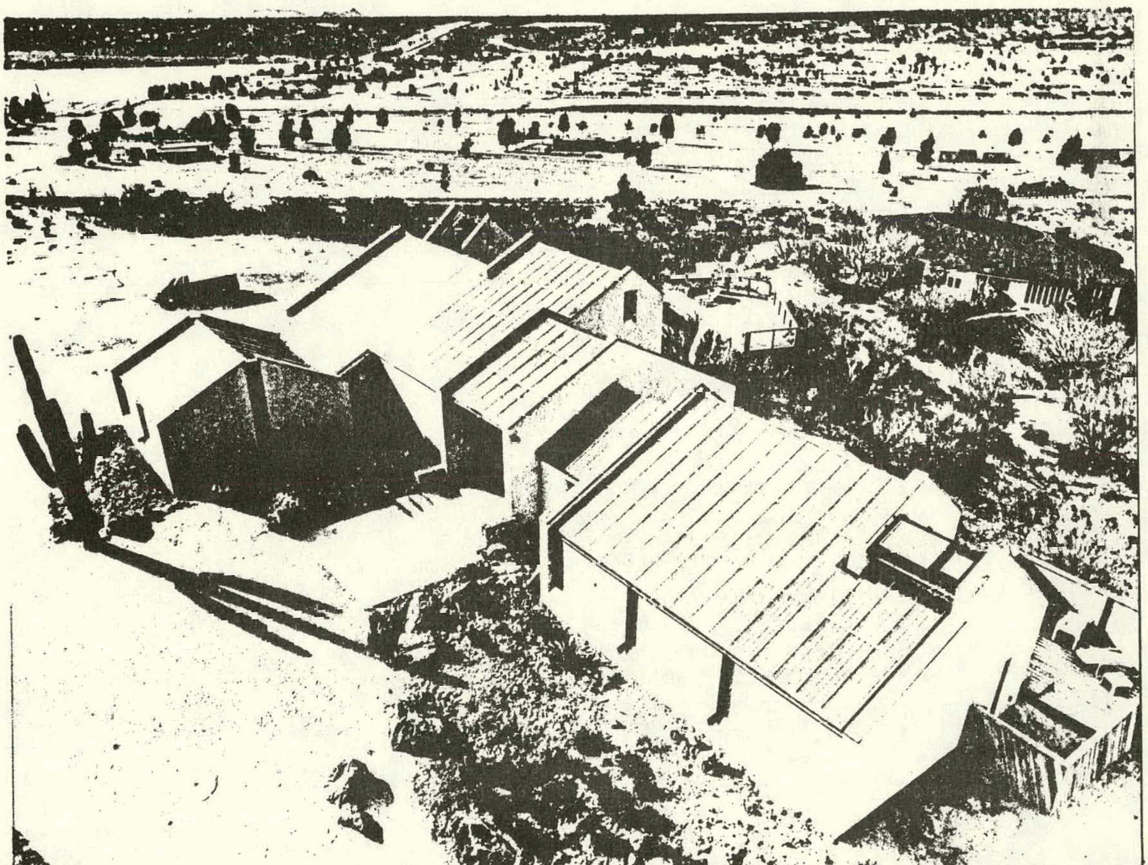


Figure 18. Copper Development Association Tucson House [108]

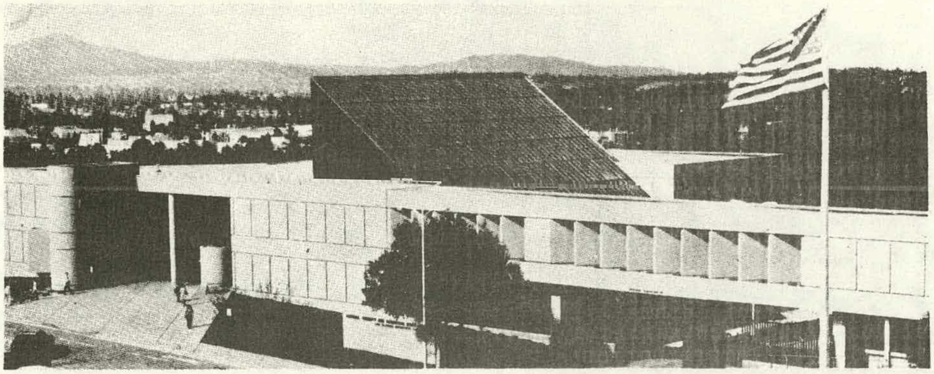


Figure 19. Los Alamos Scientific Laboratory Library and Conference Center [109]

legal implications which should be considered. In general the legal precedence appears to favor the right of the owner not to have a previously unshaded collector shaded.

The effect of shading on performance is to decrease the effective area of the collector array for portions of the day. While such performance effects are relatively straightforward, designers should also consider the potential problems of durability and reliability caused by extreme temperature gradients between shaded and unshaded portions of a collector array and/or module.

Structural features may constrain the design options of a solar system. For example, many commercial installations have large flat roof areas which may or may not support the weight of a solar collector array. The use of thermosyphon systems requires the placement of the storage unit at a higher elevation than the collectors, thus for residential roof mounted collectors, the roof structure must be capable of supporting the storage unit at an appropriate height.

The configuration of a building with respect to the solar system and the resulting locatotions of solar components (e.g., collector array, storage, chillers, etc.) can have an important effect on the length of piping/ducting runs. It must be emphasized that long piping and/or ducting runs will result in large piping/ducting heat losses. Excessively long runs can result in technically infeasible solar designs (i.e., more energy lost and/or used in the form of electricity than gained by the solar collectors). Even with proper insulation of pipes and ducts, excessive heat losses may still result in systems where long runs exist between solar components. The piping and/or ducting heat losses may be exterior or interior and thus may be partially useful (see section 4.5.4.2). Nevertheless, large piping/ducting losses lead to uncontrolled and potentially undesirable space heating.

Long piping/ducting runs also increase the frictional pressure drop. This in turn will lead to increased electrical power usage in moving the heat transfer fluid. When combined with increased heat losses, the additional electrical energy usage can result in poor performance of the solar heating and cooling system.

The importance of the architectural effect on system performance cannot be overstressed. In any system design effort for a particular installation or building, it is essential that the designer recognize that the architectural constraints of a particular building may eliminate the technical feasibility of a solar system design. This is true irrespective of whether or not an adjacent building may be particularly suited for a specific solar design and its installation.

5. PERFORMANCE EVALUATION OF SYSTEMS

5.1 PERFORMANCE EVALUATION OF DHW SYSTEMS

5.1.1 Introduction

In assessing the performance of solar domestic hot water (DHW) systems it should be noted that many of the systems (particularly the larger commercial installations) were essentially developmental and testing projects. Nevertheless, solar DHW systems have, in general, performed well. With few exceptions, system thermal efficiencies of 22 to 33 percent were achieved. Several systems (identified in Table 16 as systems 16, 17 and 20) had system overall efficiencies of 31.4 to 43.6 percent and corresponding energy savings of the three systems of approximately 0.21 million Btu/year per square foot (2.4 GJ/year$\cdot$m^2) of collector.

The cost-effectiveness of these systems can be estimated by the following: For a system costing $2000, a tax credit of 50 percent and conventional energy costs of $20 per million Btu ($19/GJ), the first year's savings in conventional energy would be about $214 for an estimated 21.4 percent return on the investment. Alternatively the cost of delivered energy varied from $9 to $23 per million Btu ($8.50 to $22/GJ) [48].

With this demonstrated feasibility of solar DHW systems, the remaining questions concern the choice of design or the type of solar DHW system to be selected. In comparing the different generic designs of DHW systems, three major considerations are paramount:

1. Durability and reliability
2. Performance, i.e., energy savings and system efficiency
3. Freeze protection as it affects performance (as well as boiling and corrosion factors and their effects on the system's performance)

Durability and reliability have been previously discussed. The emphasis here will be on energy savings and system efficiency.

5.1.2 Thermal Losses/Performance

The noteworthy exceptions to the good performance of solar DHW systems are those systems which suffered excessive thermal losses, principally from preheat and other solar storage units. Of the several systems with low system thermal efficiencies, it is noteworthy that the thermal losses, Q_L, divided by the useful collected energy, Q_c, exceeded 25 percent (i.e., more than one-fourth of the collected energy was not delivered to hot water. This is shown in Table 63.

Table 63. System Heat Losses as a Percentage of Collected Energy for Several Low Performance Systems

System	Q_L/Q_c	Possible Reason for Excessive Heat Losses
3	33	Buried preheat storage tank
6	58	Continuous circulation system
7	61	Combined space and DHW system
8	50	Combined space and DHW system
9	99	Combined space and DHW system
18	78	Recirculation loop *
22	27	Continuous circulation system
23	33	Piping runs to and from collector = 200 ft
Avg of all other DHW systems	13	(varied from 3 to 23%, the system with 23% losses had exceptionally high collector efficiency to offset the high heat losses)

* The recirculation loop was used in an office building and "The system had an extremely low summer time efficiency due to light hot water heating requirements. The collector efficiency increased from 25% in July to 41% in October due to the greater hot water heating requirements" [50].

5.1.3 Electrical Operating Energy/Performance

Silicone liquid systems used excessive electrical energy in operating the solar systems. In system 23, "the seemingly excessive pumping energy found in this system is the indirect result of the use of the silicone liquid and the particular flow meter used to meter the collector flow loop" [53]. The pumping conversion efficiency of electrical input energy to thermal energy added to the collector fluid has been estimated to be 0.74 [53]. The bottom line in Table 63 for system 23 considers this input, which causes a very substantial increase in system overall efficiency of 13.4 percent to 35.1 percent (see Table 19).

Table 20 provides additional information on the electrical energy usage of silicone liquid systems. For example in Table 20, comparisons of systems 24 and 27 show that:

System	Thermal Efficiency	Btu ($Btu/day \cdot ft^2$)	System Overall Efficiency
24 - Silicone	35.8	195	22.6
27 - Water/drain-back	42.4	58	55.7

The electrical enrgy used by the silicone liquid system is approximately 2.5 times that of the drain-back system. This leads to reduced energy savings because of the larger parasitic energy consumption of the collector pump.

E/A values of less than 40 Btu/ft^2 day (456 $kJ/m^2 \cdot day$) have provided good system overall efficiencies. System 16 had a higher value of E/A of 74 Btu/ft^2 day (843 $kJ/m^2 \cdot day$), but the excellent collector and thermal efficiency more than compensated for the higher electrical usage. It should be noted that the collector efficiency of 40.5 percent is due in part to the greater electrical usage in pumping power, higher flow rate and longer operating periods, which provided more useful collection at the expense of higher electrical usage.

Another contributor to electrical energy usage is the use of solenoid valves in drain-down systems for freeze protection. Because the valves require electrical power to remain in the operating position (in order that the valves will fail open and drain the collector liquid in the event of an electrical power failure), the energy used by the freeze protection control system was a major factor.

This is an important result and indicates the effect of some freeze protection techniques on the system performance.

Several systems were considered by one investigator [49] and had electrical usage of [48]:

Type System	Electrical Usage by: Pumps Kwh	Solenoid Valves (estimated) Kwh
Single tank, direct, liquid	68.2	91.4
Double tank, direct, liquid	88.3	91.4
Single tank, indirect, liquid	69.0	91.4
Double tank, indirect, liquid	87.1	--
Double tank, indirect, air	112.5	--
Single tank, direct, thermosyphon	--	32.4

"The direct systems had two 15w solenoid valves which used more energy than the pumps for drain-down freeze protection" [48].

5.1.4 Single/Double Tank Systems

Table 18 provides a comparison of different systems. From this information the single tank seems to show a superiority over the double tank. However, Table 18 is based on only about four months of data. A monthly comparison of two of these systems is of more interest.

Table 64 shows that the system efficiency, $(Q_u-E)/AI$, for the double tank direct system is higher during October through March. In particular, November (which had a severe drop in solar radiation intensity level -- 49 percent lower than October and 26 percent less than in December) provided a major improvomont in the double tank direct efficiency (34.1 percent) over the single tank (only 13.3 percent). Tables 21 and 22 also provide monthly data for direct (system 16) and single tank indirect (system 17) with January system overall efficiencies of 48.8

Table 64. Monthly Performance of Double Tank Direct and Single Tank Indirect DHW Systems [49]

Month	Double Tank Direct S_c	Double Tank Direct $(Q_u\text{-}E)/AI$	Single Tank Indirect S_c	Single Tank Indirect $(Q_u\text{-}E)/AI$
7	5.5	22.3	12.3	28.7
8	4.8	19.0	11.0	23.7
9	5.0	19.6	10.0	23.4
10	6.9	27.5	12.4	24.0
11	5.4	34.1	5.8	13.3
12	7.6	32.1	10.5	20.9
1	9.2	36.2	11.1	22.3
2	10.3	27.3	11.6	18.4
3	8.2	31.3	13.8	28.5
4	5.9	25.6	13.1	26.7
5	5.2	23.6	11.8	27.1
6	5.7	26.0	11.4	27.7

Note: See Section 3.2 for definitions of S_c, Q_u, E, A, and I.

percent (double) and 33.9 percent (single) and for July of 25.4 percent (double) and 42.6 percent (single).

A related factor is that the single tank system used a hot water mixing valve to temper the water from a maximum of 140°F down to 120°F (60 to 50°C). This provides for better single tank performance (particularly due to the higher temperatures experienced in summer) but at the possible risk of scalding in case of a failure of the mixing valve. A second factor is that the data in Tables 18 and 64 are based on the hot water demand profile assumed by the simplified computer design method, F-Chart. This demand profile has peaks in the morning and evening. Such a profile favors the single tank over the double tank because of the greater overnight heat losses of the double tank (the double tank has a larger surface area by which to lose heat). Had the profile been skewed to the afternoon and early evening (i.e., the major hot water demand in the afternoon and evening), the solar availability and hot water demand would have been more in sequence. This would have reduced the heat losses from all systems, on a proportional basis, and the double tank's performance would have been improved relative to the single tank's performance. This result points to the critical importance of load profile on selection of system design, especially for DHW systems.

Also note (Table 18) the excellent performance of the thermosyphon, the only passive DHW system evaluated. This was a single tank liquid system. Also, note from Table 20 that the double tank drain-back system (not a drain-down) had the highest system overall efficiency of all DHW systems of 55.7 percent. This is considerably higher than the single tank drain-down system operating under similar conditions. However, the period of performance reported was only over a 20 day period in the heating season, and therefore may not provide conclusive evidence.

5.1.5 DHW Air Systems

The double tank indirect air DHW system in Table 18 shows relatively poor performance-system efficiency of only 3.1 percent in comparison to the liquid systems of 12.5 to 22.6 percent. This is due in part to the larger collector heat losses and approximately 30 percent greater electrical usage than the similar double tank indirect liquid DHW system. It is noteworthy, however, that the annual performance [49] indicates a fractional energy savings (solar fraction) for the air system of 29.7 percent as opposed to 43.6 percent for the liquid system. (The fractional energy savings considers the energy necessary to run pumps, controls, solenoid valves, etc.). Thus the comparison in Table 18 may be biased in favor of single tank systems.

A deficiency in the available data is that there are no single tank indirect air systems evaluated. Thus no definitive conclusions on the merits of a single tank indirect air system can be made, especially when considerations of durability and reliability (freezing, boiling, corrosion, etc.) are included.

5.1.6 DHW Heating Systems - Conclusions

Based on the data presented here, several preliminary conclusions and/or recommendations can be stated. The principal conclusion is that properly designed and installed solar DHW systems perform well and provide substantial net energy savings.

Conclusions which consider the relative merits of different types of solar systems include (based on the limited data available):

1. Continuous circulation and recirculation systems cannot be recommended because of excessive heat losses and electrical energy usage. (This condition is not limited to solar systems.)

2. Silicone liquid systems have a serious disadvantage of reduced performance due to a low heat capacity (only 0.38 Btu/lb•°F as opposed to 1.00 for water and 0.84 for water glycol) and therefore excessive electrical energy usage.

3. Stand alone DHW systems perform better than combined space and DHW heating systems (BASED ON A LIMITED SAMPLE OF COMBINED SYSTEMS).

4. Single tank systems obtain higher system efficiencies in summer (or high solar radiation conditions) and double tank systems obtain higher system efficiencies in winter (or low solar conditions).

5. A single tank <u>direct</u> liquid (either pumped or thermosyphon) system performs better than single tank <u>indirect</u> systems, BUT the double tank <u>indirect</u> liquid system outperforms the double tank <u>direct</u> system.

6. <u>Drain-back</u> systems have a distinct performance advantage over <u>drain-down</u> systems because of the use of electrical energy in the drain-down's control system (for freeze protection). This may be offset in part by the need for repressurizing the drain-back system's water for subsequent delivery to the DHW distribution system.

7. Freeze protection has been a <u>critical</u> factor in the choice of DHW systems. Double tank indirect liquid and air systems have had fewer freeze protection problems when designed and installed properly.

8. The double tank indirect air systems evaluated by this handbook performed poorly. And, while the single tank air system cannot be expected to perform as well as the single tank system, freeze protection considerations may become the overriding factor. It is noteworthy, however, that recent results have been obtained which indicate equivalent performance of air-heating DHW systems and water-heating DHW systems [77].

9. Combined space and DHW heating systems may achieve better performance by separating the collector arrays for space and DHW subsystems and thus reducing the electrical pumping energy usage during the non-space heating season.

10. Space heating and/or cooling systems can effectively utilize DHW subsystems in many cases to assure year-round utilization of the solar system. However, such applications require careful design in order to avoid overheating, boiling and thermal shock problems (see Sections 2.2.1.2.1, 2.2.1.2.2, and 2.7.2).

5.2 PERFORMANCE EVALUATION OF PASSIVE SYSTEMS

5.2.1 Introduction/Assumptions

The detailed performance evaluation of passive systems is in general considerably more difficult than for active systems. This is primarily due to the fact that the major energy flows are not contained within pipes and/or ducts and are therefore much more difficult to measure. In evaluating passive solar systems it is usually necessary to resort to theoretical calculations, which are in turn based on simplifying assumptions. Because of the varying assumptions made by different evaluators and because of the enormous impact of system performance resulting from different assumptions, it is difficult to make definitive judgements on passive systems performance.

A review of Table 24 suggests that solar passive systems have, in general, performed well. However, these results are based on several important assumptions. These include (a) that all heat losses from mass walls, etc. are to the conditioned space ($Q_{losses} = Q_I$) and (b) all heat losses are useful ($\beta = 0$). Systems 37 and 40 do have non-zero β (i.e. there is some overheating of the conditioned space). Also, some investigators do not include in their calculations the heat losses by interior components to the exterior of the conditioned space. For example the heat losses from a massive thermal wall may include substantial losses through the foundation or wall perimeters and may reduce the usefulness of the heat collected by as much as 25 percent.

Another critical assumption is made in the calculation of the building load. Because passive systems typically have large south-facing window areas, the building heating load may be substantially higher than for a "comparable conventional" building because of the greater night time heat losses through the larger window areas. The effect of heat losses through the glazed south walls and the method by which it is accounted for is of major importance. For example, in the systems identified as 31, 33, 35 and 36 in Table 24, it is assumed that all night time losses through the glazing are included in the design estimate of the load. This is enormously beneficial to the apparent performance of the solar system and can be justified only if the "so-called non-solar home" could be expected to incorporate the same amount of window area. In general this latter assumption is not always true.

The definition of a passive system also constitutes an assumption. In virtually all of the "passive" systems listed in Table 24, some active elements are utilized. These include:

System	Active/Passive Elements
31	Rock bed under floor on north side of building; Fan for charging rock bed and central air distribution; Sliding glass doors between greenhouse and direct gain space
32	Rock bed under floor on north side; Fan for charging rock bed and central air distribution
33	Manually operated insulating curtains for south window wall and manually operated shutters for sky light
35	Five fans for central air distribution
36	Power to operate "beadwall"
37	Beadwall
38	Rock bed, fan for charging rock bed and central air distribution
39	3370 gallon (12,750 liter) water storage and pumps for circulation of water
40	Rock bed, fan for charging rock bed and central air distribution

It may be argued that fan distribution electrical energy requirements would also be required in a conventional non-solar building and thus should not be charged to the solar system. However this ignores the fact that the passive system may require longer operating periods for the distribution fan in order to prevent excessive temperature variations within the space.

5.2.2 Overall Performance

Passive systems have generally performed at high levels and with minimal operating problems. Storage, in whatever form, had non-useful heat losses ranging from 0 to 36 percent. Temperature variations in the conditioned space ranged from 5°F or 7°F (3°C or 4°C) for closely controlled environments, to 35°F (20°C) for a warehouse. Movable day/night insulation was effectively used to substantially reduce night time heat losses without interfering with day time collection of solar radiation. Greenhouses provided only minor contributions to the heating of the conditioned space, but did function as an effective temperature buffer between the conditioned space and the ambient.

System 31 consisted of a combination sunspace (greenhouse)/direct gain system. Massive walls and floors were heated by direct gain and/or indirectly by fans, which provided heated air to a rock bed under the floor on the north side of the building. The building used a 850 square foot (80 square meter) south-facing window to heat the building's 1056 square feet (98 square meters) of floor space.

System 35 used vertical double glazed panels on the entire south and east walls of a warehouse to heat the concrete floor and warehouse contents. Overheating protection was provided by an overhang over the glazings and by natural ventilation. Five distribution fans were used for distribution of auxiliary heat and for ventilation whenever the conditioned space temperature was below 60°F (16°C) or above 90°F (32°C).

System 36 utilized 720 square feet (67 square meters) of double glazed windows to heat a building with 1800 square feet (167 square meters) of floor space. The windows were directly in front of a drum wall (55 gallon/208 liter drums filled with water) with a "beadwall" night time insulation, to limit night time heat losses. The building utilized a concrete slab floor and reinforced concrete exterior insulated walls.

In reviewing the systems identified as 31, 33, 35 and 36:

1. Systems which use some form of movable insulation to reduce glazing losses (at non- or low solar conditions) ... are 10 to 15 percent more efficient in utilizing collected solar energy, thus illustrating the benefits of movable insulation assemblies [55].
2. "Daily building temperature variations had little effect on magnitude of energy savings" [55]. This conclusion did not consider comfort conditions or electrical energy usage by a fan for ventilation and distribution.

   Common problems noted in reference [55] included:
   a. Insufficient storage capacity to prevent overheating. System 36 had no significant problems but system 33 had a fluctuation in room temperature from 70 to 85°F (21 to 29°C) on May 7, 1979.
   b. West windows tend to cause undesirable energy gains in the spring and fall.
   c. Moisture accumulation (especially from attached greenhouses) was a problem not adequately addressed.
   d. Night time heat losses did sometimes present a problem. System 33 had a net loss of energy during the 1977-78 season because of excessive heat losses through glazings. Calculations indicated a night time insulating panel with R-2 insulation would achieve a breakeven in terms of energy in/energy out.
   e. Distribution fans were not always useful when the storage was depleted (system 35).

In general, energy savings of greater than 300 Btu per square foot per day (3420 kJ/m^2·day) were possible with comfort conditions maintained [55].

Figure 4 (see Section 3.3.1) gives the total energy flows for a 176-day period for system 32. Several factors are noteworthy [34]:

1. "The adobe mass wall is effective primarily as heat storage for the greenhouse but is less effective than expected in heating the living areas."
2. "Above a threshold level of 295 Btu/day·ft^2 (3363 kJ/m^2·day), seven percent of solar through greenhousing glazing is conducted into the living area in November-February and three percent in March-April."
3. During November-February, 41 percent of the radiation through the greenhouse GLazing is absorbed by the adobe wall, during March-April only 21 percent is absorbed by the wall.
4. "The effectiveness of the <u>fan-forced</u> rock bed (an active system component) as an important thermal element of the system has been shown emphatically confirmed." The ratio of useful heating to electrical operating energy input is about 11.0.
5. Peak temperatures in the greenhouse were reduced by forced ventilation by about 13°F (7°C).
6. The warm floor radiant heating capability increases the room temperature from approximately 57°F up to 72°F (14°C up to 18-22°C).
7. "The greenhouse is an effective solar collector." The solar radiation threshold for useful collection is about 300 Btu/day·ft^2 (3420 kJ/m^2·day) wherein about 51 percent is transferred to the house.
8. Temperature variations ranged from extremes of 28°F to 43°F (15.5°C to 23.9°C). The average clear day temperature variation was 5°F (3°C). Table 25 provides monthly performance information on system 32.

System 39 combined a roof water storage (3370 gallons, 12,755 liters) with clerestory windows and hinged insulation panels. Results include [35]:

1. The "heat storage room seems to be effective thermally but was overpowered by direct gain through south windows during the early part of the heating season, tending to overheat the space".

2. "Convective heat transfer from the warm ceiling to the air below is rather inefficient, hence the radiative mode accounts for most of the heat transfer."
3. Critical design parameters are angles of tilt of glazing (in this design, 65 degrees from horizontal) and the length of roof overhang.
4. With direct gain, room temperatures may vary by 25°F (14°C). Without direct gain the room temperature variation is about 10°F (6°C).

Table 26 provides monthly performance data for system 39.

System 40 also provided some cooling data utilizing an economizer cycle. The system charged the rock bed with cool ambient air for four hours at night and discharged it over nine hours into the conditioned space by the central air distribution system. The overall useful cooling to electrical operating energy input (fan) ratio was 3.3. See Table 27 for monthly performance values.

System 37 is noteworthy in that it is earth covered with a combination direct gain and mass wall with beadwall insulation for night time use. The storage features include 54 fifty-five gallon (208 liter) drums with a heat capacity of 22,680 Btu/°F (43 MJ/°C) plus concrete walls with a heat capacity of approximately 50,000 Btu/°F (95 MJ/°C).

Only systems 37 and 40 attempted to evaluate the usefulness of the heat losses to the conditioned space. For system 37:

Month:	Nov	Dec	Jan	Feb	Mar	Apr	May
β	.58	.19	.00	.18	.25	.24	.37

Clearly the assumption that all of the heat losses contribute to meeting the heating load is false.

System 34 was, in general, a failure [43]. The room temperature varied from 45.7 to 100°F (8 to 38°C), primarily due to a lack of storage mass [43]. An insulating curtain was planned but not installed. There "could be a net energy loss due to uninsulated glazing and insufficient storage mass" [43]. It is noteworthy that of all ten systems evaluated, only system 34 had no distinctive active and/or hybrid system components and this system by far had the poorest performance.

System 38 was another system that did not match the performance of the other passive heating systems evaluated. Several problems were identified including the need to relocate sensors and backdraft dampers in order to more effectively utilize the rock bed storage and to better distribute heat to the north rooms; the need for night time ventilation of the greenhouse for summer time heat relief; and the possible need for a reflective shade for preventing storage wall heat build up. Room temperatures were reported as high as 84°F (29°C) [43].

5.2.3 Passive Space Heating Systems - Conclusions

Passive heating systems have demonstrated their ability to provide useful heat for space conditioning purposes. However, two objectives must be addressed in the design of a passive system.

The first objective is to reduce the heat loss to the exterior (particularly through the collection surfaces).

The second objective is to minimize temperature fluctuations within the occupied space and to maintain comfort conditions.

These objectives can be met when the following design components are considered:

- o To reduce night time heat losses, some method of day/night insulation must be used (e.g., insulating curtains, movable type insulations -- automatic or manually operated, etc.), or the use of an attached greenhouse as a buffer between the conditioned space and ambient temperatures.

- o Temperature variations can be limited by the use of:
  1. Optimum thermal mass which is not subject to excessive thermal losses to the exterior. (Such a thermally massive component could also improve performance of an active heating and/or cooling system), and
  2. Hybrid, active/passive system components which provide for improved heat distribution.

- Direct gain systems without intervening mass walls, greenhouses, or other buffers, caused unacceptable temperature variations for systems contributing more than 10 to 20 percent of the heating requirements.
- Designs must consider moisture accumulation and the need for reasonable levels of natural or forced ventilation. In general, health and comfort conditions require a minimum of one-half to one air change per hour.

The only major limitation of most passive solar systems is that such systems have to be integrated with the building structure. Unfortunately, retrofit installations on an existing building are difficult and, in most cases, not feasible (except for attached greenhouses).

5.3 PERFORMANCE EVALUATION OF ACTIVE HEATING SYSTEMS

5.3.1 Introduction

A wide variety of solar active space heating systems have been evaluated and their performance reported in Tables 29, 30 and 31. The level of performance in terms of durability, reliability, energy savings, and systems efficiency varied dramatically. These variations were due to:

- Distinct and wide variations in the solar system designs
- Major differences in the quality and thoroughness of installation and maintenance procedures
- Different heating demand profiles
- The experimental nature of many of the unique and innovative systems and/or components utilized

On average, the active systems performed at unexpectedly low levels. However, several systems which received careful attention to details of design, installation, operation, and maintenance performed quite well. The major distinction between the levels of performance of "successful" and "unsuccessful" systems is directly due to the degree of adherence to proper solar and HVAC design and installation procedures. In future applications all solar active heating systems can be expected to receive this same attention to detail in all phases of the application of a system to a particular building. On this basis, therefore, the performance of two well-designed and installed systems will be considered initially.

5.3.2 Achievable Performance Levels

5.3.2.1 Residential

System 61 of Table 31 achieved a solar collector efficiency of 33.6 percent, a system thermal efficiency of 31.7 percent, and a system overall efficiency of 42.9 percent. This air-heating collector and rock bed storage combination had system heat losses of less than 10 percent of the collected solar energy and a high useful heating to electrical operating energy input ratio (i.e., S = 17.7). The electrical operating energy per unit area of collector was only E/A = 27 Btu(electric)/ft^2 (307 kJ(electric)/m^2), as compared to an average of systems 51 through 62 of 31 Btu(elec)/ft^2 (353 kJ(elec)/m^2). The electrical usage as a fraction of useful heating was E/Q_u = 5.6 percent (the average for all 13 systms was 12.1 percent).

In a subsequent year of operation, system 61 achieved comparable results with a substantially modified system. The lower system overall efficiency of 34.9 percent (as opposed to 42.9 percent for the previous year) was due to a higher solar radiation level of 1733 Btu/ft^2 (19.7 mJ/m^2•day) (as opposed to only 1499 Btu/ft^2•day; 17 MJ/m^2•day). For example, we note that [(34.9%)(1733/1499)] = 40.3%. More importantly, the modified system achieved an energy savings of Q_s = 239,100 Btu/day (252 MJ/day) with 623 ft^2 (57.8 m^2) of collector (Q_s/A = 384 Btu/day•ft^2; 4377 kJ/m^2•day), while the previous design achieved an energy savings of Q_s = 263,000 Btu/day (277 MJ/day) with 722 ft^2 (67 m^2) of collector (Q_s/A = 364 Btu/day•ft^2; 4149 kJ/m^2•day).

The result of this performance is overall annual energy savings of about 117 million Btu per year (123 GJ/year) with 73 million Btu (77 GJ) of solar heat delivered to the space heating load, 8 million Btu (8.4 GJ) of solar heat delivered to the DHW heating load (October through May only), and 18 million Btu (19 GJ) consumed in operating the solar system. The resulting savings in non-renewable energy resources, such as natural gas (with an energy content of 1000 Btu/ft^3 of gas; 37.3 MJ/m^3) is 117,000 cubic feet (3311 cubic meters) of gas per year.

5.3.2.2 Commercial

System 62 of Table 31 achieved solar collector efficiencies of 32.1 and 33.2 percent (in subsequent years), system thermal efficiencies of 31.5 and 32.3 percent, and system overall efficiencies of 41.5 and 42.9 percent. This liquid heating and cooling system for a conference center and library had system heat losses of less than 15 percent of the collected solar energy for both years and high useful heating to electrical operating energy input ratio of about 16. While the heat losses were from equipment located within the conditioned space, only 80 to 87 percent of these heat losses are considered useful in meeting the heating load. The electrical operating energy usage was not excessive, E/A = 33 Btu(elec)/day·ft^2 (367 kJ/m^2·day) and E/Q_u = 6 to 6.5 percent.

Average energy savings of Q_s/A = 450 Btu/day·ft^2 (5136 kJ/m^2·day) of collector were obtained. This resulted in overall annual energy savings of about 1440 million Btu per year (1519 GH/year), 1080 million Btu (1139 GJ) of solar heat delivered to the space heating load and 240 million Btu(elec) (253 GJ) being consumed in operating the solar system. For a conventional natural gas heating system, the total savings in non-renewable energy is approximately 1.44 million cubic feet (40,750 cubic meters) of gas per year.

5.3.3 Demonstration Program Performance

5.3.3.1 First Year

Systems 41 through 49 of Table 29 provide data on the first year operations of systems designed and installed under the National Demonstration Program, with system thermal efficiencies of 2.0 to 18.5 (and with seven systems less than eight percent). The program cannot be called an initial success. Active heating system thermal efficiencies of less than 10 percent cannot be justified.

In systems 41 through 43, the low system thermal efficiencies resulted primarily from storage heat losses. The storage efficiency (energy delivered from storage to load divided by energy delivered to storage) averaged less than 30 percent. The uncontrolled losses in one building (system 42)had the effect of increasing the room temperature from 70 to 85°F (21 to 29°C). Thus while it appears that system 42 had an acceptable system thermal efficiency, only one-third of the useful collected heat was delivered to the heating load in a controlled manner.

The second major factor in the poor performance of systems 41 through 43 was reliability problems. These problems included control problems (pumps running continuously or at wrong times) and leakage through check valves, ball-float valves, and three-way valves. Storage units were typically oversized or undersized [43].

Systems 44 through 49 had room temperature variations of 62 to 84°F (17 to 29°C) (in some cases), excessive air leaks in ducts and/or storage units, inadequate or missing backdraft dampers, heating of cold storage by the auxiliary furnace, DHW subsystem freeze ups, restrictions to flow in rock bed storage units, and in some cases an inability to charge storage with collected heat.

In one subdivision, 24 new homes were built with solar active heating systems as part of the demonstration program. (These systems are not specifically listed in Table 29.) All of these systems had major control and excessive heat loss problems. For example, the system piping was not insulated. Table 59 shows the results of this oversight in 24 homes. Almost half of the collected energy was lost, resulting in an average system thermal efficiency of 11.4 percent. As shown in Table 59, the simple expedient of insulating the piping increased the system thermal efficiency to 24.3 percent.

5.3.3.2 Subsequent Years

The major improvement in system thermal efficiency of 24 solar homes, briefly discussed above, is an important aspect of the performance of active heating solar systems in the National Demonstration Program. For example, systems 52 through 57 in Table 30 represent second generation systems (for the National Demonstration Program). These systems had average system thermal efficiencies of about 15 percent, a very significant improvement. Nevertheless, some problems still existed.

System 57, for example, lost 47 percent of the collected useful energy from a 1000 gallon (3790 liter) buried storage. A large hot water usage, when combined with a small (30 gallon) solar DHW preheat tank resulted in a very low DHW solar heating fraction. This in turn caused cycling of pumps in the DHW preheat loop, which in turn interferred with the delivery of solar space heating. The space heating subsystem had been set to deliver heat whenever the storage tank temperature was greater than 75°F (24°C). Because of the DHW pump cycling, solar space heating was usually not delivered when the storage tank temperature was less than 105°F (40°C) [61].

Other problems included excessive snow (about 30 inches) on the collector array, resulting in a seasonal drop in collector efficiency of three percent (system 53) [60], severe duct leakage in system 55 [60], and no DHW heating (because of the building being unoccupied) in system 56 [60].

System overall efficiencies averaged slightly less than 17 percent. However, three of the systems ranged from 11.0 to 12.2 percent while the remaining three ranged from 19.1 to 25.8 percent. Two of the former systems used an excessive amount of electricity (i.e., E/Q_u = 20%) while the third combined excess electrical usage (E/Q_u = 15%) with exceptionally poor collector efficiency.

5.3.4 Other System Performances

System 51 had excessive storage heat losses (79 percent of Q_c) and excessive electrical operating energy usage (27 percent of Q_u), resulting in a low system overall efficiency of 9.4 percent.

System 59 had excessive heat losses (87 percent of Q_c) and excessive electrical operating energy usage (16 to 20 percent of Q_u), resulting in a low system overall efficiency of 11.2 to 13.4 percent (in succeeding years). In the first year of operation, duct losses alone accounted for 50 percent of the useful heat collected. "The performance in terms of total energy delivered is poor" [63]. The reasons for the poor performance include several specific problems [63]:

1. Heating demand was 21 percent lower than expected.
2. Additional storage insulation and space for air plenums resulted in a smaller (34 percent reduction) storage unit than originally planned. The smaller storage mass resulted in higher storage temperatures (usually reaching 145 to 150°F (63 to 66°C) on a good day), which in turn reduced collector efficiencies to 18-19 percent.
3. Duct losses in attic were excessive (heat loss was double that expected); installer was unable to correct leakage.
4. Storage heat losses caused overheating in fall and spring.
5. Fan electrical power was excessive (480 watts at six percent efficiency). Overall COP was 7.6 to 8.9 when the fan energy is included.

System 60 had acceptable system heat losses (14 to 17 percent of Q_c) and marginal electrical operating energy usage (10 to 13 percent of Q_u). Nevertheless the system overall efficiency was adequate (23.3 to 28.9 percent). The system utilized water/glycol in the collector and a 2000 gallon (7570 liter) hot water buried storage. The storage is of particular interest in that it consisted of a buried concrete septic tank insulated with eight inches (20 cm) of polyurethane. The storage heat losses were only 9.2 percent and 5.7 percent (in succeeding years) of the energy delivered to the storage. There were effectively no difficulties in the design (only in obtaining good measurements) [64].

The heat exchanger between the collector array and the storage tank in system 60 was considered to cause an approximate six percent decrease in Q_u and about two percent in collector efficiency [64].

System 58 had unacceptably high heat losses (35 percent of Q_c) and marginally acceptable electrical operating energy usage (13 percent of Q_u), resulting in a system overall efficiency of 20.5 percent. The heat losses were due in part to an improperly insulated buried storage and to the fact that the 2400 gallon (9080 liter) storage was oversized (3.5 gallons per square foot of collector (142 liters per square meter) as opposed to about 1.5 for more optimal sizing).

System 58 also included passive space heating components, including south-facing glass, interior insulated shutters, exterior fixed louvers, and a four-inch thick concrete floor slab for thermal storage. Room temperatures regularly varied from 68 to 79°F (20 to 26°C) during periods of high insolation. "This rapid overheating and cooling of the space is due to the lack of effective thermal storage for the passive space heating system" [62]. During the summer, "it was found that, for approximately 50 percent of the time, the space thermal conditions exceeded limits due to excessive relative humidity" [62].

5.3.5 Monthly Performances of Systems

Tables 32 through 35 provide monthly performance information on systems 61, 58, 50, and 62, respectively. Figure 20 is a plot of system overall and thermal efficiencies of the four systems. It is noteworthy that the residential and commercial systems 61 and 62 have consistently high system overall and thermal efficiencies, but that the residential and commercial systems 58 and 50 have serious degradation of performance in the fall and spring, particularly as denoted by the thermal performance. Note also that systems 61 and 62 have storage volume to collector area ratios of 1.5 and 1.3 gallons per square foot of collector (61 and 53 liters per square meter). (System 61 is an air system, the 1.5 gallons per square foot is the heat capacity equivalent.) Systems 58 and 50 on the other hand have ratios of 3.6 and 2.9 gallons of storage per square foot of collector (146 and 118 liters per square meter). Oversized storage provides for excessive storage heat losses under relatively light heating load demand. Table 65 lists the effectiveness of usage of the collected energy, i.e., Q_u/Q_c. This demonstrates the importance of proper sizing of systems to load and individual components within a system.

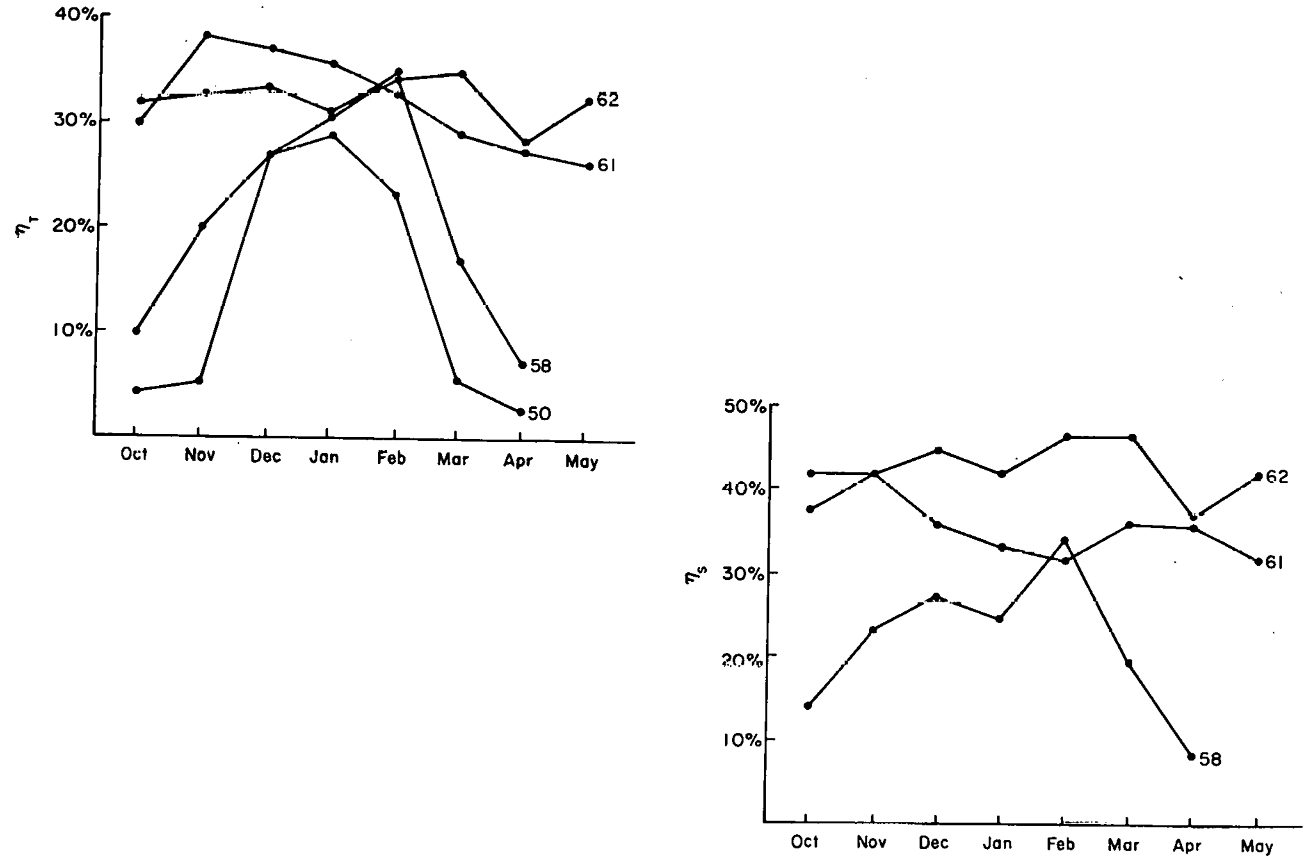


Figure 20. System Thermal and Overall Efficiencies - Monthly Values

Table 65. Q_u/Q_c (Percent)

System	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May
61	90	104	97	96	96	89	95	87
62	95	99	100	100	100	100	92	92
58	65	73	75	87	67	52	34	--
50	17	27	96	117	96	22	8	--

Q_u = Useful solar energy delivered to heating load

Q_c = Collected useful energy by array

5.3.6 Air/Liquid Systems Comparison

Two solar heating systems, located at the same site but on different buildings, offer some indication of the relative performance of solar heating systems utilizing air-heating and liquid-heating solar collectors and rock bed and hot water thermal storage units, respectively. Table 66 details some of the performance parameters for side-by-side air and liquid systems. The performance is based on four months of data (December through March) and incorporates the same performance criteria as used in Tables 29 through 31. Both systems use equivalent collectors (from the absorber plate up, flat-plate, single cover selective surface absorber), and equivalent collector array areas.

It should be noted that both systems have been substantially modified following the four months of performance data shown in Table 66. In both cases, the electrical solar operating energy usage was significantly reduced. For the air-heating system, major modification of the lower plenum of the rock bed storage reduced the pressure drop through storage by about 40 percent and was a major factor in reducing electrical usage. The liquid-heating system had major modifications to the piping and reduced the installed horsepower rating of all pumps to about one-third of the previous system. Other improvements included reduced heat losses and improved controls. Table 67 compares the month of February for the two designs of the liquid system.

Table 66. Air-Heating and Liquid-Heating Solar Systems Comparative Seasonal Performance [95]

		Air-Heating System	Liquid-Heating System
Q_c	Collected useful heat (Btu/day)*	343,700	348,400
η_c	Collection efficiency (%)	34.5	34.0
Q_u	Useful heating to load (Btu/day)*	319,300	301,100
η_T	System thermal efficiency (%)	32.1	30.2
E	Electrical energy usage (Btu/day)*	24,000	35,400
Q_s	Energy savings (Btu/day)*	439,900	365,700
η_s	System overall efficiency (%)	40.4	32.3

*Multiply Btu/day by 1.055 to obtain kJ/day

Table 67. Air and Liquid Systems Comparative Monthly (February) Performance [95]

		Air-Heating System	Liquid-Heating System	
		Original Design (1979)	Original Design (1979)	Improved Design (1980)
Q_c	Collected useful heat (Btu/day)*	362,200	378,900	482,900
η_c	Collection efficiency (%)	33.0	34.5	44.0
Q_u	Useful heating to load (Btu/day)*	340,100	318,000	366,600
η_T	System thermal efficiency (%)	31.0	29.0	33.4
E	Electrical energy usage (Btu/day)*	23,100	33,300	11,000
Q_s	Energy savings (Btu/day)*	478,100	402,100	568,700
η_s	System overall efficiency (%)	40.3	32.8	49.9

*Multiply Btu/day by 1.055 to obtain kJ/day

5.4 PERFORMANCE EVALUATION OF SOLAR COOLING SYSTEMS

5.4.1 Overview

The performance of solar cooling systems was in general very disappointing. Most of the solar cooling systems were net energy losers, two with exceptionally large net energy losses. Two systems (identified as systems 78B and 79 in Table 38), however, performed well and achieved significant energy savings. Many of the causes for the poor performance of the other systems were eliminated or substantially reduced in systems 78B and 79. Therefore these two high performance systems (one residential and one commercial sized), can serve as examples of the potential performance for all solar cooling systems.

Systems 78B and 79 had system thermal efficiencies of 12 and 12.4 percent. Their system overall efficiencies were 11.2 and 5.1 to 5.3 percent, respectively. The residential-sized system (system 78B) achieved an annual energy savings of approximately 16 million Btu/year (16.8 GJ/year), or about 0.03 million Btu/year•ft^2 (.34 GJ/m^2•year) of collector. The commercial-sized system (system 79) achieved an annual energy savings of approximately 130 million Btu/year (137 GJ/year), or about 0.02 million Btu/year•ft^2 (.22 GJ/m^2•year) of collector. In comparing these savings with various heating systems, it should be noted that the cooling seasons are only about three months long.

A detailed analysis of these systems is discussed below. However, because of the importance of several conclusions that arise from this more detailed discussion, they are presented here at the outset, i.e.,:

> Given the realistic potential improvements in the solar cooling systems 78B and 79 and the modified calculation of potential real energy savings by these solar systems, we may conclude that solar cooling systems can achieve savings in non-renewable energy resources of:

Potential Annual Energy Savings by Solar Cooling	Residential	Commercial	
Collector area (ft^2)	631	7705	(1)
million Btu/year	69.1	781.0	(2)
million Btu/year•ft^2	0.11	0.10	(3)
Ft^3 of natural gas	69,000	781,000	(4)

(1) Multiply by .0929 to obtain square meters
(2) Multiply by 1.055 to obtain GJ/year
(3) Multiply by 11.4 to obtain GJ/year•m^2
(4) Multiply by 0.0283 to obtain cubic meters

The six systems listed in Table 37 provide a clear indication of the reasons for the problems of those solar cooling systems which failed to achieve positive net energy gains. Specific problems on a case-by-case basis are delineated below.

Major causes of poor performance were:

1. Excessive solar system heat losses, including:
   a) Thermal storage losses to interior
   b) Thermal storage losses to ground (buried storage)
   c) Thermal storage losses to exterior
   d) Piping and other component heat losses
2. Excessive electrical energy consumption for operating solar system
3. Improper control strategy for operating solar system
4. Poor integration of collectors with specific system and/or cooling requirements
5. Poor chiller performance caused by a lack of consideration of the chiller's internal control system

5.4.2 System 71

The major factor in system 71 was the excessive heat losses from the thermal storage. The unit was buried and improperly insulated, resulting in an effective R-value of the storage of R = 2.7 hr•ft^2•°F/Btu (effectively the ground U-factor only). This resulted in 49 percent of the collected solar energy being lost by storage.

Table 39 shows some monthly data for system 71. Of particular interest is that July with the lowest insolation also resulted in the best monthly system thermal performance for the season. The reason is clearly due to the relatively lower heat losses during that month.

This in turn resulted from a better correlation between solar availability and cooling demand (which reduced the time that useful heat was in storage and thus reduced the heat losses). Note that the electrical energy usage was not significantly reduced from June and August levels.

Buried hot storage (high temperature 100 to 200°F; 38 to 93°C) is normally a poor design choice, particularly for cooling (buried chilled water storage is probably acceptable). Table 68 provides an estimate of the temperature differential, ΔT, between the operating temperature of storage and the ambient for different storage locations and seasons. For a given insulating value of the storage insulation, the amount of heat losses are directly correlated with ΔT. We can therefore compare the heat losses from an exterior (i.e., above ground) storage with that from buried storage by:

$$\frac{\text{Exterior above ground storage heat loss}}{\text{Buried storage heat loss}} \approx \frac{105°F}{135°F} = 0.778$$

Thus the exterior storage has about 75 to 80 percent of the heat losses that would result from a buried storage. Heat losses from a buried storage will depend, of course, on ground conditions. Underground dry chambers may offer a realistic alternative.

Table 68. Temperature Differentials, ΔT, Between Storage and Ambient for Different Storage Locations and Seasons

	Cooling Season		Heating Season	
	Ambient Temperature T_a* (°F)	Operating Temperature of Storage (°F)	Ambient Temperature T_a* (°F)	Operating Temperature of Storage (°F)
T_{min}		175-195		100-150
ΔT (exterior storage)	80-90	95-115	20-40	60-130
ΔT (interior storage)	70	105-125	70	30-80
ΔT (buried storage)	50-60	125-145	40-50	50-110

*Ambient temperature of material surrounding storage
°C= (°F-32)/1.8

It should be noted that system 77, which had positive energy savings, also utilized a buried storage. In this case the storage tank was a vertical cylinder with a capacity of 3000 gallons (11,355 liters) and filled to 2800 gallons (10,600 liters) and was insulated with sprayed-on urethane foam insulation on top and sides and an unknown amount of "Gilsulate" granular poured-in-place insulation beneath [71]. Initially this storage lost 20 percent of the collected solar energy but this value was later reduced to less than nine percent. However, heat losses from the piping between the buried storage and the chiller added an additional 12 and 4 percent, respectively, heat loss of the useful collected heat. These total heat losses are therefore still significant (albeit not necessarily disastrous). Had the storage been located above ground (exterior), the ΔT between storage and ambient would have been decreased from about 120°F (66°C) for the buried storage to about 90°F (50°C) for the above ground exterior storage. This would have increased the system overall efficiency for system 77B from 3.2 to 3.5 percent.

An important distinction between systems 71 and 77 is that 71 is located in Florida, with a high water table, and 77 is located in Arizona in a very arid region.

System 71 also had a heat rejection mechanism for instances when the storage temperature exceeded 220°F (104°C). This system was apparently seldom used.

The end result was that system 71 had a thermal system efficiency of 7.7 percent. It is highly unlikely that a solar cooling system can be expected to be technically feasible if the thermal system efficiency is less than 10 percent. Because system 71 utilized an excessive amount of electricity (i.e., E/A for system 71 was approximately 161 Btu(elec)/day·ft^2 (1835 kJ/m^2·day); the system overall efficiency was negative. The electrical usage per square foot of collector was over five times the average for active heating systems (31 Btu(elec)/day·ft^2; 353 kJ/m^2·day), while other cooling systems averaged E/A values of 54 Btu(elec)/day·ft^2(615 GJ/m^2·day) (neglecting the clearly excessive values of 529 and 576). It is noteworthy that one system had an E/A value of 26.6 Btu(elec)/day·ft^2 (303 kJ/m^2·day).

"All measured electrical appears high" [68]. This may in fact be true and may be due to faulty measurements (wattage appears far too high for horsepower). This residential system utilized for its various pumping requirements:

Pump	Flow Rate	Pumping Horsepower[3]
Collector	16.6 gpm [1]	1/4
Load (storage to chiller)	10.4 gpm [1]	1/6
Cooling tower	10.2 gpm [1]	1/6
Chilled water	7.2 gpm [1]	1/6
Chiller	--	1/4
Fan	1100 cfm [2]	1/3
Cooling tower fan		1/3
		5/3 hp

[1] Multiply by .063 to obtain l/s
[2] Multiply by .472 to obtain l/s
[3] Multiply by .746 to obtain kW

Neglecting the fan power this constitutes 4/3 hp (.99kW), which can be compared to a slightly smaller system (system 78 with 631 square feet (58.6 m^2) of collector versus 714 square feet (66.3 m^2) of collector for system 71) which had a total horsepower of 0.74 (.55 kw) (a conventional 3-ton heat pump might utilize 3 to 4 horsepower; 2.2 to 3 kW). The electrical usage of the two systems was 115,000 and 16,800 Btu(elec)/day (121,360 and 17,730 kJ/day). This would imply that the reported electrical usage [67,68] is either too high or that the system controls were not optimized properly or a combination of the two reasons.

In any case, the combination of excessive heat losses and excessive electrical energy usage reduced the energy savings of system 71 to a negative value and an overall system efficiency of -19.1 percent. Had the electrical usage been reduced to one-fourth of the actual value (which seems possible), the overall system efficiency would have been +2.0 percent.

5.4.3 System 72

System 72 in 1978 had an extremely low thermal system efficiency of only 1.9 percent. This was due to excessive heat losses, poor collector performance, and a low COP of the solar chiller. The excessive heat losses were due in part to the large storage unit (10,000 gallons, 37,850 liters) in relation to the collector area (3840 ft^2; 357 m^2). This represents a storage volume to collector area ratio of 2.6 gallons per square foot (.106 l/m^2) (as compared to more typical values of 1.0 to 1.5 gallons per square foot).

The collector efficiencies of 14.8 and 12.0 percent were due to the use of concentrating collectors in a low to intermediate temperature application (120-200°F; 50-90°C). The collector daily operating efficiency (defined as the useful energy collected, divided by the solar radiation incident during the period when the collector pump was operating) was 47.6 and 43.9 percent over a two year period [69,70]. However, the inability of the concentrating collector array to utilize effectively: (1) the diffuse radiation components and (2) the early morning and late evening direct radiation (causing significant end losses) caused the substantial reduction in collector daily performance.

The poor chiller performance experienced was virtually eliminated by the second cooling season. It should be noted that the solar system provided about 31 percent of the cooling load, utilizing both hot and cold storage. This provides for optimum use of the solar energy available for meeting the cooling load.

5.4.4 Systems 73 and 74

Systems 73 and 74 had good-to-excellent average daily collector efficiencies and excellent heat loss to collected energy ratios (i.e., $(Q_E+Q_I)/Q_c$ = 9.8 and 7.3 percent, respectively). In system 73 the reduced losses may be attributed in part to the use of a 6000 gallon (22,700 liter) hot storage (1.64 gal/ft^2 of collector) and a 2000 gallon (7570 liter) cold storage (both insulated with four inches of urethane foam). In system 74, there was no storage and all losses were piping and other component-related. The non-use of storage, however, tends to result in considerable mismatch between solar heat availability and cooling demand, which in turn leads to unproductive cycling of the cooling unit.

The 25 and 100 ton, respectively, chiller performances, however, indicated an average COP of the chillers of about 0.34 to 0.37. In many periods the chillers achieved COP's of 0.65 to 0.72 [70]. The poor average performance was in general due to low condensing water temperatures and a lack of control to account for this effect.

An absorption cycle chiller contains two internal circulation loops (refrigerant and absorbent) and three external circulation loops (hot water, chilled water and condensing water). Most absorption chillers contain internal controls which monitor the temperatures of these external circulation loops and control the concentrations and, to a lesser extent, the flow rates of the internal loops. This internal control system is used to maximize the chiller performance and to prevent damage to the chiller when these temperatures rise above or fall below the chiller's operating range. While most system designers recognize the minimum temperature requirement of the energy supplied to the generator of the chiller, many do not recognize the effects on the chiller's performance due to variations in the temperatures of either the chilled water or the condensing water.

Table 69 shows the effects of low condensing water temperature on the COP of the chiller in system 74 [70]. As can be seen from the table, the COP of the chiller was below 0.5 until noon (due to either cycling or degradation of the unit), even though the hot water temperature was approximately 190°F (90°C) and the temperature of the chilled water exiting the evaporator

Table 69. System 74 Absorption Chiller Performance (March 5, 1979) [70]

Hour Ending	Cooling Produced (tons)*	COP	T_g	(T_c)	T_t
8 am	5.2	0.05	180	49	68
9 am	7.8	0.07	188	53	72
10 am	19.2	0.18	190	53	75
11 am	35.5	0.37	189	56	77
12 noon	41.1	0.49	190	60	84
1 pm	44.6	0.55	190	61	84
2 pm	46.6	0.53	190	61	83
3 pm	43.4	0.51	190	61	83
4 pm	43.5	0.52	189	60	83
5 pm	42.1	0.59	189	60	83
6 pm	34.2	0.52	188	59	82

T_g = Temperature to generator
T_c = Chilled water
T_t = Condensing water temperature
°C = (°F-32)/1.8
*Multiply tons by 3.52 to obtain kW

was greater than 50°F (10°C). It is important to note, however, that the condensing water temperature was below 80°F (27°C) until noon. The internal controls for this chiller monitor the condensing water temperature and by-pass the absorbent around the absorber if this condensing water temperature is less than 80°F (27°C),, thus causing a significant reduction in the amount of cooling produced. Under the full 100 ton load conditions, sufficient would be rejected by the cooling tower in the first hour of operation to quickly raise the condensing water temperature to 80°F (27°C). However, under the partial load conditions caused by operating the chiller at below the design hot water loop temperature of 230°F (110°C), four hours were required to raise the condensing water temperature to above 80°F (27°C). The performance of this chiller can be increased by preventing the condensing water temperature to remain below 80°F (27°C) for this extended period [70].

Systems 73 and 74 also experienced disastrously high solar operating electrical energy requirements. The E/A values were 529 and 576 Btu(elec)/day·ft^2 (6030-6566 kJ/m^2·day), respectively, which are about twenty times greater than the best system reported in Table 41, and about ten times higher than reasonable. It is noteworthy that both systems had relatively high system thermal efficiencies (coincidentally both were 11.4 percent). The electrical usage, however, resulted in these two solar systems using more electrical energy than the total solar energy delivered to cooling.

Had the chiller controls been properly integrated into the system in order to maintain an average COP of about 0.6, and the E/A values reduced to about 50 Btu/day·ft^2(570 kJ/m^2·day), the system overall efficiencies would have been increased to 12 to 14 percent! In addition, with a better chiller, COP ≈ 0.65 to 0.8 could be obtained.

5.4.5 System 75

System 75 had poor collector performance, excessive heat losses (about 40 percent of Q_c) and poor chiller performance. The result was a system thermal efficiency of only 1.7 percent. The E/A value of 45.5 Btu(elec)/day·ft^2 (519 kJ/m^2·day) was acceptable.

The poor collector performance may be due to inadequate filling of the collector array (see section on collector array performance) [24]. The excessive storage heat losses are due to a slight oversizing of storage (20,000 gallons (75,700 liters) for 11,000 square feet (1022 m^2) of collector area) and poor insulation.

5.4.6 System 76

System 76 had adequate collector efficiencies, excessive heat losses (41 percent of Q_c), and good chiller performance. The storage heat losses resulted from oversizing of storage (30,000 gallons (113,550 liters) for 12,660 square feet (1177 m^2) of collector, i.e., 2.37 gallons per square foot (97 liters per square meter)!) and poor insulation. Temperature differences of 70°F (39°C) have been achieved in the storage (due in part to different rates of heat losses) but also due to "short circuiting" of the flow through the tank.

5.4.7 Systems 77 and 78

Systems 77 and 78 are residential-sized solar cooling installations and provide excellent examples of improved performance with modified system designs for the same installation. Systems were initially operating properly but project staffs determined to improve the performance (and in fact did). It is particularly noteworthy that both systems utilized the identical collector arrays (i.e., without any modifications) in each of the respective system designs. System design improvements caused collector efficiencies to increase from 12.4 to 18.0 percent (with a 6.5 percent increase in solar radiation - system 77), and from 29.2 to 33.9 percent (with a 2.2 percent decrease in solar radiation - system 78). The relatively low performance of the solar collector array of system 77 may be due to the fact that it was assembled in place as a combination roof/collector system and that it was built in 1975, prior to significant inmprovements in collector array technology.

Improvements in system efficiencies (thermal and overall) due to reduction in solar system heat losses are also noteworthy. System 77 reduced its heat losses to collected energy ratio from 34.3 to 13.7 percent and had a subsequent increase in system thermal efficiency of from 5.7 to 8.2 percent. System 78 had a less significant decrease in heat losses (25.9 percent of Q_c to 21.2 percent of Q_c) but reduced the interior heat losses from 15.2 percent of Q_c to 6.2 percent of Q_c. Because of the requirement for the solar system to remove system heat losses from the conditioned space, this significant reduction in interior heat losses was a major factor in improving the system thermal efficiency from 8.7 to 12.0 percent.

It should be noted that both systems 77 and 78 suffered significant reductions in chiller performance. In system 77 the two original units were replaced by two improved units (by the same manufacturer) in the summer of 1978. The lower operating temperature requirements of the new units allowed for about 75 percent more energy collection. This was due to the fact that lower temperature requirements imply lower heat losses (which was also aided by substantially reduced piping), which in turn led to an energy savings increase of sixfold. The lower chiller COP was apparently due to non-condensible gas accumulation and some overfiring (input temperatures to generator of chiller of 185°F (85°C) instead of 170°F (77°C)) [71]. "Steady-state measurements made in late August 1979 indicate that a COP of 0.75 is regularly obtainable when the proper input temperatures are maintained" [71]. Such an improvement would increase the system overall efficiency from 1.8 to 6.2 percent.

Electrical energy usage was fairly consistent for system 77 (rising slightly from E/A = 52.3 to E/A = 55.1 Btu(elec)/day·ft^2; 596 to 628 kJ/m^2·day) but had a very pronounced improvement in system 78 (reduced from E/A = 84 to E/A = 26.6 Btu(elec)/day·ft^2; 957 to 303 kJ/m^2·day) System 78's very low E/A value is a primary factor in its high system overall efficiency.

5.4.8 System 79

System 79 had two years of fair collector performance, moderately limited heat losses (16.5 to 17.9 percent of Q_c), excellent chiller performance (for both a lithium bromide absorption chiller and a Rankine cycle compression unit), and electrical operating energy requirements which had an E/A value of 60 Btu(elec)/day·ft^2 (684 kJ/m^2·day).

System 79 includes a 5000 gallon (18,900 liter) hot storage and 10,000 gallon (37,850 liter) cool storage (which is converted during the heating season to 10,000 gallon hot storage). The ratio of hot storage volume to collector area is 0.65 gallons per square foot (26.4 l/m^2), a particularly interesting number. The chillers include an 85 ton lithium bromide absorption unit and a 77 ton Rankine cycle unit. The system also includes night evaporative cooling of the large cool storage tank and heat recovery for fresh air during the heating season.

Table 70 details the relative performance of the absorption and Rankine chillers. Common to both units was the excessive electrical energy requirement. "The big energy consumers are the collector pump and cooling tower (67 percent of the total). By changing the collector fluid from paraffinic oil to glycol/water and by using a cooling tower with a propeller type fan instead of a squirrel type fan and placing it outside at ground level, the power of these three items could be reduced by at least 50 percent. The system COP would then be increased to over 5.0 (from 3.6)" [66].

Table 70. Chiller Performance and Parasitic Power Requirements [66]

1979 Cooling Season

		Absorption Unit	Rankine Unit
1.	Hot water input	132.6 GJ	112.5 GJ
2.	Chilled water output	86.6	78.6
3.	Collector pump	7.51	7.56
4.	Heat exchanger pump	0.67	0.69
5.	Collector hot water pump	3.27	3.09
6.	Chilled water pump	1.29	1.15
7.	Cooling tower pump	3.30	3.34
8.	Cooling tower fan	5.04	4.55
9.	Chiller power	2.73	2.37
10.	Total power	23.82	22.76
	COP (chiller)	0.658	0.699
	$S_c = Q_u C/E$	3.635	3.455
	S - Modified by replacing paraffinic oil with water/glycol, i.e, reducing items 3, 7, and 8 by 50 percent	5.35	5.23

The use of paraffinic oil as the collector fluid may also be responsible for the relatively poor collector performance.

Table 40 shows monthly performance values. Note that August has a better than average collector efficiency (24 percent), heat losses of only 15.5 percent of Q_c, a subsequent system thermal efficiency of 14.4 percent, E/A = 66 Btu(elec)/day·ft^2 (752 kJ/m^2·day), and finally a system overall efficiency of 6.9 percent. Note also that July and August had virtually identical solar insolation but significantly different efficiencies.

5.4.9 Comparison with Conventional Cooling Systems

5.4.9.1 Residential

Utilizing system 78B as an example, we can estimate the potential performance of a solar residential cooling system and compare this performance to a conventional cooling design. Design modifications to the system include: the improvement of chiller performance by utilizing better control techniques, the elimination of all interior heat losses by removing the system component to the exterior, reducing these heat losses by 10 percent (achievable in part by the relocation of the system components to the exterior), and reducing electrical energy requirements (by reducing piping runs to cooling tower and eliminating heat exchanger between storage and collector). These modifications and the resulting change in parameters is shown in Table 71.

Table 71. Comparison of Observed and Predicted Solar Cooling Performance, Systems 78B and 79

		Residential		Commercial	
		Actual System Observed	Ideal System Predicted	Actual System Observed	Ideal System Predicted
IA	Solar insolation times collector area (million Btu/day) *	1.00	1.00	14.5	14.5
Q_c	Collected solar energy (1000 Btu/day)*	339	339	3033	4640
η_c	Collector efficiency (%)	33.9	33.9	20.9	32.0 @
Q_E	Exterior heat losses (1000 Btu/day)*	50.9	70.0	544	832
Q_T	Interior heat losses (1000 Btu/day)*	21.1	--	--	--
Q_u	Useful heat to chiller (1000 Btu/day)*	227	269	2489	3808
C	Chiller coefficient of performance	0.527	0.65	0.69	0.72
η_T	System thermal efficiency (%)	12.0	17.5	12.0	18.9
	Collector pump energy (1000 Btu/day)*	4.0	2.0	159.0	80.0
	Circulating pump energy (1000 Btu/day)*	2.7	2.7	64.0	64.0
	Cooling subsystem (1000 Btu/day)* (chilled water and cooling tower pumps and cooling tower fan)	10.1	6.5	240.5	0.0 (assumed to be the same as conventional)
E	Electrical solar operating energy (1000 Btu/day)*	16.8	11.2	463.5	144.0
S	Solar useful heat to cooling unit/ electrical usage	7.1	15.6	3.8	19.0/12.5**
C_c	Conventional unit coefficient of performance	--	1.8	--	--
η_A	Conventional unit efficiency ($\eta = C_c\eta_E$)	0.65	0.47	0.65	0.72
Q_s	Energy savings by solar (1000 Btu/day)* ($Q_s = Q_u/\eta_A - E/\eta_E$)	119.5	328.9	859.6	3245.2
η_s	System overall efficiency (%)	11.2	31.5	5.3	21.6
Number of days in cooling season (day/year)		135	210	150	240
Annual energy savings (million Btu/year)*		16.13	69.1	128.9	781.0
Annual energy savings/collector area (million Btu/year·ft^2) @@		.026	0.11	.017	0.10

** See text
@ See text
* Multiply Btu/day by 1.055 to obtain kJ/day
@@ Multiply Btu/year·ft^2 by 11.4 to obtain kJ/year·m^2

5.4.9.2 Commercial

System 79 can also be modified slightly in order to show the potential performance of a solar commercial-sized cooling system. The investigator has already noted that the replacement of the collector heat transfer liquid (a paraffinic oil) with a water/glycol solution would cut the collector pump electrical energy requirements by half [66]. We should also expect a major improvement in collector performance. Because the residential and commercial chillers have similar input temperature requirements and the climatic factors are similar (system 79 has 19 percent more solar radiation than system 78B), we will assume a collector efficiency of 32 percent. Note that system heat losses will also probably increase by an equivalent amount.

In comparing the solar and conventional cooling systems we must realize that the conventional unit for a commercial system is normally an absorption chiller and thus has the same energy usage in the chiller, cooling tower, etc. Therefore, because energy savings is the difference between conventional and solar energy usage, this component of electrical energy usage can either be deleted (for comparison purposes) or incorporated into the conventional unit's efficiency. For simplicity we will delete the energy requirements for the chiller (i.e., the chilled water pump, cooling tower pump, cooling tower fan, and chiller power).

The results of these modifications are shown in Table 70.

5.4.9.3 Conclusion

We note that our assumptions of potential improvements have yielded approximately equal values of Q_u/AI (i.e., 26.9 and 26.3 percent for the residential and commercial systems, respectively). This is in accordance with the fact that these two collection systems should be approximately equal in efficiency. These values are also in line with the efficiencies of well-designed solar active space heating systems (i.e., Q_u/AI for selected active heating systems was 22.0 to 32.3 percent). When the chiller's coefficient of performance is taken into account, the thermal efficiencies of the cooling systems are then reduced to 17.5 and 18.9 percent.

The substantial reduction in electrical solar operating energy usage provides for substantial increases in the ratio of solar useful heat to cooling unit to electrical solar operating energy, S. The residential system effectively doubled S (7.1 to 15.6), while the commercial system had an even more dramatic improvement. However, for the commercial system the chiller power should be included. After some reduction by replacement of the type cooling tower fan, we would expect a value of S for the commercial system of about 12.5.

The more noteworthy effect is the combined improvement due to improved thermal performance and reduced electrical energy usage. The overall energy savings for the residential system was nearly tripled (i.e., from 119,500 Btu/day to 328,900 Btu/day; 126,000 to 347,000 kJ/day). For the commercial system the overall energy savings was nearly quadrupled (i.e., from 859,600 Btu/day to 3,254,200 Btu/day; 906,900 to 3,433,200 kJ/day). The system overall efficiencies show the result.

Because of the limited cooling degree days at the two locations of systems 78B and 79, the annual energy savings per square foot of collector area is significantly smaller than that for the various heating systems. If we assume, however, a more substantial period of cooling requirements (typical of southern, moderate altitude locations), we obtain annual energy savings per unit area of collector of 0.11 and 0.10 million Btu/year·ft^2 (1.25 and 1.14 GJ/year·m^2) of collector for the residential and commercial systems, respectively. For natural gas with an energy content of 1000 Btu per cubic foot (37,300 kJ/m^3) of gas, the residential and commercial systems' annual savings in natural gas were 69.1 and 781.0 thousand cubic feet (1,955 and 22,102 m^3) of gas/year, respectively.

5.4.10 Solar Cooling Systems - Conclusions

In order for solar cooling systems to be technically and economically feasible, all major factors listed in Section 5.4.1 must be accounted for. Design requirements for any solar cooling design must include: Minimizing of heat losses and electrical energy operating requirements, intelligent use of internal and system controls, proper integration of all components with system and load requirements, and carefully designed control systems (in order to maximize the chiller and array performance).

Electrical usage should be limited for solar system operations to values of E/A not to exceed 60 to 70 Btu(elec)/day·ft^2 (684 to 798 kJ/m^2·day) and preferably in the range of 20 to 40 Btu(elec)/day·ft^2 (228 to 456 kJ/m^2·day).

Integrating the various solar components into a complete solar cooling system must be done with care. The use of concentrating collectors with lithium bromide absorption chillers has not necessarily resulted in the most appropriate integration of components. System 72 (see Table 37) achieved some of the lowest collector efficiencies, and the concentrating collectors used in the system described in Table 77 were responsible for the zero solar cooling in that system. It is noteworthy that one concentrating collector (without a defocus control) resulted in the thermal fluid catching fire [8].

In discussing systems 72 through 76, one report noted that "The factors which cause the majority of the operational problems are ... caused by inexperience in integrating components into a solar energy system. This integration requires a control system which must be designed using an overall system approach to ensure the system efficiency is maximized" [70].

It is noteworthy that system 79 used a flat-plate collector array to operate a Rankine cycle cooling system at reasonable system efficiency.

Solar cooling system overall efficiencies of 20 to 35 percent are attainable when careful attention to design is a prerequisite.

APPENDIX A
Management and Logistics

Reference: U.S. Department of Energy, "Solar Heating and Cooling Project Experiences Handbook", Preliminry issue. Prepared under contract number EC-78-C-01-4131

1. A management plan to complete the project in all phases should be prepared. It should define areas of responsibility of all team members.

2. Assign one project engineer or supervisor to be responsible for the complete program. The same supervisor, starting with the initial concept and continuing through operation, should demonstrate strong technical and administrative skills, thus eliminating many of the problems observed in previous projects. Do not rely upon scattered members of a team to provide for project management. Engage a firm technically knowledgeable and who can assume full responsibility for project control.

3. Attempt to obtain firm contractual bids from all bidders. Maintain open communication during any bidding process to ensure proper understanding with the bidding contractors.

4. Utilize local micro-climatic data in design, if possible.

5. Consider the code requirements for conventional systems when interfacing with solar-assisted systems. To avoid later delays or misunderstandings, consult with insurance underwriters during the design phase.

6. Review project summaries showing previous design concepts.

7. Plan on developing realistic cost estimates and cost trade-off studies on centralized versus unitary systems.

8. Obtain site approval from proper authorities in early phase of design. Evaluate thoroughly initial project site details for minimum site preparation cost.

9. Consider shop labor assembly of some system components as opposed to field installation of all components as a means to reduce cost.

10. Control costs by keeping up-to-date on engineering changes and their effect on both performance and budget. Conventional design/construction procedures have evolved from cost-effective considerations. Deviations or delays are nearly always very costly. Major overruns can and will occur unless a very disciplined approach to completing the project is followed.

11. Consider the severe environmental conditions that exist near salt water. Particular attention should be paid to extra corrosion protection for metal support structure, assembly hardware, external electronics, and exposed metal portions of the collector as well as buried metal pipe. Requirements for special pre-assembly treatment as well as post-assembly treatment should be completely evaluated. Address potential freeze protection as required by local conditions at conceptual design.

12. Evaluate the installation experience of other installers of the specific collector under consideration. Cost saving procedures and problems on a workman's level are often available.

13. Provide the installation crew with detailed installation instructions for the collector and support structure. The instructions must be understandable and, if possible, provisions should be made for "go/no go" checks. Special hardware should be considered as a means to avoid misalignment. Corrective action can be taken in early construction phases if detected by a competent construction/installation review. Due to language and site location problems, do not assume that conventional "installation manuals" are in all cases adequate.

14. Provide in the project management structure for adequate and professionally competent engineering review of the construction progress and quality. Adverse comments on construction quality or progress must be resolved quickly. Do not permit cost-effective corrective recommendations by the design engineer to go uncorrected by construction personnel.

15. Require adequate supervision during handling and installation of collectors. Provide in a management plan adequate provision to handle shipping damage to collectors.

16. Assure during the design phase that all installation requirements meet local codes.

17. Specify that the equipment supplier will provide complete instructions for mounting equipment. All information necessary for installation should be readily available from the supplier. Where the collectors are an integral part of the roof, responsibility for tightness of the roof should be a part of the same contract.

18. Consider advantages of preassembling groups of collectors and hoisting to roof by crane or helicopter.

19. Maintain a proper inventory of spare parts to preclude partial shutdown of collector arrays.

20. To minimize costs, consider:
    (a) Bidding collector subsystem as a separate package
    (b) Including job performance penalty in contract for failure to meet time schedule
    (c) Require testing before approval
    (d) When estimating cost of construction, be aware of the possibility of charges resulting from concern about uniqueness of project or "fear factor" of solar.

21. Plan to provide maintenance and operation manuals for all systems. Have a plan to periodically verify proper system operation and correct as necessary.

22. Coordinate carefully all requirements imposed by owner.

23. When working with local, state and federal governments, be aware of all the contract and working requirements. Do not establish unrealistic project milestone or schedule dates which do not allow for interaction between various team participants such as subcontractors, etc. Read and carefully understand the requirements of contractual agreements. Do not rely upon past experiences to provide information regarding present contractual requirements. Ask questions of the responsible agency with regard to any question of interpretation. Do not rely on hearsay.

24. Respond to design review action items in a timely manner to avoid delays in design construction.

25. Where aesthetic considerations are a serious concern, architectural, electrical, mechanical and solar contractor teams should work together to achieve a design acceptable to all parties.

APPENDIX B

EXAMPLES OF SPECIFIC PROBLEMS ENCOUNTERED AND CORRECTIVE ACTION TAKEN

Obtained from

NASA Marshall Space Flight Center, Alabama

SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00104A

SUBJECT: Freeze-Up of a Drain-Back System

LOCATION: North Georgia Area Planning & Development Commission, Dalton, Georgia

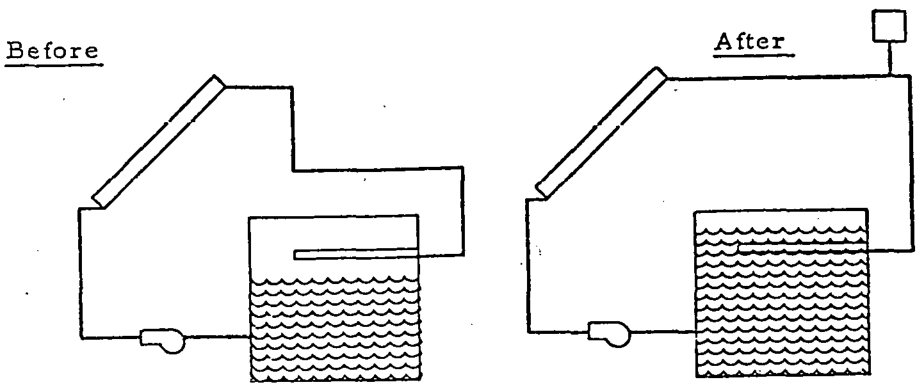
APPLICABILITY: System Designed Without Vacuum Breakers

PROBLEM: A freeze-up was experienced on this project when water vapor was drawn into the system and condensed. This condensation accumulated then froze before being allowed to drain back to the storage tank. The results were fifty failures (split copper lines) in supply, return and collector passage ways. The water vapor was allowed to enter the system from the top of the storage tank and the isolation valves, at the ends of the collector rows, were uninsulated allowing the vapor to freeze.

CORRECTIVE ACTION:

1. Vacuum breakers installed at the end of each collector row to allow ambient air into the system instead of water vapor from the storage tank, when the system is drained.
2. The isolation valves at the end of each row in the supply and return lines were insulated to protect against freezing in this area.
3. The water level in the tank was raised until the return was below the top of the water.

REMARKS:



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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00100B

SUBJECT: Separation of Glazing from Collector Housing

LOCATION: Huntsville Senior Citizens Center, Huntsville, Alabama

APPLICABILITY: Suncatcher, Model H2, Flat-Plate Collector
Manufactured by Solar Unlimited

PROBLEM: The Model H2 collector is approximately 26 ft. in length and approx. 2 1/2 feet wide. In this installation, the collectors are mounted at a tilt angle of 60° with the 26 ft. length in a vertical orientation. The collector housing is a light weight concrete shell. Glazing consists of a single pane of low iron tempered glass. Separation of the glazing from the collector housing has been experienced in a significant number of the collectors installed. The separation occurs at the bottom of the collectors resulting in thermal losses and moisure entering the collector.

CORRECTIVE ACTION:

A silicone adhesive has been applied by the manufacturer to re-seal the glazing to the collector housing.

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SOLAR HEATING AND COOLING
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ADVISORY CIRCULAR

April 19, 1979
AC00101A

SUBJECT: Hydronic Heating Coil Freeze-Up

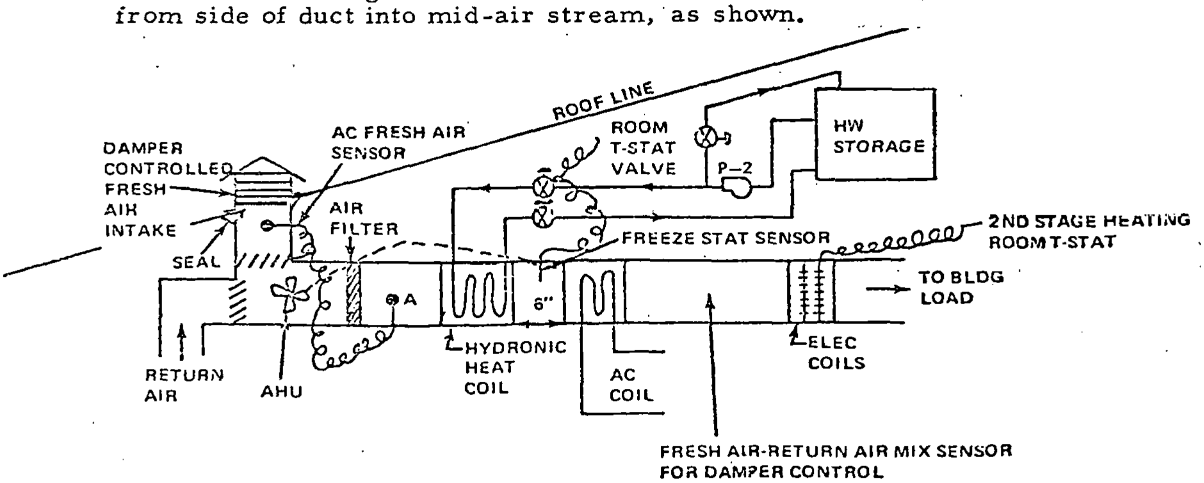
LOCATION: Telex Communications, Blue Earth, Minnesota

APPLICABILITY: Space Heating--Improper Solar System Integration With Conventional Air Conditioning (Existing) System

PROBLEM: Improper systems integration between the retrofitted solar-fired space heating system and the conventional vapor compression air conditioning system resulted in both systems operating at the same time. The freeze stat sensor located between the heating and the air conditioning coil was designed to turn off the air handling unit (AHU) blower when the temperature falls to 35° F. If the outside air dampers failed to close, this would protect the hydronic coil from freeze-up. The AHU blower was off and the room T-stat valve failed in the closed position and with the proximity of the freeze stat sensor to the nearby AC coil and with it still cooling, the heating coil froze and ruptured. (See Diagram below.)

CORRECTIVE ACTION:

1. Air mix sensor moved upstream of hydronic heating coil to point A.
2. Room T-stat valves changed to fail-open rather than closed in order to circulate warm water from storage (pump P-2 runs continuously) to hydronic coil.
3. Air conditioning and heating systems interlocked so both cannot operate simultaneously.
4. Freeze-state sensor inter-connected to close outside air damper.
5. Outside air ducting at roof line sealed and air conditioning fresh air sensor moved from side of duct into mid-air stream, as shown.



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SOLAR HEATING AND COOLING
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April 19, 1979
AC00102A

SUBJECT: Heat Exchanger (Collector Loop) Freeze-Up

LOCATION: Mt. Rushmore National Memorial, South Dakota

APPLICABILITY: Leaking 3-Way Diverter Valve (Inlet Loop)

PROBLEM: Instrumentation data indicated that the two temperature probes on the collector side of the heat exchanger registered near 32° F during the night of a severe winter storm. Subsequently, the mechanical contractor disassembled the heat exchanger on site and found the water end cover plate and two tubes cracked from freezing. All data points indicated a leaking 3-way diverter valve which resulted in thermosyphoning of the antifreeze (50% glycol) through the exchanger during the extreme cold weather.

CORRECTIVE ACTION:
The mechanical contractor replaced the cover plate and brazed the water tubes to repair the heat exchanger. From past experience with the 3-way diverter valves, the solar designer recommended putting a positive solenoid valve closure in the collector loop above the 3-way valve to be actuated by the control system to preclude thermosyphoning.

REMARKS:

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SOLAR HEATING AND COOLING
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April 23, 1979
AC00105A

SUBJECT: Collector Pump Control Sensor Location

LOCATION: Stephens College, Columbia, Missouri

APPLICABILITY: Active Liquid Solar Systems

PROBLEM: The collector temperature sensor in the glycol return line was located quite a distance downstream from the collectors. The control logic called for energizing the collector pump when this sensor was 10^{o} F above tank temperature. This seldom occurs with the sensor in this location. To compound the problem, the mechanical contractor jumpered around the control so the pump ran night and day, resulting in a near freeze-up of the secondary (storage water) side of the tube in shell heat exchanger.

CORRECTIVE ACTION:

Controls were modified so that collector pump operation is controlled by the absorber plate/tank temperature differential.

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April 19, 1979
AC00106A

SUBJECT: Collector Loop Pump Controller

LOCATION: Brandon Swimming Association, Brandon, Florida

APPLICABILITY: Pool Heating

PROBLEM: A 20° F differential temperature measurement between a probe located at the bottom of the diving well (deep end of the pool) and the collector plate temperature (at night this reading approximated outside ambient air) was turning on the collector pump in cyclic fashion.

CORRECTIVE ACTION:
A temporary fix was achieved by inserting a photoelectric cell in the collector pump power line to prevent nighttime turn-on. A permanent fix involved removing the photo-cell and modifying the control module by removing the differential measurement and installing a fixed reference point temperature probe (set at 110° F) on the collector absorber plate to energize the pool heating loop.

REMARKS:
Cooling (absorption chiller), heating, and hot water are first in priority with regards to receiving solar heated water. The pool heating mode is activated only after the first priority requirements for heat have been satisfied.

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SOLAR HEATING AND COOLING
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April 23, 1979
AC00107A

SUBJECT: Contamination of Potable Water by Storage Tank Lining

LOCATION: Thompson Motel (Restaurant), Taylor, Texas

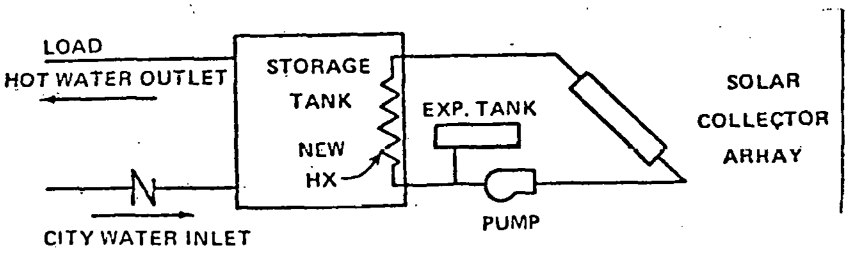
APPLICABILITY: Phenolic-Epoxy Paint, Plasite No. 7155HHB (High Temperature Corrosion Resistant Lining for Steel Tanks for High Purity Water) Manufactured by Wisconsin Protective Coating Corporation, Green Bay, Wisconsin

PROBLEM: Food and water in the motel restaurant took on an unusual taste and odor. Water samples from storage tank were analyzed and 1.7 milligrams per liter of "free phenol" was found. Levels of 0.1 to 50 MG/L are toxic to fish. Phenol had leached out into the water from two layers of phenol-epoxy paint which lined the interior wall of the domestic hot water storage tank; with no expansion tank or check valve, the heater water backed up into the cold water line supplying the motel restaurant.

CORRECTIVE ACTION:

1. Storage tank vendor replaced the existing tank with a new tank utilizing an FDA approved phenolic epoxy lining manufactured by Exon Chemicals. ("Rust-Ban EP6839").
2. A check-valve was added to prevent heated water from backing up into the cold water supply line.
3. A heat exchanger and expansion tank were added to the collector loop to allow the addition of propylene glycol or suntemp for freeze protection.

REMARKS:



Piping Diagram For Domestic Hot Water Heating

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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00108A

SUBJECT: Leaking Rubber Hoses on Collector Headers

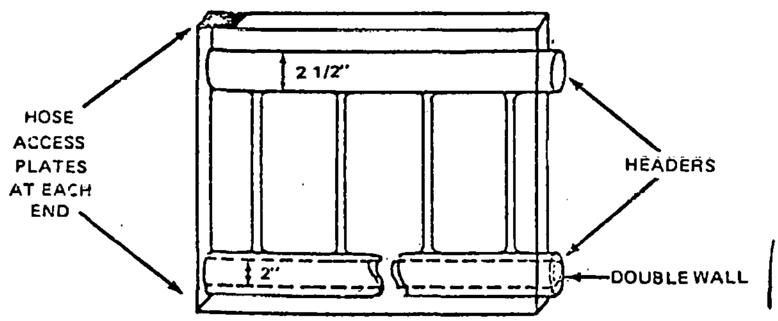
LOCATION: Telex Communications, Blue Earth, Minnesota

APPLICABILITY: ITC Mark III Flat Plate Collectors Connected with a Flexible Silicone Rubber Coolant Hose, Manufactured by Purosil, Part No. 70-262

PROBLEM: The Solar designer (ITC) chose the rubber hose over a copper pipe because of: (1) Its ability to flex as the copper header pipe expands and contracts, and (2) its ability to handle misalignment. With the required clamps in place, more than half the hose length is under clamps with only about one inch left to expand and contract. The four layers of fabric reinforcement gives the hose strength but creates a problem in that it will not stretch or compress longitudinally. This becomes significant in a row of 36 collectors where thermal expansion can amount to more than three inches. The leak problem is aggravated by the large diameter of the headers (2 and 2 1/2 inches). Thermal cycling of the smooth wall header (no ferrule on ends) allows the hose to creep. The automotive type screw clamp then looses its grip. One of the headers can not be swedged because of the double wall pipe (pipe within a pipe) configuration utilized in each collector panel, as illustrated in the sketch below.

CORRECTIVE ACTION:
A more flexible hose (fewer layers of supporting fabric) is being sought with a flex pleat (hump) midway in its length. The ability of such a configuration to readily expand and contract longitudinally together with a ferrule added to the header ends should prevent the hose from creeping off. Hose manufacturers are being asked to send samples to MSFC for testing. Different clamps are also being investigated. Various methods of attaching a lip for a ferrule effect on the end of the copper header will be included in the test. The selected hose, clamp, and ferrule will be identified in a later Advisory Circular.

ITC MARK III COLLECTOR PANEL SKETCH



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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00109A

SUBJECT: Flat Plate Collector Freeze-Up

LOCATION: Telex Communications, Blue Earth, Minnesota

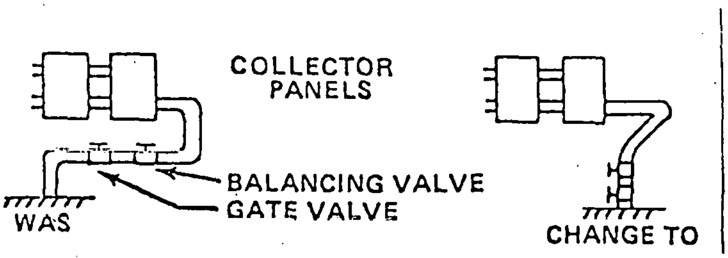
APPLICABILITY: Drain-Down Systems - For Freeze Protection, InterTechnology Corporation's Collector Model Mark III - Solar Designer is ITC

PROBLEM: On Dec. 4, 1978, collector rows 7 and 10 experienced severe freeze damage. These rows were valved off and the system turned back on on Dec. 12, 1978. Row 5 froze on Dec. 31, 1978. At that time, the system was shut down by the owner until design deficiencies were corrected. The three rows contained 108 collectors. The owner estimated that 10 to 20 collectors would have to be sent back to ITC for repair due to extensive damage resulting from the freezing water which failed to drain from the inlet header (2" diameter) which expanded longitudinally approximately 2 1/2" and, in some instances, sheared the collector riser tubes from the header. The system did not completely drain-down during the severe weather and, upon inspection, it was found that the heat taped vacuum breakers did not completely open and the horizontal uninsulated valves in the inlet header restricted drain-back flow.

CORRECTIVE ACTION:

1. Standpipes installed to replace existing vents and vacuum breakers at the end of each row (3/4" diameter standpipes 2 to 3 ft. long).
2. Two-inch diameter standpipes added to each end of 4" line into which all collector return headers drain to a vertical position.
3. Horizontal valves on inlet line were moved to a vertical position. (See sketch below.)
4. Outside valves and (2) standpipes on return line were wrapped with new design heat tape.
5. Outside valves (30) and standpipes were insulated.
6. Control system modified to prevent short cycling of collector loop pump.

REMARKS:



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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00110A

SUBJECT: Control Problem

LOCATION: Telex Communications, Blue Earth, Minnesota

APPLICABILITY: Pressure Controller for Single Pump versus Dual Pump Operation in the Storage-to-Load Loop

PROBLEM: Solar heated water is pumped from storage-to-load with either one or two pumps working in parallel. Each pump has a capacity of 140 GPM and is driven by a 7 1/2 horsepower electric motor. For nominal loads a single pump is used. As the heating demand (load) increases and pressure drops in the inlet line to the heating coil, the second pump is energized by a pressure sensitive control device. With both pumps operating, the pressure builds back up to the maximum pressure limit where one pump cuts off. This resulted in the system cycling rapidly from single to dual pump operation regardless of high and low pressure settings.

CORRECTIVE ACTION:

A variable time delay was added by Johnson Controls to the pressure controller to give the system pressure dynamics time to equalize. When the second pump cut in both flow rate and pressure jumped up past the maximum set point. With the addition of a time delay, the system has enough time for the flow dynamics to equalize before the electronic controller monitors the pressure level. This stopped the second pump from cycling on and off in rapid fashion.

REMARKS:

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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00111A

SUBJECT: Collector Absorber Coating and Glazing

LOCATION: ARATEX Services, Inc., Fresno, California

APPLICABILITY: Flat Plate Collectors Produced by Ying Manufacturing Company

PROBLEM: The collector absorber coating blistered, peeled and outgassed resulting in damage to the absorber and the Lexan glazing. The damage was initiated by an extended stagnation period. The collectors were installed 10 - 12 weeks prior to system operation with ambient air temperatures in excess of 100° F, resulting in absorber plate temperatures in excess of 400° F. Subsequently, unusually heavy rain storms contributed further to the absorber coating and Lexan glazing damage.

CORRECTIVE ACTION:

REMARKS:

To correct the problem at ARATEX, Ying proposes the following:

1. Strip glazing and old absorber coating.
2. Prepare aluminum surface and repaint with a superior "semi-flexible epoxy based flat black paint" (epoxy resin EE-37, Hardner EC-1), manufactured by Guardsman Chemical, or equivalent.
3. Install new Lexar glazing containing improved ultraviolet stablizers using an improved resealing technique with butyl tape.

Ying collectors with the proposed absorber coating are currently being tested in the DOE/MSFC Collector Test Program. Collector test reports and a detailed refurbishment procedure, prepared by Ying, will be evaluated before the work is approved.

When the refurbishment activity is approved, an updated Advisory Circular will be published.

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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 19, 1979
AC00112A

SUBJECT: Heat Exchanger (Collector Loop) Freeze-Up

LOCATION: William Tao & Associates, Inc., St. Louis, Missouri

APPLICABILITY: Closed Loop Collector System

PROBLEM:

The contractor reported the system was shutdown because of a leak in the pump seal. Further investigation revealed that the heat exchanger in the collector loop had burst--probably due to thermosyphoning and subsequent freezing. The freezing probably occurred when the temperature was about 10° F.

The collector system is closed loop with approximately 40% glycol. The possibility of freezing due to thermosyphoning had previously been discussed with the contractor; however, he had decided it was not worth the additional $1,000 plumbing cost.

CORRECTIVE ACTION:

A new heat exchanger and new motorized positive sealing cut-off valve to prevent thermosyphoning is being installed.

REMARKS:

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AC 205 / 453-2054

SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

April 23, 1979
AC00113

SUBJECT: Heat Dump Control Sensor Location

LOCATION: Stephens College, Columbia, Missouri

APPLICABILITY: Active Liquid Solar Systems

PROBLEM: The temperature sensor that activated the collector loop heat dump was located on the upstream side of the heat dump. This resulted in excessive heat dumping and the occasional transfer of heat from storage to the collector loop.

CORRECTIVE ACTION: The control sensor was relocated downstream from the heat dump.

REMARKS:

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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

July 13, 1979
MAC00114A

SUBJECT: Excessive Pressure in Collector Loop

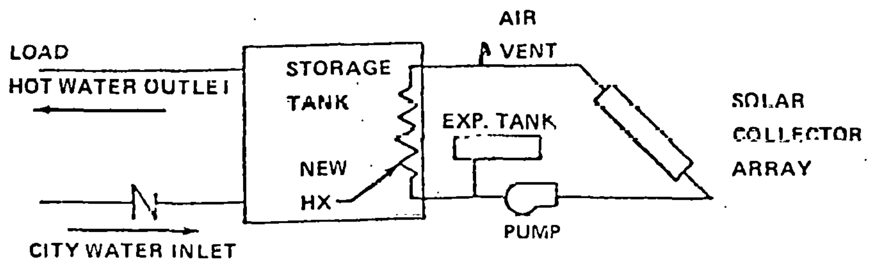
LOCATION: Thompson Motel, Taylor Texas

APPLICABILITY: Closed Loop Collector Systems

PROBLEM: Previous drain-down system was converted to a closed system utilizing propylene glycol for freeze protection. A heat exchanger and expansion tank were installed in the collector-storage tank loop. When the system was activated, excessive pressure caused the pressure relief valves in the collector loop to open, and steam was vented. This resulted from air being trapped in the system causing flashing in the hot collector as a result of low flow.

CORRECTIVE ACTION: An air vent was installed at the high point in the return line. This eliminated the flashing and over pressurization problem allowing the air to escape from the system (see sketch).

REMARKS:



PIPING DIAGRAM FOR DOMESTIC HOT WATER HEATING

NASA
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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

Jan. 10, 1980
MAC00115A

SUBJECT: Debonding of Absorber Plate and Riser Tubing

LOCATION: Alabama Power Company, Montevallo, Alabama
Page Jackson School, Charles Town, West Virginia
Virginia Wade Elementary School, Coral Gables, Florida

APPLICABILITY: PPG Solar Collector Panels, Dual Glazed with Selective Coated Absorber Plate.

PROBLEM: After removal of collectors at Alabama Power and Page Jackson School, examination revealed that the copper tubing which had been soldered to the copper absorber plate had debonded. Distortion and warping of the absorber plate resulting from thermal expansion was especially evident at the Page Jackson installation. This condition is apparent in many of the collectors remaining in place which indicates that absorber plate/riser tube debonding has probably occurred in these collectors. This has resulted in a reduction in the efficiency of the collectors. Collectors from the Virginia Wade School, most of which were never installed, also had severe tube debonding from the absorber plate observed on a test setup at the site.

CORRECTIVE ACTION: The PPG collectors were rejected as unacceptable by the engineering designers for the Virginia Wade School because of the above problems. The other sites are pursuing warranty.

REMARKS: Tests have been performed at Argonne National Labs to determine the causes of the solder problem. Their summary states "Failure analysis of a PPG flat-plate collector removed from the Page Jackson Elementary School's solar heating-and-cooling system, a DOE-sponsored project, was conducted. The heat-transfer-fluid (HTF) passage tubes had become separated from the absorber plate. Tests with the solder showed it to be essentially pure tin with a melting point of 451° F (233° C). Visual examination revealed very little adhesion of the solder to the tubes; most of the solder was on the plate. Some evidence of solder melting or softening during the system's operation was also noticed. Stress-analysis studies showed that relatively small stresses were present on the tubes due to thermal gradients. It appears that the soldering of tubes to the panel during initial fabrication was incomplete along most of the tube-panel interfaces. Moreover, temperatures close to or in excess of the solder's melting point (450° F) were reached during operation. Thus, the partially soldered tubes became completely separated from the absorber panel."

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SOLAR HEATING AND COOLING
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ADVISORY CIRCULAR

Jan. 10, 1980
MAC00117A

SUBJECT: Control Logic for Collector to Storage Loop Pump

LOCATION: Alabama Power Company, Montevallo, Alabama

APPLICABILITY: All Active Solar Systems

PROBLEM: The primary control for the operation of the collector to storage loop pump is a differential controller measuring the delta temperature between the inlet and outlet of the collector array. Both sensors were located in the control room in the piping going to and from the collector array. A cycling clock was utilized to start the pump with a time delay to allow the pipe temperatures to stabilize and then the differential controller would compare the delta temperature between collector inlet and outlet governing pump operation. This control logic has not operated satisfactory which has resulted in random operation of the pump throughout the day and night.

CORRECTIVE ACTION: The control wiring was changed to measure the temperature between the collecter absorber plate and the water in the bottom of the hot storage tank. A new seven-day clock has been added to the solar pump circuit to allow solar operation during the weekend when the building is unoccupied.

REMARKS: Probes for the Δ temperature controllers controlling the operation of solar pumps should be hard mounted on the collector absorber back or on the face but under a sunshield and in the storage tank. Use of the sunshield will prevent a false temperature sensing due to direct exposure of the probe to sunlight which will give a higher temperature reading than the shaded plate and thus the transport fluid.

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Jan. 10, 1980
MAC00118A

SUBJECT: Improper Hot Water Tank Storage Capacity

LOCATION: Alabama Power Company, Montevallo, Alabama

APPLICABILITY: All Solar Cooling Systems

PROBLEM: The solar system was designed to operate a 25 ton absorption chiller from the hot water storage tank which has a capacity of 8000 gallons. This storage capacity is grossly oversized for the collector array area ($2340 ft^2$) in this installation. Therefore, the collectors were unable to raise the temperature of the storage tank to the level necessary to operate the absorption chiller.

CORRECTIVE ACTION: A major redesign was necessary to bypass the existing 8000 gallon storage tank in order for the absorption chiller to be operated directly from the collectors. A 500 gallon surge tank was also installed between the collectors and the chiller to prevent rapid cycling of the chiller due to the passage of small clouds.

REMARKS: This problem was compounded by the control logic problem addressed in Advisory Circular MAC00117A and the collector problem addressed in Advisory Circular MAC00115A.

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SOLAR HEATING AND COOLING
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July 3, 1979
MAC 00119

SUBJECT: Concrete Water Storage Tanks

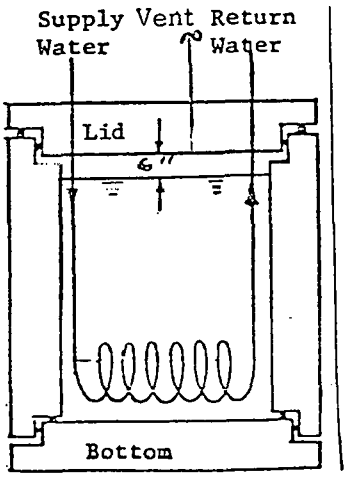
LOCATION: Travis-Braun & Assocs., Dallas, Texas

APPLICABILITY: Concrete water storage tanks made up of reinforced concrete pipe with concrete slab lids held in place by gravity.

PROBLEM: The concrete water storage tanks (each approximately 2300 gallons) were under a residual head pressure of 15 psi when the system was filled. The concrete lids were forced up by this hydrostatic pressure. With the heating loop pump operating the pressure would have increased to ≈40 psi.

CORRECTIVE ACTION: The tanks were isolated from the heating loop pressure and vented to the atmosphere. A heat exchanger was installed in each storage tank in the heating loop piping. The storage tanks were then filled with water to within 6" of the top and operated at atmospheric pressure.

REMARKS:



STORAGE TANK
WITH HEAT EXCHANGER

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July 3, 1979
MAC 00120

SUBJECT: Water Leaks Through Collector Flashing Joints

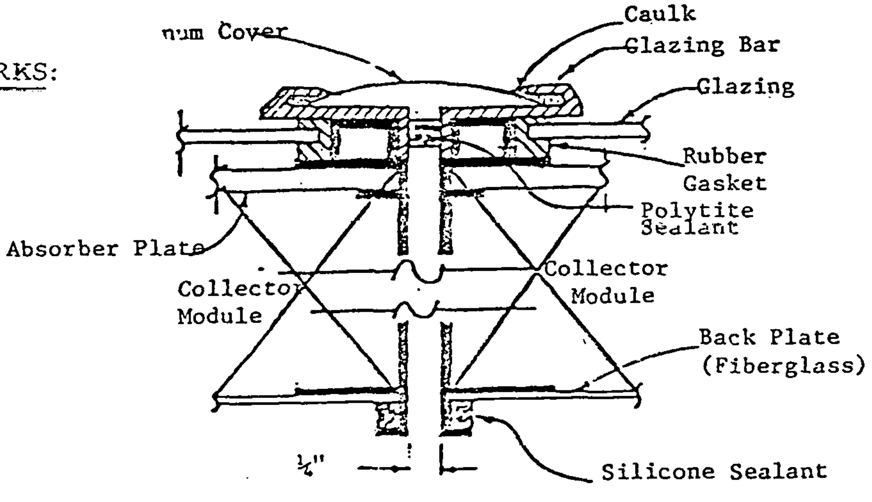
LOCATION: Travis-Braun & Assocs., Dallas, Texas

APPLICABILITY: Solar collector modules which serve as a weatherproof roofing system.

PROBLEM: There are 28 collectors which also serve as the roof over the building lobby. Each collector is 3 by 19 feet, so there is approximately 730 linear feet of flashing joints on the roofing system. Approximately five leaks occurred between the collector modules when there was a south blowing rain. During a light rain, no leaks could be observed. The collectors expand approximately 3/8" over their 19 foot length due to thermal expansion. Sealing of the joint was by sponge rubber gaskets (polytite) compressed from 1/2" to 1/4" between the module cases (See Sketch); however, the collector spacing was greater than 1/4" in some instances. Additional protection was provided by an aluminum cover over the joint.

CORRECTIVE ACTION: The flashing joints were made watertight by the application of a generous amount of silicone sealant under and around the aluminum cover. Expansion and contraction provisions for the collector modules should be provided for in the basic design. Where collectors are an integral part of the roof, uniform spacing is imperative during installation. Prior to acceptance leak tests of the joints should be specified as is done for curtain walls.

REMARKS:



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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
ADVISORY CIRCULAR

August 6, 1979
MAC00121

SUBJECT: Stalling of Magnetically Coupled Pumps

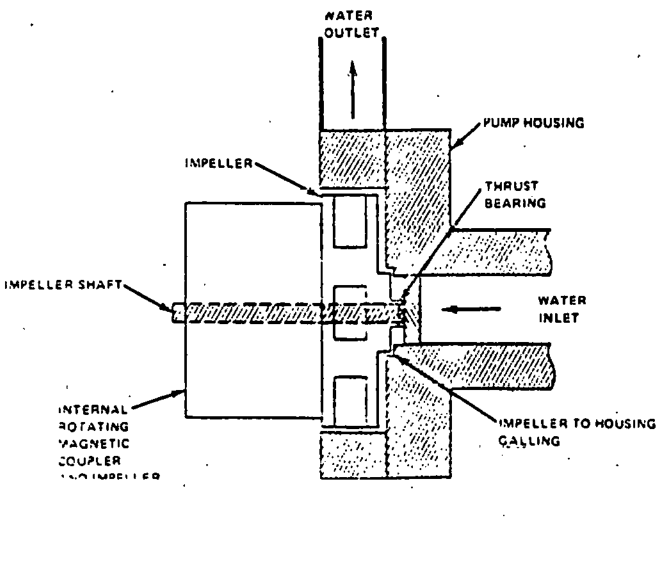
LOCATION: Key West, FL

APPLICABILITY: All solar systems utilizing or proposing to use magnetically coupled pumps.

PROBLEM: After approximately one year of successful operation, the solar system randomly experienced loss of circulation between storage and the collectors. A site inspection revealed that the magnetically coupled water pump was loosing its coupling resulting in a no-flow condition. Disassembly of the pump revealed the impeller thrust bearing to be worn so that the impeller was rubbing on the pump housing enough to cause galling and eventual complete lockup of the impeller. Once the magnetic coupling was lost, it would not pickup the load until power to the motor was shutoff and the motor stopped.

CORRECTIVE ACTION: The magnetically coupled pump was replaced with a direct-drive hot water circulator pump.

REMARKS: If the above problem is experienced with this type pump in a solar system, it should be immediately dismantled and the impeller and thrust bearing inspected for wear or galling.



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SOLAR HEATING AND COOLING
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October 29, 1979
MAC00122

SUBJECT: Solar System Controller

LOCATION: Charlotte, North Carolina

APPLICABILITY: All Solar Systems

PROBLEM: In order to maintain control component commonality with the previously installed HVAC System, a pneumatic differential controller was installed to provide flow control (pump and diverting valve) to and from the solar array. The accuracy and stability of this controller was not suitable for the operation of a solar array and the system operatior was very erratic.

CORRECTIVE ACTION: Following numerous attempts to adjust the pneumatic controller, a solid state adjustable ΔTemperature controller, developed for solar applications, was installed.

REMARKS: Control of the solar collection system should be performed by a field adjustable solid state ΔTemperature controller designed for solar applications.

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SOLAR HEATING AND COOLING
COMMERCIAL DEMONSTRATION PROGRAM
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October 29, 1979
MAC00123

SUBJECT: Solar Loop Pump Controller

LOCATION: Mt. Rushmore, S. D.

APPLICABILITY: All active Hydronic Solar Systems

PROBLEM: The control of the solar pump operation was designed and installed to be a collector plate temperature (125°F.) turnon and a collector plate temperature (100°F.) turnoff. This allowed the solar array pump to operate at times when the solar storage temperature was much higher than the collector plate, resulting in a net loss of collected energy and excessive pump run time which consumed additional energy.

CORRECTIVE ACTION: The Single Temperature Probe Controller was replaced with a two probe Δ Temperature Controller measuring the temperature differential between the storage tank and the collector plate. This has resulted in a more efficient operation of the collector array and prevented the loss of energy from storage.

REMARKS:

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SOLAR HEATING AND COOLING
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ADVISORY CIRCULAR

December 10, 1979
MAC 00124

SUBJECT: Leaking Fiberglass Storage Tanks

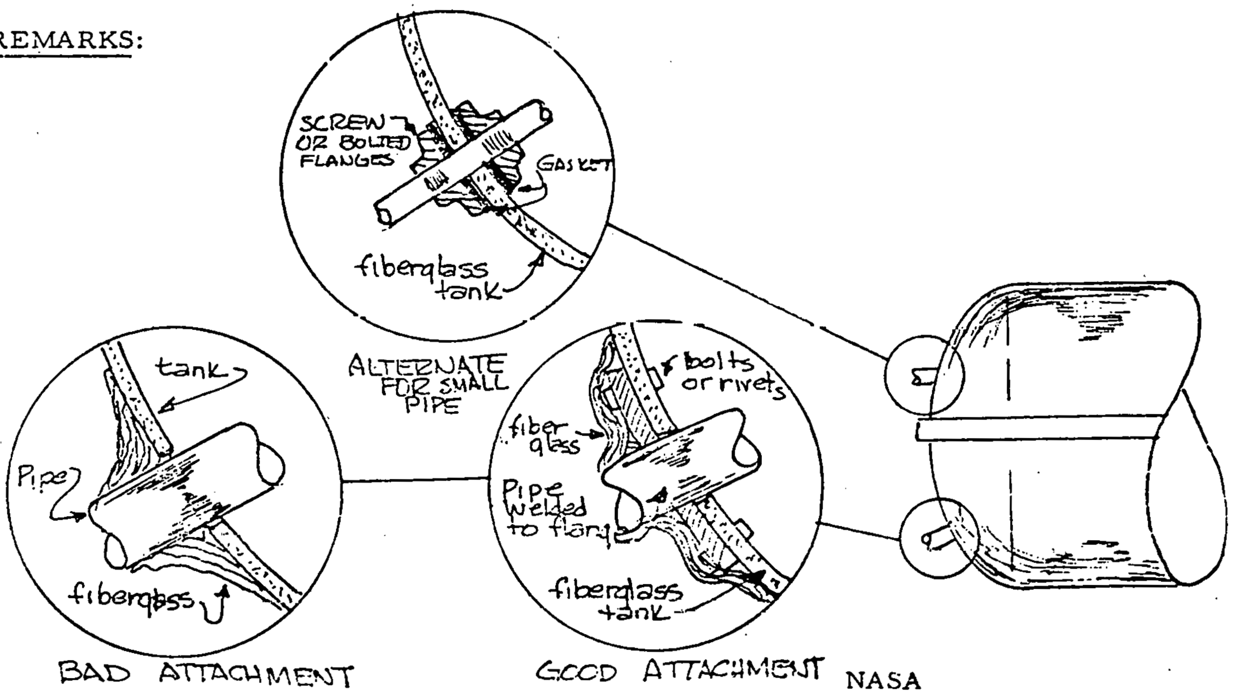
LOCATION: Yosemite National Park & Department of Transportation, Pueblo, CO

APPLICABILITY: All fiberglass storage tanks

PROBLEM: Leaks occurred around penetrations installed by the contractor. The lack of compensation for expansion and stress characteristics of fiberglass resulted in numerous fractures.

CORRECTIVE ACTION: Repairs made on the sites in question were field repairs and not positive cures for the basic problem. In most cases, additional glass cloth and resin were layered on the leaking area. The correct mounting procedures must occur before the tank is installed and plumbing attached. This generally means that the penetration must be adequately attached to the fiberglass shell with a flange of sufficient size to distribute the stress. The flange can then be attached with fasteners and fiberglassd or possibly attached on a semi-permanent basis with soft gaskets and double flanges. See the sketches below for suggested attachment methods. When possible penetrations should be made from above to avoid such problems.

REMARKS:



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SOLAR HEATING AND COOLING
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December 10, 1979
MAC 00125

SUBJECT: Leaking Collectors

LOCATION: Yosemite National Park, Yosemite, CA

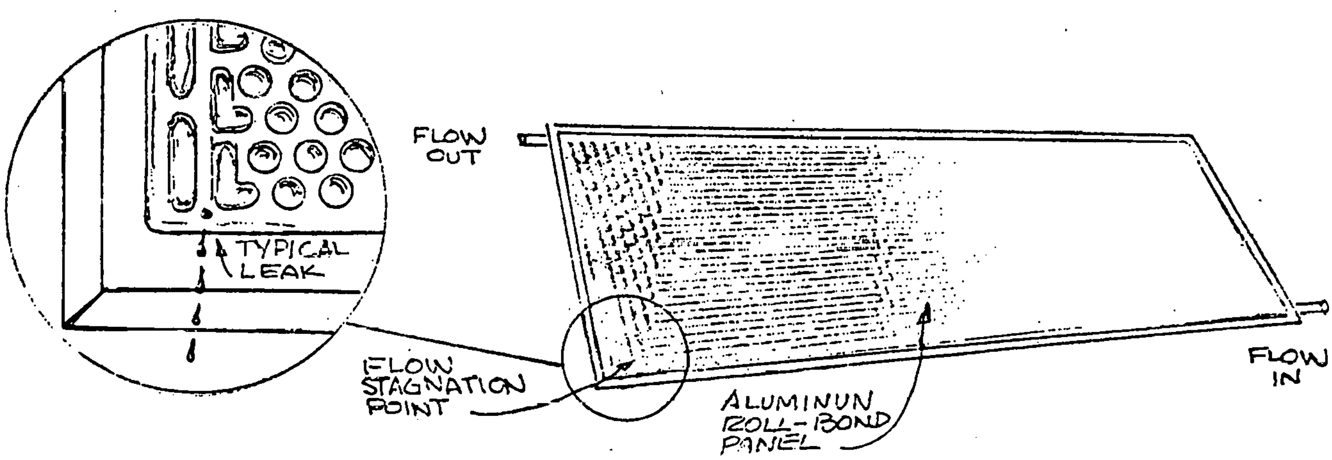
APPLICABILITY: Colt aluminum roll bond collectors mounted horizontally with oil transport fluid.

PROBLEM: The collectors developed leaks as shown on the sketch. Due to the orientation of the collector and the absorber construction, a stagnation or low flow region developed. This area accumulated what moisture there was in the system and with probable copper particle contamination from drilling holes, etc., a galvanic action occurred with the aluminum. This resulted in the subsequent leaks.

CORRECTIVE ACTION: The collectors with aluminum absorber panels were replaced with copper "roll-bond" units. The oil transport fluid of Thermia 33 was replaced with a lower viscosity, inhibited Dialia AX providing increased corrosion resistance. The replacement oil was circulated continuously through a bank of six temporary filters to remove any moisture or solid particles. The filters were changed twice during the four hour filtering period and the fluid continuously monitored through the clear filter housings. Each collector bank was individually filtered by valving off the others. This resulted in a high velocity flow in each individual bank.

This system is a sister system to one located in Pueblo, Colorado, which has not experienced any collector failure. The difference appears to have been the prevention of contamination in the Pueblo system that did not occur in the Yosemite system. Care must be taken when plumbing a system and filling, notably when reactive metals similar to aluminum are used.

REMARKS:



SOLAR HEATING AND COOLING
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ADVISORY CIRCULAR

January 3, 1980
MAC 00126

SUBJECT: Failure of Collector Controller

LOCATION: Kaw Valley State Bank, Topeka, KN

APPLICABILITY: General Electric evacuated tube collector systems with external "solar integrator" sensor or any similar system.

PROBLEM: General Electric uses a "solar integrator" set at a solar insolation of 35 BTU/ft^2-hr to turn on its evacuated system(P1). The integrator doesn't sense the collector temperature but only the available energy. In this failure, the collectors were turned on when covered with ice and snow and the subfreezing glycol solution pumped through a heat exchanger interfacing with water. The HEX-froze and ruptured, subsequently allowing the glycol solution in the collectors to be diluted with enough water to freeze and rupture the collector tubes.

CORRECTIVE ACTION: The original collector plumbing was as shown in figure 1 in the remarks section. The system was modified as shown in figure 2. This modification consisted of installing a 2" diverting valve that is temperature controlled as to normally pass flow from "A" to "C". On a temperature rise to 75° F, the valve will position "A" to "B" allowing the collector fluid to be circulated through the heat exchanger. When the temperature drops to 65° F, the valve will return to the "A" to "C" position. A final safety feature is controller S-2 which will cut off the collector pump if it sees a temperature of 40° F or less. The system will then remain off until manually reset. This combination of safety features should prevent any chance of freeze up from occurring.

REMARKS:

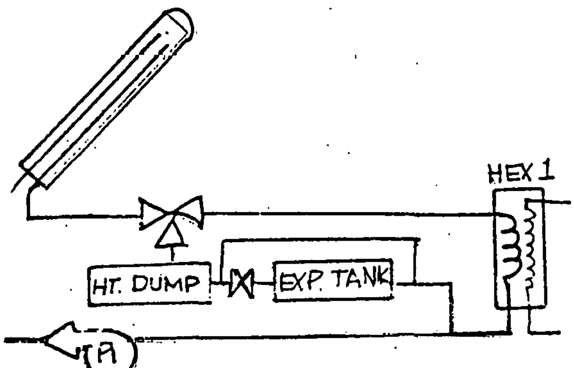

FIGURE 1

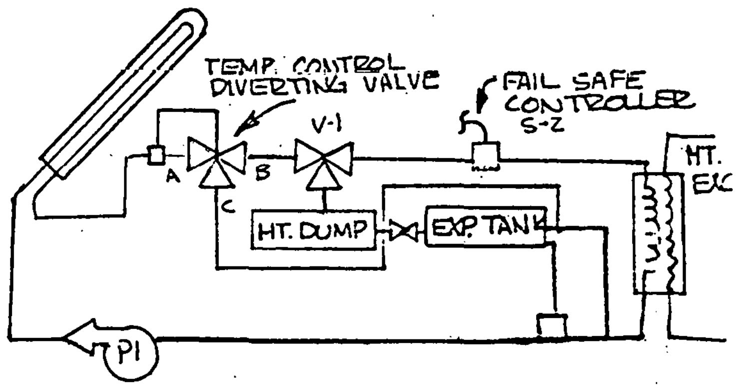

FIGURE 2

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SOLAR HEATING AND COOLING
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January 4, 1980
MAC 00127

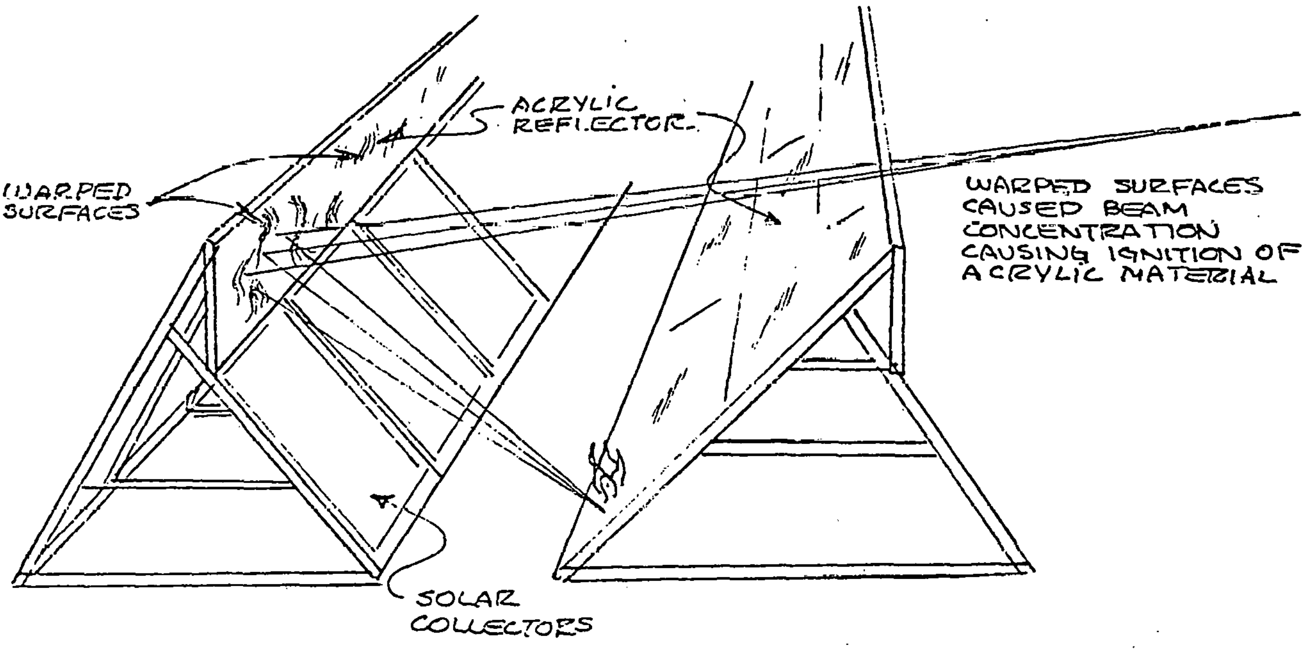
SUBJECT: Potential Fire Hazard Associated with Solar Reflectors Fabricated from Acrylic Materials

LOCATION: Alabama Power Company, Montevallo, Alabama

APPLICABILITY: All Solar Systems with Acrylic Solar Reflectors

PROBLEM: A section of aluminized acrylic reflectors, ignited and burned. The fire was caused by one section of reflectors that had warped, concentrating the sun's rays on the edge of another reflector. Ignition appears to have occurred when the flash point of the thin layer of aluminized acrylic, which had separated from the heavier (1/4" thick) acrylic used as a backing material, was exceeded by the concentration of sunlight from the warped reflectors. The thin layer of acrylic then ignited the thin layer of aluminum, used as a reflective surface, and the heavier (1/4") acrylic backing which resulted in an intense fire.

CORRECTIVE ACTION: The reflectors were removed from the installation.



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APPENDIX C

ANALYTICAL BACKGROUND FOR THE PERFORMANCE EVALUATION OF A SOLAR HEATING AND COOLING SYSTEM

APPENDIX C

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ANALYTICAL BACKGROUND FOR THE EVALUATION OF THE PERFORMANCE OF A SOLAR HEATING AND COOLING SYSTEM

C.1 SOLAR COLLECTOR PERFORMANCE

The Hottel and Whillier collector equation has been historically used to evaluate solar collector performance. It is usually given as follows:

$$q_c = F_R A \, [I'(\tau\alpha) - U_L \, (T_i - T_a)] \qquad \text{(C1)}$$

where:

q_c = Rate of energy collection by a flat-plate collector, Btu/hr

F_R = Solar collector heat removal factor, dimensionless

A = Solar collector area, ft^2 (m^2)

I' = Solar radiation incident on the (tilted) solar collector surface, $Btu/hr \cdot ft^2$ (w/m^2)

$(\tau\alpha)$ = Product of the cover transmittance and the plate absorptance accounting for dirt and shading, dimensionless

U_L = Solar collector overall energy loss coefficient, $Btu/hr \cdot ft^2 \cdot °F$ ($w/m^2 \cdot °C$)

T_i = Fluid temperature at the solar collector inlet, °F (°C)

T_a = Ambient temperature, °F (°C)

More recently, Ward [C1] derived and experimentally verified the following equation for solar collector performance:

$$\frac{q_c}{A} = I'(\tau\alpha) - U_L \, (T_f - T_a) - cM(\Delta T_f/\Delta t) \qquad \text{(C2)}$$

where:

T_f = Solar collector outlet fluid temperature, °F

c = Specific heat of the solar collector materials and fluid, $Btu/lb \cdot °F$ ($kJ/kg \cdot °C$)

M = Mass of the collector and fluid per unit area, lb/ft^2 (kg/m^2)

ΔT_f = Change in collector outlet fluid temperature during the time interval Δt, °F (°C)

Δt = Time interval, hr

For steady-state conditions, $\Delta T_f/\Delta t = 0$, and equation (C2) reduces to:

$$q_c = A \, [I'(\tau\alpha) - U_L \, (T_f - T_a)] \qquad \text{(C3)}$$

In equation (C1), the quantity F_R is:

$$F_R = \frac{\dot{m}c_p}{U_L + \dot{m}c_p} \qquad \text{(C4)}$$

where:

$\dot{m}$ = Mass flow rate of collector fluid per unit area of collector, lbs/hr·ft^2 (kg/hr·m^2)

c_p = Specific heat (at constant pressure) of the collector fluid, Btu/lb·°F (kJ/kg·°C)

Experimentally the heat output of a solar collector is determined using:

$$\frac{q_c}{A} = \dot{m}c_p(T_f - T_i) \tag{C5}$$

$$= \dot{m}c_p(\Delta T) \tag{C6}$$

where $\Delta T = T_f - T_i$.

C.1.1 Solar Collector Utilizability

Klein [C2] states that Equation (C1) indicates that there is a minimum level of solar radiation required to maintain the collector plate at the temperature of the entering collector fluid. He called this solar radiation intensity the critical level and stated that it could be found by setting q_c in Equation (C1) equal to zero, i.e.

$$I_c = \frac{U_L(T_i - T_a)}{(\tau\alpha)} \tag{C7}$$

where:

I_c = Critical solar radiation level, Btu/hr ·ft^2 (kJ/hr·m^2)

C.1.2 Collectible Solar Radiation

The authors of this report agree in principal with the general concept introduced by Klein; however they believe that Equation (C2) should be used to determine the minimum solar radiation intensity that is collectible. By setting $q_c = 0$ in Equation (C2), one obtains:

$$I_o = \frac{U_L(T_f-T_a)}{(\tau\alpha)} + \frac{cM(\Delta T_f/\Delta t)}{(\tau\alpha)} = \frac{U_L(T_i-T_a+\Delta T)}{(\tau\alpha)} + \frac{cM(\Delta T_f/\Delta t)}{(\tau\alpha)} \tag{C8}$$

where:

I_0 = The minimum solar radiation intensity that is collectible, Btu/hr·ft^2 (kJ/hr·m^2)

Equation (C8) shows that, on a typical day, the value of I_o in the morning is greater than that in the evening. However, for ease of analysis, assume that the term, $cM(\Delta T_f/\Delta t)/(\tau\alpha)$, is negligible compared to I_o. When this is the case, then Equation (C8) reduces to:

$$I_o = \frac{U_L(T_f-T_a)}{(\tau\alpha)} = \frac{U_L(T_i - T_a + \Delta T)}{(\tau\alpha)} \tag{C9a}$$

Clearly, under all practical conditions, $I_o > I_c$.

The minimum fluid temperature that is useful varies with the season (heating or cooling) and with the type of solar system. For example, the minimum useful liquid temperature for solar cooling is not likely to be less than 170°F (75°C). On the other hand, the minimum useful liquid temperature for solar heating could be as little as 72°F (22°C) if the heat lost from storage is useful. For an air heating solar system, the minimum useful air temperature is about 100°F (38°C). For heating hot water, temperatures above 40°F (4°C) may be useful in the winter, but higher temperatures will be required in the summer. In any event, T_f must be greater than any of these temperatures because of piping (duct) heat losses and temperature drops across heat exchangers.

Because of the importance of controls in the performance of solar heating and cooling systems, it is necessaary to consider the relationship between T_f in Equation (C9) and the minimum useful temperatures discussed above. In general heat exchangers, piping and/or ducting heat losses, and other considerations ensure that $T_f > T_{min}$ (where T_{min} is the minimum useful temperature for cooling (~170°F; 77°C), space heating (70-100°F; 20-40°C), DHW heating (~40-60°F; 5-15°C, etc.). The difference between T_f and T_{min} is in general, just the control set point for the temperature differential between collector outlet and storage (or load), which energizes or de-energizes the collector pump or blower. This control setting has been found experimentally to be the control set point to turn the pump (or blower) OFF. This set point is defined as ΔT_c; and when incorporated into Equation (C9a), we obtain:

$$I_o = \frac{U_L(T_f - T_a)}{(\tau\alpha)} = \frac{U_L(T_{min} + \Delta T_c - T_a)}{(\tau\alpha)} \qquad \text{(C9b)}$$

Substitution of Equation (C9) into Equation (C3) gives:

$$q_c = A\ (\tau\alpha)\ [I' - I_o] \qquad \text{(C10)}$$

The development of the ϕ-curve method proceeds by using an equation similar to Equation (C10) to determine the useful energy collection for a given hour of the day averaged over a long-term (usually a month). The total quantity of energy collected during a time interval of t hr is:

$$q_c t = At\ (\tau\alpha)\ [I' - I_o] \qquad \text{(C11)}$$

The average useful energy collection, E_i, for a given hour (i) of the day averaged over N days is:

$$E_i = \frac{\Sigma\ q_c t}{N} = \frac{A(\tau\alpha)t}{N} \sum_{j=1}^{j=N} (I' - I_o)^+_{ij} \qquad \text{(C12)}$$

where:

$$[E_i] = \text{Btu (kJ)}$$

In Equation (C12), t = 1 hour, the plus superscript is used to indicate that negative values of $(I' - I_o)_{ij}$ are not considered, and N is usually the number of days in the month. The total quantity of solar radiation incident on the tilted surface of the solar collector during the time interval t is:

$$E_s = At \sum_{j=1}^{N} (I')_j \qquad \text{(C13)}$$

The average quantity of solar radiation incident on the tilted solar collector surface E_{si} for a given hour (i) of the day averaged over N days is:

$$E_{si} = \frac{\Sigma\ E_s}{N} = \frac{At}{N} \sum_{j=1}^{j=N} I'_{ij} = AI'_i \qquad \text{(C14)}$$

By definition, ψ_i is the fraction of the long-term average hourly solar radiation, I'_i, that is above the intensity I_c:

$$\phi_i = \frac{1}{N\ I'_i} \sum_{j=1}^{j=N} (I' - I_c)^+_{ij} \qquad \text{(C15)}$$

In a similar fashion, the authors of this report are defining Φ_i, to be the fraction of the long-term average hourly solar radiation, I_i', that is above the intensity I_o:

$$\Phi_i = \frac{1}{N\ I_i'} \sum_{j=1}^{j=N} (I' - I_o)^+_{ij} \tag{C16}$$

The ratio (E_i/E_{si}) is the fraction of the total solar radiation that is useful:

$$\frac{E_i}{E_{si}} = \Phi_i\ (\tau\alpha) \tag{C17}$$

Substitution of Equation (C16) into Equation (C12) gives:

$$E_i = A(\tau\alpha)t\ I_i'\ \Phi_i \tag{C18}$$

The total daily useful energy gain, Q_c', (the collectible solar energy) is then the sum of the n hourly contributions where n is the number of hours between sunrise and sunset:

$$Q_c' = \sum_{i=1}^{i=24} E_i \tag{C19}$$

Substituting Equation (C18) into Equation (C19) gives:

$$Q_c' = A(\tau\alpha)t \sum_{i=1}^{i=24} I_i'\ \Phi_i \tag{C20}$$

and again t = 1 hour in Equation (C20).

Whillier [C3] showed that ϕ is a function of (I_c/I_i'), month of the year, and location. The relation between ϕ and (I_c/I_i') is independent of the time of day. Whillier also showed that, on the average, solar radiation is usually symmetrical about solar noon, so that it is only necessary to determine ϕ for each hourly interval from solar noon rather than for each hour of the day.

Liu and Jordan [C4, C5, C6] introduced the quantity, K, which they defined as follows:

$$K = \frac{\text{Monthly average solar radiation on a horizontal surface}}{\text{Monthly average solar extraterrestrial radiation}}$$

They also show that $\phi = f(I_c/I_i', K)$ and that this relationship is independent of month of the year or location. They also developed a method of including the effect of collector tilt.

Comparison of Equations (C15) and (C16) shows that they are identical in concept and differ only in the magnitude of I that is deemed useful. Consequently $\Phi = f(I_o/I_i', K)$ and this relationship is independent of time of year and location.

C.1.3 Klein's $\bar{\phi}$-Charts

Klein [C2] stated that the total useful energy gain over an extended period (such as a month) can be determined by summing the hourly contributions over the entire period. In other words, each of the n N hourly contributions are added together. Equation (C11) can be used for this purpose when t = 1 hour:

$$Q_M = \sum_{1}^{mN} E = A(\tau\alpha)t \sum_{1}^{mN} (I' - I_o)^+ \tag{C21}$$

In an equation similar to Equation (C21), Klein states that I_c is evaluated using the daytime average ambient temperature but, unfortunately, such records are usually not available. If the average daily solar radiation intensity incident on the tilted solar collector surface is defined as I ($Btu/ft^2 \cdot day$; $kJ/m^2 \cdot day$), then $I = nI'$, where n is the number of hours per day of sunlight. Therefore, the total solar radiation incident on the collector surface during the month is NI $Btu/ft^2 \cdot month$, where N is the number of days per month.

Klein [C2] defined the monthly average daily utilizability, $\bar{\phi}$, as:

$$\bar{\phi} = \frac{\sum_{1}^{nN} (I' - I_c)^+}{NI}$$

where $\bar{\phi}$ is the fraction of the total solar radiation during the month that is above the level I_c. In like manner, the authors define $\bar{\Phi}$ to be the fraction of the total solar radiation during the month that is above the intensity I_o:

$$\bar{\Phi} = \frac{\sum_{1}^{nN} (I' - I_c)^+}{NI} \qquad \text{(C22)}$$

Accordingly, Equation (C22) can be substituted into Equation (C21) to obtain:

$$Q_M = A(\tau\alpha)t\,\bar{\Phi}N\,I \qquad \text{(C23)}$$

C.1.4 Solar Collectors in Series

Equation (C1) can be rewritten to obtain the solar collector instantaneous efficiency, η:

$$\eta = \frac{\dot{q}_c}{AI'} = F_R[(\tau\alpha) - U_L \frac{(T_i - T_a)}{I'}] \qquad \text{(C24)}$$

Solar collector test results on single panels are often expressed in terms of the intercept and slope of Equation (C24). Based on this slope intercept test data on a single collector panel, Oonk, Jones and Cole-Appel [C7] developed a method for extending these test results to predict the performance of N solar collector panels in series. Consequently, the solar collector efficiency of N panels in series is:

$$\eta_N = \eta\, C_N \qquad \text{(C25)}$$

where:

$$C_N = \frac{1}{N\psi}[1 - (1 - \psi)^N] \qquad \text{(C26)}$$

where:

$$\psi = \frac{F_R\, U_L}{\dot{m}c_p} \qquad \text{(C27)}$$

so that ψ and C_N are both dimensionless. Substituting Equation (C4) into Equation (C27) gives:

$$\psi = \frac{U_L}{U_L + \dot{m}c_p} \qquad \text{(C28)}$$

From Equation (C28) it is clear that as $mc_p \rightarrow 0$, $\psi \rightarrow 1$, and $C_N \rightarrow 1/N$. Also from Equation (C26):

$$C_1 = 1,\ C_2 = 1 - \frac{\psi}{2},\ C_3 = 1 - \psi + \frac{\psi^2}{3},\ \text{etc.} \tag{C29}$$

Equation (C2) can also be rewritten in terms of solar collector efficiency:

$$\eta = (\tau\alpha) - U_L \frac{T_f - T_a}{I'} - \frac{cM}{I'} \ (\Delta T_f/\Delta t) \tag{C30}$$

It is clear from Equation (C29) that it is very desirable to make ψ as small as possible.

C.1.4.1 Calculation of C_2 for CSU Solar House III

The manufacturer's data indicates that $F_R(\tau\alpha) = 0.8$ and that $F_R U_L = 17.58$ kJ/hr m^2 °C. The recommended value of $\dot{m}c_p$ for these solar collectors is 196.06 kJ/hr m^2 °C. Using Equation (C27),

$$\psi = 17.58/196.06 = 0.0897$$

and using Equation (C29),

$$C_2 = 1 - 0.0897/2 = 0.955.$$

C.1.4.2 Calculation of F_R, $(\tau\alpha)$, and U_L for CSU Solar House III

Equating Equations (C27) and (C28) and solving for U_L one obtains:

$$U_L = \frac{\psi \dot{m}c_p}{1-\psi} = \frac{F_R U_L \dot{m}c_p}{\dot{m}c_p - F_R U_L} \tag{C31}$$

Using Equation (C31),

$$U_L = \frac{(17.58)(196.06)}{196.06 - 17.58} = 19.4 \text{ kJ/hr } m^2 \text{ °C}$$

Therefore:

$$F_R = \frac{F_R U_L}{U_L} = \frac{17.58}{19.40} = 0.906$$

and

$$(\tau\alpha) = \frac{F_R(\tau\alpha)}{F_R} = \frac{0.8}{0.906} = 0.883$$

C.1.4.3 Heat Loss from Interconnections Between N Solar Collectors in Series

Using ΔT_i to represent the temperature increase across each solar collector and using ΔT_j to represent the temperature loss across each interconnection, the total temperature increase across an array of N solar collectors in series is:

$$T = N\ \Delta T_i - (N - 1)\ \Delta T_j$$

Similarly, the total rate of energy collection is:

$$q_c = N\ q_i - (N - 1) q_i \tag{C32}$$

Using (ua) to represent the heat loss coefficient of the interconnection in Btu/hr·°F, then:

$$q_i = (ua)(T_j - T_a) \tag{C33}$$

Clearly $T_i < T_j < T_f$ in a practical situation. In general, however, T_j will be nearer T_f than T_i. However, one can with fair accuracy assume that for two collectors in series (j = 1):

$$T_i = \frac{T_i + T_f}{2} \tag{C34}$$

Likewise for three collectors in series, one can assume that:

$$T_1 = \frac{3}{4} T_i + \frac{T_f}{4}, \quad T_2 = \frac{T_i}{4} + \frac{3}{4} T_f \tag{C35}$$

For four collectors in series, the temperatures of the interconnections would be:

$$T_1 = \frac{3}{4} T_i + \frac{T_f}{4}, \; T_2 = \frac{T_i + T_f}{2}, \; T_3 = \frac{T_i}{4} + \frac{3}{4} T_f \tag{C36}$$

Therefore, whether one has two, three, four or more collectors in series, the average temperature of the interconnections is (from Equations C34, C35 or C36):

$$T_j = \frac{T_i + T_f}{2} \tag{C37}$$

Consequently Equation (C33) becomes:

$$q_j = (ua)\left(\frac{T_i + T_f}{2} - T_a\right) \tag{C38}$$

Substituting Equations (C3) and (C38) into Equation (C32), one obtains:

$$q_c = NAI'(\tau\alpha) - NU_LA(T_f - T_a) - (N - 1)(ua)\left(\frac{T_i + T_f}{2} - T_a\right) \tag{C39}$$

Comparing the individual parameters of the last two terms, it is obvious that:

$$N > N - 1 \tag{C40}$$

$$U_L > u \tag{C41}$$

$$A >> a \tag{C42}$$

$$(T_f - T) > \left(\frac{T_i + T_f}{2} - T\right) \tag{C43}$$

so that, in general,

$$NUA(T_f - T_a) >>>>> (N - 1)(ua)\left(\frac{T_i + T_f}{2} - T_a\right) \tag{C44}$$

so that the term, $(N - 1)(ua)(T_i + T_f/2 - T_a)$ is completely negligible in comparison with $NUA(T_f - T_a)$. This is true whenever 1/u is greater than about 4 hr·ft^2·°F/Btu (0.7 m^2·°C/w) (i.e., the R - value = 4 or more).

C.1.5 Solar Collector Array Efficiency

Q_c is defined as the daily average useful energy collected by the solar collector over a specified period of one month, Btu/day·ft^2 (kJ/m^2·day).

$$Q_c \equiv C_N \, Q_M/M - (N-1) \, q_j \tag{C45}$$

where

C_N accounts for collector modules in series and is given by Equation (C26),

Q_M is given by Equation (C23),

M is the number of days in the month,

N is the number of collector modules in series, and

q_j accounts for the heat losses from the collector modules in series interconnections, and is given by Equation (C38).

The daily solar collector efficiency is defined as:

$$\eta_c = \frac{Q_c}{AI} \tag{C46}$$

C.2 COMPARISON OF PERFORMANCE

The coefficient of performance, C, expresses the effectiveness of a heating or cooling system. It is a dimensionless ratio defined by the expression:

$$C = \frac{\text{Useful heating or cooling in units of energy}}{\text{Energy supplied to the heating or cooling system}} \tag{C47}$$

For a simple theoretical absorption refrigeration system [C8]:

$$C_{max} = \frac{T_E \, (T_G - T_o)}{(T_0 - T_E) \, T_G} \tag{C48}$$

where:

T_E = The refrigerated substance absolute temperature, °R

T_G = Generator heating medium absolute temperature, °R

T_o = Environmental absolute temperature, °R

For a solar cooling system, Equation (C47) can be rewritten as follows:

$$C = E_R/E_H \tag{C49}$$

where:

E_R = Energy removed from the air, Btu (kJ)

E_H = Heat energy supplied to the solar cooling unit, Btu (kJ)

The energy lost from the solar system to the conditioned space, E_L, reduces the performance of a solar cooling system in two ways: (1) it adds to the load, E_R, and (2) it reduces the amount of energy available to drive the solar cooling system. Therefore the total effect on the solar cooling system is as follows: Using E_T to represent the total effect of E_L, then:

$$E_T = E_L + \frac{E_L}{C} = E_L \left(1 + \frac{1}{C}\right) \quad \text{(C50)}$$

Defining the ratio:

$$\frac{E_T}{E_L} = \beta = 1 + \frac{1}{C} \quad \text{(C51)}$$

From the foregoing, it can be seen that, for a solar heating system, $\beta \simeq 0$. Energy losses from a solar cooling system to the conditioned space (E_L) on the other hand, must be multiplied by β to obtain the total energy drain on the solar cooling system. For a typical solar cooling C of 0.5, from Equation (C51) it is clear that $\beta = 3$. On the other hand, most of the heat lost from the solar heating system to the conditioned space is useful even if this causes the conditioned space temperature to temporarily rise above that desired, because the excess energy is stored in the conditioned space thermal mass. However, it is highly desirable to minimize uncontrolled heat losses year round; otherwise an active solar heating system degenerates into an uncontrolled passive solar heating system. For this reason it is worth defining a term that quantitatively measures the degree of uncontrolled heat supplied by a solar heating system:

$$H = \frac{\text{Controlled solar heat delivered to the conditioned space}}{\text{Total solar heat delivered to the conditioned space}} \quad \text{(C52)}$$

Therefore, for an ideal active system, $H = 1$. For a passive system, $H = 0$, and for a hybrid system, $0 < H < 1$. Obviously, many active systems will fall into the hybrid category, but it is desirable that H be as close to one as is practical.

Using Q_c to represent the solar energy collected by the solar collector, Btu/day, then from Equations (C23) and (C45), where we have neglected q_j,

$$Q_c = A(\tau\alpha)t\,\bar{\Phi}\,I\,C_N \quad \text{(C53)}$$

Using Q_H to represent the controlled solar energy delivered to the load in Btu/day, then for solar heating:

$$Q_H = Q_c - Q_E - Q_I \quad \text{(heating)} \quad \text{(C54)}$$

where:

Q_E = Solar system heat losses to the exterior of the conditioned space, Btu/day

Q_I = Solar system heat losses to the interior of the conditioned space, Btu/day

The total useful solar heat delivered to the conditioned space is:

$$Q_u = Q_c - Q_E \quad \text{(heating)} \quad \text{(C55)}$$

during the winter. The total solar heat delivered to the cooling unit during the summer is:

$$Q_x = Q_c - Q_E - Q_I \quad \text{(cooling)} \quad \text{(C56)}$$

However, heat losses to the interior of the conditioned space are an addition to the cooling load of Q_I/C. Therefore, the net useful solar heat used for cooling is:

$$Q_u = Q_x - Q_I/C \quad \text{(cooling)} \quad \text{(C57)}$$

and so for solar cooling:

$$Q_u = Q_c - Q_E - Q_I \left(1 + \frac{1}{C}\right) \quad \text{(cooling)} \quad \text{(C58)}$$

Keeping in mind that in the winter $\beta \simeq 0$ and in the summer β is given by Equation (C51), one can write one equation to replace both Equations (C55) and (C58):

$$Q_u = Q_c - Q_E - Q_I \beta \qquad \text{(heating and cooling)} \tag{C59}$$

By definition (Equation C52):

$$H = \frac{Q_H}{Q_c - Q_E} = 1 - \frac{Q_I}{Q_c - Q_E} \qquad \text{(heating)} \tag{C60}$$

both Q_I and Q_E may include heat losses from the heat storage unit, piping and/or ducting, heat exchangers, pumps and/or blowers, and other solar system components.

C.3 SOLAR SYSTEM EFFICIENCY

The total energy available to the solar system is AI (Btu/day; kJ/day). Therefore, the solar system thermal efficiency is (for cooling):

$$\eta_T = \frac{Q_u C}{AI} \qquad \text{(cooling)} \tag{C61}$$

Using E to represent the quantity of electrical energy consumed by the solar system (Btu(elec)/day; kJ(elec)/day), then:

$$E = (3600)(\frac{\text{kw-hr}}{\text{day}}) \tag{C62}$$

Using η_E to represent the efficiency of electric power generation, transmission, and distribution, then the total energy, Q_e, required to furnish E Btu(elec)/day (kJ(elec)/day) of electric energy is:

$$Q_e = E/\eta_E \tag{C63}$$

where $\eta_E \simeq 0.25$, so that $Q_e \simeq 4E$.

Using η_F to represent furnace efficiency, the total energy required to deliver the energy Q_u delivered by the solar heating system is:

$$Q_F = Q_u/\eta_F \tag{C64}$$

where $\eta_F \simeq 0.6$ for a gas furnace and about 0.55 for an oil furnace.

For solar heating, Equation (C61) becomes:

$$\eta_T = \frac{Q_u}{AI} \qquad \text{(heating)} \tag{C61A}$$

In the following discussion, C_c will be used to represent the coefficient of performance of a conventional mechanical vapor compression unit. If Q_p is the heat delivered by a heat pump, then the electrical energy required is $Q_p/C_c = E$. Consequently, the total energy Q_e required is given by Equation (C63) or:

$$Q_e = \frac{Q_p}{C_c \eta_E} \qquad \text{(heating and cooling)} \tag{C65}$$

The same relationship would be true if Q_p was the cooling furnished by a heat pump or a conventional refrigeration air conditioning compressor. In other words, where the heat delivered or the heat removed, Q_p, is accomplished by a mechanical vapor compression unit, then the total energy required is given by Equation (C65).

C.3.1 Fraction of the Heating Load Furnished by Solar Energy

Using Q_L to represent the total heating load in Btu/day and f to represent the fraction of the total heating load furnished by solar energy, then $f\ Q_L$ is the quantity of solar energy delivered to the heating load. Therefore $(1 - f)Q_L$ is the quantity of auxiliary energy required and:

$$Q_L = f\ Q_L + (1 - f)\ Q_L \qquad \text{(heating and cooling)} \qquad \text{(C66)}$$

and from Equation (C55):

$$f\ Q_L = Q_u = Q_c - Q_E \qquad \text{(heating)} \qquad \text{(C67)}$$

and:

$$f = \frac{Q_u}{Q_L} = \frac{Q_c - Q_E}{Q_L} \qquad \text{(heating)} \qquad \text{(C68)}$$

Equation (C68) is the conventional representation for the determination of f and often Q_E is assumed to be zero.

If the actual building heat load is Q_L Btu/day (kJ/day), then from Equation (C64), the amount of conventional energy required is:

$$Q_F = Q_L/\eta_F \qquad \text{(heating)} \qquad \text{(C69)}$$

If a heat pump is used, the amount of energy required is (from Equation C65):

$$Q_e = \frac{Q_L}{C_c \eta_E} \qquad \text{(heating)} \qquad \text{(C70)}$$

Therefore Equation (C66) must be rewritten to take into account Equations (C69) and (C70).

Using Q_u to represent the solar contribution to the total heating load and Q_η to represent the conventional contribution to the total heating load, and Q_L to represent the total heating load (all in Btu/day; kJ/day), then Equation (C66) can be rewritten:

$$Q_L = Q_u + Q_\eta \qquad \text{(heating)} \qquad \text{(C71)}$$

or

$$Q_\eta = Q_L - Q_u \qquad \text{(heating)} \qquad \text{(C72)}$$

In order to get Q_η, the quantity of energy required (from Equation C69 and C70) is either:

$$Q_F = Q_\eta/\eta_F \qquad \text{(heating)} \qquad \text{(C73A)}$$

or

$$Q_e = Q_\eta/C_c\eta_E \qquad \text{(heating)} \qquad \text{(C73B)}$$

so that the actual total building energy requirement, Q_A, in Btu/day (kJ/day) is:

$$Q_A = Q_u + Q_F = Q_u + \frac{(Q_L - Q_u)}{\eta_F} \qquad \text{(heating)} \tag{C74}$$

or

$$Q_A = Q_u + Q_e = Q_u + \frac{(Q_L - Q_u)}{C_c \eta_F} \qquad \text{(heating)} \tag{C75}$$

Equations (C74) and (C75) can be written:

$$Q_A = Q_u + \frac{(Q_L - Q_u)}{\eta_A} \qquad \text{(heating)} \tag{C76}$$

where:

$\eta_A = \eta_F$ for a furnace

$\eta_A = C_c \eta_E$ for a heat pump

$\eta_A = \eta_E$ for electric resistance heating

However, the total energy, Q_T, in Btu/day required to satisfy a heating load of Q_L Btu/day is:

$$Q_T = Q_u + \frac{(Q_L - Q_u)}{\eta_A} + \frac{E}{\eta_E} \qquad \text{(heating)} \tag{C76A}$$

where E is the electrical energy required to operate the solar heating system.

The conventional energy required for heating the building is $(Q_L - Q_u)/\eta_A$. If no solar energy was used to heat the building, then the conventional energy requirement would be Q_L/η_A. Therefore the gross energy saved by using the solar energy, Q_G, in Btu/day is:

$$Q_G = \frac{Q_L}{\eta_A} - \frac{(Q_L - Q_u)}{\eta_A} = \frac{Q_u}{\eta_A} \qquad \text{(heating)} \tag{C77}$$

The net energy saved by using solar energy, Q_s, in Btu/day (using Equation C63) is:

$$Q_s = Q_G - \frac{E}{\eta_E} = \frac{Q_u}{\eta_A} - \frac{E}{\eta_E} \qquad \text{(heating)} \tag{C78}$$

For solar heating, Q_u is given by Equation (C55), so:

$$Q_s = \frac{Q_c - Q_E}{\eta_A} - \frac{E}{\eta_E} \qquad \text{(heating)} \tag{C78A}$$

In addition we define S as the ratio of the useful solar heating to the electrical solar operating energy used, i.e.:

$$S = Q_u/E \qquad \text{(heating)} \tag{C79}$$

The overall solar system efficiency is:

$$\eta_s = \frac{Q_s}{AI + E/\eta_E} \qquad \text{(heating)} \qquad \text{(C80)}$$

C.3.2 Fraction of the Cooling Load Furnished by Solar Energy

Using Q_L to represent the total cooling load in Btu/day (kJ/day) and f to represent the fraction of the total cooling load furnished by solar energy, then Equation (C66) can be used. Using C to represent the coefficient of performance of an absorption solar cooling unit then:

$$fQ_L = C\ Q_u \qquad \text{(cooling)} \qquad \text{(C81)}$$

Substitution of Equation (C59) into Equation (C81) gives:

$$fQ_L = C\ Q_c - C\ Q_E - C\ Q_I\ (1 + 1/C) \qquad \text{(C82)}$$

$$= C(Q_c - Q_E)\ - Q_I(C + 1) \qquad \text{(cooling)} \qquad \text{(C83)}$$

Therefore, from Equation (C81):

$$f = \frac{C\ Q_u}{Q_L} \qquad \text{(cooling)} \qquad \text{(C84)}$$

Also the right hand term of Equation (C66) is:

$$(1 - f)Q_L = Q_p = C_c\eta_E Q_e \qquad \text{(cooling)} \qquad \text{(C85)}$$

Substituting Equations (C81) and (C85) into Equation (C66) gives:

$$Q_L = C\ Q_u + C_c\eta_E Q_e \qquad \text{(cooling)} \qquad \text{(C86)}$$

However, the total energy, Q_T, in Btu/day required to achieve a cooling of Q_L Btu/day (kJ/day) is:

$$Q_T = Q_u + Q_e + E/\eta_E \qquad \text{(cooling)} \qquad \text{(C87)}$$

Substituting Equation (C86) into Equations (C87) gives:

$$Q_T = Q_u + \frac{(Q_L - C\ Q_u)}{C_c\ \eta_E} + \frac{E}{\eta_E} \qquad \text{(cooling)} \qquad \text{(C87A)}$$

If no solar energy was used to cool the building, then the conventional energy requirement would be $Q_L/C_c\eta_E$. However, using solar energy to cool the building requires the following amount of conventional (non-solar energy):

$$\frac{Q_L - C\ Q_u}{C_c\ \eta_E} + \frac{E}{\eta_E} \qquad \text{(cooling)} \qquad \text{(C88)}$$

Therefore the net energy saved by using solar energy, Q_s, is:

$$Q_s = \frac{Q_L}{C_c\ E} - \left[\frac{Q_L - C\ Q_u}{C_c\ \eta_e} + \frac{E}{\eta_E}\right] \qquad \text{(C89)}$$

$$Q_s = \frac{C\ Q_u}{C_c\ \eta_E} - \frac{E}{\eta_E} \qquad \text{(cooling)} \qquad \text{(C90)}$$

For solar cooling, Q_u is given by Equation (C58) so substituting Equation (C58) into Equation (C90) gives:

$$Q_s = \frac{C}{C_C\ \eta_E} \left[Q_c - Q_E - Q_I\ (1 + 1/C) \right] - \frac{E}{\eta_E} \qquad \text{(C91)}$$

As before:

$$S = Q_u C/E \qquad \text{(cooling)} \qquad \text{(C92)}$$

The overall solar system efficiency is given by Equation (C80).

C.3.3 Equations Common to Both Solar Heating and Cooling

$$Q_u = Q_c - Q_E - Q_I \beta \qquad \text{(C59)}$$

where:

$\beta \approx 0$ for solar heating

β is given by Equation (C51) for solar cooling, i.e. $\beta = 1 + 1/C$

$$\eta_T = \frac{Q_u C}{AI} \qquad \text{(C61)}$$

where:

C = 1 for solar heating

$$f = \frac{C\ Q_u}{Q_L} \qquad \text{(C84)}$$

where:

C = 1 for solar heating

$$Q_T = Q_u + \frac{(Q_L - C\ Q_u)}{\eta_A} + \frac{E}{\eta_E} \qquad \text{(C87A)}$$

where:

C = 1 for solar heating

Q_T = Total energy required to satisfy a cooling or heating load of Q_L Btu/day

$\eta_A = \eta_F$ for a furnace

$\eta_A = \eta_E$ for electric resistance heating

$\eta_A = C_c\ \eta_E$ for a heat pump or conventional air conditioning unit

E = Electrical energy required to operate the solar heating or cooling system

C_c = Coefficient of Performance of conventional air conditioning unit or heat pump.

The energy saved by using solar energy for heating or cooling (in Btu/day; kJ/day) is:

$$Q_s = \frac{C\ Q_u}{\eta_A} - \frac{E}{\eta_E} \tag{C90}$$

where:

C = 1 for solar heating

For solar heating and cooling, the ratio of useful solar heating and/or cooling

$$S = \frac{Q_u\ C}{E} \tag{C92}$$

where:

C = 1 for solar heating

For both heating and cooling the overall solar system efficiency is:

$$\eta_s = \frac{Q_s}{AI + E/\eta_E} \tag{C80}$$

C.4 COMPARISON OF SOLAR HEATING AND SOLAR COOLING

It is worth noting that if electric resistance heating is avoided, η_A has about the same value for both heating and cooling, namely about 0.5. On the other hand, $\beta = 0$ for solar heating and β is about 3 for solar cooling. Another big difference is that C = 1 for solar heating and is about 0.5 for solar cooling. Consequently, if all other things are equal:

$$Q_u(\text{solar heating}) - Q_u(\text{solar cooling}) \simeq 3\ Q_I \tag{C93}$$

Another way of expressing Equation (C93) is:

$$\frac{Q_u(\text{solar heating})}{Q_u(\text{solar cooling})} \simeq 1 + 3\frac{Q_I}{Q_u} \tag{C94}$$

Consequently,

$$\frac{\eta_T(\text{solar heating})}{\eta_T(\text{solar cooling})} \simeq 2 + 6\frac{Q_I}{Q_u} \tag{C95}$$

In addition, for identical heating and cooling loads, Q_L:

$$\frac{f(\text{solar heating})}{f(\text{solar cooling})} \simeq 2 + 6\frac{Q_I}{Q_u} \tag{C96}$$

The ratio of the solar system's S value is:

$$\frac{S(\text{solar heating})}{S(\text{solar cooling})} \simeq 2 + 6\frac{Q_I}{Q_u} \tag{C97}$$

for the same electrical power consumption, E.

If the electric power consumption, E, is negligible then:

$$\frac{Q_s(\text{solar heating})}{Q_s(\text{solar cooling})} \simeq 2 + 6\,\frac{Q_I}{Q_u} \tag{C98}$$

so that:

$$\frac{\eta_s(\text{solar heating})}{\eta_s(\text{solar cooling})} \simeq 2 + 6\,\frac{Q_I}{Q_u} \tag{C99}$$

In summary, every conceivable parameter of performance (η_T, f, S, Q_s, η_s) for solar heating is at least double that for solar cooling even if $Q_I = 0$. Even if Q_I is only 17 percent of Q_u, this ratio is triple, etc. In terms of conventional energy saved, solar heating has at least double the potential of solar cooling. This analysis also clearly shows that for solar cooling Q_I must be zero!

These conclusions can be maintained, if we choose different values of the parameters. For example, assume:

Heating	Cooling	Heating and Cooling
$T_a = 5°C$	$T_a = 30°C$	$\eta_E = 0.25$
$T_i = 48°C$	$T_i = 88°C$	$C_c = 2$
$T_{min} = 40°C$	$T_{min} = 80°C$	$\eta_F = 0.7$
$C = 1$	$C = 0.68$	$\Delta T = 5°C$
$\beta = 0$	$\beta = 1 + 1/C = 2.5$	$\Delta T_c = 8°C$
$E = .05\ Q_u$	$E = .075\ Q_u$	
$\eta_A = 0.7$	$\eta_A = 0.5$	

With these assumptions (where primes denote cooling), we obtain:

$$I_o = 0.75\ I_o' \tag{C100}$$

$$Q_c = 0.75\ Q_c' \tag{C101}$$

$$Q_E = 0.75\ Q_E' \tag{C102}$$

$$Q_I = 0.375\ Q_I' \tag{C103}$$

$$\frac{Q_u}{Q_u'} = 0.75 + 5\,\frac{Q_I'}{Q_u'} \tag{C104}$$

$$\frac{\eta_T}{\eta_T'} = 1.5 + 10\,\frac{Q_I'}{Q_u'} \tag{C105}$$

$$\frac{Q_s}{Q_s'} = 1.125 + 7.5\,\frac{Q_I'}{Q_u'} \tag{C106}$$

$$\frac{\eta_s}{\eta_s'} = \frac{1.5 + 10\ Q_I'/Q_u'}{1 - 0.3\ Q_I'/Q_u'} \tag{C107}$$

When $Q_I' = (12.5\ \text{percent})\ Q_u'$,

$$Q_s = 2.1\ Q_s' \tag{C108}$$

$$\eta_s = 2.9\ \eta_s' \tag{C109}$$

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APPENDIX D
Solar System Costs

D.1 COSTS

Most of the information in this section is based on recent solar system cost studies by the Department of Energy, Mueller Associates, Inc., and Others [D1, D2, D3]. These data were based on commercial demonstration projects from the National Solar Data Network. Because of this source of data, it should be noted that many of the cost figures from these federally funded projects may be significantly higher than privately funded projects.

The data base includes accurate total system construction costs for 24 demonstration projects. These sites were not selected to be representative of all solar energy systems. A very large sample of systems would be needed to completely understand all the operant factors affecting solar system costs. Nonetheless, data on the 24 sites do provide a basis for some preliminary conclusions.

D.2 ASSUMPTIONS AND OTHER RELEVANT INFORMATION

1. Only one passive system is included in the study since few commercial demonstration projects utilize passive solar energy.

2. The costs data were obtained during visits to the sites, discussions with system designers, construction contractors and owners, and review of DOE vouchers.

3. The costs presented herein have been modified in order to account for differences in the contractual costs of the projects and the bare costs, by reducing the contractual costs by the amount of a standard overheat and profit rate.

4. All costs are expressed in 1977 dollars.

5. The cost figures do not include the design, instrumentation, or auxiliary energy system costs.

Further, the reader is reminded that, incomparing the costs of various systems and subsystems, "cheaper" is not necessarily "better".

D.3 TOTAL SYSTEM COST ANALYSIS

Table D1 summarizes the different project types. Table D2 describes the projects for which accurate total system and subsystem costs were obtained. Table D3 summarizes the category cost breakdowns for the 24 sites grouped according to type of application with costs presented as dollars per unit collector area and percent of the total system cost.

Table D1. Summary of Project Types

Description	Number of Sites
Passive	1
Heating	10
Process water	5
Heating and cooling	8
New	10
Retrofit	14
Air type collectors	5
Liquid type collectors	18

PROJECT NAME	APPLICATION[a]	LOCATION	CONSTRUC-TION[b]	COLLECTOR AREA, SQ.FT[c]	COST REPORT #[d]
1. Aberdeen First Baptist Church	SH,HW (AIR)	Aberdeen, SD	N	1,260	SOLAR/2070-79/60
2. Aratex Industrial Lounge	HW	Fresno, CA	R	6,350	SOLAR/2008-78/60
3. Billings Shipping Warehouse	SH (LIQ)	Billings, MT	R	1,660	SOLAR/2066-79/60
4. Blakedale Professional Center Office Bldg.	SH,HW (LIQ)	Greenwood, SC	N	928	SOLAR/2014-79/60
5. Charlotte Area Health Center	SH,HW (LIQ)	Charlotte, NC	R	3,950	SOLAR/2010-79/60
6. Columbia Gas Office Bldg.	SH & C, HW	Columbus, OH	R	2,978	SOLAR/2068-79/60
7. Concord Municipal Light Building Garage	SH (AIR)	Concord, MA	R	1,737	SOLAR/2048-79/60
8. DuCat Investments Warehouse	SH (AIR)	Kansas City, KS	N	7,000	SOLAR/2069-79/60
9. Hogates Restaurant	HW	Washington, DC	R	5,840	SOLAR/2028-79/60
10. Howard Grove School	SH (AIR)	Howard's Grove, WI	N	2,357	SOLAR/2041-79/60
11. Ingham Co. Medical Care Facility	HW	Okemos, MI	N	9,374	SOLAR/2056-79/60
12. Iris Images Photo Processing Lab	HW	Mill Valley, CA	N	600	SOLAR/2005-78/60
13. Irvine School	SH & C, HW	Irvine, CA	R	5,000	SOLAR/2021-79/60
14. Kalwall Warehouse	SH (PAS)	Manchester, NH	R	1.750	SOLAR/2015-70/60
15. Louden Co. School	HW	Leesburg, VA	N	1,169	SOLAR/2016-77/60
16. Terrell E. Moseley Office Bldg.	SH, HW (LIQ)	Lynchburg, VA	R	376	SOLAR/2011-78/60
17. Mount Rushmore Memorial Visitor's Center	SH & C	Keystone, SD	R	1,728	SOLAR/2019-79/60
18. North Hampton Recreational Center	SH & C	Dallas, TX	R	3,660	SOLAR/2039-79/60
19. Page Jackson School	SH & C	Charlestown, WV	N	10,943	SOLAR/2036-79/60
20. Radian Office Building	SH & C	Austin, TX	R	350	SOLAR/2002-78/60
21. Reedy Creek Utilites Office Building	SH & C, HW	Lake Buena Vista, FL	N	3,840	SOLAR/2018-79/60
22. Scattergood School Gymnasium	SH,HW (AIR)	West Branch, IA	N	2,240	SOLAR/2003-78/60
23. Telex Communications Assembly Plant	SH (LIQ)	Blue Earth, MN	R	10,700	SOLAR/2033-79/60
24. Trinity University Sports Complex & Dormitory	SH & C, HW	San Antonio, TX	R	15,633	SOLAR/2004-78/60

[a] SH = Space Heating, HW = Hot Water, SH & C = Space Heating and Cooling, (AIR) = Air Systems, (LIQ) = Liquid Systems, (PAS) = Passive System.

[b] N = New construction, R = Retrofit construction

[c] Net aperture area

[d] Available from the Technical Information Center, P.O. Box 62, Oak Ridge, Tennessee 37830. Project Description Documents have report numbers identical to Cost Report Numbers, except the last two digits are 50.

Table D2. Basic Information for Cost Analyzed Projects

D.3.1 Conclusions Drawn from Table D3

1. The unit cost for each application type varies considerably. Therefore one should be careful in making gross estimations in cost such as "heating systems in commercial building cost $ X/square foot of collector area, based on this data. There are, however, some indications [D4] that privately funded DHW systems cost in the area of $20 per square foot ($215 per square meter) and that heating and cooling systems average approximately $30 per square foot ($322 per square meter).

2. For most of the subcategories, very large variations are evident in unit costs:

Subsystem	Range of Costs per Square Foot of Collector Area @
Collector	$10 - $163
Support structure	$2.40 - $26
Piping, ductwork, insulation	$3.20 - $32

@Multiply by 10.7 to obtain $/m^2

Each system has unique characteristics that affect these category costs. In some extreme cases the incremental subsystem costs cannot be justified.

APPLICATION	PROJECT	TOTAL[a] SYSTEM COST, $	TOTAL[b] SYSTEM $/SQ FT	COLLECTOR		SUPPORT STRUCTURE		PIPING DUCTWORK, & INSULATION		HEATING COOLING/ EQUIPMENT		STORAGE		ELECTRICAL & CONTROL		GENERAL CONSTRUCTION	
				$/SQFT	%[c]	$/SQFT[d]	%	$/SQFT	%	$/SQFT	%	$/SQFT	%	$/SQFT	%	$/SQFT	%
Passive	Kalwall	13,530	7.7	6.1	79	--	--	--	--	0.6	8	0.8	10	0.3	3	--	--
Process Hot Water	Aratex	18,240	29.0	11.7	40	2.8	10	10.4	36	--	--	2.4	8	1.3	4	1.0	3
	Iris Images	18,115	30.2	10.5	35	7.2	24	4.0	13	--	--	2.1	7	1.8	6	4.4	15
	Ingham Co.	337,260	36.0	10.3	29	11.2	31	11.2	31	--	--	1.2	3	1.7	5	0.8	2
	Hogate's	283,955	48.6	18.2	37	8.7	18	13.1	27	--	--	4.1	8	1.1	2	3.4	7
	Loudoun Co.	70,940	60.7	21.8	36	17.2	28	12.2	20	--	--	5.4	9	3.3	5	0.8	1
	Average	---	40.9	14.5	35	9.4	23	10.2	25	--	--	3.0	7	1.8	4	2.1	5
Liquid Space Heating	Moseley	14,295	38.0	13.8	36	3.3	9	6.3	17	5.2	14	4.1	11	4.6	12	0.8	2
	Telex	462,305	43.2	17.3	40	6.6	15	13.9	32	1.0	2	1.5	3	2.0	5	0.9	2
	Billings	77,430	46.6	20.4	44	7.8	17	13.3	29	0.9	2	1.8	4	2.5	5	--	--
	Charlotte	249,195	60.8	18.0	30	5.9	10	24.8	41	0.2	<1	3.3	5	3.3	5	5.1	8
	Blakedale	60,510	65.2	12.0	18	15.4	24	21.0	32	--	--	8.2	13	8.5	13	0.1	<1
	Average	---	50.8	16.3	32	7.8	15	15.9	31	1.5	3	3.8	7	4.2	8	1.4	3
Air Space Heating	Howard's Grove	53,300	22.6	13.4	59	2.4	11	3.2	14	--	--	2.7	12	0.9	4	--	--
	DuCat	252,435	36.1	21.9	61	9.3	26	3.5	10	--	--	--	--	1.4	4	--	--
	Aberdeen	62,595	49.7	24.8	50	1.3	3	13.1	26	--	--	5.6	11	3.3	7	0.9	2
	Scattergood	113,825	50.8	24.5	48	11.5	23	6.0	12	2.1	4	3.8	7	2.7	5	0.2	<1
	Concord	118,740	68.4	18.6	27	25.7	38	12.3	18	--	--	5.2	8	6.2	9	0.4	1
	Average	---	45.5	20.6	45	10.0	22	7.6	17	0.4	1	3.5	8	2.9	6	0.3	1
Space Heating & Cooling	Page Jackson	526,845	48.1	15.9	33	15.8	33	8.4	17	3.3	7	1.5	3	0.3	1	3.0	6
	Irvine	285,190	57.0	28.3	50	7.7	14	15.0	26	--	--	--	--	4.8	8	1.3	2
	Trinity	958,210	61.3	29.1	47	5.5	9	13.2	22	2.9	5	4.2	7	2.9	5	3.6	6
	Mt. Rushmore	130,705	75.8	21.2	28	13.9	18	16.9	22	8.6	11	3.6	5	6.6	9	5.0	7
	North Hampton	384,680	105.1	21.4	20	6.3	6	31.5	30	7.4	7	8.9	8	16.2	15	13.4	13
	Columbia Gas	352,260	118.3	53.0	45	10.2	9	17.1	14	--	--	4.5	4	14.6	12	18.9	16
	Radian	42,150	120.4	37.4	31	15.7	13	30.9	26	14.4	12	7.7	6	9.8	8	4.8	4
	Reedy Creek	625,985	163.0	108.4	67	4.7	3	18.9	12	4.7	3	10.9	7	5.6	3	10.4	6
	Average	---	93.6	39.3	42	10.0	11	19.0	20	5.2	6	5.2	6	7.6	8	7.6	8
	Total Average	---	60.1	24.1	40	9.0	15	13.4	22	2.1	3	3.9	6	4.4	7	3.3	5

[a] All costs in 1977 $'s, costs include overhead and profit
[b] Dollars per square foot of Net collector area
[c] Percent of total system cost
[d] -- No category costs incurred

Table D3. Summary of System Costs

3. The percent of the total system cost values are relatively uniform within most of the cost categorics.

4. Storage costs are fairly consistent with only a few systems varying significantly from the average of $3.90 per square foot ($41.73 per square meter) of collector area.

Figure D1 illustrates the relationship between the total system unit costs for the 24 units for the five different system types.

D.3.2 Conclusions Drawn from Figure D1

1. Process water systems seem to be the least expensive active solar application on a per square foot basis, followed closely by space heating applications.

2. Space heating and cooling applications are much more expensive, on the average, than the other system types. However, there are exceptions to these general rules. Some cooling applications have been installed at lower relative cost than some process water systems.

3. There is a very large range of costs for every application type.

Figure D2 shows total system costs divided into two main groups, new and retrofit applications.

D.3.3 Conclusions Drawn from Figure D2

1. All but one of the cooling projects were retrofit. This fact may account for some of the higher average costs of the cooling projects in this sample. If no cooling

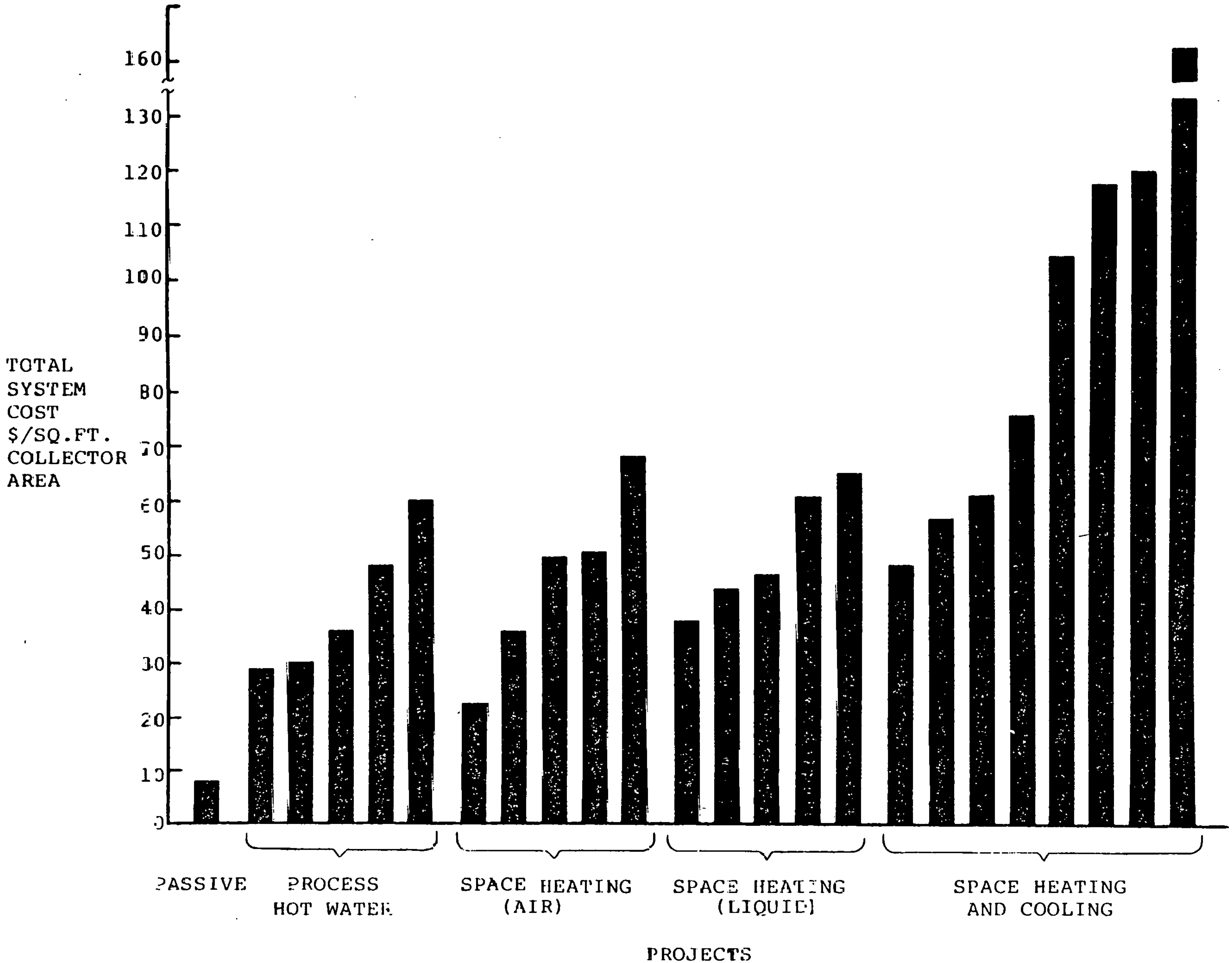

Figure D1. Total System Costs Grouped by System Type

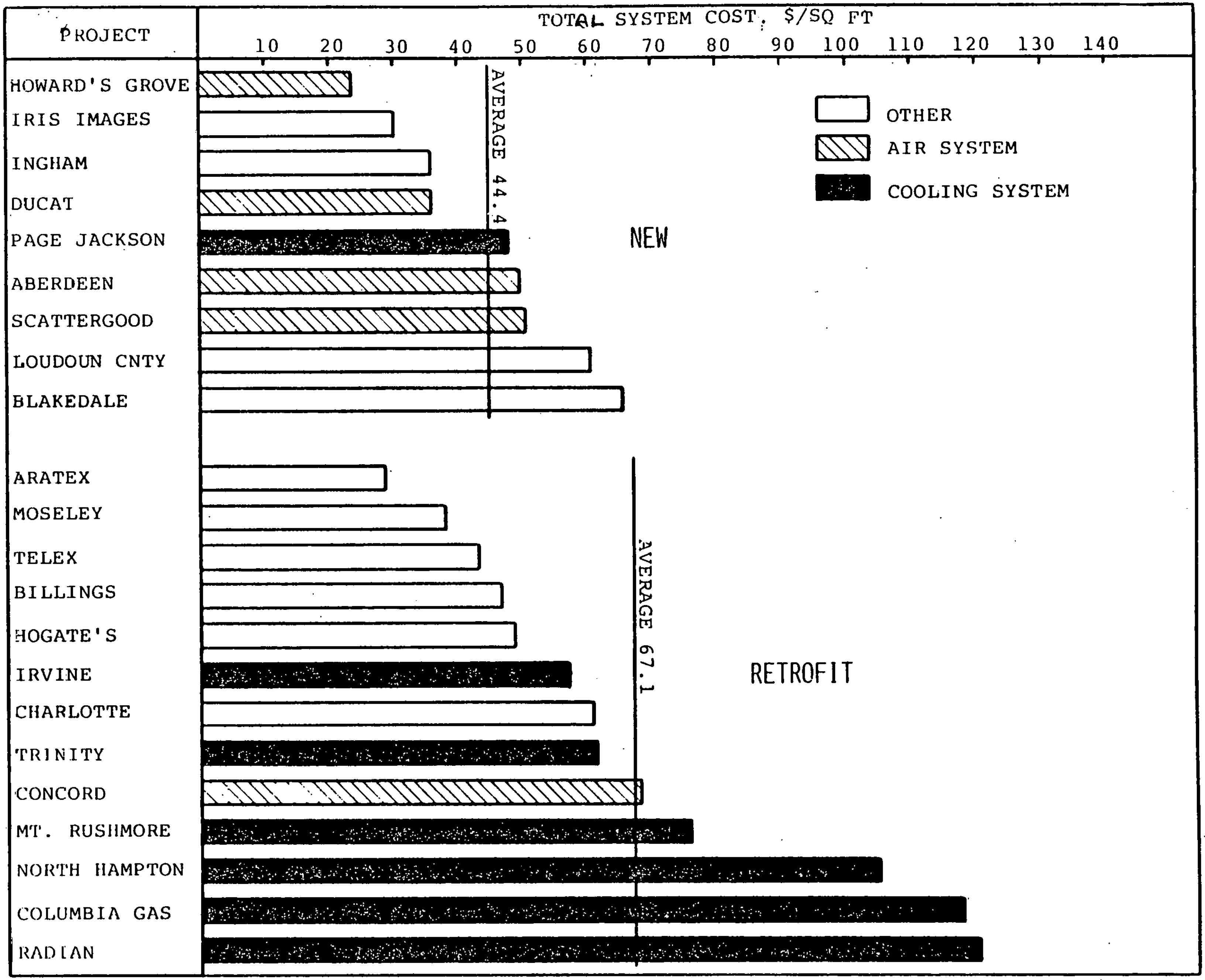

Figure D2. Total System Cost for New and Retrofit Systems projects are considered, the average cost is $44 per square foot (470 per square meter) and $48 per square foot ($513 per square meter) for retrofit systems.

2. All but one of the five air type space heating systems are new systems. This probably reflects the greater difficulty perceived for retrofitting air systems.

3. Given all the factors, a conclusion may be made that retrofit systems are only slightly more costly than systems in new construction.

D.3.4 Other Conclusions

1. Within the range of systems studied, economies of scale were found to be a relatively unimportant cost factor.

2. Regional cost variation influences were not found to be significant in the system costs studies.

3. Site-built flat-plate collectors ($8 per square foot; $85.60 per square meter) were found to be less expensive than the other flat-plaate collectors (material costs):

Collector Type	Average Collector Costs per square foot @@
Site-built, flat-plate	$8
Single glazed, flat black, flat-plate	$12
Double glazed, flat black, flat-plate	$16
Single glazed, selective, flat-plate	$16
Double glazed, selective, flat-plate	$16

@@Multiply $/ft^2 by 10.7 to obtain $/m^2

4. Within tho limited data studied, it was found that in all cases concentrating tracking collectors and the single evacuated tubular collector were more expensive per square foot than the flat-plate collectors.

5. Piping, ductwork, and insulation costs for systems in new construction (average cost of $8.88 per square foot of collector area; $951 per square meter) were found to be almost half that in retrofit systems (average cost of $18.16 per square foot of collector area; $194 per square meter). For various system types, piping, ductwork, and insulation costs varied as a percentage of total costs (as shown below):

Piping, Ductwork and Insulation Costs for	Average Costs per Square Foot of Collector @@
Air systems	$ 7.60 (17%)
Liquid type heating systems	$15.90 (31%)
Heating and cooling systems	$19.00

@@Multiply by 10.7 to obtain $/m^2

6. Storage costs are strongly dependent on the type of storage vessel used. Unpressurized systems storage costs were found to be lower. However, they represent a higher percentage of total system cost. This implies that pressurization probably impacts other costs as well as storage system costs.

Storage Vessel	Average Cost per Unit Storage Capacity ($/MBtu·°F) @
Unpressurized steel	143
Fiberglass	181
Pressurized steel	288
Rock bins	300
Residential water heater type	420

@Multiply $/Btu·°F by 10.2 to obtain $/kJ·°C

7. Storage tank location:
   (a) Buried tanks were found to be least expensive but only slightly less expensive than exterior tanks. (The added cost of piping to exterior or buried tanks was not considered and they should be added in calculating costs.)
   (b) Tanks placed within the buildings were, on the average, found to be significantly more expensive than others. (The analysis did not charge the cost of building space to the costs of interior storage.)

8. Controls and electrical costs were found to average seven percent of total system costs (Range: 1% to 13%) and averaged $4.40 per square foot ($47 per square meter) of collector area. The costs per actuator or controller were found to range from $325 to $925 (ignoring unusual data). Pumps, blowers, automatic valves, motorized dampers were all considered actuators. Note: Large variations in system costs were observed in all types. The reader is advised to be aware of the limited sample size and its implications regarding the validity of the averages presented.

Struss, et al [D3] in evaluating the economics of hotel/motel solar hot water projects made several additional claims:

1. With the proper application, a selective surface can be more cost-effective than a non-selective surface. Note that struss's sample did not include site-built collectors.

2. Projects cost between $100 and $200 per million Btu per year ($95 and $190 GJ per year).

3. Average design costs were 10% of the total system cost.

4. Wood support structures for collectors cost about half the cost of other types of support structures.

5. Liquid storage tanks averaged $1.50/gallon ($.39/liter).

6. Project material costs (excluding collectors) averaged about $6 per square foot ($64.50 per square meter) of collector area.

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