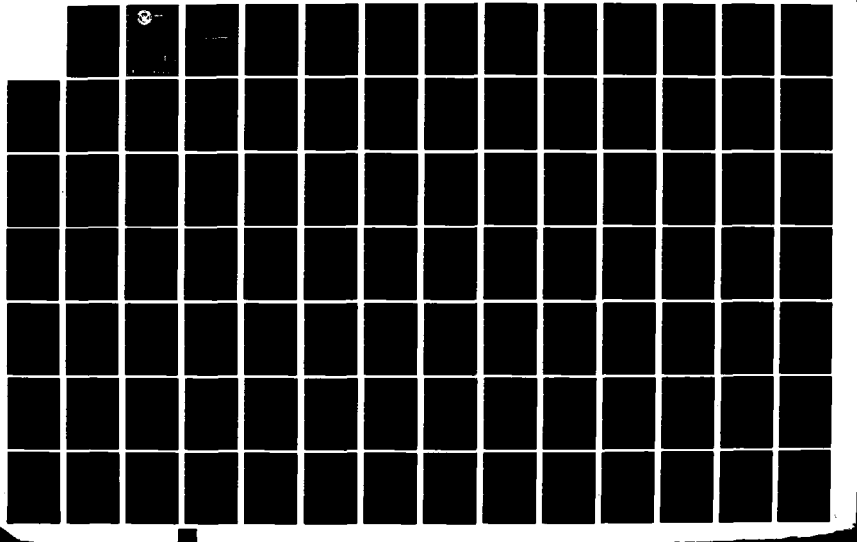
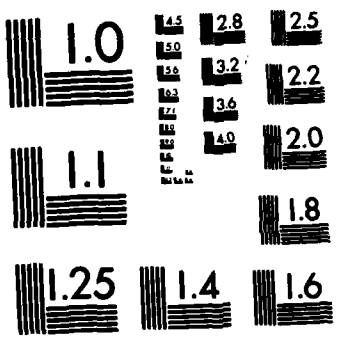
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METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

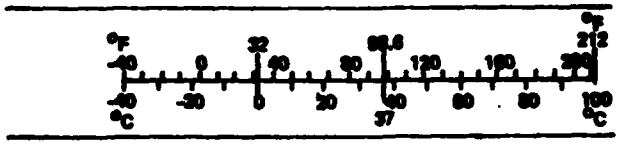
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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The purpose of this handbook is to provide information and criteria necessary for the selection and sizing of energy storage technologies for use at U.S. Naval facilities. The handbook gives Naval base personnel procedures and information to select the most viable energy storage options to provide the space conditioning (heating and cooling) and domestic hot water

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needs of their facility. The handbook may also be used by contractors, installers, designers, engineers, architects, and manufacturers who intend to enter the energy storage business.

The handbook is organized into three major sections: a general section, a technical section, and an example section. While a technical background is assumed for the latter two sections, the general section is simply written and can serve as an introduction to the field of energy storage. The technical section examines the following energy storage technologies: sensible heat storage, latent heat storage, cold storage, thermochemical storage, mechanical storage, pumped hydro storage, and electrochemical storage. The example section is limited to thermal storage and includes examples for: water tank storage, rockbed storage, latent heat storage, and cold water storage.

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We also thank the manufacturers who have corrected our discussion about their products, or allowed us to use drawings of their products or to refer to their trademarks and products in the sample specifications. These manufacturers are named on the pages where their drawings, trademarks, or product names appear.

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ABSTRACT

The purpose of this handbook is to provide information and criteria necessary for the selection and sizing of energy storage technologies for use at U.S. Naval facilities. The handbook gives naval base personnel procedures and information to select the most viable energy storage options to provide the space conditioning (heating and cooling) and domestic hot water needs of their facility. The handbook may also be used by contractors, installers, designers, engineers, architects, and manufacturers who intend to enter the energy storage business.

The handbook is organized into three major sections: a general section, a technical section, and an example section. While a technical background is assumed for the latter two sections, the general section is simply written and can serve as an introduction to the field of energy storage. The technical section examines the following energy storage technologies: sensible heat storage, latent heat storage, cold storage, thermochemical storage, mechanical storage, pumped hydro storage, and electrochemical storage. The example section is limited to thermal storage and includes examples for: water tank storage, rockbed storage, latent heat storage, and cold water storage.

1.0 INTRODUCTION

1.1 Purpose and Scope

The purpose of this handbook is to provide information and criteria necessary for the selection and sizing of energy storage technologies for use at U.S. Naval facilities. In other words it will provide naval base personnel with the appropriate procedures and information to select the most viable energy storage options to provide the space conditioning (heating and cooling) and domestic hot water needs of their facility. Transportation energy requirements and energy needs of military engagements are beyond the scope of this handbook.

Our national recognition of solar energy as a feasible energy source has enhanced the need for energy storage. The energy storage systems described in this handbook however should be considered whenever energy supply does not match energy demand. The technical aspects of energy storage are stressed. Although actually of much broader application, the technical information presented will be geared towards climates typical of continental United States. Constraints on implementation, such as building codes and environmental considerations, are mentioned only in passing as cautions not to be overlooked in any design.

The handbook is organized into three major sections: a general section, a technical section, and an example section. While a technical background is assumed for the latter two sections, the general section is simply written and can serve as an introduction to the field of energy storage. A glossary defines most of the technical terms used.

The general section first outlines the available energy storage technologies in Chapter 2, and then, in Chapter 3, explains how to select an appropriate system.

The technical section in Chapters 4-9 examines thermal storage technologies, including hot storage, cold storage, sensible heat, latent heat, and reversible chemical reactions. Mechanical storage, pumped hydro storage, and electrical storage are reviewed in Chapters 10-12 respectively. The discussion of each energy storage technology includes both what is currently available and likely future directions. Detailed sizing information, explaining how to determine the appropriate storage capacity, is included only for technologies that are currently available and easily purchased. Further information on each technology may be found in the references at the end of each chapter and in the sources listed in the bibliography.

The example section is limited to thermal energy storage. First, the currently available design and sizing methods are described in Chapter 13. This is followed by a set of design and sizing worksheets in Chapter 14. A detailed description of the use of each worksheet is provided through the four solved problems provided in Chapter 15. The worksheets are designed to enable engineers to conduct basic feasibility studies, initial sizing calculations, and cost benefit analyses for the use of thermal storage in both new and retrofit (remodelling-type) applications.

Data and information are presented in English units (e.g. ft and BTU). In the text and tables metric equivalents (e.g. m and J) enclosed in parentheses follow quantities given in English units. An appendix on units and conversion units is included.

1.2 How to Use This Handbook

This handbook was written for use by three types of readers: (1) the person with little or no technical background needing to survey energy storage technology, (2) the person with a technical background needing to understand the state of the art in one or more areas of energy storage technology, and (3) the person with a technical background needing to size a thermal storage system for a particular application. A schematic flowpath on the use of this handbook for each type of reader is presented in Table 1-1.

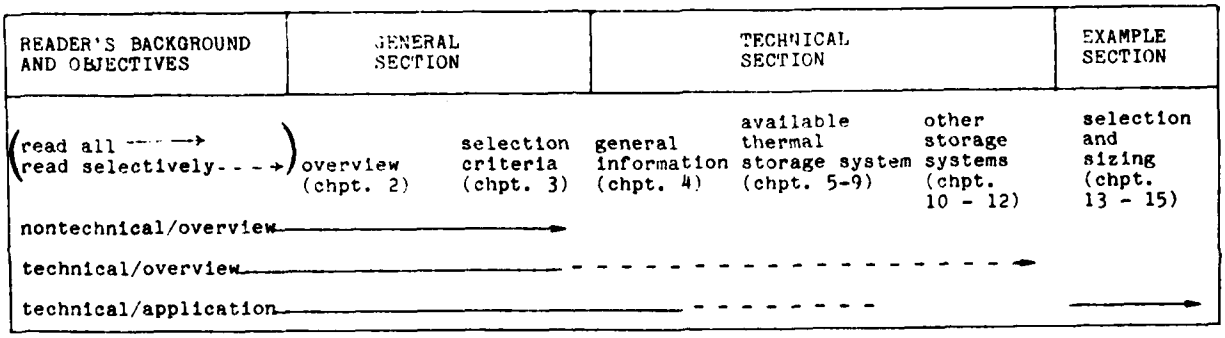


Table 1-1 Suggested Procedure for Using Handbook

(1) The nontechnical reader can receive a general introduction to energy storage technology by reading Chapter 2. A better understanding can then be obtained by studying the references at the end of Chapter 2 and some of the sources listed in the bibliography. The general feasibility of a given type of storage technology for a particular application can be determined by the nontechnical user from the information in Chapter 3, comparing and evaluating different energy storage systems.

(2) A person with a technical background wanting to thoroughly understand one or more areas of energy storage technology should first read Chapter 2, and possibly Chapter 3, to compare that system with other storage technologies. He or she can then read the appropriate chapter in the technical section (Chapters 4-12) for detailed information on a given technology. Each chapter in the technical section develops the basic concepts for each technology, surveys the state of the art, and indicates likely future developments. The references at the end of each chapter suggest suitable textbooks and current technical reports for further study.

(3) A person with a technical background and a need to size a thermal energy system for a particular application is advised to first read Chapters

2-3 so that the appropriate storage technology is selected. The user should then read Chapter 4 on the general features of thermal storage, together with one or more of Chapters 5-9 to become familiar with currently available systems for the chosen technology. Finally, this user should read Chapters 13-15 and perform the worksheet calculations appropriate to the particular application and storage technology. If further sizing calculations are necessary, some of the methods suggested in Chapter 13 may be useful. References to installation procedures are included so that bidding and specification documents may be prepared with minimal extra effort.

The preceding paragraph and Table 1-1 provide a general format for technical use. The following paragraphs, Table 1-2, and Table 1-3 provide a format to users who are either source- or application-oriented. For example, if the available energy source is waste heat the reader can use this manual to ascertain the most appropriate end use. Conversely, if the specific objective is domestic hot water the handbook will help the user determine the most appropriate source and storage system available.

Source Oriented Study

SOURCE	OVERVIEW	STORAGE SELECTION	TECHNICAL REFERENCE	WORKSHEETS	APPLICATIONS
SOLAR THERMAL	Section 2.2.1	water tank → Sections	5.2, 5.3 → 14.1,	15.1	domestic hot water
		latent heat → Section	7.4 → 14.3,	15.3	
		water tank → Sections	5.2, 5.3 → 14.1,	15.1	domestic space heating
		latent heat → Section	7.4 → 14.3,	15.3	
		rock bed → Sections	6.2, 6.3, 6.4 → 14.2,	15.2	
		water tank → Sections	5.2, 5.3 → 14.1,	15.1	domestic space cooling
		latent heat → Section	7.4 → 14.3,	15.3	
		latent cold → Section	2.6, Chpt. 8 → 14.4,	15.4	
WASTE HEAT	Section 2.2.5	water tank → Sections	5.2, 5.3 → 14.1,	15.1	domestic hot water
		latent heat → Section	7.4 → 14.3,	15.3	
		water tank → Sections	5.2, 5.3 → 14.1,	15.1	domestic space heating
		latent heat → Section	5.4 → 14.3,	15.3	
		rock bed → Section	6.2, 6.3, 6.4 → 14.2,	15.2	
		water tank → Sections	5.2, 5.3 → 14.1,	15.1	domestic space cooling
		latent heat → Section	7.4 → 14.3,	15.3	
		latent cold → Section	2.6, Chpt. 8 → 14.4,	15.4	
OFF-PEAK ELECTRICITY	Section 2.2.4	water tank → Section	2.3, 5.2, 5.3 → 14.1,	15.1	domestic hot water
		latent heat → Section	2.4, 7.4 → 14.3,	15.3	
		water tank → Section	2.3, 5.2, 5.3 → 14.1,	15.1	domestic space heating
		latent heat → Sections	2.5, 7.4 → 14.3,	15.3	
		water tank → Sections	2.3, 5.2, 5.3 → 14.1,	15.1	domestic space cooling
		latent heat → Sections	2.5, 7.4 → 14.3,	15.3	
		latent cold → Sections	2.6, Chpt. 8 → 14.4,	15.4	

Table 1-2 Source-Oriented Procedure for Using Handbook

Table 1-2 provides a procedure for using the handbook for source-oriented users. Examples of this procedure follow below.

Solar Thermal. Read section 2.2.1 for a brief overview of this energy source and scan references at the chapter end for further information. Next examine Table 3-2 for an overview of the suitability of different energy storage technologies for this energy resource. Examine Table 3-1 for assistance in selecting an application appropriate to the storage type. Choose several appropriate paths.

Since various alternative source/storage/application scenarios have now been specified, the Table of Contents will give further direction to reading content.

If an attractive pathway was solar-thermal/water-tank/domestic-hot-water, the reader could profitably study chapter 5, design example 14-1, and complete associated worksheets by using example problem 15.1 as a guide. Appendices B,C,E, and G will also be particularly useful.

Waste Heat. Waste heat, as an energy source, is discussed in section 2.2.5. Study protocol is analogous to that presented for solar thermal. By using Tables 3-1 and 3-2, determine several viable source/storage/application options. Example problem 15.4 discusses in detail a waste-heat/absorption-chiller/space-cooling sequence. Section 8-1, design example 14-1, and example problem 15-4 will direct the reader through a satisfactory completion of design specifications on worksheets in 14.4.

Application-Oriented Study

APPLICATION	OVERVIEW	STORAGE SELECTION	TECHNICAL REFERENCE	WORKSHEETS	SOURCE
DOMESTIC HOT WATER	Chapt. 2	water tank	Sections 5.2,5.3	14.1,	15.1 → Solar Thermal, Waste Heat
		latent heat	Section 7.4	14.3,	15.3 → Off-peak Electricity
DOMESTIC SPACE HEATING	Chapt. 2	water tank	Sections 5.2,5.3	14.1,	15.1 → Off-peak Electricity, Solar Thermal, Waste Heat
		latent heat	Section 7.4	14.3,	15.3 →
		rockbed	Section 6.2,6.3,6.4	14.2,	15.2 →
DOMESTIC SPACE COOLING	Chapt. 2	latent cold	Chapt. 8	14.4,	15.4 → Ambient Thermal
		water tank	Sections 5.2,5,3	14.1,	15.1 → Off-Peak Electricity, Solar Thermal, Waste Heat
		latent heat	Section 7.4	14.3,	15.3 →

Table 1-3 Application-Oriented Procedure for Using Handbook

Table 1-3 provides a procedure for using the handbook for application-oriented users. Examples of this procedure follow below.

Domestic Hot Water. Scan chapter 2 for a brief overview of the book and scan references for further information sources specifically on hot water. The two most viable storage options, as summarized in tables 3-1 and 3-2, are water tank and latent heat. Either of these can be effectively coupled with one of three sources. As shown in Table 1-3, these are solar thermal, waste heat, or off-peak electricity. Appropriate reading selections are also shown in Table 1-3 for this example.

Domestic Space Cooling. Table 1-3 summarizes the best study regime to understand and evaluate this energy application. Unlike the case of domestic hot water, note that there is not symmetrical coupling between storage options and energy sources. Latent cold can be effectively coupled to only one of four different sources; latent heat and latent heat storage can only be coupled to three sources.

2.0 OVERVIEW OF AVAILABLE ENERGY STORAGE

2.1 General Features of Energy Storage

Generally speaking, energy storage is necessary when the supply of energy does not match the energy demand, or load. If the source is derived from a solar collector, this energy is only available during the day when the sun is shining. If part of the load is at night or on a cloudy day, then energy collected when the sun is shining must be stored for use at these times.

Storage may also be desirable when a number of energy sources are available but when the most cost-effective source is only available part of the time. Depending on the generation equipment available and the load profile of their customers, electric utilities will often sell electricity at different rates during different parts of the day. If electricity is used to power air conditioning equipment to cool a building, for example, it may prove more economical not to buy electricity during the periods of highest rates (on-peak hours) and cool the building directly. Instead the air conditioning equipment would be powered only during the period of lowest rates (off-peak hours) to cool a storage tank of water. The chilled water would then be circulated through the cooling system whenever cooling is required. This discussion presupposes that the difference in electricity rate structure would be large enough to make the additional cost of storage economical.

Energy storage is also necessary when reliability of energy supply is important. A typical example of this is the installation of battery powered lights in buildings where many rooms do not have an outside window. During normal operation the batteries are kept charged by the building's electrical system. During power failure, the batteries would power adequate lighting to allow safe evacuation of the building.

Since 1975, there has been growing appreciation for the importance of considering energy quality in making optimal use of energy flows. Energy quality can be defined as the maximum amount of work that can be produced from the energy of a system; it is related to the level of energy density and temperature in the system. In general, a concentrated form of energy or a large temperature indicates a high energy quality. Since more work can be produced from high quality than from an equivalent amount of low quality energy, it is more versatile and more valuable. Some tasks, such as producing mechanical motion, are best suited to high quality energy, while other tasks, such as the space heating of buildings, are better suited to low quality energy. The optimal use of energy is usually obtained by matching the energy quality of the source with the energy quality of the task.

When energy flows or is converted from one form to another, some is lost to the environment. The fraction of the original energy retained after the energy conversion is called the conversion efficiency. The maximum obtainable conversion efficiency varies dramatically, depending on the type of conversion. The energy storage chosen for a particular application should limit the number of energy conversions and follow conversion paths where efficiencies are relatively high.

2.2 Energy Sources

The types of energy sources available strongly influence the selection of an energy system. These are classified either as primary, such as fossil fuels or solar energy which occur naturally; or as secondary, such as electricity, which is manmade. The next five sections discuss the currently available energy sources that are likely to require energy storage.

2.2.1 Solar Thermal

The sun radiates solar energy that is equivalent to that emitted from a heat source at a temperature of approximately 6000°C. Solar thermal energy is the heat energy produced when an object absorbs solar radiation. Solar radiation received at the ground is composed of two parts: direct radiation from the immediate direction of the sun and diffuse radiation from molecular scattering in the atmosphere, clouds, and objects on earth.

Solar energy has two major limitations: (1) Solar energy has a low flux density, that is, only small amounts of energy fall on each square foot (square meter) of surface exposed to the sun. This requires the commitment of a large collection surface area, making the collector the most expensive part of any solar system. The amount of solar flux varies considerably with geographic area; average amounts are generally higher for higher altitudes and sunnier locations. A solar energy availability map for the conterminous United States is presented in Appendix C. (2) Solar energy is intermittant, making storage essential in most applications. Sunshine is not available at night and is severely minimized during bad weather. There is also seasonal variation; solar flux is reduced during the winter for latitudes typical of the United States.

The type of solar collector used depends upon the required temperature associated with the load or energy demand. Concentrators are required for high temperatures and tracking is necessary for the highest temperatures. For low temperatures, flat-plate collectors are generally less expensive than concentrating collectors. Collection efficiency varies with temperature for each type of collector but always decreases with higher temperatures. Many types of fluids and fluid combinations are possible for the circulating fluid in solar collectors but air and water are the most common, especially in low temperatures.

2.2.2 Photovoltaic

Photovoltaic conversion is the direct transformation of solar radiation into electricity. When energetic photons (particles of light) are absorbed at the interface between certain types of materials, a voltage is produced. If the material pair is connected to an external load, an electric current is generated. The device consisting of the material pair and external connectors is called a solar cell.

Since photovoltaic conversion derives energy from solar radiation, the limitations discussed in the previous section for solar thermal devices, that is, low flux density and intermittent availability, also apply to solar cells. Originally developed for the space program, commercially available solar cells made from single crystal silicon can currently achieve a conversion efficiency of about 15%. Although the conversion efficiency is lower than for solar

thermal conversion the energy produced is high quality electricity. Solar cells are currently economical only in remote areas where it is too costly to run electrical transmission lines or transport fossil fuels. One attempt to reduce cost is to use a relatively inexpensive concentrating system to focus sunlight on a highly efficient (conversion efficiency between 20% and 28%), but expensive, gallium-arsenide solar cell. Other current research efforts include trying to improve the conversion efficiencies of other materials, e.g. polycrystalline silicon, amorphous silicon, and cadmium-sulfide, and also to reduce associated material and fabrication costs.

2.2.3 Wind

Wind energy is derived from the uneven heating of the earth's surface by solar radiation. A wind turbine converts the kinetic energy of moving air into mechanical motion, usually in the form of a rotating shaft. The rotating shaft can then drive a pump or an electric generator.

The available power in wind energy is proportional to the cube of the wind speed, that is, if the speed is doubled, the available power is eight times greater. This relationship puts a premium on potential sites that have large average wind speeds. Some of these sites can be identified on the average wind availability map in Appendix D. Wind speeds do vary considerably, however, both daily and seasonally, and significant differences may be found between locations only a mile apart. In general, wind speeds are larger with increased height from the ground. Rough terrain lowers wind speeds near the ground. The most effective locations for wind turbines are likely to be on flat open country, on tops of hills, or places where the topography directs the prevailing winds through narrow channels.

There are currently a large number of commercially available small and medium size wind turbines. Economic analyses have shown that large machines are more cost effective, but a significant number of large machines built to date have been prone to mechanical failure.

2.2.4 Off-Peak Electricity

Electricity demand by utility customers varies considerably during the day. The largest demand is in the late afternoon and early evening and the smallest demand is in the period after midnight and before sunrise. As net demand for electricity grows, utilities are often hard pressed to meet the entire electrical needs of their customers. It is increasingly common for utilities to offer significantly reduced rates for electricity consumption during the off-peak hours. Users benefit financially by buying the off-peak electricity and storing the energy for use during the peak demand period during the next day. This technique has been most widely used with thermal energy storage.

2.2.5 Waste Heat

Recovery of waste heat normally lost as (1) unburned but combustible materials (waste fuel), (2) sensible heat discharged from drainpipes (drain water) and (3) sensible and latent heat exhausted from stacks (flue gas) has been practiced for decades in energy intensive industries. The economic attractiveness of heat recovery has been increasing along with rising fuel

costs; the pay-back period for boiler gas-to-gas heat recovery is now two to three years.

Millions of Btu's can be reclaimed, as waste fuel, drain water, and flue gas are converted to energy sources (see Figure 2-1). Notice that only two major types of heat recovery equipment are necessary: the combustion equipment which eliminates discharge of combustible materials and the heat transfer equipment.

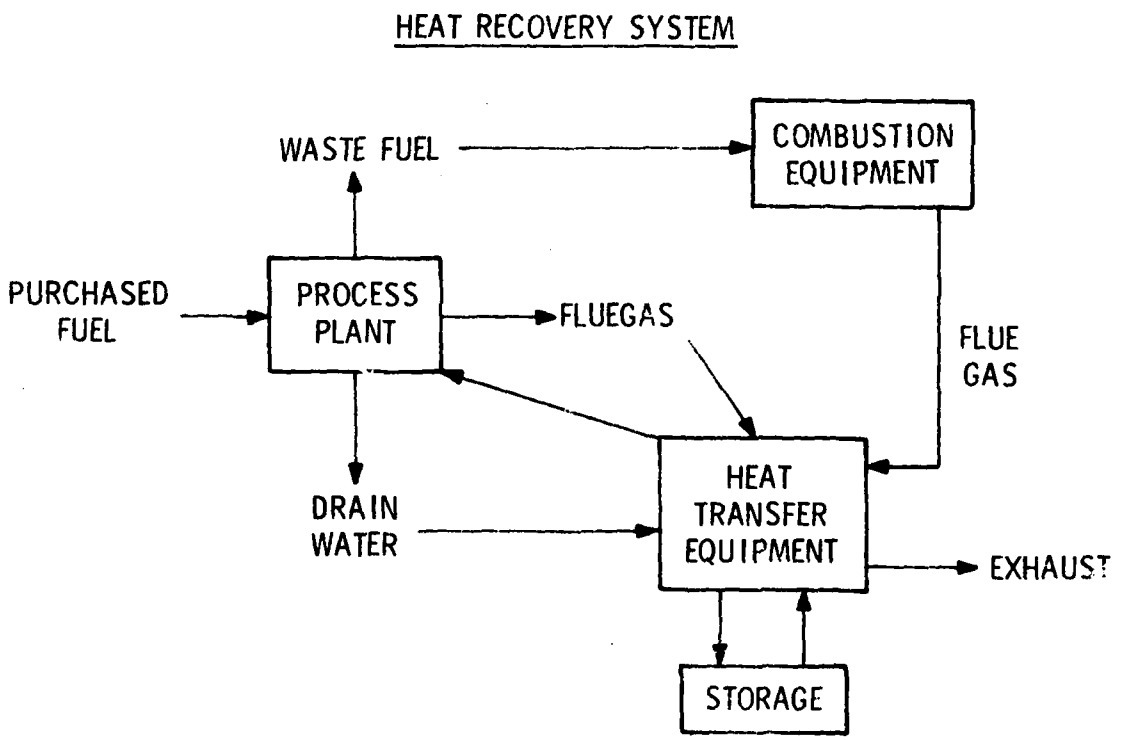


Figure 2-1. Typical process plant energy flow showing possible modes of heat recovery

Heat Transfer Equipment Options

The same heat transfer equipment is used to minimize energy losses from waste fuel, drain water, and flue gas. Actual selection of the heat recovery technology depends on the characteristics of application, process plant requirements, and economic need. There are five major types extensively used:

- Sensible heat exchangers are classified into gas-to-gas (waste fuel recovery), gas-to-liquid (flue gas recovery), and liquid-to-liquid (drain water recovery) systems.

- Steam generation equipment (waste heat boilers) can produce low pressure steam to be used for space heating or absorption air conditioning.

- Condensing type heat exchangers are used to recover the latent heat of water (550 Kcal/Kg) or organic contaminants (for benzene, 120 Kcal/Kg) as well as reduce hydrocarbon emissions.

- Chiller-type heat exchangers can exploit low temperature flue gases, waste fuel, and liquids in absorption refrigeration processes.

- Direct contact heat exchangers are exemplified by bubbling flue gas through water and thus transferring the flue gas sensible heat to vaporized water.

Waste Heat Recovery Systems

The following are well known waste heat recovery systems:

- Co-generation, in one operation, produces both heat and electricity. As Table 2-1 illustrates, boilers are about 80% efficient; there is much usable energy in the flue gas going up the stack.

- Heat pumps are capable of upgrading the escaping flue gas and using it for higher temperature applications. A heat pump system can be based on a chiller-type heat exchanger utilizing a vapor-compression or absorption cycle (see section 8.1).

- Waste incineration, with appropriate heat exchangers, can generate heat that can be used to preheat waste or combustion air or can be transferred to other in-plant locations. Waste heat boilers are often coupled with solid or fume waste incinerators in England. Some liquid waste incinerators currently use direct contact heat exchangers.

Boiler Output Capacity (Q) (10^6 Btu/hr)	Mean Capacity (10^6 Btu/hr)	Average Annual Fuel Consumption (gal/boiler)	Mean Boiler Efficiency (%)	No. of Boilers
$0.35 \leq Q < 5.0$[a]	1.8	13,500	79.8	1,800[b]
$5.0 \leq Q < 20.0$	9.3	107,00	79.2	344
$Q \geq 20.0$	60.0	95[illegible],000	80.7	261

[a]Mean annual hours of burner operation is 850 hours.

[b]Estimated number of boilers.

Adapted from J. Ashley. Automatic Boiler Control Options for Naval Shore Boiler Plants. Civil Engineering Laboratory Technical Manual 63-79-23, Naval Construction Battalion Center, Port Hueneme, CA, December 1979.

Table 2-1 Efficiency of Naval Shore Boilers

2.3 Sensible Heat

In a sensible heat storage system, the heat flowing into storage raises the temperature of the storage material. When needed, heat is removed from storage and the temperature of the storage material drops. Sensible heat may be used in thermal storage situations whenever the match between storage and load allows substantial temperature changes in the storage material. Many variations of sensible heat storage systems are possible, but these systems always include (1) a heat storage material, (2) a well-insulated container, and (3) provisions for efficiently adding or removing heat. Both liquids and solids are commonly used for sensible heat storage.

Heat losses from storage are directly proportional to the temperature difference between storage and the environment. To minimize heat losses, sensible heat storage must (1) be of short duration, (2) be heavily insulated, (3) operate at temperatures close to the ambient temperature or (4) be a large system where the surface/volume ratio is small.

Sensible heat storage systems are discussed in more detail in Chapters 4-6.

2.3.1 Liquid

The temperature range of sensible heat storage systems using liquids as the storage material is restricted by the melting point and boiling point of the liquid. The desired storage temperature then determines the appropriate liquid; water is the most common choice below 212°F (100°C). Organic liquids, high molecular weight oils, and pressurized water are effective at higher temperatures. For the highest temperatures, liquid metals such as sodium or molten inorganic salts are used. Liquids require containment vessels with considerable integrity to prevent leakage.

Figure 2-2 illustrates the use of a typical liquid storage system, with a solar collector supplying the input heat, and building space heating serving as the load. Water tanks are commonly used for storage in space heating and domestic hot water use due to their commercial availability and installation simplicity in retrofit instances. In some situations the water used for storage can be circulated both in the solar collector and also in hydronic space heating networks which are hot water radiators or water pipes buried in the floor. Air space heating systems require a heat exchanger between the liquid storage and the air. To prevent contamination of potable water in domestic hot water systems a heat exchanger is required between the collector loop and storage. A double wall heat exchanger is required by most building codes if the collector fluid is other than potable water.

2.3.2 Solid

Solid-phase storage has an advantage over liquid storage in allowing higher temperature changes of the storage material. Most common solid storage materials do not melt readily and can be used at temperatures associated with concentrating collectors. Since solid materials have little tendency to flow, minor leakage from the containment vessel is less of a problem than for liquid storage.

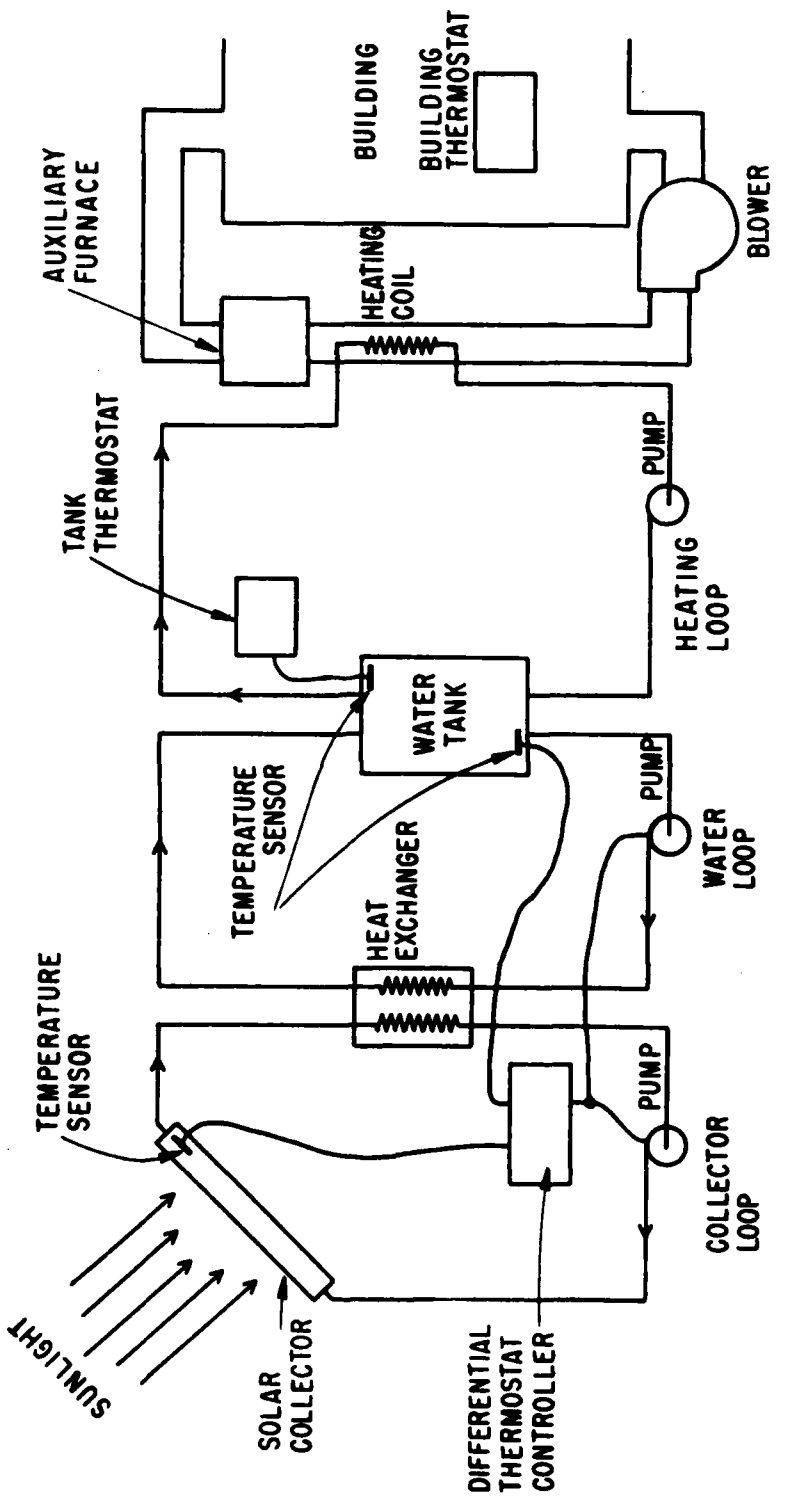


Figure 2-2. Typical Liquid-based System

Figure 2-3 illustrates the use of a typical solid material storage system. A solar air collector is the energy source, and a forced air heating system is the load. Due to the prevalence of forced air furnaces this type of system is common in many residential retrofits. Among the most common solid-phase storage systems are rock beds. Air is the most common heat transfer fluid used with rock beds but liquids can also be used in some situations. For example, heated water from inexpensive solar collectors, trickled through a storage medium of porous concrete, has proven cost effective in heating greenhouses. One experimental solar-powered air-conditioning system uses oil to transfer 300°F (150°C) heat to rock storage.

2.4 Sensible Cold

In a sensible cold storage system, heat is removed from storage, thus lowering the temperature of the storage material. When cooling is desired heat is moved from the load to storage, and the temperature of the storage rises. The cold storage may consist of chilled water that is circulated upon demand in a building's air conditioning system. Alternatively, the cold storage could consist of cold earth or rocks that are used as a heat sink to improve heat pump efficiency. Sensible cold storage is discussed in Chapter 8.

Commercial chillers operate more efficiently when run for long periods of time. Coupling cold storage to chillers effectively reduces the chiller cycling frequency, especially in periods of low demand. A more continuous chiller operation also reduces the amount of installed cooling capacity needed, that is, the size of the chiller can be smaller.

In temperate climates it may be practical to use the cold winter air to remove heat from storage, thereby reducing the cost of producing the cold. If the cooling load is in the summer, seasonal storage is then necessary, requiring a larger amount of storage and an increased amount of insulation. Several systems of this type are presently undergoing reseach and development.

A limitation of sensible cold storage is the decreased efficiency realized in producing very cold storage temperatures. The efficiency of pumping heat from a cold source to a hot source decreases as the temperature difference between the two sources increases. If pumping heat to the ambient air is the method of producing the cold storage, there is a minimum temperature, beyond which it is impractical to refrigerate the cold storage. During most periods when space cooling is needed the ambient air is warm.

The useful temperature range of water in sensible cold storage is usually less than in sensible heat storage, requiring a larger volume of storage for an equivalent amount of energy stored. In many geographical regions a large fraction of the space cooling load is devoted to dehumidification. This imposes an upper temperature limit on the chilled water, as temperatures above 55°F (13°C) cannot reduce the humidity to a comfortable level. A lower temperature limit, several degrees above the freezing point (32°F or 0°C), is imposed with commonly used chillers to avoid freezing ice on the chiller coils.

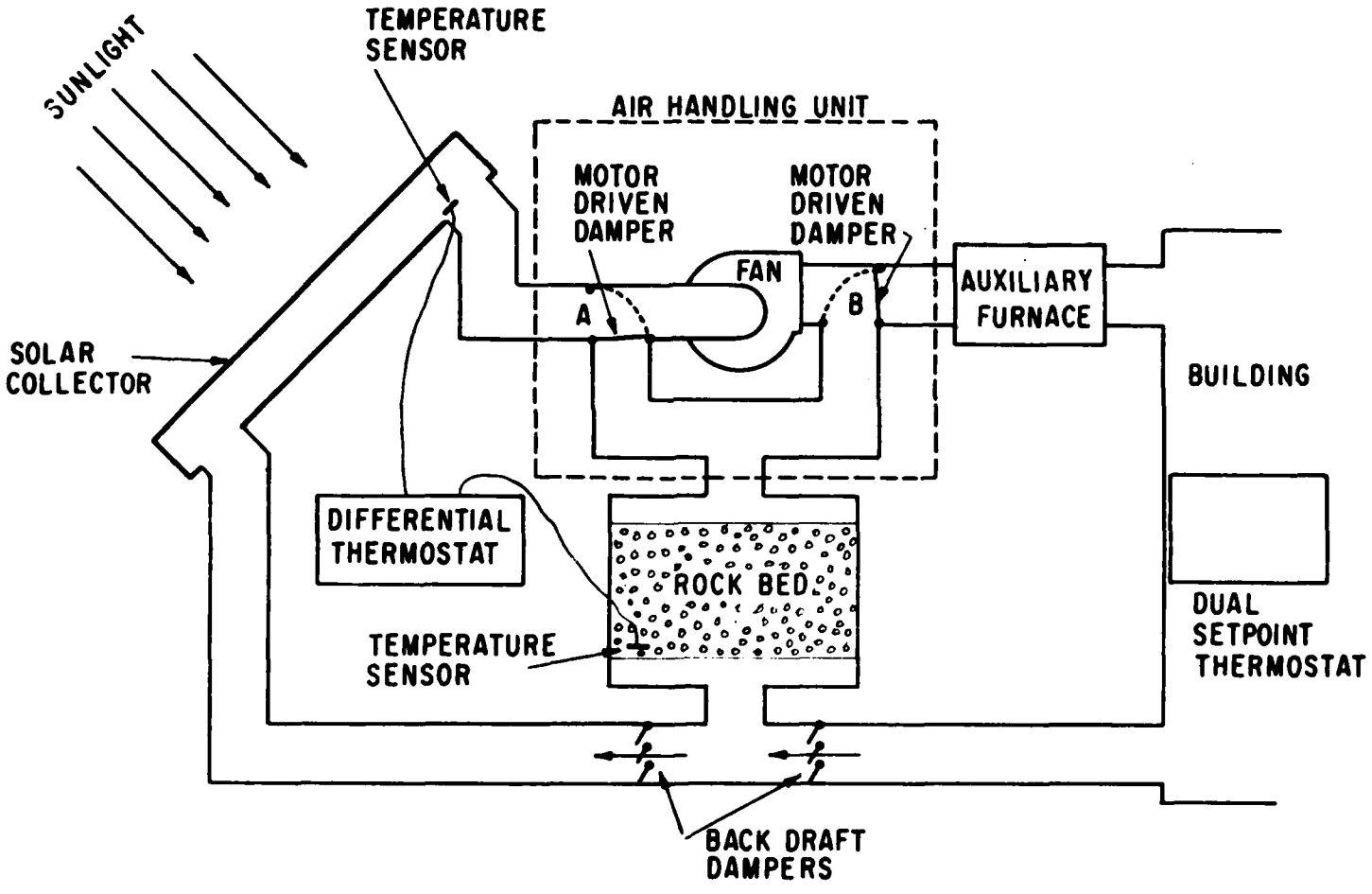


Figure 2-3. Typical Solid Material Storage System

2.5 Latent Heat

In all latent heat storage systems the storage medium undergoes a phase change, for example, melting from solid to liquid, in which it absorbs or releases heat without changing temperature. If the phase change used is a melting-freezing cycle, then heat flowing into storage melts the material, while heat removed from storage refreezes the material. Latent heat is the amount of energy absorbed or released during this complete phase change for a given amount of phase change material (PCM).

One advantage of latent heat storage systems is the compactness of the storage unit. The volume of PCM required to store a given amount of latent heat is considerably less than the volume of sensible heat storage material required to store the same amount of sensible heat. This allows greater flexibility in choosing a location for the storage unit. Further, since the unit is small, much less insulation is required to minimize heat loss. In actual practice this advantage is diminished by the need for large heat exchangers to overcome the thermal resistance of solid PCM.

A second advantage is that the temperature in a phase change storage system remains nearly constant throughout the charge-discharge cycle in which heat is absorbed and later released by the PCM. If the melting point of the PCM is chosen so that the storage unit provides heat at slightly above the minimum temperature required by the system, the output from a solar collector need be only a few degrees warmer. By contrast, a sensible heat storage system typically operates at 35° to 65°F (20° to 35°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.

Latent heat storage systems are discussed in more detail in Chapter 7.

2.6 Latent Cold

Latent cold resembles latent heat storage in its utilization of a phase change for the release or absorption of heat. In a freeze-melt cycle the latent cold storage system is charged when heat flows from storage, freezing the storage material. The system is discharged when heat flows from the load into storage, remelting the storage material.

While there are several inorganic salts that melt at low temperatures, ice is the most common latent cold storage material. Ice may be produced in various ways, including heat pump processes, winter production in ponds, and using snow making machines. The latter two methods use cold ambient air and are best suited for seasonal cold storage.

Latent cold storage is discussed in more detail in Chapter 8.

2.7 Passive Solar Heating and Cooling

A passive solar energy system uses the building structure itself as solar collector, energy storage, and transfer mechanism, with a minimal amount of mechanical equipment. Passive space heating systems have proven very cost effective in new construction. Although passive solar heating and cooling is

beyond the scope of this handbook it is important enough to deserve a complete handbook by itself.

2.8 Solar Ponds

A solar pond is a combination solar collector and energy storage device that is capable of storing heat for long periods of time — in some cases for many months. A solar pond, shown schematically in Figure 2-4, consists of three layers of water. Solar energy is collected and stored in the bottom layer called the storage zone. Ordinarily, the hot water at the bottom would tend to rise and dissipate the heat, but the bottom layer is nearly saturated with salt. The increased density due to the salt overcomes the tendency for the hot water to rise. The salt content of the middle layer or gradient zone varies from nearly saturated at the bottom to nearly fresh at the top. Here too, the salt overcomes the tendency for hot water to rise. Although water is not considered a good insulator, the 3-6 ft. thick (1-2 m) middle layer is as effective as 2.5-5 inches (6.2-12.5 cm) of fiberglass.

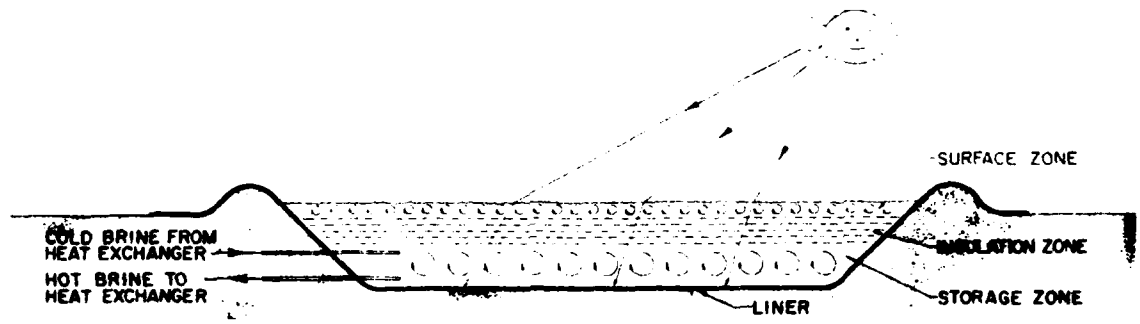
Energy can be collected in the summer and stored until winter when demand is the greatest. The temperature at the bottom of the pond can reach the boiling point if heat is not extracted. Solar ponds supplying low-temperature heat can have an overall efficiency of 20-30%. A major source of thermal loss is the top layer, which is about 4-8 inches (10-20 cm) thick, under ideal conditions. The thickness of the top layer can be increased by wind stirring and other effects. Methods of reducing thermal losses by reducing the thickness of the top layer are one of many subjects of solar pond research.

Solar ponds are potentially a very inexpensive way of collecting and storing solar energy, particularly in locations where salt is inexpensive, but much research and development is necessary before solar ponds can become fully viable.

2.9 Aquifers

Aquifers are beds of earth or porous rock containing water and since they are the source of much of our potable water, can be readily found under most populated areas. In this storage method nature provides the containment vessel, the aquifer itself, in which large amounts of thermal energy, either hot or cold, may be stored on a seasonal basis.

An aquifer system should contain at least two wells in the same geological formation, separated by several hundred feet. This is the minimum for a system providing either heating or cooling. One well is the injection or storage well and another is the supply well. In a heating system, water is pumped from the supply wells to the heater and then injected into the storage wells. In cold weather, hot water is withdrawn from the storage wells, used for heating, and the resulting warm water is returned to the supply wells. In a cooling system, water from the supply wells is chilled and injected into the storage wells. When cooling is needed, water is withdrawn from the storage wells, used for cooling, and the warmed water is returned to the supply wells. Ground water levels are not affected since the total amount of water in the aquifer is constant.



SALT-GRADIENT SOLAR POND

Figure 2-4. Schematic Concept of Solar Pond

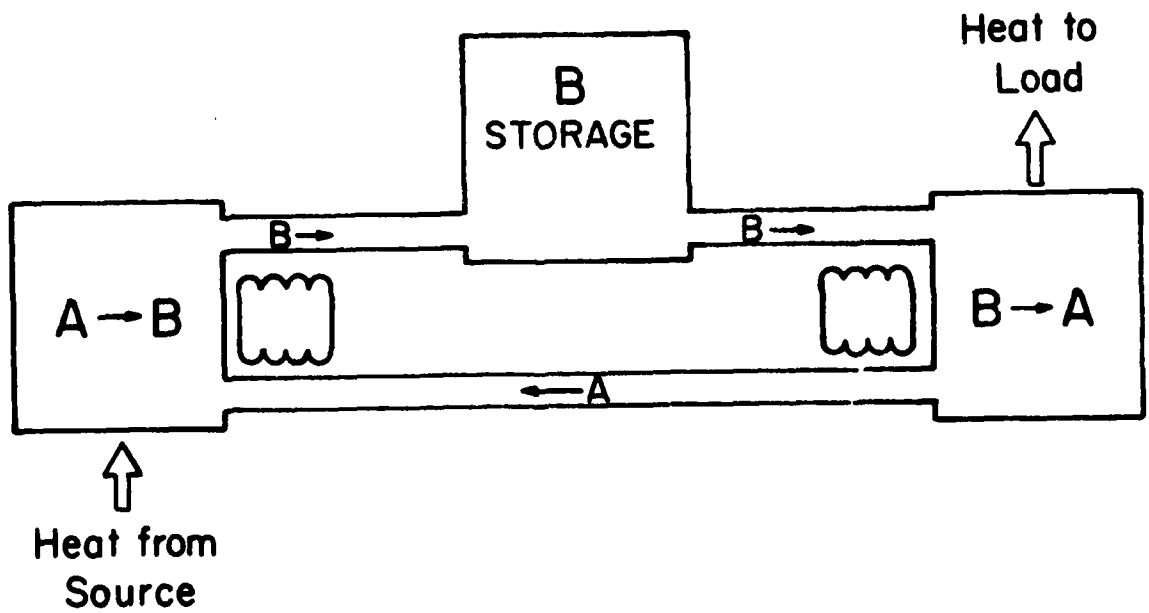


Figure 2-5. Typical Thermochemical Storage System

In a system for both heating and cooling, a minimum of four wells would normally be used - one hot supply, one hot injection, one cold supply, and one cold injection.

Since the storage temperatures in heating system aquifers can be as low as 120°F (50°C) a low-temperature, low-cost solar collector can be used as the heater. Since seasonal storage is used, the collector can be retired during the coldest months. Conversely, in cooling system aquifers the winter's chill can be exploited. By using a simple device like a spray pond it is possible in most of the United States to produce large amounts of cold water at low cost. Recovery of 80% of the heat or cold stored in an aquifer is possible.

2.10 Reversible Thermochemical Reactions

Thermochemical energy storage is generated when heat-absorbing chemical reactions are exploited that can later be reversed to regenerate the original heat energy upon demand. These reversible chemical reactions are generally capable of storing energy in a much smaller volume than in sensible heat storage.

A typical thermochemical storage system is illustrated schematically in Figure 2-5. The original energy is absorbed in the chemical reactor at the left of the Figure where the heat is converted into chemical bond energy by changing chemical A into chemical B. Both A and B may consist of more than one type of molecule, and there may be catalysts present in the reactor. The products B are transported via pipeline to storage. If B consists of more than one chemical, it may be necessary to store each one separately. When heat is needed, B is piped into the chemical reactor at the right of Figure 2-5, where it is changed back into A, at the same time giving off useful heat energy. Chemical A is transported back to the original reactor to complete the cycle. Storing chemical A to provide a continuous supply of feed stock to the reactor at the left may also be necessary. Note that heat exchangers just outside both chemical reactors keep most of the sensible heat from entering the pipeline. This underscores one of the advantages of energy storage in reversible thermochemical reactions - both the transport of energy and the storage of energy can be accomplished at relatively low temperatures, minimizing the thermal losses.

There are a wide variety of chemical reactions suitable for thermochemical energy storage. The selection of a suitable reaction must take into account the corrosiveness of the chemicals to both the reactor and storage facilities, cost, reaction temperatures, and overall efficiency of the process. Most thermochemical storage methods are still in their infancy and not yet commercially available. Reversible thermochemical reactions are discussed in more detail in Chapter 9.

2.11 Mechanical Storage

Mechanical energy storage exploits (1) the kinetic energy of moving matter as exemplified by rotating flywheels, or (2) the stored potential energy of matter under compression, as in the case of compressed air storage. Mechanical storage is discussed in more detail in Chapter 10.

A flywheel is a massive wheel that stores rotational kinetic energy.

Flywheels have traditionally been used to smooth out power from cyclic engines and to adjust for uneven loads. When storing electricity, a flywheel is generally powered by connecting a shaft from its center to a dynamo which acts either as motor, increasing the rotational velocity, or as a generator in which case the flywheel slows down and supplies energy. Flywheels are capable of energy recovery efficiencies of 80-90% and energy densities of 30-40 BTU/lb (70-90 kJ/kg). The energy density is limited by the strength of the flywheel material, which limits the maximum rotational speed.

Compressed air storage is an old concept that has seen little implementation. Compressed air can be stored in pressure tanks, or natural reservoirs such as salt domes, mine caverns, depleted oil or gas fields, or bound aquifers. It can then be used to power pneumatic machinery or to drive a turbine for electricity generation. Compressed air storage is not without its problems. These include: (1) air/heat leakage from the container, (2) upper limits placed on maximum pressure by container structural strength, and (3) fluctuations in power output with pressure variations.

2.12 Hydro Storage

The basis of pumped hydro storage is the gravitational potential energy of water pumped to a higher elevation. This energy is later recovered by allowing the water to drop through a turbine driving an electrical generator. Many above ground hydro storage systems have been built, and their characteristics are well known. Utilities, for example, use pumped hydro storage systems as an inexpensive method to level peak load demands and have realized energy recovery efficiencies of 65-80%. It has also been found that the cost of hydro storage decreases as the elevation of the pumped water increases. Thus, in the above ground systems steep hills or mountains are essential to meet the topographical requirements of (1) access to 2 large storage reservoirs which are (2) separated by several hundred feet in vertical distance and (3) are nearly adjacent horizontally. Below ground storage, using natural subterranean cavities as the lower reservoirs, are an alternative. Hydro storage is discussed further in Chapter 11.

2.13 Electrochemical Storage

In electrochemical energy storage, electrons from an electricity flow are used to drive a reversible chemical reaction. When energy from storage is needed, the reaction is allowed to go in the reverse direction, generating electricity. Electrochemical energy storage is characterized by a lack of moving parts, rapid electrical response, compactness and modularity; the storage battery exemplifies these features.

There are various types of storage batteries in use today, with the most common being the lead-acid type used in most cars. Lead-acid batteries consist of two plates, one of lead and one of lead dioxide, immersed in a weak sulfuric-acid solution functioning as the electrolyte; plates and electrolyte are encased together in a cell. On top of the plates are contacts so that electric current can be drawn from them.

In theory it is possible to completely discharge and recharge a storage battery repeatedly. In practice, especially for lead-acid batteries, such a deep discharge impairs the ability of the battery to become fully recharged.

Batteries with the ability to sustain many deep discharge cycles are usually more expensive.

A fuel cell resembles a battery in concept. The difference is that the chemical reagents are not incorporated as part of the cell but are instead fed into the cell as the reaction proceeds. A common fuel cell reaction is the formation of water from hydrogen and oxygen. Typically, the hydrogen and oxygen are earlier derived from the process of electrolysis in which electricity is used to break water, H_2O, into its constituents, H and O. The hydrogen and oxygen are then stored separately in molecular form until electricity is needed when they are reintroduced to the fuel cell and converted back to water.

A characteristic of all electrochemical storage, whether battery or fuel cell, is that electricity is produced as direct current (D.C.). However most common appliances and electrical systems use alternating current (A.C.). Consequently the use of batteries for electrical energy usually requires an inverter to change the D.C. electricity to A.C.

Electrochemical storage is discussed further in Chapter 12.

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3.0 TECHNOLOGY SELECTION CRITERIA

In this chapter, several parameters will be discussed that will guide the reader in selecting the best energy storage technology for a given application. All of the general storage technologies are compared with each other for each criterion.

Any particular application can conceivably be categorized by a specific subset of the criteria. The comparisons of the different technologies made for the various parameters should allow selection of one or more appropriate technologies for the particular application. Sizing and other technical details for a particular technology can be found in one of the later chapters.

3.1 Systems Approach

It must be emphasized strongly that energy storage cannot be considered by itself, but must be considered in the context of a complete system including energy supply and energy use. It must also be emphasized that supplying a given amount of energy upon demand is never the ultimate goal. Rather, the goal is always to fulfill a basic human need such as provision of a comfortable living space. Usually there are many different ways to fulfill these needs, each way requiring a different amount of energy use and a different type system to provide the energy. The system chosen may depend on:

- cost
- reliability
- space limitations
- degree to which it fulfills the basic need

A common example of the benefits of a system approach is in the space heating of a single family residence. Under current construction practice, it is almost always more cost effective to first thoroughly insulate the house to reduce the heat load than to provide an alternative energy source and try to heat a poorly insulated structure.

Human comfort in a spatial environment depends on a number of factors. These include: air temperature, humidity, radiant temperature of surrounding walls, velocity of surrounding air, type of clothing worn, and activity level. The effects of these factors on human comfort are summarized in considerable detail by P. O. Fanger in Thermal Comfort - Analysis and Applications in Environmental Engineering, McGraw-Hill, 1972. All of the above-mentioned factors will influence the type of storage system used for a given application, as well as the size such a storage unit should take.

3.2 Applications

The suitability of the various storage technologies for several different applications is summarized in Table 3-1. In general, mechanical (compressed air, flywheels, and pumped hydro) and battery storage is best suited for elelctrical applications, while thermal storage (water tanks, rock beds, latent heat, latent cold, and thermochemical) is best suited for thermal

TABLE 3-1: Suitability of Storage Technologies According to Application

type of storage / application	water tank	rock bed	latent heat	latent cold	thermo-chemical	comp. air	flywheel	pumped hydro	battery
domestic hot water	**		**		*				
space heat --small, medium	**	**	**		*				
--medium, large	**	*	**		*				
space cooling	**	*	**	**	**				
process heat (incl.pre-electricity generation)	**	*	**		**	*			
electricity -long term -small, medium					*	*		*	*
--short term --small, medium						*	**	*	**
--long term --large						*		**	*
--short term --large						**	*	**	**

** - excellent suitability; * - some suitability; - little suitability

applications. The efficiency of converting from electricity to thermal storage and back to electricity is low, so thermal storage is inappropriate for electrical applications, unless perhaps it is storage for the original thermal source. When off-peak electricity is available for electrical resistance heating or operating a heat pump, it is usually more effective to use the electricity immediately and store the thermal energy rather than storing the electrical energy. Since the cost of electricity is roughly 2 1/2 to 3 times the cost of thermal energy; mechanical and battery storage, when available, are almost exclusively used for electrical applications. Hot thermal storage can be used in cooling applications by providing the high temperature necessary to drive an absorption chiller. Thermal storage can also be used in space cooling by providing a cold temperature heat sink.

Water Tank

In the present chapter, water tank storage usually refers to sensible heat storage by water in tanks, but may include other liquids, as well as pond and aquifer containers. Water tank storage is a mature technology that can be used for almost all thermal energy applications. If high temperature storage is needed, the water must be pressurized. Alternatively, other liquids such as oils or molten salts could be used. Water tank storage is particularly well suited to domestic hot water applications, since water tanks are already used for this purpose.

Rock Bed

In the present chapter, rock bed storage usually refers to a packed bed of fist-sized rocks that is usually coupled to an air-based solar collector, but it may also include any solid sensible heat storage medium. Rock bed storage is also a mature technology that can be used over a wide temperature range for most thermal energy applications. Rock beds can be easily implemented in space heating applications by passing the air to be heated directly through the warm rock bed.

Latent Heat

Latent heat storage can be used for most thermal applications. Since there are many phase change materials, each with a different melting point, latent heat storage is available for a wide range of temperatures. Most latent heat storage methods are still in the development stages.

Latent Cold

Ice storage is a mature latent cold storage technology that is suitable for space cooling. Ice storage is particularly attractive for larger cooling systems. Other latent cold storage systems are still under development.

Thermochemical

Thermochemical storage is particularly attractive for situations in which heat energy must be stored for long periods of time (months) or transmitted long distances (miles). It may even be attractive for long thermal storage in electricity applications. Unfortunately, thermochemical storage is still mostly in the research and development stages.

Compressed Air

Compressed air storage can be used for some thermal applications, since air is heated somewhat when compressed. If underground caverns are used, there is a minimum size that is economical. This developing technology is best used for medium to large power plant scale applications.

Flywheel

Flywheels can store energy for electrical applications for short time periods. Individual units are limited in size. Current development is directed towards lowering the cost of this otherwise mature technology.

Pumped Hydro

Pumped hydro storage can be used for either long or short-term storage, but economics impose a minimum size limitation. Above-ground pumped hydro is a mature technology and is one of the least costly of the storage technologies when the geography is suitable.

Battery

Batteries are a mature, but currently expensive, technology for electricity storage. Their modularity makes them suitable for both small and large applications. They are best suited for shorter term storage, because most battery types will slowly self-discharge.

3.3 Sources

The suitability of coupling the various storage technologies to different energy sources is summarized in Table 3-2. Thermal energy sources are usually best suited for thermal storage, while wind and off-peak electricity are more versatile.

Solar Thermal

Solar collectors can be easily coupled to any thermal storage technology. Gas or liquid can serve as the collector working fluid. For most circumstances so far, the conversion of solar thermal energy to mechanical or electrical energy has not proven economical.

Ambient Air

The use of ambient temperatures can be used to heat or cool water or rocks, but this usually requires taking advantage of yearly temperature fluctuations with seasonal energy storage. Using winter conditions to make ice for summer cooling is one of the best uses of ambient thermal conditions.

Waste Heat

Like solar thermal, waste heat can be used effectively with almost any thermal storage technology. Waste heat is often left over from electricity generation and in these cases, is usually not of high enough quality for economical coupling to mechanical or battery storage.

TABLE 3-2: Suitability of Coupling Storage Technology to Energy Resource

source / type of storage	solar thermal	ambient thermal	waste heat	off wind	photo-voltaic	off-peak electric-ity
water tank	**	*	**	*		**
rockbed	**	*	**	*		*
latent heat	**		**	*		**
latent cold	**	**	**	*		**
thermo-chemical	**		**	*		*
compressed air				**	**	**
flywheel				**	**	**
pumped hydro				**	**	**
battery				**	**	**

** - excellent suitability
* - some suitability
- little suitability

Wind

Wind energy can be coupled to any of the storage technologies. Wind energy is mechanical in nature and is best coupled to mechanical or battery storage. However, the efficiency of wind machines is high enough that in some cases they can be used to mechanically drive a heat pump or even provide electric resistance heating.

Photovoltaic

The output of photovoltaic devices is electricity; and so, photovoltaics are most suitably coupled to mechanical or battery storage. The efficiencies of solar cells are much lower than solar thermal collectors; hence, photovoltaics are not suitable for coupling with thermal storage. The combination of photovoltaics and solar thermal in one device is still in the research stage.

Off-Peak Electricity

If excess electricity is available, it can be coupled effectively with any of the storage technologies.

3.4 Technology Characteristics

Energy storage technologies can be broadly characterized by:

- type of energy
- density
- temperature
- length of storage
- efficiency
- lifetime
- capital cost
- size
- stage of technical development

Table 3-3 lists these characteristics for the storage technologies discussed in Chapters 4-12. The values listed in Table 3-3 are representative and should be used only for the roughest of comparisons. There is usually a considerable range within each technology for each characteristic. Further, the value of one characteristic often depends on the value of another characteristic. For example, costs per unit energy stored are usually lower for larger systems, energy density of water tank and rock bed systems is higher if a larger temperature range is used, etc.

Density

Energy storage density is seldom a major consideration unless space is at a premium. Energy densities for water tanks and rock beds will typically vary up or down from the values listed in Table 3-3 by a factor of 2 or sometimes 3, while the other technologies may vary by as much as a factor of 8. Underground pumped hydro usually has an energy density approximately 10 times higher than the value for surface pumped hydro given in the Table.

TABLE 3-3: Representative Characteristics of Energy Storage Technologies

characteristic / type of storage	Type of Energy	Density Btu/ft^3	Temperature range °F	Length of storage	Efficiency	Lifetime year	Capital cost \$/10^6BTU	Minimize size 10^6BTU	Development stage
Water tank	Q	6000	0 to 600	day to months	0.80	20	3000	0.02	mature
Rock bed	Q	3000	-100 to 2000	day to months	0.80	20	3000	0.02	mature & developing
latent heat	Q	10,000	60 to 600	day to week	0.80	5	4000	0.02	developing
latent cold	Q	3000	0 - 70	day to year	0.80	20	4000	0.02	mature & developing
thermochemical	Q	20,000	0 - 1000	day to years	0.70	?	6000	?	research & developing
compressed air	Q/W	900	0 - 600	day to month	0.70	20	3000	2.0	developing
flywheel	W	4500	--	hours to days	0.80	20	60,000	∿0	developing
pumped hydro	W	25	--	day to years	0.75	20	1500	0.5	above-ground-mature/below-ground-mostly mature
battery	W	25,000	40 - 840	hours to months	0.70	5	21,000	∿0	mature & developing

Q = thermal
W = work

Temperature

The temperature of energy storage is important for thermal applications, because the storage container must usually be thermally insulated. Matching the storage temperature to the temperature of the load minimizes the required insulation and helps to maximize the efficiency of storage.

Storage Time

The only storage technologies that have unlimited storage times are thermochemical storage and some types of batteries. Thermochemical storage is still in the research and development stage, and batteries with long storage times are currently expensive. Other thermal storage technologies will slowly lose their stored energy by conduction through the container walls. The longer the energy is stored, the more energy is lost, and the lower the storage efficiency. This loss can be minimized in seasonal storage devices only by using large systems with small area/volume ratios or by using extensive amounts of insulation. Compressed air storage loses energy by air leakage from the container. Flywheels lose energy through friction of the moving parts. Pumped hydro can lose energy from exposed upper reservoirs through evaporation. Most types of batteries will lose energy through self-discharge caused by slow chemical reactions inside the battery cells.

Efficiency

The efficiency of an energy storage device can be defined as the net energy delivered to the load by storage divided by the energy delivered to storage by the source. The efficiency of energy storage devices must be evaluated in the energy system in which they are used. Efficiency can be much lower than that listed in Table 3-3 if the storage is not properly integrated into the system. For example, improper sizing of heat exchangers can significantly reduce the efficiency of thermal storage devices.

Lifetime

The lifetime of a mature energy storage technology device should be at least 20 years. An exception is battery storage, where a typical lifetime is closer to 5 years, although longer lifetimes are available at a higher cost.

Cost

The initial capital cost within an energy storage technology can vary widely. For example, underground pumped hydro is considerably more expensive than surface pumped hydro. Nickle-cadmium batteries are more expensive than lead-acid batteries. With the exception of batteries, which are modular, larger systems are generally less expensive than smaller systems of the same type.

Size

There is usually no upper size limitation for any of the storage technologies, since modular units of the largest practical size can always be built. Technologically, there is also no lower limit to size, although there

is usually a lower size for economic reasons. The two technologies which are most restricted to larger sizes are compressed air storage in underground caverns and pumped hydro, which needs a relatively large vertical distance (10 feet minimum) and storage reservoir to be practical.

Development Stage

Thermochemical storage is the least developed of the technologies, with the first practical devices expected to be available in the near future. Latent heat storage has several systems that are commercially available, but none of these have yet demonstrated a long lifetime while maintaining good efficiency. Most of the other storage technologies listed in Table 3-3 have developed some mature methods which have been found pratical for at least some applications. Improvements are continually being made to all the storage technologies.

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4.0 GENERAL INFORMATION ON SENSIBLE THERMAL ENERGY STORAGE

In a sensible thermal storage system, the heat flowing into storage raises the temperature of the storage material. When heat is removed from storage the temperature of the storage material drops. The most common sensible thermal storage systems are built around water tanks or rock beds. Water tanks are commonly used in space heating and domestic hot water applications and are discussed in detail in Chapter 5. Rockbeds are commonly used in space heating applications and are discussed in detail in Chapter 6. A general discussion of sensible thermal storage may be found in Sections 2.3 and 2.4.

We will first describe some general characteristics of sensible heat storage materials--heat capacity and daily operating range, which affect the size of the storage device, and temperature stratification, which affects its performance. We will then discuss situating and insulating the storage device and moving heat into and out of storage.

4.1 Heat Capacity

Heat, or thermal, capacity is a material's ability to store sensible heat. Sensible heat is characterized by utilizing the heat capacity of a material and a temperature rise. In the English system of units it is measured in terms of the number of British thermal units (Btu) required to raise the temperature of 1 pound of the material by 1 degree Fahrenheit. The basic equation for the amount of heat stored per mass of material is given by:

$$\frac{Q}{m} = c_p \Delta T \qquad (4\text{-}1)$$

where $\frac{Q}{m}$ = quantity of heat in mass m (ie $\frac{Btu}{lb.}$)

c_p = heat capacity ($\frac{Btu}{lb°F}$)

ΔT = temperature rise above system's minimum (°F) temp.

Water has a heat capacity of 1 Btu per pound per degree Fahrenheit (Btu/lb·°F). Most other materials have a lower heat capacity than water, rock, for example, has a heat capacity of about 0.21 Btu/lb·°F. In metric (SI)

units, the heat capacity of water is 4184 joules per kilogram per degree Celsius (J/kg·°C), and the heat capacity of rock is about 879 J/kg·°C.

The amount of sensible heat that materials can store per unit mass is not the only basis by which storage materials' thermal capacities are compared. Engineers frequently work with a derived quantity known as the volumetric heat capacity, which is found by multiplying the material's heat capacity by its density. The volumetric heat capacity describes the quantity of sensible heat the material can store per unit volume for every degree of temperature change.

In this case, the basic equation is

$$\frac{Q}{V} = \rho c_p \Delta T \qquad (4\text{-}2)$$

where $\frac{Q}{V}$ = quantity in volume V (i.e. $\frac{Btu}{ft^3}$)

ρc_p = density · heat capacity

= volumetric heat capacity

Table 4-1 gives the heat capacities and volumetric heat capacities of some common storage materials. The voids referred to in the table are the spaces that exist between fragments of piled up rock or other loose material. The proportion of these spaces to the total volume of the rock bed is called the void fraction. Numerous experiments have shown that loose materials pack with a void fraction of from 20 to 30 percent. These loose materials behave as if their thermal capacities were reduced by this same percentage.

As you can see in Table 4-1, 1 cubic foot of water can store 62.4 Btu for every degree Fahrenheit of temperature rise. A cubic foot of rock packed to a 30 percent void fraction can store only 24.3 Btu per degree of temperature rise. Therefore, to store the same quantity of heat over the same temperature range the rock bed's volume would have to be 2.6 times greater than that of the water tank.

4.2 Temperature Range

Within a sensible heat storage system, the characteristic temperature range is also closely related to the size of the storage device. Both air- and liquid-based systems typically operate over a daily temperature range of under 60°F (33°C) on a sunny winter day. The exact range is highly variable from system to system, season to season, and day to day. Factors that influence the daily temperature range include the amount of

sunshine available, the size of the storage device, the heat capacity of the storage material, the demand for heat, the type of system, the way the system is connected to the load, and the temperature limitations of materials in the system.

Table 4-1. Sensible Heat Storage Materials

Material	Density, ρ lb/ft^3 (kg/m^3)	Heat Capacity, c_p Btu/lb·°F (J/kg·°C)	Volumetric Heat Capacity, ρc_p Btu/ft^3·°F (MJ/m^3·°C) No Voids	30% Voids (ρc x .7)
Water	62.4 (1000)	1.00 (4180)	62.4 (4.18)	-
Scrap Iron	489 (7830)	0.11 (460)	53.8 (3.61)	37.7 (2.53)
Scrap Aluminum	168 (2690)	0.22 (920)	36.96 (2.48)	25.9 (1.74)
Scrap Concrete	140 (2240)	0.27 (1130)	27.8 (1.86)	26.5 (1.78)
Rock	167 (2680)	0.21 (879)	34.7 (2.33)	24.3 (1.63)
Brick	140 (2240)	0.21 (879)	29.4 (1.97)	20.6 (1.38)

Note: SI units in parentheses follow English units.

The daily temperature range, $T_{max} - T_{min}$, is related by the following equation to the amount of usable heat, Q, measured in Btu (joules), stored in the device:

$$Q = m\, c_p(T_{max} - T_{min}). \qquad (4\text{-}3)$$

Here m is the mass of the storage material in pounds (kilograms) and c_p is the heat capacity in Btu/lb·°F (J/kg °C) of the storage medium.

Example 4-1: Suppose you must store 400,000 Btu from the solar collectors on a sunny winter day, and you want to limit the daily temperature range on that day to 40°F. How much water will be required?

Solving Equation 4-1 for m and substituting the values for stored heat, daily temperature range, and heat capacity yields the required mass of water.

$$m = \frac{Q}{C_p(T_{max} - T_{min})} \qquad (4-4)$$

$$= \frac{400{,}000 \text{ Btu}}{1 \frac{\text{Btu}}{\text{lb.}^\circ\text{F}} \times 40^\circ\text{F}}$$

$$= 10{,}000 \text{ lb (4540 kg) of water.}$$

Since one gallon of water weighs 8.34 pounds, this amounts to 1200 gallons of water.

4.3 Thermal Stratification

When heat is delivered into a sensible heat storage material, thermal stratification can occur. That is, some parts of the storage material can become hotter than other parts. In theory, perfect stratification in a system designed to take advantage of stratification can perform 5 to 10 percent more efficiently than a thermally mixed system. The type of system used determines the practical usefulness of thermal stratification.

Stratification is relatively easy to achieve in rock beds with vertical air flow. The solar-heated air enters at the top of the rock bed and flows downward through many winding paths, losing most of its heat by the time it leaves the bottom of the rock bed. Thus, the rock bed is hot at the top but relatively cool at the bottom. Since rocks can't move, this temperature differential is easily maintained. The collector operates at a lower, more efficient average temperature than it would if there were no stratification, and consequently more useful heat can be collected. Thermal stratification from top to bottom of a rock bed compensates for the lower efficiency of air-type collectors compared with liquid-type collectors.

Thermal stratification can also occur in water tanks, but pumping the water to the collector or the heat exchanger tends to cause the hotter and cooler water to mix, destroying stratification. Even natural convection from a coil-in-tank heat exchanger can upset the stratification process.

4.4 Location

In principle, the storage device should be adjacent to the solar collectors to minimize the collector-to-storage losses. Since the solar collectors are usually mounted on the roof, the roof or attic might seem

a logical site for the storage container, and it is a common practice to store solar-heated domestic hot water on rooftops in the tropics. If thermal storage is to be used for space heating, however, the storage unit's weight makes such a location impractical. Moreover, the climate in the United States frequently causes builders to seek more sheltered locations for thermal storage devices--usually basements, crawl spaces, garages, or underground. The following sections describe the advantages and disadvantages of these various locations; Tables 4-2 and 4-3 summarize this information.

Basement

Basements or other heated indoor areas are good locations for thermal storage devices. Tanks or rock beds in heated areas generally require less insulation than in outdoor installations, and the building protects the storage device from weathering. Since any heat lost from storage escapes into heated areas these losses are not considered a threat to the solar system during the heating season.

The main disadvantage of indoor storage devices is that they take up valuable space. In addition, heat lost from storage into the living area is uncontrollable. In summer, the losses from a hot storage device can increase the air conditioning load unless the storage area is well ventilated or the system is shut down for the summer.

Although a basement will generally provide enough room for a water tank to be installed, getting a large tank into the building is a major problem. Cast-in-place concrete tanks and plastic-lined wooden tanks can be assembled inside existing buildings. Although steel and fiberglass tanks can be installed in basements of new construction, repair or replacement of these tanks would be expensive. For this reason, we do not recommend installing a large one-piece tank in a basement.

If you plan to install a rock bed in a basement, you must consider the floor-to-ceiling space available. Most basements have 7 feet of clearance between the floor and the ceiling. By careful design, you can keep a rock bed within this size limit; but in some cases you will have to remove part of the basement floor slab to gain more clearance.

Appropriate foundations must be developed to support the extreme weight of any storage device. A structural engineer should determine what type of footings are required to support the storage unit's weight.

Crawl Space

Heat losses from storage devices in unheated locations such as the crawl space are irrecoverable. This is a disadvantage during the heating season, but a potential advantage during the air conditioning season. If the crawl space is well ventilated and insulated from the air conditioned rooms, heat escaping from the storage device will not add to the air conditioning load.

Table 4-2. Advantages and Disadvantages of Storage Locations

Advantages		
Utility Room or Basement	Unheated Garage	Crawl Space
• Insulation requirement is minimal	• Insulation is protected from weather	• Insulation is protected from weather
• Insulation is protected from weather	• Leaks are easily detected	• Thermal losses may contribute to building heat in winter
• Thermal losses contribute to building heat in winter	• Access for repairs is easy	
• Leaks are easily detected	• Steel or FRP tanks can be installed in an existing garage	
• Access for repairs is relatively easy		

Disadvantages		
Utility Room or Basement	Unheated Garage	Crawl Space
• Living space is reduced	• Garage space is reduced	• Thermal losses may add to air conditioning load in summer
• Thermal losses add to air conditioning load in summer	• Extra insulation is required	• Access is difficult for retrofit or repairs
• Leaks may damage building interior	• Freeze protection is required in most of U.S.	• Tank shape may require extra insulation
• Steel or FRP tanks are difficult to install in an existing building	• Leaks may damage the garage	
	• Thermal losses cannot be recovered	

Table 4-2 (continued)

Advantages	
Outdoors, Above Grade	Outdoors, Below Grade
• Access is easy	• Thermal losses do not add to air conditioning load
• Thermal losses do not add to air conditioning load	• Storage unit does not reduce living space

Disadvantages	
Outdoors, Above Grade	Outdoors, Below Grade
• Extra insulation is required	• Access for repairs is difficult
• Weather protection is required	• Groundwater may cause problems
• Thermal losses cannot be recovered	• Thermal losses cannot be recovered
• Freeze protection is required in most of U.S.	• Vermin may burrow into insulation
• Vermin may burrow into insulation	• Careful design is required to ensure sufficient net positive suction head for pump

Most crawl spaces do not have enough vertical clearance for a rock bed with vertical air flow to be installed. Although rock beds with horizontal air flow have been built, maintaining uniform air flow and thermal stratification in them is more difficult than in rock beds with vertical air flow. Therefore, we do not recommend installing a rock bed in a crawl space that does not have an unusually large amount of vertical clearance.

To install a storage device in a crawl space, follow the recommendations for below grade installations. Taking into account the difficulties of installing a storage device in an existing crawl space, this location should be considered primarily for new construction. Even in this case, the cramped working quarters could make repair costly.

Garage

The garage is an excellent location for steel or fiberglass water tanks, since they can be easily removed and replaced through the large door. Other types of water tanks, as well as rock beds, can also be used here. The garage protects the storage device from the weather.

Tanks in unheated garages must be protected from freezing. Heat losses from storage devices in this location are irrecoverable, and the storage device reduces the space available for vehicles.

Outdoors

The storage device can be located outside the building either above or below grade. In either case, the storage container must be well insulated and built on a good foundation. The ducts or pipes to and from the storage unit must be insulated, weatherproofed, and possibly waterproofed. Vermin are occasionally a problem in outdoor tanks. Use vermin-proof insulation or surround the insulation with half-inch mesh wire cloth. Heat losses from outdoor storage devices are irrecoverable.

Above Grade

In an aboveground installation, the storage device should be placed inside a shelter or covered with roofing and siding. This protection must be designed in compliance with local codes for wind resistance, snow loads, and seismic loads.

Water tanks in unheated locations must be protected from freezing. In moderate climates, an electric immersion heater will do this satisfactorily.

Below Grade

The main problem with buried storage devices is that groundwater can soak the insulation (even if it is closed-cell foam). In several instances, the resulting high losses have forced owners to abandon underground storage devices.

Table 4-3. Storage Location, Applicability, and Special Requirements

Storage Location	Applicability		Special Requirements				
	New Building	Retrofit	Weather-proof Insulation	Extra Insulation	Freeze Protection	Protection from Groundwater	Long-Lifetime Components
Utility Room or Basement	yes	X	no	no	no	no	yes
Unheated Garage	yes	yes	no	yes	yes	no	no
Crawl Space	yes	X	no	yes	no	no	yes
Outdoors, Above Grade	yes	yes	yes	yes	yes	no	yes
Outdoors, Below Grade	yes	yes	yes	yes	X	yes	yes

Note: Items marked "X" must be determined by the individual situation.

The following guidelines should be followed if you choose to install the storage device underground.

- If possible, avoid burying the storage device so deep that the highest level of groundwater (usually in Spring) rises above the bottom of the insulation.
- Use pea gravel beneath the foundation to allow water to drain.
- Use waterproof insulation, such as closed-cell foam, even in dry areas. We recommend that for an underground storage device you use twice the insulation thickness specified in the following section, because groundwater can degrade the performance of even a closed-cell foam. Urethane foams in contact with earth tend to degrade rapidly.
- Do not rely upon dry earth for insulation.
- If possible, direct rainwater away from the storage unit. Avoid situations where undisturbed soil (especially clay) surrounding the hole where the storage unit is installed forms a catch-basin for rainwater.
- Position tank vents so that neither rainwater nor groundwater can enter the tank. If the lid is separate from the body of the storage device, the joint must be above the highest groundwater or floodwater level. A waterproof barrier must be installed to direct rainwater away from the joint.

If a substantial portion of the tank is buried below the freeze line, no freeze protection will be required.

4.5 Limiting Heat Loss

Characteristics of Thermal Insulation

The most important characteristic of thermal insulation is its resistance to the flow of heat. This resistance is given the symbol R and has units of degrees Fahrenheit-square feet-hours per Btu ($°F \cdot ft^2 \cdot hr/Btu$), or degrees Celsius-square meter per watt ($°C \cdot m^2/W$). The larger the value of R, the greater the insulation's resistance to heat flow. Since many types of insulation can be applied in various thicknesses, it is convenient to use the R-value per unit thickness, which is given the symbol r in this manual and has units of degrees Fahrenheit-square feet-hours per Btu per inch ($°F \cdot ft^2 \cdot hr/Btu \cdot in$), or degrees Celsius-square meters per watt per centimeter ($°C \cdot m^2/W \cdot cm$).

Table 4-4 gives values of r for various common insulating and building materials. Several types of insulation have characteristics that make them especially suitable for some applications but unsuitable for others.

Fiberglass and mineral wool are inexpensive general-purpose insulation materials. Because they insulate by trapping air between their fibers, they tend to lose their insulating value when compressed. That is, a 5-1/2-inch-thick batt of fiberglass compressed to fit into a 3-1/2-inch cavity will

work no better than a 3-1/2-inch-thick batt. Because these materials are porous, they must be protected from wind, and they cannot be used in locations where they might become soaked with water.

Cellulose fill is a very inexpensive insulation, but it must be fireproofed to be used safely in buildings. It cannot be used where it might become wet.

Vermiculite and perlite are expanded mineral products that are used as loose fills. They are fireproof and can withstand high temperatures. As loose fills they should not be used where they might become wet, but they are sometimes used as a part of the aggregate in concrete to make a waterproof insulation.

Glass foam is a relatively expensive material that is waterproof and fire-proof, resists crushing, and can withstand high temperatures.

Polystyrene foam is a versatile material that can be purchased as beads for loose fill or as boards. Its disadvantages are that it is weak, cannot withstand high temperatures, is flammable, and is easily destroyed by solvents. Despite these disadvantages, closed-cell polystyrene foam is excellent for underground applications. Like other plastics, polystyrene foam must be protected from weathering and sunlight. Urethane foam has properties similar to those of polystyrene foam, but it can support slightly heavier loads, can withstand higher temperatures, and is much more resistant to solvents.

Urea-formaldehyde foams are most frequently applied as a two-part, spray-on mixture. Because the composition and density of spray-on foams can be difficult to control, we recommend reducing the r-value given in Table 4-4 (which was obtained in a laboratory under ideal conditions) by half. Users of urea-formaldehyde insulation have experienced several types of problems, including shrinking and sagging, deterioration in underground applications, and offensive odors caused by the formaldehyde. Because of the potential health hazards, use of urea-formaldehyde is not recommended and is forbidden by some building codes.

Because most of the insulation materials used in construction use trapped air to restrict the flow of heat, their r-values are likely to approach or exceed the r-value of still air of 5.56 $^{\circ}F \cdot ft^2 \cdot hr/Btu \cdot in$ (0.386 $^{\circ}C \cdot m^2/W \cdot cm$). A few closed-cell foams have been expanded with gases (such as refrigerants 11 or 12) which have a higher thermal resistance than air. When these foams are new, their resistance to the flow of heat can exceed that of air, but as they age the gases diffuse out and are replaced by air, causing their r-values to diminish.

A similar reduction in resistance to flow of heat occurs when foams are used in a wet environment. Water diffuses into the foam and replaces part of the trapped air. Because of this problem, we recommend using only half the r-values listed in Table 4-4 for closed-cell insulation when it is used underground or in other wet locations.

Table 4-4. Densities and r-values of Common Building and Insulating Materials

Material	Density, lb/ft^3 (kg/m^3)	r-value, °F·ft^2·hr/Btu·in (°C·m^2/W·cm)
Acoustic Tile	18.0 (288.)	2.53 (0.175)
Aluminum (1100 Alloy)	171.0 (2740)	0.000651[b] (45×10^{-6})
Asbestos-Cement Board	120.0 (1920)	0.26[b] (0.018)
Brick: Common	120.0 (1920)	0.20[b] (0.014)
Face	130.0 (2080)	0.11[b] (0.0076)
Cellulose Fill	2.5-3.0 (40-48)	3.70[b] (0.257)
Cement (Mortar or Plaster with Sand)	16.0 (1860)	0.20[b] (0.014)
Concrete, Medium Weight		
Dried Aggregate	140.0 (2240)	0.11[b] (0.0076)
Undried Aggregate	140.0 (2240)	0.08[b] (0.0055)
Concrete, Heavy Weight	80.0 (1280)	0.40[b] (0.028)
Concrete, Light Weight	30.0 (481)	1.11[b] (0.077)
Concrete Block, Heavy Weight		
4-inch	101.0 (1620)	0.18[a] (0.012)
6-inch	85.0 (1360)	0.15[a] (0.010)
8-inch	69.0 (1110)	0.13[a] (0.009)
12-inch	76.0 (1220)	0.11[a] (0.0076)
Concrete Block, Medium Weight		
4-inch	76.0 (1220)	0.28[a] (0.019)
6-inch	65.0 (1040)	0.23[a] (0.016)
8-inch	53.0 (849)	0.18[a] (0.012)
12-inch	58.0 (929)	0.18[a] (0.012)
Concete Block, Light Weight		
4-inch	65.0 (1040)	0.33[a] (0.023)
6-inch	55.0 (881)	0.30[a] (0.021)
8-inch	45.0 (721)	0.25[a] (0.017)
12-inch	49.0 (785)	0.19[a] (0.013)
Fiberglass		
Batt		3.15[b] (0.218)
Rigid, Organic Bonded	4-[illegible]0 (64-144)	4.00[b] (0.277)

[a]Source: R .M. Graven and P. R. Hirsch. DOE-1 Users' Manual. Argonne National Laboratory Report ANL/ENG-77-04, November 1977.

[b]Source: American Society of Heating, Refrigerating and Air Conditioning Engineers. ASHRAE Handbook of Fundamentals, 1972, pp. 360-363.

Table 4-4 (continued)

Material	Density, lb/ft^3 (kg/m^3)	r-value, °F·ft^2·hr/Btu·in (°C·m^2/W·cm)
Glass Foam	9.0 (144)	2.50[b] (0.173)
Gypsum or Plaster Board	50.0 (801)	0.90[b] (0.062)
Gypsum Plaster		
Light Weight Aggregate	45.0 (721)	0.63[b] (0.044)
Sand Aggregate	105.0 (1680)	0.18[b] (0.012)
Hard Board		
Medium Density Siding	40.0 (641)	1.53[a] (0.106)
Medium Density Other	50.0 (801)	1.37[a] (0.095)
High Density Standard Tempered	55.0 (881)	1.22[a] (0.085)
Insulation Board		
Sheathing	18.0 (288)	2.63[a] (0.182)
Shingle Backer	18.0 (288)	2.52[a] (0.175)
Nail Base Sheathing	25.0 (400)	2.28[a] (0.158)
Mineral Board, Preformed	-	3.47[a] (0.241)
Mineral Wool/Fiber		
Batt	-	3.33[a] (0.231)
Fill	-	3.09[a] (0.214)
Particle Board		
Low Density		1.85[a] (0.128)
Medium Density		0.11[a] (0.0076)
High Density		0.08[a] (0.0055)
Underlayment		0.46[a] (0.032)
Perlite Expanded with Refrigerant-11	5.0-8.0 (80-128)	2.70[b] (0.187)
Polystyrene		
Board, Expanded	1.8 (29)	4.00[b] (0.277)
Molded Beads	1.0 (16)	3.57[b] (0.248)
Boards, Expanded with Refrigerant-12	2.2-3.5 (35-56)	5.00-5.26[b] (0.347-0.365)
Polyurethane Expanded with Refrigerant-11	1.5-2.5 (25-40)	6.25[b] (0.433)
Roof Insulation, Preformed	16.0 (256)	2.78[a] (0.193)
Steel (Mild)	489.0 (7830)	0.00318[b] (0.22×10^{-3})
Urea-Formaldehyde	0.7 (11)	4.1[a] (0.289)
Vermiculite, Expanded	7.0-8.2 (112-131)	2.13[b] (0.148)
	4.0-6.0 (64-96)	2.27[b] (0.157)
Wood, Soft (Fir, Pine, etc.)	32.0 (513)	1.25[b] (0.087)
Wood, Hard (Maple, Oak, etc.)	45.0 (721)	0.91[b] (0.063)

R-value of Multiple Layers

Most insulation consists of multiple layers. The combined R-value is the surface resistance, if any, between solid materials and air plus the sum of the R-values of the layers, or:

$$R = R_S + s_1 r_1 + s_2 r_2 + s_3 r_3 + \ldots \qquad (4\text{-}5)$$

where:

R_S is the surface resistance.

s_1, s_2, and s_3 are thicknesses of each insulation layer.

r_1, r_2, and r_3 are the thermal resistances per unit of thickness for each insulation layer.

Surface resistance will only be present when a surface of the insulation is exposed to air. Insulation in contact with earth or water will not have surface resistance; however, exposed air ducts can have surface resistance on the inside as well as on the outside. Surface resistances are tabulated in Table 4-5.

Table 4-5. Surface Resistances for Nonreflective Surfaces

Wind, mph (m/sec)	Position of Surface	Direction of Heat Flow	Surface Resistance, $°F \cdot ft^2 \cdot hr/Btu$ ($°C \cdot m^2/W$)
0	horizontal	upward	0.61 (0.11)
0	45° slope	upward	0.62 (0.11)
0	vertical	horizontal	0.68 (0.12)
0	45° slope	downward	0.76 (0.13)
0	horizontal	downward	0.92 (0.16)
7.5 (3.4)	any	any	0.17 (0.030)
15.0 (6.7)	any	any	0.25 (0.044)

Source: American Society of Heating, Refrigerating and Air-Conditioning Engineers. ASHRAE Handbook of Fundamentals, 1972, p. 357.

Example 4-2: The vertical wall of a rock bin is constructed with the following layers: 1/2 inch of gypsum board, 3/4 inch of plywood, 5-1/2 inches of fiberglass batt, and 1/2 inch of gypsum board. What is the combined R-value of the insulation?

The r-value for each layer is multiplied by the layer's thickness, and the products are added to the surface resistance on the outer wall, as shown in Figure 4-1. The combined R-value is 19.84 $°F \cdot ft^2 \cdot hr/Btu$ (3.50 $°C \cdot m^2/W$).

Example 4-3: Suppose the cross-section used for Example 4-2 is taken through a 2 by 6 pine stud instead of through the fiberglass insulation. What is the combined R-value, including the stud?

The stud has an r-value of 1.25 $°F \cdot ft^2 \cdot hr/Btu \cdot in$, while the fiberglass had an r-value of 3.15 $°F \cdot ft^2 \cdot hr/Btu \cdot in$. The surface resistance is 0.68 $°F \cdot ft^2 \cdot hr/Btu$, and the combined R-value is:

$$R = 0.50 \times 0.90 + 0.75 \times 1.25 + 5.50 \times 1.25 + 0.50 \times 0.90 + 0.68$$

$$= 9.39 \ °F \cdot ft^2 \cdot hr/Btu \ (1.65 \ °C \cdot m^2/W).$$

This example will be used in the next section to illustrate the effect of parallel heat loss.

Parallel Heat Loss

When heat can escape by several paths, each with a different R-value, it is most convenient to work with the thermal transmittance of the insulation. This quality, given the symbol U, has units of $Btu/hr \cdot ft^2 \cdot °F$ ($W/m^2/°C$) and is often called the U-value. Mathematically, the thermal transmittance of insulation is:

$$U = \frac{1}{A}\left(\frac{A_1}{R_1} + \frac{A_2}{R_2} + \frac{A_3}{R_3} + \ldots\right) \qquad (4\text{-}6)$$

where:

R_1, R_2, and R_3 are R-values for each path of heat escape.

A_1, A_2, and A_3 are the cross-sectional areas of each path.

A is the sum of A_1, A_2, A_3, and so on.

The lower the thermal transmittance, the less heat flow through the insulation.

Example 4-4: Calculate the thermal transmittance of a 4 foot by 4 foot stud wall with layers as shown in Figure 4-1. The 2 by 6 studs are located on 16-inch centers as shown in Figure 4-2.

Path 1 through the fiberglass has an R-value of R_1 = 19.84 °F·ft²·hr/Btu (3.50 °C·m²/W) as calculated in Example 4-2 and an area of A_1 = 13.12 ft² (1.22 m²) as shown in Figure 4-2. Path 2 through the studs has an R-value of R_2 = 9.39 °F·ft²·hr/Btu (1.65 °C·m²/W) as calculated in Example 4-3 and an area of A_2 = 2.88 ft² (0.27m²). Substituting these numbers into Equation 4-6 yields the thermal transmittance of the stud wall.

$$U = \frac{1}{16}\left(\frac{13.12}{19.84} + \frac{2.88}{9.39}\right)$$

$$= 0.0605 \text{ Btu/hr·ft}^2\text{·°F } (0.343 \text{ W/m}^2\text{·°C}).$$

Notice that the parallel heat loss through the studs increases the transmittance by 20 percent over the path through the fiberglass alone. This illustrates the importance of accounting for the parallel heat losses.

Brackets, supports, and mountings of storage units, heat exchangers, pipes, and ducts are among the most often neglected paths for external heat losses. These losses tend to be significant even in systems that are well insulated in other respects. Some suggestions for reducing the losses are shown in Figure 4-3.

SMACNA and HUD Insulation Standards

The HUD Intermediate Minimum Property Standards specify that the storage device should be insulated so that losses during a 24-hour period do not exceed 10 percent of the storage capacity. Because the energy-saving benefits of insulation are so great, we recommend that storage devices be insulated to comply with the SMACNA (Sheet Metal and Air Conditioning Contractors' National Association) standard of a 2-percent loss in 12 hours and that the HUD standard be used only if all the following conditions are met:

- All the heat that escapes from storage heats the building.
- The solar system is shut down in the summer or the area around the storage device can be ventilated so that heat losses do not add to the air conditioning load.
- The storage device is used only to supply space heating.
- Uncontrolled heat losses from storage overheating the building are occasionally tolerable.

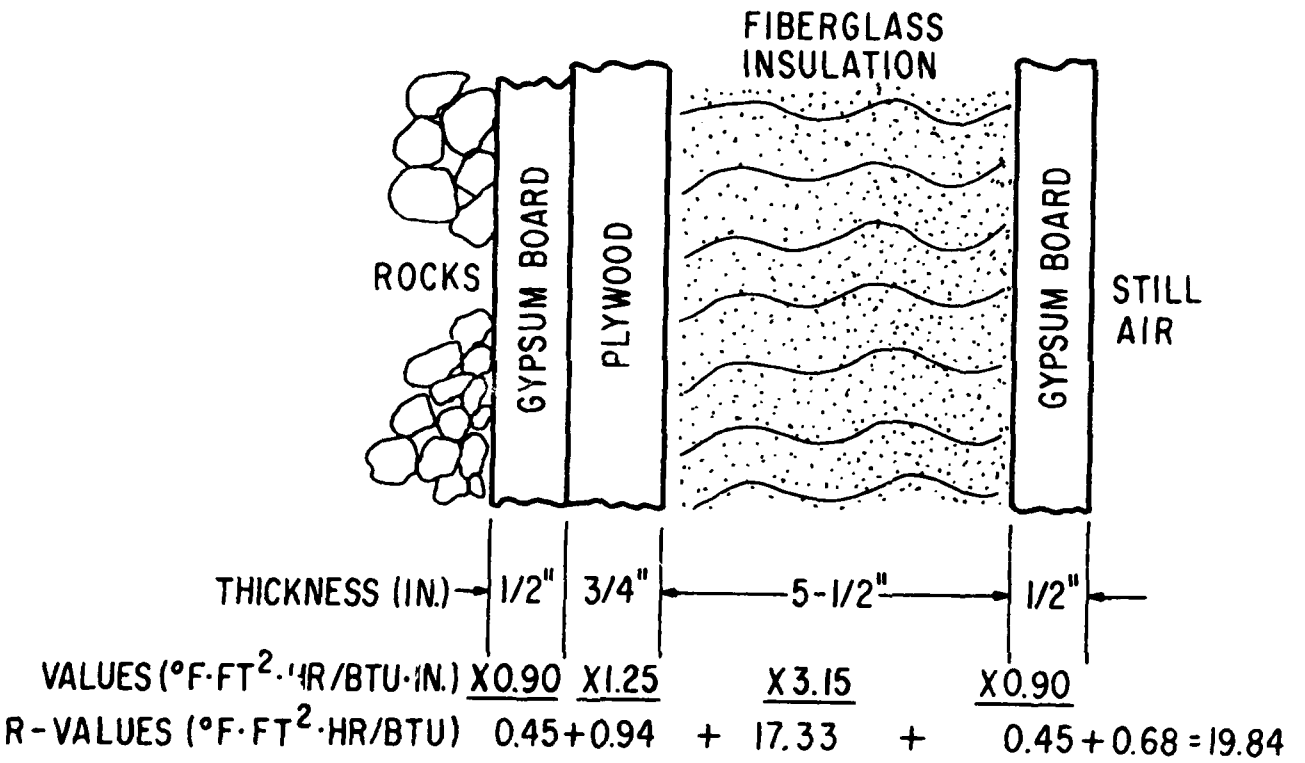


Figure 4-1. R-value of Multilayered Insulation

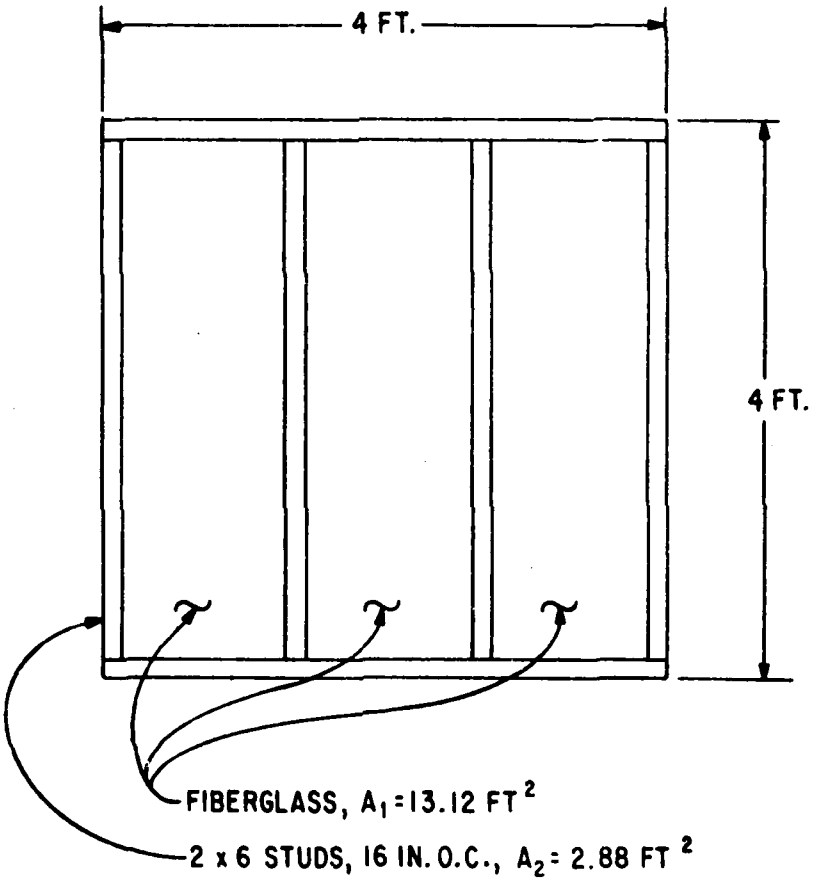


Figure 4-2. Layout of a 4 Foot by 4 Foot Stud Wall

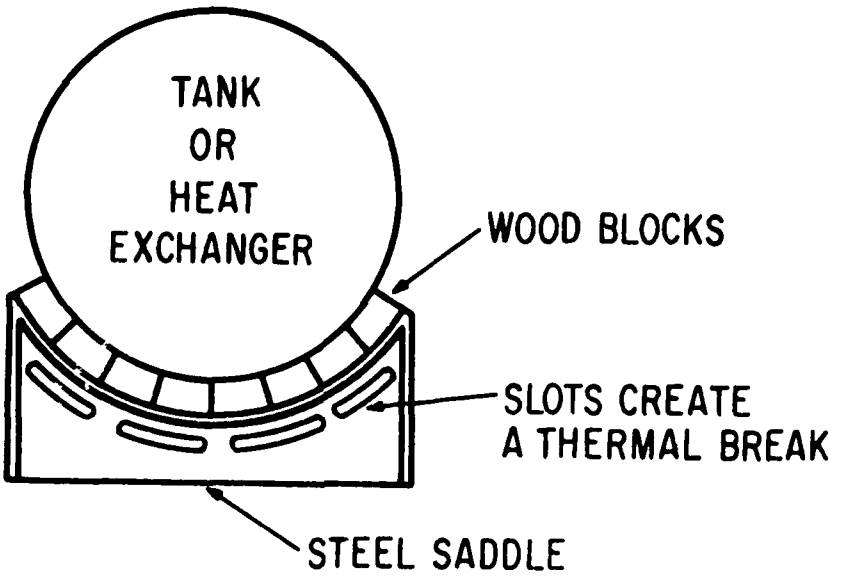


Figure 4-3. Methods for Reducing Parallel Heat Losses

The average rate of heat loss from storage, Q'_{avg}, is given by:

$$Q'_{avg} = UA\ (T_{avg} - T_a) \qquad (4\text{-}7)$$

where:

U is the overall thermal transmittance of the insulation on the storage unit as calculated in Equation 4-6.

A is the exposed surface area of the storage unit.

T_{avg} is the average temperature in the storage unit.

T_a is the ambient temperature surrounding the storage unit.[1]

The standards require that no more than a specified fraction, f, of the energy stored, Q, can be lost in a certain time period, t. (The energy stored is calculated from either Equation 4-3 or the methods of Chapter 13.) In the form of an equation, this statement is:

$$Q'_{avg}t = fQ\ . \qquad (4\text{-}8)$$

Combining Equations 4-7 and 4-8 and solving yields the maximum allowable thermal transmittance:

$$U = \frac{fQ}{At(T_{avg} - T_a)}\ . \qquad (4\text{-}9)$$

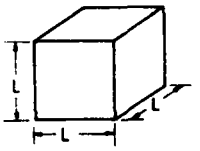
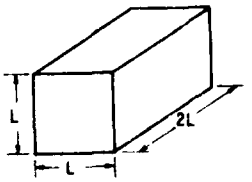
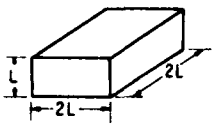
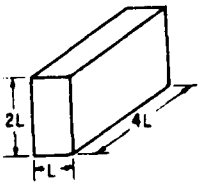
This is a general equation that can be used with all types of storage for either the SMACNA or HUD standards.

To simplify the calculation of maximum allowable thermal transmittance, we have supplied the insulation factor fQ/At in Tables 4-6 through 4-9. The following assumptions have been used in preparing the tables:

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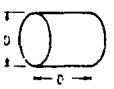
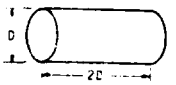
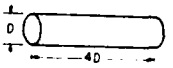
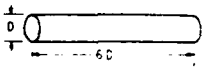
[1] For "ambient temperature," use the average temperature of the storage device's surroundings during the season when the storage device will he heated Assume an ambient temperature of about 68°F (20°C) for storage devices in heated areas. For unheated locations, find out the average outdoor temperature during the heating season from the nearest weather bureau office. For underground ambient temperature, assume that the ground temperature rises linearly from the average heating season outdoor temperature at the surface to the average annual outdoor temperature at a point 20 feet underground. This is a rough approximation to the ground temperature, but it is sufficiently accurate for our purposes here.

Table 4-6. Insulation Factor fQ/At for Rectangular Water Tanks

Size, gallons (liters)	Shape			
				
250 (946)	2.87 (9.05)	2.66 (8.39)	2.53 (7.98)	2.51 (7.92)
500 (1893)	3.62 (11.4)	3.35 (10.6)	3.19 (10.1)	3.17 (10.0)
750 (2839)	4.14 (13.1)	3.83 (12.1)	3.65 (11.5)	3.62 (11.4)
1000 (3785)	4.56 (14.4)	4.22 (13.3)	4.02 (12.7)	3.99 (12.6)
1500 (5678)	5.22 (16.5)	4.83 (15.2)	4.60 (14.5)	4.57 (14.4)
2000 (7571)	5.74 (18.1)	5.32 (16.8)	5.06 (16.0)	5.02 (15.8)
3000 (11356)	6.57 (20.7)	6.09 (19.2)	5.80 (18.3)	5.75 (18.1)
4000 (15142)	7.24 (22.8)	6.70 (21.1)	6.38 (20.1)	6.33 (20.0)
5000 (18927)	7.79 (24.6)	7.22 (22.8)	6.87 (21.7)	6.82 (21.5)

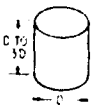
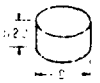
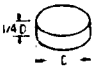
Note: Table values are for a 2-percent loss in 12 hours with an assumed daily temperature range of 60°F (33°C). Table units are Btu/hr·ft^2 (W/m^2). To obtain the maximum allowable thermal transmittance for the side and top insulation, divide the insulation factor by the difference between the average storage temperature and the ambient temperature. The maximum allowable thermal transmittance for the bottom insulation is assumed to be twice that on the top and sides.

Table 4-7. Insulation Factor fQ/At for Horizontal Cylindrical Water Tanks

Size, gallons (liters)	Shape 			
250 (946)	3.63 (11.5)	3.46 (10.9)	3.05 (9.62)	2.77 (8.74)
500 (1893)	4.57 (14.4)	4.36 (13.8)	4.84 (12.1)	3.49 (11.0)
750 (2839)	5.24 (16.5)	4.99 (15.7)	4.40 (13.9)	3.99 (12.6)
1000 (3785)	5.76 (18.2)	5.49 (17.3)	4.84 (15.3)	4.39 (13.8)
1500 (5678)	6.60 (20.8)	6.28 (19.8)	5.54 (17.5)	5.03 (15.9)
2000 (7571)	7.26 (22.9)	6.92 (21.8)	6.10 (19.2)	5.53 (17.4)
3000	8.31 (26.2)	7.92 (25.0)	6.98 (22.0)	6.33 (20.0)
4000 (15142)	9.15 (28.9)	8.71 (27.5)	7.68 (24.2)	6.97 (22.0)
5000 (18927)	9.86 (31.1)	9.39 (29.6)	8.28 (26.1)	7.51 (23.7)

Note: Table values are for a 2-percent loss in 12 hours with an assumed daily temperature range of 60°F (33°C). Table units are Btu/hr·ft^2 (W/m^2). To obtain the correct maximum allowable thermal transmittance for the insulation, divide the insulation factor by the difference between the average storage temperature and the ambient temperature.

Table 4-8. Insulation Factor fQ/At for Vertical Cylindrical Water Tanks

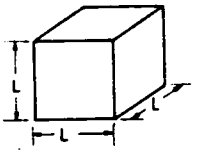
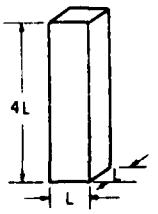
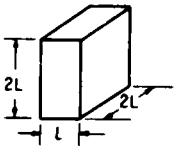
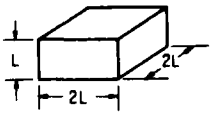
Size, gallons (liters)	Shape			
				
80 (303)	2.10[a, b] (6.62)	1.88[a] (5.93)	2.15 (6.78)	1.97 (6.21)
120 (454)	2.39 (7.54)	2.15 (6.78)	2.46 (7.76)	2.26 (7.13)
250 (946)	3.07 (9.68)	2.74 (8.64)	2.46 (9.91)	2.88 (9.09)
500 (1893)	3.87 (12.2)	3.46 (10.9)	3.96 (12.5)	3.63 (11.5)
750 (2839)	4.43 (14.0)	3.96 (12.5)	4.53 (14.3)	4.16 (13.1)
1000 (3785)	4.87 (15.4)	4.36 (13.8)	4.99 (15.7)	4.57 (14.4)
1500 (5678)	5.58 (17.6)	4.99 (15.7)	5.71 (18.0)	5.24 (16.5)
2000 (7571)	6.13 (19.3)	5.49 (17.3)	6.28 (19.8)	5.76 (18.2)
3000 (11356)	7.03 (22.2)	6.28 (19.8)	7.19 (22.7)	6.60 (20.8)
4000 (15142)	7.73 (24.4)	6.92 (21.8)	7.92 (25.0)	7.26 (22.9)
5000 (18927)	8.33 (26.3)	7.45 (23.5)	8.53 (26.9)	7.82 (24.7)

Note: Table values are for a 2 percent loss in 12 hours with an assumed daily temperature range of 60°F (33°C). Table units are Btu/hr·ft^2 (W/m^2). To obtain the maximum allowable thermal transmittance of insulation, divide the insulation factor by the difference between the average storage temperature and the ambient temperature.

[a]The maximum allowable thermal transmittance of the bottom insulation is assumed to be twice that on the top and sides for tanks specified by the first two columns.

[b]The first column is applicable to all tanks with height of one to three times the diameter. Insulation factors for most domestic hot water tanks can be found in the first column.

Table 4-9. Insulation Factor fQ/At for Rock Beds

Volume of Rock,[b] ft^3 (m^3)	Shape of Rock Bed[a]			
				
100 (2.83)	1.22 (3.82)	1.20 (3.79)	1.25 (3.88)	1.03 (3.22)
150 (4.25)	1.39 (4.35)	1.39 (4.32)	1.43 (4.45)	1.18 (3.69)
200 (5.66)	1.54 (4.79)	1.52 (4.76)	1.56 (4.89)	1.32 (4.07)
300 (8.50)	1.75 (5.49)	1.75 (5.46)	1.79 (5.62)	1.49 (4.64)
400 (11.33)	1.92 (6.03)	1.92 (5.99)	1.96 (6.18)	1.64 (5.11)
500 (14.16)	2.08 (6.50)	2.08 (6.47)	2.13 (6.66)	1.79 (5.52)
600 (16.99)	2.22 (6.91)	2.17 (6.85)	2.27 (7.07)	1.89 (5.87)
800 (22.65)	2.44 (7.60)	2.44 (7.54)	2.50 (7.79)	2.08 (6.44)
1000 (28.32)	2.63 (8.20)	2.63 (8.14)	2.70 (8.39)	2.22 (6.94)

Note: Table values are for a 2-percent loss in 12 hours with an assumed daily temperature range of 5° F (28°C). Table units are in $Btu/ft^2 \cdot hr$ (W/m^2). To obtain the maximum allowable thermal transmittance for the side and top insulation, divide the insulation factor by the difference between the average storage temperature and the ambient temperature. The maximum allowable thermal transmittance for the bottom insulation is assumed to be twice that on the top and sides.

[a,b] The insulation is assumed to cover both plena, but the volume and shapes given are for the rocks only.

- f = 0.02 (2 percent) and t = 12 hours, per SMACNA standards.
- The daily temperature range ($T_{max} - T_{min}$) is 60°F (33°C) for water tanks and 50°F (28°C) for rock beds.
- Several container shapes (Tables 4-6, 4-8, and 4-9) have only half as much insulation on the bottom as on the top and sides. The thermal transmittance calculated from the tabulated values of fQ/At apply to the top and sides of the container.
- Rock beds have a 30 percent void fraction and an allowance for the volume of the plena (air spaces) at the top and bottom.

To use the tables, find the vertical column that best represents the shape of your storage unit. Find the insulation factor fQ/At in the row corresponding to the storage unit's size. It may be necessary to interpolate between rows or columns. Divide the insulation factor by the difference between average storage temperature and ambient temperature to obtain the maximum allowable thermal transmittance, U.

Besides controlling thermal losses, insulation also limits the exposed surface temperature to prevent burns. The insulation factors given in Tables 4-6 through 4-9 will limit the temperature on the outer surface of the insulation to 140°F (60°C) or less.

Example 4-5: Calculate the maximum allowable thermal transmittance for the 375 cubic-foot (10.6 cubic-meter) rock bed shown in Figure 4-4. The rock bed will have an average temperature of 115°F (46°C) and will be located in a 68°F (20°C) basement.

The space occupied by the rocks is 102 inches x 96 inches x 66 inches (2.59 meters x 2.44 meters x 1.68 meters); its shape lies between the shapes listed in the first and fourth columns of Table 4.9. The size lies between the 300- and 400-cubic-foot rows. Interpolating between the rows and columns gives a value of fQ/At = 1.74 Btu/hr·ft^2 (5.48 W/m^2). Substituting these numbers into Equation 4-9 gives the maximum allowable thermal conductance on the top and sides:

$$U = \frac{1.74}{115-68} = 0.0370 \text{ Btu/hr·ft}^{2}\text{·°F}.$$

The insulation in Example 4-4, with U = 0.0605 Btu/hr·ft^2·°F, would be inadequate in this case.

Since the storage unit is to be located in a basement the insulation requirement can be relaxed to a 10 percent loss in 24 hours if the other requirements in this section are satisfied. The maximum allowable thermal conductance can be adjusted by multiplying it by 10/2 to change the fraction lost from 2 percent to 10 percent and by 12/24 to change the time period from 12 hours to 24 hours:

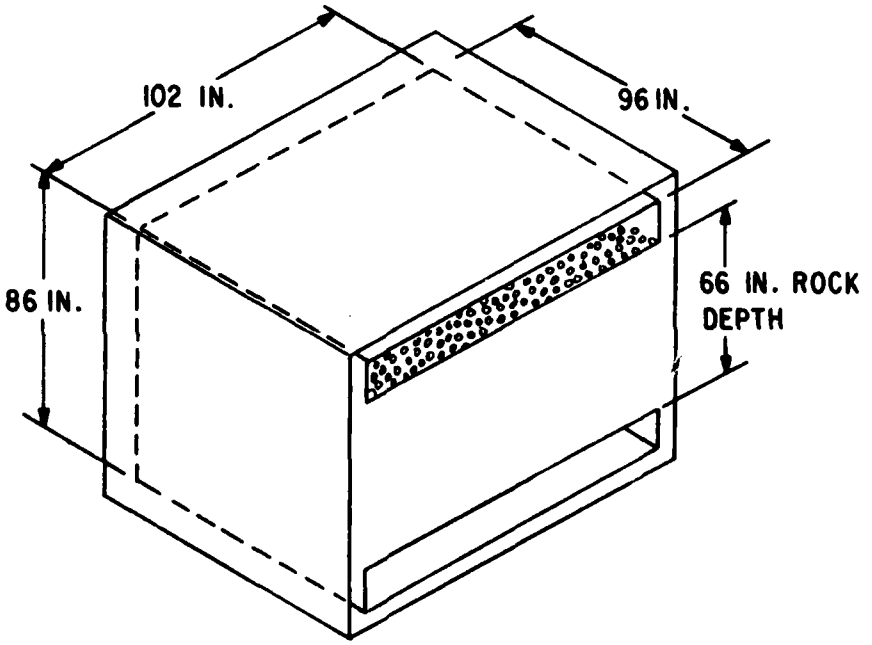


Figure 4-4. Inside Dimensions of a 375 Cubic-Foot Rock Bin

$$U = \frac{10}{2} \times \frac{12}{24} \times 0.0370 = 0.0926 \text{ Btu/hr} \cdot \text{ft}^2 \cdot {}^\circ\text{F}.$$

In this case, the insulation calculated in Example 4-4 would be adequate.

Insulating Pipes and Ducts

Heat can be lost not only from the storage device but also as it is moved into and out of storage. These losses include:

- Losses between the collector and the storage unit (charging losses).
- Losses between the storage unit and the heating load (discharging losses).

To minimize these losses, (1) the piping system from collector to storage must be well insulated and have weather protection, and (2) the piping or ductwork from storage to load must be kept as short as possible and be well insulated.

The Polytechnic Institute of New York recommends using R-4 insulation for pipes less than 1 inch in diameter and R-6 insulation for pipes 1 to 4 inches in diameter.

4.6 Heat Exchangers

Heat exchangers are devices that transfer heat from one fluid to another while preventing mixing of the two fluids. Hot fluid flows on one side of a metal barrier and heats a cold fluid flowing on the other side. In order for heat to be transferred the hot fluid must be hotter than the cold fluid directly across the barrier. This necessary temperature difference leads to a loss of overall system efficiency each time a heat exchanger is used. Heat exchangers typically used in solar energy systems are shown in Figures 4-5 through 4-10.

Heat Exchangers in Liquid-based Systems

The collectors must be protected against freezing in the winter. If anti-freeze is used for protection, a liquid-to-liquid heat exchanger must be installed as shown in Figure 2-2 to separate the heat transfer fluid from the water in storage. Because there must be a temperature difference from the collector side to the storage side of the heat exchanger, the collector

must operate at a higher, less efficient temperature than in a system without a heat exchanger between collector and storage. Thus, the heat exchanger imposes a performance penalty on the system.[2]

The collector-to-storage heat exchanger can be as simple as a coiled tube immersed in the storage tank (Figure 4-5) or wrapped around the outside of the tank (Figure 4-6). Figure 4-7 shows a wraparound shell heat exchanger.

The types of heat exchangers represented in Figures 4-5 through 4-7 rely on natural convection to move the water inside the tank past the heat exchange surface. If the tank is large, say several hundred gallons, natural convection is an inefficient means of transferring heat. Shell-and-tube heat exchangers (Figures 4-8 and 4-9) are often used in this case, and a pump circulates the water between the tank and the heat exchanger.

In the simple heating system shown in Figure 2-2, a heat exchanger (liquid-to-air) is needed to transfer heat to the building. This heat exchanger is often a finned-tube unit (Figure 4-10) inserted in an air duct. Another common type of finned-tube heat exchanger is the A-frame type (not shown). Less frequently used alternatives are baseboard convectors, radiant heating coils, and individual fancoil units.

Heat Exchangers in Air-based Systems

In contrast with the liquid-based system, the air-based system (Figure 2-3) does not need separate heat exchangers between the collectors and the heating load. The rock bed is both the storage device and the collector-to-load heat exchanger.

Air-based collector systems used to heat domestic hot water must use an air-to-liquid heat exchanger. Such a heat exchanger usually consists of finned water tubing in the air-handling duct (Figure 4-10) similar to the finned tube heat exchanger used in air ducts of liquid-based systems.

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[2]Another method of protecting the collectors is to drain them whenever there is danger of freezing weather. This method, known as the draindown system, is one of the most efficient solar collection systems available. Details of the draindown system and its many variations are available in systems design manuals such as ITT's Solar Systems Design Manual.

A draindown system must be totally foolproof. Pipes must be carefully pitched and collectors carefully selected to ensure that all of the water will drain when it should. A single failure can ruin the collectors. Many designers prefer to use antifreeze in the collectors rather than risk this catastrophe. The designer must decide whether to pay the penalties of lower performance and higher first cost for an antifreeze system in return for less risk of an expensive failure.

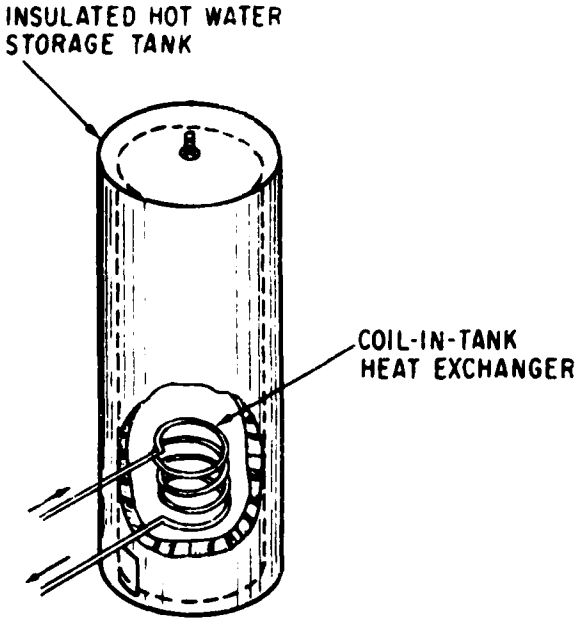


Figure 4-5. Typical Coil-in-Tank Heat Exchanger

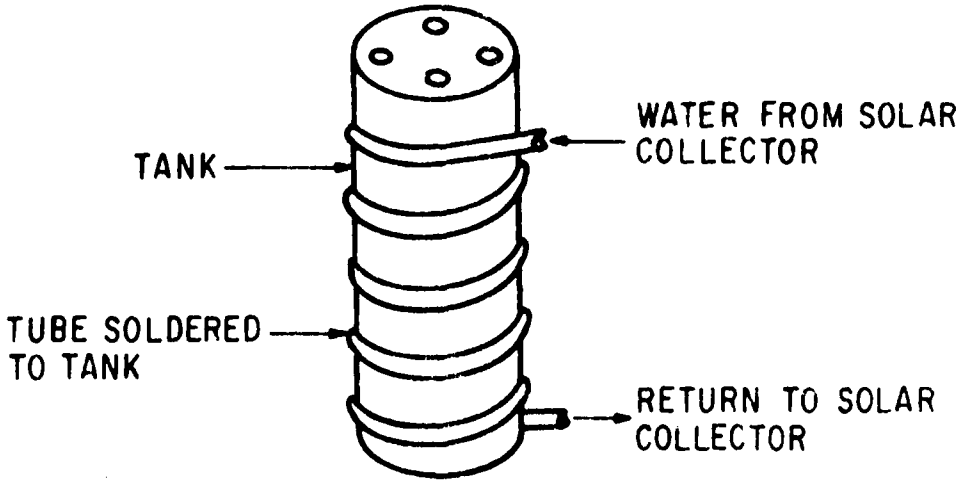


Figure 4-6. Schematic Drawing of Wraparound (Traced Tank) Heat Exchanger

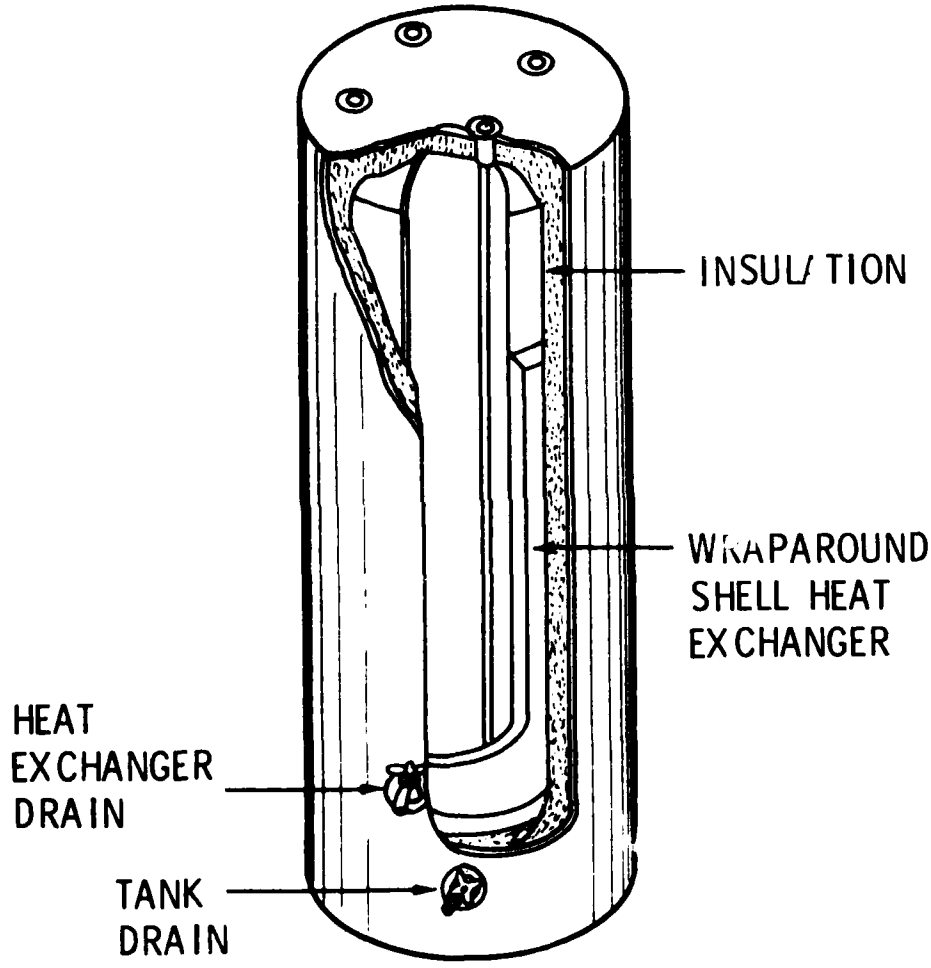


Figure 4-7. Wraparound Heat Exchanger

A pressure-bonded metal plate with integral fluid passageways is clamped around the outside of the storage tank.

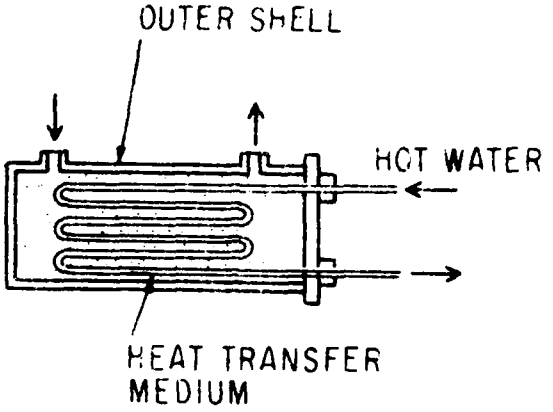


Figure 4-8. Shell-and-Tube Heat Exchanger

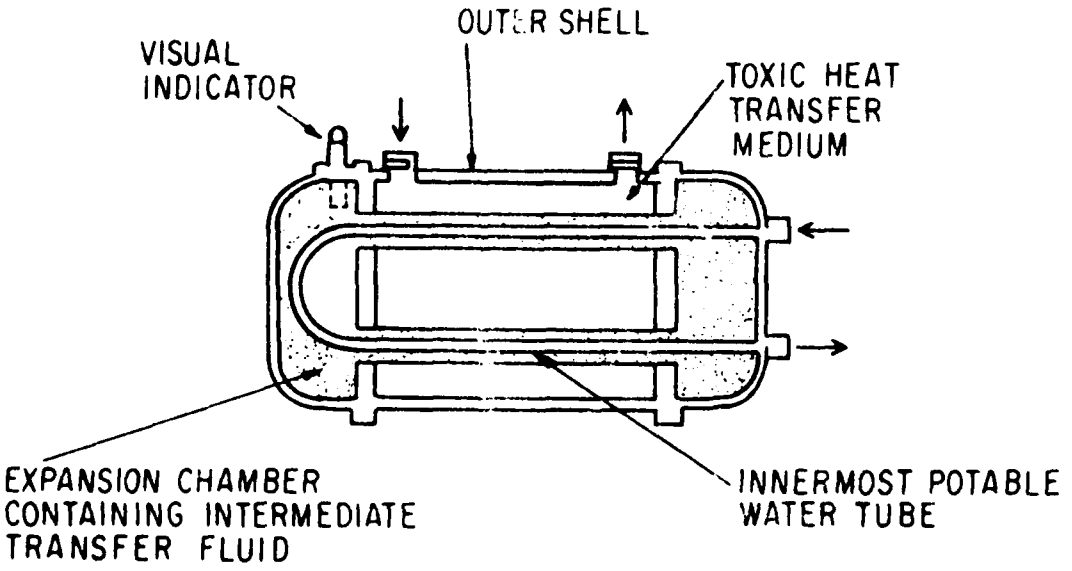


Figure 4-9. Shell-and-Double-Tube Heat Exchanger

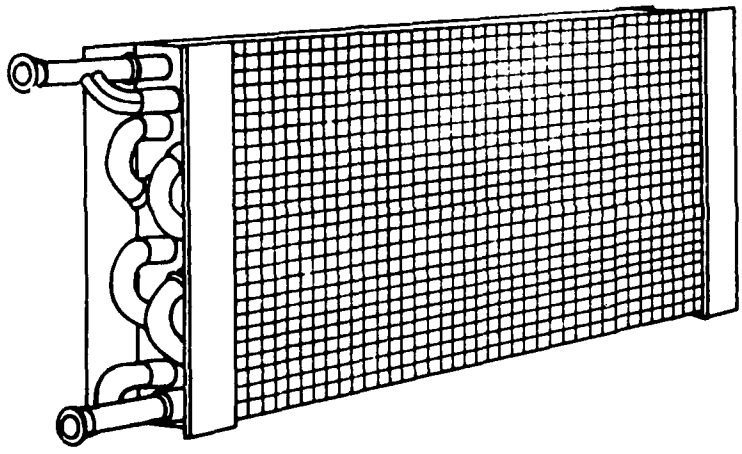


Figure 4-10. Typical Liquid-to-Air or Air-to-Liquid Heat Exchanger

Drawing courtesy of Bohn Heat Transfer Division, Gulf-Western Manufacturing Company, Danville, Illinois

Heat Exchanger Effectiveness

Using heat exchangers imposes a penalty on the solar space heating system. A collector-to-storage heat exchanger forces the collector to operate at a higher temperature than in a system without the heat exchanger. Similarly, the storage-to-load heat exchanger forces the storage system to operate at a higher temperature than would be required if that heat exchanger could be eliminated.

In order to calculate how efficiently a system with heat exchangers will perform, determine the penalty imposed on the system by the heat exchangers. This penalty, called heat exchanger effectiveness, is defined as the actual rate of heat exchange divided by the rate of heat exchange of a perfect, infinitely large heat exchanger.

Since there is no perfect, infinitely large heat exchanger, the designer's task is to choose the size of heat exchanger that will minimize the overall cost of the system. Most heat exchanger manufacturers can select the properly sized heat exchanger given the following information about the system:

- The physical characteristics of the two fluids in the heat exchanger.
- The amount of heat to be transferred.
- The flow rates on both sides of the heat exchanger.
- The approach temperature difference, defined as the difference between the temperatures of the hot fluid entering the heat exchanger and the heated fluid leaving the heat exchanger.

A form for specifying heat exchangers is provided in Section 14.1, Worksheet I.

4.7 Symbols Used

Main Symbols

A	area, ft^2 (m^2)
C_p	heat capacity, Btu/lb·°F (J/kg·°C)
f	fraction of energy stored
m	mass, lb (kg)
Q	amount of heat stored, Btu (J)
Q'	rate of heat flow, Btu/hr (W)
R	thermal resistance to flow of heat, $°F \cdot ft^2 \cdot hr/Btu$ ($°C \cdot m^2/W$)
r	resistance to flow of heat per unit thickness, $°F \cdot ft^2 \cdot hr/Btu \cdot in$ ($°C \cdot m^2/W \cdot cm$)
s	thickness of an insulation layer, in. (cm)
T	temperature, °F (°C)
t	time, hr (sec)
U	thermal transmittance, $Btu/hr \cdot ft^2 \cdot °F$ ($W/m^2 \cdot °C$)

Subscripts

a	ambient condition
avg	an averaged quantity
max	a maximum quantity
min	a minimum quantity
s	surface condition
1,2,3	first, second, third, (and subsequent) layers of insulation or paths for heat flow

4.8 References

- American Society of Heating, Refrigerating and Air-Conditioning Engineers, 345 East 47th Street, New York, New York 10017.
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5.0 SENSIBLE HEAT STORAGE IN LIQUIDS

Sensible heat storage in liquids was mentioned briefly in Sections 2.3.1, 2.8, and 2.9. General features of sensible heat storage were discussed in Chapter 4. In this Chapter technical details of liquid sensible heat storage are explored, with the discussion concentrating on water tank storage.

5.1 Possible Liquid Systems

The choice of an appropriate storage liquid depends ultimately on total system cost but is determined mainly by the required temperature range of the storage.

5.1.1 Low Temperature Liquids

Water in unpressurized tanks is the usual storage liquid when the maximum storage temperature remains below 212°F (100°C). Water is usually abundant and therefore inexpensive, has high thermal conductivity, low viscosity, and has one of the largest heat capacities of any sensible heat storage material. The effect of the latter property is that a smaller volume of water is required for a given amount of heat storage. The major drawbacks of water are its high freezing temperature, its expansion upon freezing, and its corrosive effect on common engineering materials (except copper). Its low boiling point can cause high pressure within solar collector systems under zero flow conditions. Water itself has no adverse biological or environmental effects, but some corrosion inhibitors (such as hydrazines) used with water are toxic. Water used for storage can often be circulated directly in solar collectors, eliminating the heat exchanger between collector and storage. Indeed, one of the simplest water-based solar collectors, the breadbox collector, consists of a tank or plastic bag filled with water and placed in sunlight. Low temperature water storage in water tanks is a mature technology, and one of the most cost effective.

Water storage in natural containers, such as ponds and aquifers was discussed briefly in Sections 2.8 and 2.9 respectively. Such systems are potentially lower in cost then water tank storage. The lack of insulation of these natural containers requires large sizes to reduce the percentage of heat loss - requiring sizes more appropriate to seasonal heat storage for large systems. The large size requirement has so far made experimental systems few in number, making a technical discussion of these systems inappropriate at this time.

Other liquids are ordinarily used only as nonfreezing heat transfer fluids. Because these alternative liquids cost considerably more than water - from $2 to $20/gal, they are used to prevent the collector from freezing and are separated from the main storage tanks containing water by a heat exchanger. The thermal properties of these fluids (heat capacity, thermal conductivity, and viscosity) are less desirable than those of water,

but corrosive action is usually less. Since most of the alternative liquids are toxic to some degree, most plumbing codes require that toxic solutions be separated from potable water by double-walled heat exchangers.

Water/glycol solutions are commonly used in solar collectors, usually with corrosion inhibitors. Ethylene glycol is the most common and one of the most toxic. The boiling and flash points of aqueous ethylene glycol mixtures are low and can easily be reached under zero flow conditions in solar collectors. Propylene glycol is also common and is less toxic but more viscous than ethylene glycol. Both diethylene and triethylene glycol have been used in industrial applications. These fluids have higher boiling points but are more expensive and more toxic than ethylene glycol.

Petroleum oils are generally designed for higher temperatures, but the viscosity of some may be low enough for low temperature operation. The toxicity of these fluids is relatively low. Silicone fluids have low toxicity, low corrosiveness, and good long term stability. Their major drawbacks are high cost, high viscosity, and a greater tendency for leakage through fittings and seals. Numerous other heat transfer fluids are commercially available. Some of these are discussed in Appendix E.

The most cost effective and most common storage liquid used for space heating or domestic hot water heating is water in unpressurized tanks. Many other different systems have been proposed, but, so far, those few that are commercially available are cost effective only in very limited applications.

5.1.2 High Temperature Liquids

Several technologies exist for storage temperatures above the normal boiling point of water (212°F (100°C)). Pressurized water has been used for many years in the utility industry. The higher the temperature, the higher the required pressure, and the more costly the pressure vessel. Organic oils can be used for storage temperatures up to 570°F (300°C). A nitrogen cover gas is required to protect the oil from degradation, but even with the cover gas, degradation is rapid above 570°F. Liquid salts, such as Hitec (produced by Du Pont) can be used at slightly higher temperatures. Hitec has a maximum temperature of 1000°F (540°C). For temperature as high as 1400°F (760°C) the utility industry has used liquid metals such as sodium. Liquid metals at these temperatures require containers and heat exchangers made of stainless steel. In general, for all liquid sensible heat storage, the higher the temperature, the more restrictive are the container requirements, with a resulting higher cost for storage. Possible exceptions exist for large systems in natural containers. For example, salt cavern storage of hot oils may be both technologically and economically feasible.

5.2 Tank Containment Systems

Before selecting a particular type of storage tank, a number of variables must be considered. These include:

- Size and shape.
- Material.
- Location.
- Insulation.
- Corrosion.
- Cost.
- Leak protection.
- Protective coating.
- Installation.
- Pressure and temperature limits.

Each individual liquid-based storage system has characteristic requirements. The temperatures listed below are subject to variation and are intended only as a general guide. Direct space heating usually requires storage temperatures no higher than 160°F (71°C), although some hydronic heating systems require higher temperatures. Most direct space heating systems use unpressurized tanks. Solar-assisted heat pump systems typically operate at less than 100°F (38°C); this dramatically reduces the amount of insulation required on the tank. Absorption air conditioning systems use cold storage at temperatures below 55°F (13°C) or hot storage, sometimes requiring a pressurized vessel, above 170°F (77°C).

Storage temperatures for load management may resemble storage for direct space heating systems if the tank is heated electrically at off-peak rates. If a load management system uses heat pumps to move heat from one part of the building to another or to air condition during the day and heat at night, the temperature requirements will be similar to the temperature requirements for a solar-assisted heat pump.

Direct heating of potable water usually requires temperatures of less than 140°F (60°C) and a pressurized tank. Preheating the water usually requires temperatures under 120°F (49°C). If the water main feeds directly into the preheat tank, this tank must be pressurized. If, instead, the water main feeds into a heat exchange coil which passes through the preheat tank, the tank need not be pressurized. It is not unusual for domestic hot water systems to operate at higher temperatures than those listed here.

Various tank materials that can meet these temperature and pressure requirements are available. Tested materials, such as steel, fiberglass, concrete, or wood with plastic lining should be used to avoid the risks inherent in using materials that have not been proven. Advantages and disadvantages of each type of tank are shown in Table 5-1. All types of tanks can be purchased or constructed in any size likely to be used in storage systems.

Normally, only one tank is used for storage in space heating systems. Where the properly sized tank is unavailable, or where space restrictions dictate the use of smaller tanks in place of a larger one, two or more storage tanks can be used. Two small tanks cost more than a large one, however, and a multiple tank system requires more insulation because it exposes a larger surface area than a single tank.

5.2.1 Tank Types

Table 5-1. Advantages and Disadvantages of Tank Types

ADVANTAGES Steel Tank	Fiberglass Tank	Concrete Tank	Wooden Tank with Liner
Steel tanks 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.	Concrete tanks may be cast in place or may be precast.	Indoor installation is easy.
Connections to plumbing are easy to make.	Some tanks are designed specifically for solar energy storage.		
Some steel tanks are designed specifically for solar energy storage.	Fiberglass does not rust or corrode.		

DISADVANTAGES Steel Tank	Fiberglass Tank	Concrete Tank	Wooden Tank with Liner
Complete tanks are difficult to install indoors	Maximum temperature is limited, even with special resins.	Careful design is required to avoid cracks, leaks, and excessive cost.	Maximum temperature is limited.
Steel tanks are subject to rust and corrosion.	Fiberglass tanks are relatively expensive.	Concrete tanks must not be pressurized.	Wooden tanks must not be pressurized.
Steel tanks are relatively expensive.	Complete tanks are difficult to install indoors.	Connections to plumbing are difficult to make leaktight.	Wooden tanks are not suitable for underground installation.
	Fiberglass tanks must not be pressurized.		

5.2.1.1 Steel Tanks

The principal advantages of steel tanks are the relative ease of fabricating them to ASME Pressure Vessel Code requirements, the ease of attaching pipes and fittings, and the amount of experience available with steel tanks. The main problem with steel tanks is corrosion. The difficulty of installing them in enclosed buildings and their cost are also problems.

Four types of corrosion present the greatest threat to steel tanks: electrochemical corrosion, oxidation (rusting), galvanic corrosion, and pitting. Six methods of protection, which can be used singly or in combination, are (1) sealing the system, (2) adding chemical corrosion inhibitors, (3) using protective coatings or liners, (4) using cathodic protection, (5) increasing the metal's thickness, and (6) using noncorroding alloys. Since corrosion rates double with each 20°F (10°C) increase in temperature, limiting the maximum temperature will prolong the tank's life.

Electrochemical Corrosion

Electrochemical corrosion of a metal is mainly governed by two conditions: the pH of the liquid (electrolyte) and the electric potential of the metal. The pH is a measure of acidity or alkalinity of the liquid; an acid solution has a pH lower than 7, an alkali has a pH higher than 7, and a neutral solution has a pH of 7. The electric potential of the metal is usually given relative to that of hydrogen, which is always present in water.

Marcel Pourbaix has produced diagrams such as Figure 5-1 showing corrosion conditions for various metals as function of pH and electrical potential. As shown in Figure 5-1, under normal conditions the pH is 7, and iron has an electric potential of -0.44 volts relative to hydrogen. This state, Point X, is in a region where corrosion can occur.

Three methods of preventing corrosion can move Point X. (1) Cathodic protection can move it down, (2) alkalinization can move it to the right, and (3) anodic protection can move it up.

(1) Cathodic protection, the most reliable method, supplies a sacrificial anode. It can be used effectively to protect either the inside or the outside of the tank. This system usually uses an anode, a metal bar selected to become oxidized (lose electrons) in preference to the structural metals. The sacrificial anode must be submerged in water and electrically connected to the tank. Since protection ends when the anode has completely dissolved, the anode should be inspected annually and replaced if necessary.

In choosing a sacrificial anode, you must choose a metal that is more reactive than the steel tank. The most commonly used anodes are made of magnesium, aluminum, or zinc. A magnesium bar is an appropriate anode for

most storage tanks. Aluminum has a tendency to form a protective coating over itself, reducing the effectiveness of the protection. Zinc, whether as a metal bar, as galvanizing, or as an additive to epoxy or paint, is not recommended for most thermal energy storage applications. At temperatures above approximately 155°F (68°C), the roles of zinc and steel reverse, so that the steel is sacrificed instead of being protected. Zinc can provide effective protection if the maximum allowable tank temperature is kept below about 140°F (60°C), however.

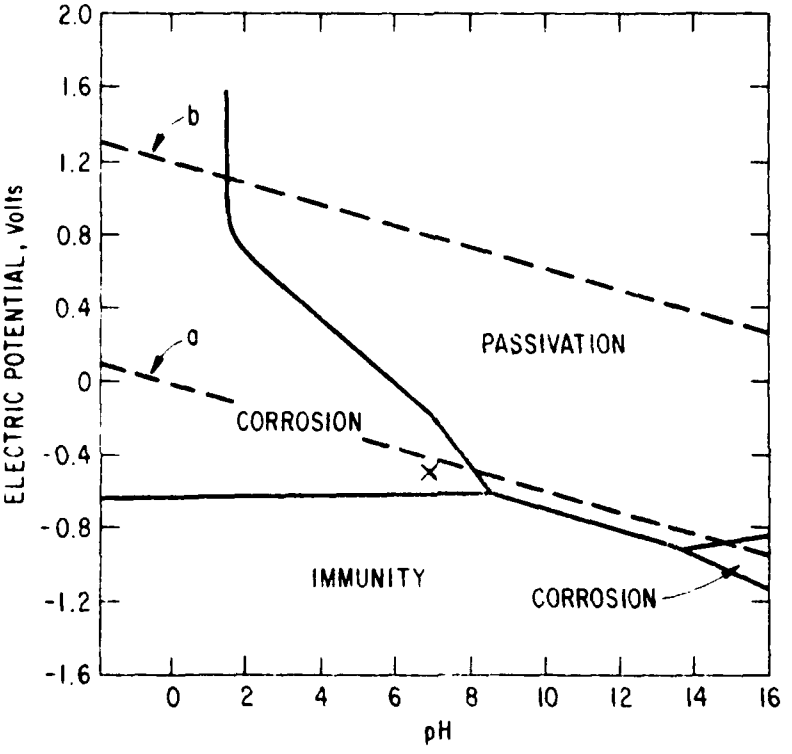


Figure 5-1. Pourbaix Diagram for Iron at 77°F (25°C)

Adapted from Marcel Pourbaix, translated from the French by James A Franklin. Atlas of Electrochemical Equilibria in Aqueous Solutions. Pergamon Press, 1966.

Cathodic protection works particularly well if the tank is lined with epoxy, glass, or hydraulic stone. The coating protects most of the steel and leaves only a few small uncoated areas.

(2) Alkalinization changes the pH. Steel is relatively inactive in an alkaline environment; adding lime, caustic soda, or trisodium phosphate to maintain a pH of 9.5-12 will protect steel from electrochemical corrosion. Increasing the pH of the water also moves iron into the region of passivation, (see Figure 5-1) in which a protective film forms over its surface. Maintain the required alkalinity by periodically testing and adjusting the pH. Alkalinization works well for vented or sealed tanks but the added chemicals do make the water unpotable. Do not use alkalinization if aluminum parts will be exposed to the water, since aluminum corrodes quickly in an alkaline environment.

In most situations, cathodic protection should not be combined with strong alkalinization. In one case the alkalinization may cause a protective coating to form over the anode, but in another it may cause rapid consumption of the anode. In both cases the effectiveness of the protection will be decreased.

(3) Anodic protection passivates or shields the anodic sites - the metal surfaces where oxidation is occurring. Anodic inhibitors, such as chromate, react with the metal forming a comparatively hard, longlasting film. Anodic protection is rarely used for several reasons. It does not stop corrosion completely, increased corrosion rates can occur if the potential of the steel is raised too high, and imperfections in the passivated surface can lead to pit corrosion. For these reasons, we do not recommend anodic protection.

Oxidation

Oxygen can enter a tank in two ways: It can be dissolved in water that enters the tank, or it can enter through the tank's air vent. Besides causing rust, oxygen tends to catalyze other types of corrosion.

One method of stopping oxidation is to seal the system so that no air or water can enter it. The oxygen initially present in the system will quickly be removed as it reacts with parts of the system. Although minor rusting may occur at that time, no further rusting can take place after the oxygen has reacted. Sealing the system requires that all of the components be able to withstand the pressure generated when the system is heated. The tank should be designed to the American Society of Mechanical Engineers' (ASME) Pressure Vessel Code or local code requirements and provided with a pressure relief valve. With some system configurations, an expansion tank may also be required. The main disadvantage of a sealed system is its cost; but if the system must operate close to or above the normal boiling point, the tank must be pressurized anyway. The effective protection from oxidation brought about by sealing is an added bonus.

Oxygen dissolved in water flows continuously into domestic hot water tanks. Thus, even though the tank is pressurized, oxygen cannot be excluded from it. Linings of glass or hydraulic stone limit the contact of steel and oxygen, thus limiting the extent of rusting. In tanks larger than 120 gallons

(454 liters), glass or hydraulic stone can be expensive. For these tanks, four interior coatings of baked-on phenolic epoxy should be sufficient.

Similarly four coats of baked-on phenolic epoxy are suggested to protect the interior of vented tanks. If a baked-on treatment is unavailable, four coats of two-part epoxy can be specified, although it is less effective than the baked-on coating. Sodium sulfite added to the water will scavenge oxygen; it must be periodically tested and replenished. Although sodium sulfite is the most commonly used scavenger, both SUR-GARD™ and ELIMIN-OX™, relatively new products marketed by Nalco Chemical Company (Oakbrook, IL), have superior chemical activity.

If the tank is to be located underground, we recommend two coats of coal-tar epoxy on the outside. For indoor or aboveground locations, a coat of primer followed by two coats of enamel provides adequate outer protection.

Galvanic Corrosion

Galvanic corrosion occurs when dissimilar metals in an electrolytic solution are in electrical contact with each other. This often occurs when a copper fitting is screwed into a steel tank. The water in the tank serves as the electrolyte. The more reactive metal (steel) dissolves in the vicinity of the less reactive metal (copper), and the usual result is a leak in the system.

Galvanic corrosion can be minimized by electrically insulating dissimilar metals from each other. Use dielectric bushings to connect pipes to tanks, and gaskets or pads to insulate other components. Eliminate other electrical connections between dissimilar metals. A common ground, for example, would defeat the effort to insulate dissimilar metals from each other. Use a volt-ohm meter (VOM) to measure the electrical resistance between components before the system is filled with water. A resistance of more than 1000 ohms indicates adequate insulation, but a resistance of less than 100 ohms means that an electrical connection still exists. Because the dissimilar metals plus the electrolyte form a battery, testing after the system has been filled with water will give false resistance measurements.

The dissimilar metal combinations encountered most frequently in solar systems are iron-copper, iron-brass, aluminum-iron, aluminum-copper, aluminum-brass, and zinc-iron. The more reactive metal is listed first in each case except for the zinc-iron combination. Zinc is more reactive than iron below about 155°F (68°C) but less reactive above that temperature.

Pitting

Pitting is a localized corrosion in which small-diameter holes penetrate the base metal. Localized galvanic corrosion produces a pit that can quickly penetrate the more reactive metal. In a storage system, a typical source of the less reactive metal ions is a copper pipe. The copper ions circulate in the water until they plate out on the steel tank and cause a pit.

Another type of pit corrosion is believed to be caused by small imperfections in a passivated surface. In these and other areas of incomplete film coverage such as crevices and screw threads, the anodic metal surface is exposed - rapid oxidation occurs and deeper pitting results. Cathodic protection as discussed earlier is more effective against pitting than alkalinization or anodic protection. By using a sacrificial anode it is not dependent upon the effectiveness of a protective film to cover all anodic sites. The presence of some ions, especially chloride ions, tends to encourage this form of pitting as well as other types of corrosion by forming a chloride concentration cell.

Other Methods of Protection

Occasionally, the thickness of the walls in steel tanks is increased for corrosion protection. Although the use of thicker metal does not stop corrosion, it does increase the length of time before the corrosion causes system failure. Use of thicker metal is often combined with alkalinization.

Although rarely encountered in solar energy storage systems because of high first cost, stainless steel alloys form a passivated surface condition that protects the metal from corrosion. Cathodic protection will destroy that surface condition and should not be used with stainless steel.

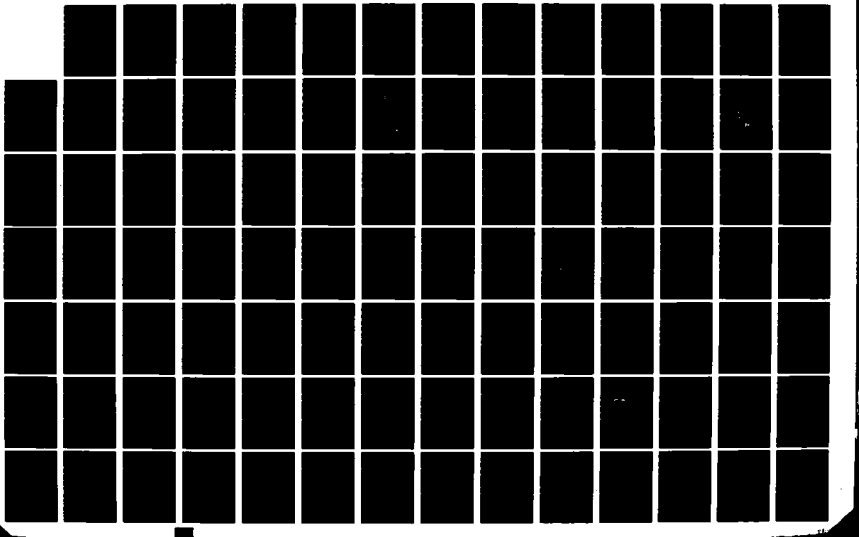
Recommendations for Steel Tanks

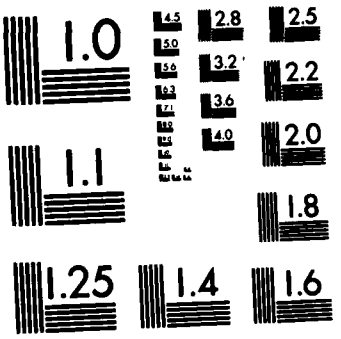
We recommend the following for most steel tanks used in space heating applications:

- Use four coats of baked-on phenolic epoxy on the inside of the tank.
- Use a magnesium bar as cathodic protection.
- Electrically insulate dissimilar metals, except the magnesium anode, with dielectric bushings, gaskets, or pads.
- Use sodium sulfite to scavenge oxygen.
- Do not use chromate-type corrosion inhibitors. They are highly toxic, carcinogenic, cause damage to a common type of pump seal, and are difficult to dispose of properly. Since chromates act by passivating the steel surface, their effectiveness is much reduced when combined with cathodic protection.
- If possible, limit the maximum tank temperature to 160°F (71°C) or less.
- Protect the outside of a buried tank with two coats of coal-tar epoxy and a magnesium anode. Use a coat of primer and two coats of enamel to protect the outside of abovegrade tanks.
- If the tank will be pressurized, it must comply with Section VIII of the ASME Boiler and Pressure Vessel Code or local codes.
- Do not install large steel tanks in basements, crawl spaces, or other locations where building modifications would be necessary to replace them.

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5.2.1.2 Fiberglass-Reinforced Plastic Tanks

Both factory-insulated and on-site-insulated fiberglass-reinforced plastic (FRP) tanks are available and have been successfully used in solar energy installations. The main advantage of FRP tanks is that they do not corrode.

Factory-insulated FRP tanks, specifically designed for solar energy storage, are generally preferable to on-site-insulated tanks both economically and in ease of installation. A typical factory insulated FRP tank consists of an inner FRP shell covered with 2 to 4 inches (5 to 10 cm) of urethane insulation protected by an outer FRP shell. They can be used outdoors above or below grade or in a garage; avoid locations where major building modifications would make tank replacement necessary.

Installation of on-site-insulated FRP tanks requires much more care than that of factory-insulated FRP tanks. Since the fiberglass is thin, the tanks must be protected from punctures and must be carefully supported to prevent rupture.

The extra labor required to insulate and install an on-site-insulated tank may make a factory-insulated tank less expensive. Some installations, however, can benefit from the greater variety of sizes and shapes that are available in on-site-insulated tanks.

Nearly all FRP tanks have two limitations.

- They must not be pressurized or subjected to a vacuum inside the tank unless they are specifically designed for it. A vent will ensure that the tank is not subjected to these conditions.
- Their temperatures must never exceed the limit specified by the manufacturer, thereby voiding the warranty and damaging the tank. Since the temperature limit is nearly always below the boiling point, the system controller should be adjusted to stop heat addition to the tank at 5°F (3°C) below the manufacturer's temperature limit. Since many controllers do not have a provision for limiting the tank temperature, you must select the controller carefully.

The temperature limitation is determined by the type of resin used to make the tank. Ordinary polyester resins have a limitation of 160°F (71°C) -- suitable only for low-temperature tanks. With premium quality resins, the temperature limitation can be raised to 180-200°F (82-93°C). Consult the tank manufacturer for details.

Before accepting delivery of an FRP tank, inspect it carefully. Do not accept delivery of a tank that has been dropped or shows any signs of physical damage. Before installing the tank, inspect the gel coat on its inner surface. If the gel coat is cracked or if fibers are exposed, hot water can break the bond between the glass fibers and the resin, which will lead to premature tank failure.

5.2.1.3 Concrete Tanks

Concrete tanks for solar energy storage can be divided into two categories: cast-in-place tanks and precast tanks. Precast tanks include those designed primarily to be used as septic tanks and utility vaults. Major advantages of concrete tanks are that they are relatively inexpensive as long as their shape is kept simple, the mass of concrete becomes part of the storage system, and concrete is a readily available construction material. Concrete also has considerable resistance to underground loads. Because concrete can be cast in almost any shape, it is a good material for retrofit installations. Concrete is also fireproof and usually corrosion resistant. However check underlying ground formation for the possibility of acid attack on the concrete. Sulfuric acid formation from weathering pyrite will badly corrode concrete.

Concrete does have several disadvantages, however. It is subject to capillary action so water can seep through cracks and joints unless the tank is lined. Leakproof connections through the tank walls are often difficult to make. Plaster coatings, which are adequate for cisterns, have a tendency to crack with the fluctuating temperatures of thermal storage materials. Seepage that can be tolerated in a cistern or swimming pool will degrade the insulation around a tank. We recommend using either a spray-on butyl rubber coating 30 to 50 mils (0.75 to 1.25 millimeters) thick on the inside of the tank or a replaceable liner of the type used for wooden tanks.

The weight of a concrete tank may be either an advantage or a disadvantage, depending on the situation. Special footings may be required to carry the weight, particularly if the tank is close to a load-bearing wall. It is possible to design the tank into a corner of a basement wall if the foundation and wall are designed to carry the added load. This technique is more amenable to new construction than to retrofitting.

The weight of concrete is an advantage in underground storage containers since it can help prevent the tank from being buoyed out of the ground by high groundwater.

Do not install precast concrete tanks in locations where major building modifications would be required to install or replace them. Cast-in-place tanks can be installed in some locations that do not permit installation of precast tanks.

5.2.1.4 Wooden or Multicomponent Tanks with Plastic Liners

Wooden multicomponent tanks can be either custom designed or purchased as kits. Their main advantages are low cost and easy indoor installation. The disadvantages are that inexpensive plastic liners have temperature limitations and the tanks are usually intended for indoor locations only.

One available kit makes a vinyl-lined, 2000-gallon (7600-liter) cylindrical tank.[1] According to the manufacturer's instructions, the tank, made of 3/8-inch CDX plywood and reinforced with steel bands, can be installed with simple hand tools. The kit includes insulation for the bottom, sides, and cover, as well as a 1-inch PVC compression fitting. The maximum allowable temperature inside the tank is 160°F (71°F).

Another type of kit uses lock-together panels of 4-inch-thick (10-centimeter-thick) urethane foam sandwiched between steel facings.[2] The plastic liner is rated for 180°F (82°C) continuous service and an aluminum roof is available for outdoor containers. Sizes from 500 to 2000 gallons (1900 to 7600 liters) are available.

Liners should be about 1 to 3 percent larger than the inside of the container to avoid the possibility of stressing the seams. With some types of plastic, it may be difficult to fabricate corners; but they can be folded from a flat sheet, if necessary. When installing the liner, you must be careful to remove all sharp edges, burrs, splinters, and debris that might puncture it. Avoid working inside the tank with the liner in place; if you must work there, remove shoes, belt buckles, tools, and other objects that could puncture the liner. Properties of several plastics are listed below.

Polyvinyl Chloride (PVC)

PVC is one of the least expensive and easiest liner materials to work with, and there is considerable experience with it in solar systems, industrial hot water processes, swimming pools, and water beds. Seams can be made by dielectric sealing, a more reliable process than heat sealing or cementing. Repairs can be made with patches and adhesives.

Specify a material thickness in the range of 30 to 60 mils (0.75 to 1.50 millimeters), and make liners slightly oversized to avoid stressing the seams. Maximum tank temperature should be limited to 160°F (71°C), although a few special compositions can tolerate a water temperature of 180°F (82°C). Lifetime varies from 6 to 15 years, with about 8 years being typical. Failure is finally caused by leaching of plasticizers. When this occurs, the plastic becomes brittle and cracks appear, usually at the corners.

Ethylene Propylene Diene (EPDM)

EPDM is a rubber-like material that can withstand boiling water. It is more expensive than PVC. Joints, especially corners, are more difficult to fabricate, since dielectric sealing cannot be used. EPDM can be patched with adhesives, but patching is more difficult than for PVC. A thickness in the 30 to 60 mil (0.75 to 1.50 millimeter) range is preferred.

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[1] Acorn Structures, Inc., Concord, Massachusetts 01742.

[2] Bally Case and Cooler, Inc., Bally, Pennsylvania 19503.

Butyl Rubber

Butyl rubber is a reasonably durable material that is less expensive than EPDM but more expensive than PVC. Although seams can be vulcanized the process is more difficult than dielectric sealing. Butyl rubber can be patched with adhesives. Its sheet form is more durable than its spray-on form.

Polyethylene and Polypropylene

Polyethylene and polypropylene are inexpensive in flat sheets, but joints are difficult to fabricate, and patching is difficult at best. Adhesives perform poorly on polyethylene and polypropylene because the plastics are unaffected by most ordinary solvents. The materials tend to be stiff if their thickness is more than 30 mils (0.75 millimeters). Polypropylene is capable of withstanding higher temperatures than polyethylene, but since there is little experience with using these materials in solar systems they are not a first choice.

Chlorosulfonated Polyethylene

Chlorosulfonated polyethylene is usually laminated with a scrim (a mesh fabric) of another material to give it dimensional stability. Flat sheets are generally available, but fabricated corners may be difficult to obtain. Some users have reported separation of the material from the scrim.

Chlorinated Polyvinyl Chloride (CPVC)

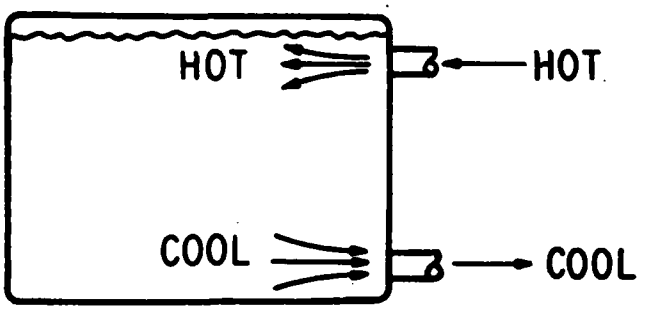
Like PVC, CPVC is normally rigid, but it can be made flexible by adding plasticizers. The plasticizers are subject to leaching, as they are with PVC; therefore, CPVC has no advantages over PVC.

Summary of Liners

For most solar heating systems there is no need for temperatures above 160°F. Therefore, a polyvinyl chloride liner is the best choice despite the need to replace it at least once during the life of the system. There are considerable differences in the reliability of PVC liner fabricators. As a minimum the fabricator should (1) use only dielectrically sealed seams and (2) have a proven success record fabricating liners for industrial hot water processes and solar system tanks. If the tank will be used above 160°F, an EPDM liner is the best choice. Consult with the liner supplier before designing the tank since difficulties in fabricating corners may dictate the shape of the tank.

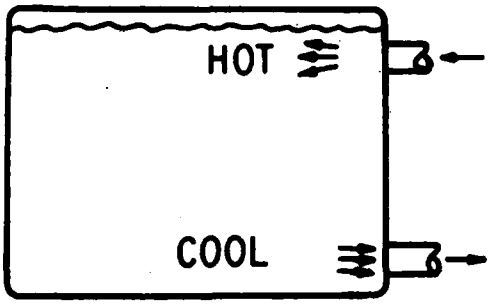
5.2.2 Thermally Stratified Tanks

As mentioned in Chapter 4, a storage system with thermal stratification can be as much as 10% more efficient than a thermally mixed system. Stratification in storage tanks can be fostered by the methods shown in Figures 5-2 a-d. Using a horizontal inlet and outlet and low velocity flows

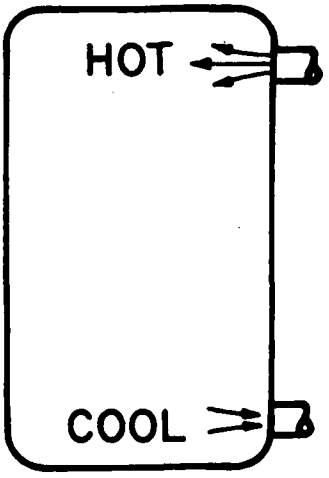


a. Horizontal Flow

Hot water should enter or leave at the top; cold water should enter or leave at the bottom.

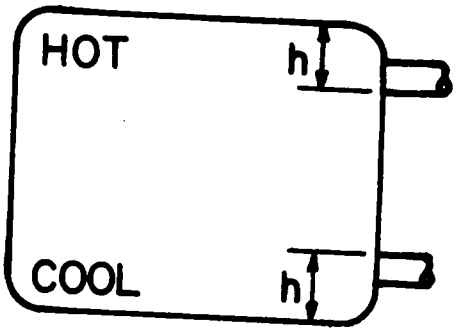


b. Low-Velocity Flow



c. A tall tank is easier to stratify than a short one.

Figure 5-2. Methods of Promoting Thermal Stratification in Water Tanks



d. Minimize the distances from the top of the tank to the bottom of the upper connection and from the top of the lower connection to the bottom of the tank.

Figure 5-2. (Continued)

(Figures 5-2, a and b) is so easy to do that it should be standard practice in all liquid-based systems. A tank can have a height as much as four times its diameter (Figure 5-2 c) without increasing thermal losses very much. Avoid the design mistakes illustrated in Figures 5-3 a-d.

The most important parameter characterizing tank stratification is the Richardson number, a dimensionless number defined as

$$R_i = \frac{g\beta L \Delta T}{u^2} \qquad (5\text{-}1)$$

where g is the gravitational constant, = 32.2 ft/sec^2,

β = volumetric thermal expansion coefficient in 1/°F,

L = height of tank in ft,

ΔT = temperature difference from top to bottom of tank in °F,

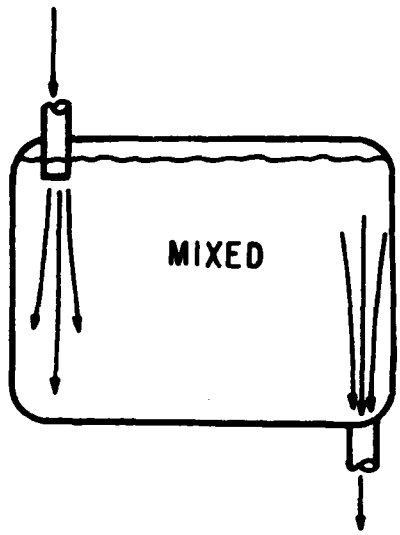
u = fluid inlet velocity in ft/sec.

The tank contents will tend to stratify for Richardson numbers larger than the critical value of 0.25, but will mix for smaller Richardson numbers*. From equation 5-1 it is apparent that a tall tank with a large temperature differential and a low flow velocity (Figures 5-2, b and c) will have the greatest tendency to stratify. In practice it is easy to design tanks with a Richardson number greater than 3, insuring good stratification.

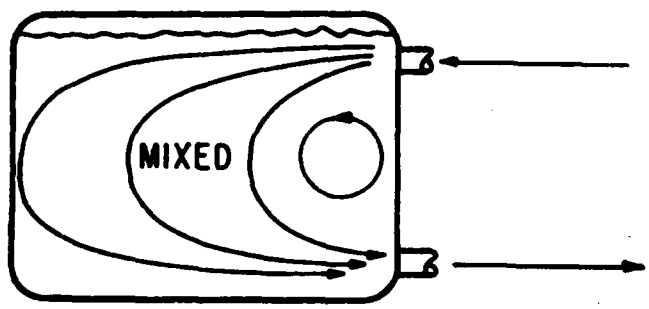
Experiments at Argonne National Laboratory have shown that the tank can be divided into three regions: (1) a mixing zone near the inlet, (2) a nonmixing zone in the center of the tank, and (3) a mixing zone near the outlet. The nonmixing zone contains the thermocline, which is the zone separating hot and cold fluid. Because the spread of the thermocline takes place slowly, overall performance of the tank is governed by the size of the mixing zones. The mixing zones should be made small by the methods illustrated in Figures 5-2, c and d.

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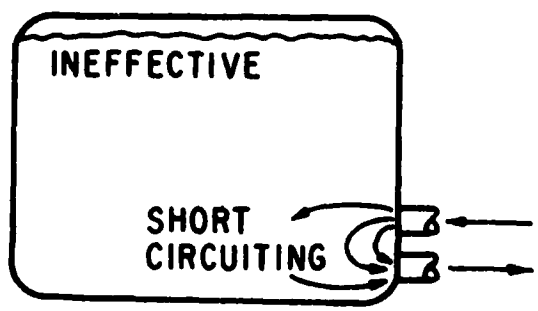
*Reference: J. S. Turner, Buoyancy Effects in Fluids, 313-337, Cambridge University Press, 1973.



a. Vertical flow disrupts thermal stratification.

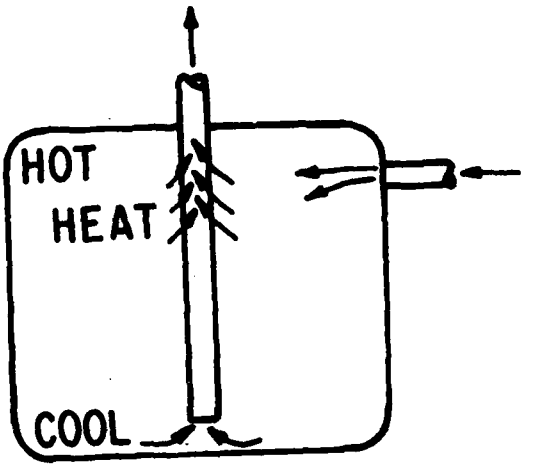


b. High-velocity flow causes turbulence, mixing, and short-circuiting.



c. Inlet and outlet close together causes short-circuiting. Parts of storage medium far from inlet and outlet may be ineffective.

Figure 5-3. Water Tank Design Mistakes



d. Cool water flowing through a dip tube cools the hot water at the top of the tank.

Figure 5-3. (Continued)

Modifications for Solar Systems

Maintenance of stratification in tanks requires modification of the coventional solar collection strategy. The following three rules are necessary to achieve the benefits of stratification:

(1) Allow no more than one tank volume of water per day to pass through the collector or collector heat exchanger.

(2) Vary the flow rate through the collector or collector heat exchanger so that its outlet temperature remains constant.

(3) Reduce the outlet temperature of the collector or collector heat exchanger to the minimum required end-use temperature plus a few degrees to allow for thermal losses and heat exchange processes. This third rule is made possible by the first two rules and results in higher system efficiency.

The strategy specified by the three rules can be accomplished by using a centrifugal pump and installing a thermostatic valve similar to an automobile thermostat at the outlet of the collector*. Because water returned from the collector or collector heat exchanger is always at least as hot as the water at the top of the tank, a high Richardson number and good stratification are maintained.

The conventional solar collection strategy violates rules (1) and (2) by having a constant flowrate that allows as many as five tank volumes per day to pass through the collector or collector heat exchanger**. If the day begins with a well-stratified tank, the pump will draw cold water from the bottom of the tank at startup. Because of the high flowrate and the low solar intensity in the early morning hours, the water will be returned to the tank only a few degrees warmer than the bottom of the tank, and much colder than the water at the top of the tank. The returning water sinks to the bottom, mixing somewhat with the water at the top of the tank, but mostly warming the tank from the bottom upward. As the temperature at the bottom rises, the Richardson number for the tank decreases. When the critical Richardson number is reached, typically about midmorning, the tank suddenly mixes. The very low Richardson number of the fully mixed tank ensures that the tank will remain mixed until long after the pump is turned off. Even if stratification could be maintained under a conventional strategy, the maximum stratification (measured in degrees of temperature difference) is only about 1/5 of the stratification that can be achieved with rule 1.

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*Footnote: One such valve is made by H & H Precision Products, In-Sink-Erator Division, Emerson Electric Co., 25 Canfield Road, Cedar Grove, NJ 07009. The purpose of this footnote is not to be construed as endorsement of any particular company's product.

**Reference: W. A. Beckman, S. A. Klein, and J. A. Duffie, Solar Heating Design by the f-Chart Method, 4,5 John Wiley & Sons, New York, 1977.

5.2.3 Pumps

Pumps are used to circulate heat transfer fluids in all liquid-based solar systems except thermosiphon systems. This section discusses the types of pumps available and how to select a pump for a specific application.

Two types of pumps are readily available on the market. One is the positive displacement pump, characterized by a low flow rate and high head. (Head, a term used throughout this section, is another word for pressure, which can be measured in the height of liquid that the pressure can support in a vertical pipe.) The positive displacement pump is rarely used in solar systems, which do not need a high head. If you use a positive displacement pump you will need a relief valve on the output side to prevent excessive pressure from mounting if a pipe becomes plugged.

The other readily available type is the centrifugal pump, characterized by a low head and high flow rate capacity, used in most solar systems and available with a wide variety of flow rates. It can be sealed against leaks in three ways: with adjustable packing, with a mechanical seal, or with a magnetic coupling.

The adjustable-packing seal, shown in Figure 5-4, is the least desirable sealing system because it requires frequent inspection and adjustment and is prone to leak. This system should be avoided where access to the pump is difficult, where leakage from the seal could cause system failure, where leakage could create a hazard to people, or where antifreeze fluids are used. Antifreeze solutions, especially silicone oils and, to a lesser extent, glycol solutions, have an affinity for leaks.

The mechanical (or face) seal, shown in Figure 5-5, consists of two carefully polished surfaces pressed together by a spring. One of the surfaces is part of the pump shaft, and the other is sealed against the pump housing with an O ring. A minute amount of leakage, so small that it evaporates before becoming visible, lubricates the polished surfaces.

Considerable experience with hydronic heating systems has shown that mechanical seals can last the lifetime of the system without requiring adjustments. The system must be kept clean, for grit in the water can easily scratch the polished surfaces. Chromate-type corrosion inhibitors have caused failures when the leakage evaporated and deposited hard chromate crystals between the polished surfaces. High temperatures and pressures also cause premature failure. Some antifreeze fluids, such as silicone oils, tend to leak excessively with mechanical seals.

A pump that uses a magnetic coupling, shown in Figure 5-6, has no troublesome rotating seals. Instead, the rotor of the electric motor and its bearings are placed entirely inside an extension of the pump housing. The stator of the electric motor fits outside the pump housing and drives the

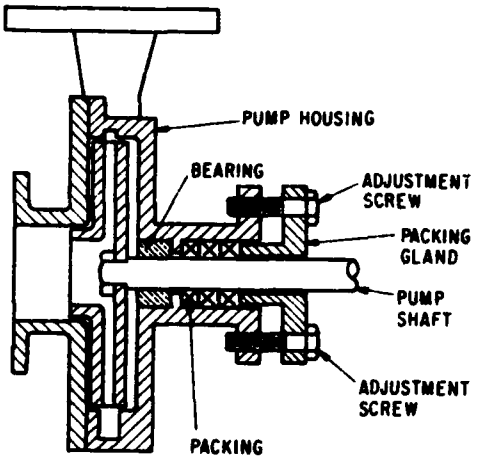


Figure 5-4. Cross-section through an Adjustable-packing Seal

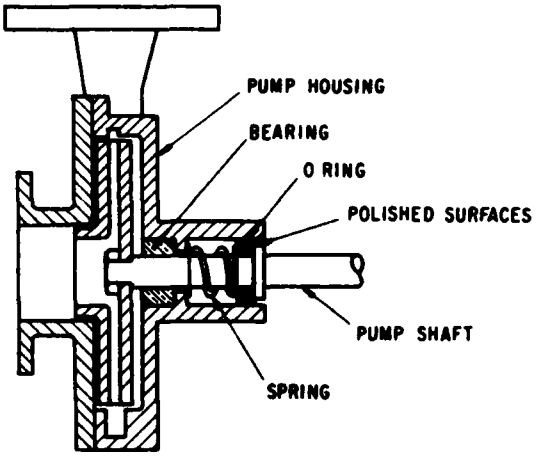


Figure 5-5. Cross-section through a Mechanical Seal

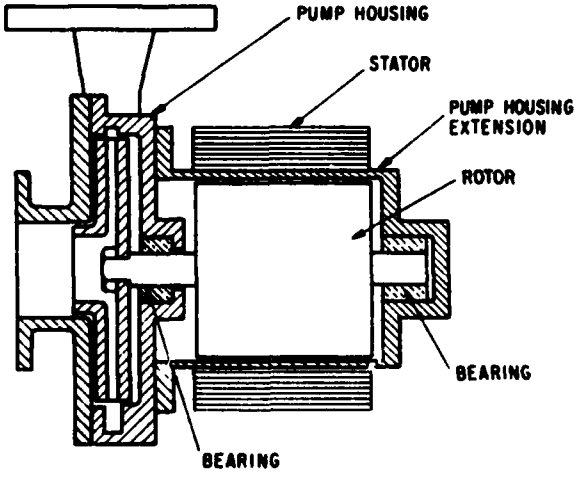


Figure 5-6. Cross-section through a Magnetically Coupled Pump

rotor with a rotating magnetic field. In some designs a set of rotating magnets replaces the stator, and an external electric motor turns a shaft that rotates the magnets. Magnetically-coupled pumps can be expected to last the lifetime of the system.

Figure 5-7 shows typical performance data for a large (140 gallons per minute or 8.8 liters per second) pump. Performance data for each pump model are available from manufacturers and distributors. Although complete data such as that in Figure 5-7 is useful to the designer, data for pumps of the size used in residential systems (less than 10 gallons per minute or 0.63 liters per second) often lack information on net positive suction head (NPSH) requirements and power consumption. Figure 5-7 will be used to illustrate how the complete set of curves, if available, is used.

The designer must first consider the upper set of curves, which gives the head versus flow. Figure 5-7 shows six head versus flow-rate curves corresponding to six impeller diameters. Manufacturers of large pumps can supply several impeller sizes to fit each pump model in order to match the pump characteristics to the system characteristics. Usually only one size of impeller is available for small pumps. Also shown on the upper set of curves are the pump efficiency and the required motor horsepower.

To use the upper set of curves in selecting a pump, you must first determine the flow rate and the head loss (pressure drop) at which the system must operate. Data on flow rate and head loss through the collectors is available from the collector manufacturer. For a heat exchanger loop, the flow rate, q, in cubic feet per hour (cubic meters per second) is:

$$q = \frac{Q'}{\rho \, C_p \, \Delta T} \qquad (5-2)$$

where:

- Q' is the heating rate in Btu per hour (watts).
- C_p is the heat capacity of the heat transfer fluid in Btu per pound per degree Fahrenheit (joules per kilogram per degree Celsius).
- ρ is the fluid density in lb/ft^3 (kg/m^3).
- ΔT is the temperature change of the heat transfer fluid in degrees Fahrenheit (degrees Celsius).

Divide q in cubic feet per hour by 8.02 to obtain q in gallons per minute. (Multiply q in cubic meters per second by 1000 to obtain q in liters per second.) Data on head loss through heat exchangers are available from the manufacturer.

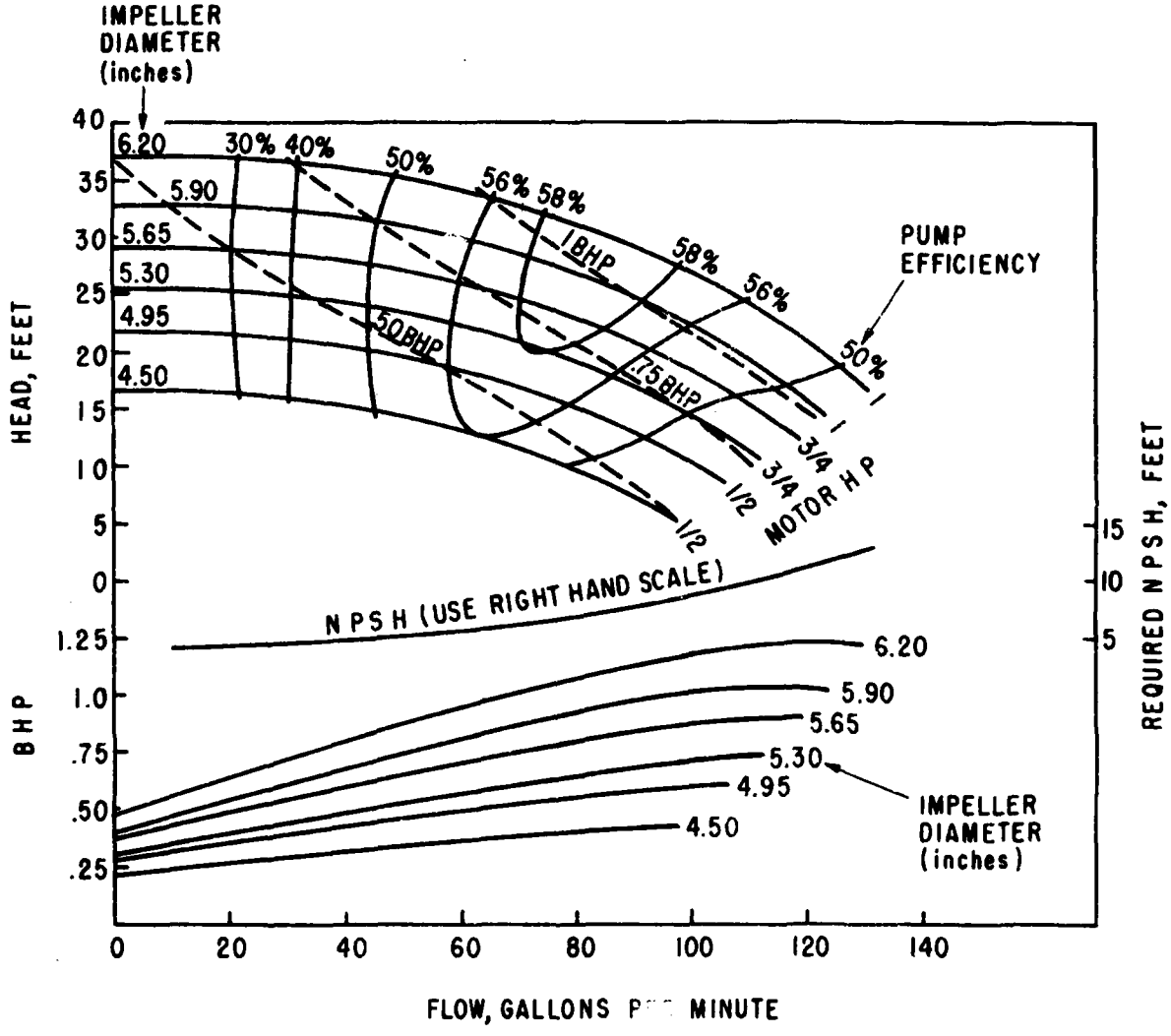


Figure 5-7. Pump Performance Data

Based on a drawing supplied by Taco, Inc., Cranston, Rhode Island.

You will also need to estimate the head loss caused by friction in pipes,[3] and if the system is the open-drop draindown type,[4] you will need to determine the difference in height between the top of the collector and the water level in the tank. The required pumping head, H_p, is equal to the sum of the head losses around the loop.

$$H_p = Z + H_f + H_x + H_c \qquad (5\text{-}3)$$

where:

Z is the difference in height between the top of the collector and the liquid level in the tank (for open-drop draindown systems only).

H_f is the head loss caused by friction in the pipe.

H_x is the head loss in the heat exchanger.

H_c is the head loss in the collector.

All terms in Equation 5-3 are expressed in feet (meters).

By plotting the required flow rate and pumping head on performance curves of several different pump models, select the pump that meets or exceeds the pumping head requirements at the required flow rate.

The second consideration in selecting a pump is that it should be capable of filling the initially empty system. That is, the pump head at zero flow rate should exceed the difference in height between the top of the loop and the level of the liquid in the reservoir. By plotting this height difference on the vertical axis of the pump performance curves, determine if a pump will meet this requirement.

Pump Power

The lower set of curves in Figure 5-7 gives the power consumed by the pump versus flow rate and impeller diameter. After selecting a pump, an impeller diameter, and a flow rate, you can use the lower set of curves to determine the size of the motor required by the pump. In a well-designed system, the total pumping power for the system should not exceed 1-1/2 percent

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[3]Details of pipe friction calculations are available from several sources, including the ASHRAE Handbook of Fundamentals, Chapter 26, "Pipe Sizing"; Flow of Fluids by the Crane Company; and textbooks such as Fluid Mechanics by Victor Streeter and Benjamin Wylie.

[4]An open-drop draindown system is vented at the highest point in the system by having a vent valve at the highest point or by having the collector to storage return pipe larger than the pump to collector pipe.

of the solar power being collected. (1 HP = 2546 Btu per hour; 1 watt = 3.41 Btu per hour.)

Net Positive Suction Head

The middle curve in Figure 5-7 gives the pump's net positive suction head (NPSH) requirement. Net positive suction head is the absolute head at the pump inlet minus the vapor head of the liquid being pumped. If the available NPSH does not exceed the pump's NPSH requirement, cavitation can destroy the pump in a short time. Most systems have sufficient NPSH, but all systems must be checked for this requirement.

The available NPSH for your system can be calculated from Equation 5-4.

$$NPSH = Z - H_f - H_v + H_\alpha + H_p \qquad (5\text{-}4)$$

where:

- Z is the difference in height in feet (meters) between the liquid level in the reservoir and the pump inlet. If the pump inlet is above the level of the reservoir, Z will be a negative number.
- H_f is the head loss in feet (meters) caused by friction in the pipe leading to the pump inlet.
- H_v is the vapor pressure head of the liquid in the reservoir in feet (meters) at the maximum allowable temperature. The relationship between H_v and temperature for water is shown in Figure 5-8. Multiply pressure in psi by 0.43 to obtain feet of water. (Multiply pressure in Pa by 1.02×10^{-4} to obtain meters of water.)
- H_α is the atmospheric pressure in feet (meters) of water. Figure 5-9 gives the relationship between H_a and altitude.
- H_p is the minimum pressure in the tank in feet (meters) of water if the system is not vented to the atmosphere.

Pump Materials

Pumps are usually made from iron, bronze, or stainless steel. Iron will quickly rust in an open system, but it can be used in a closed system with corrosion inhibitors. Both bronze and stainless steel pumps can give long service life.

5.2.4 Other Components

Hand-Operated Valves

Solar systems generally use the same types of hand-operated valves commonly found in residential water systems. There are two common types of valves, the globe valve and the gate valve. The globe valve (Figure 5-10)

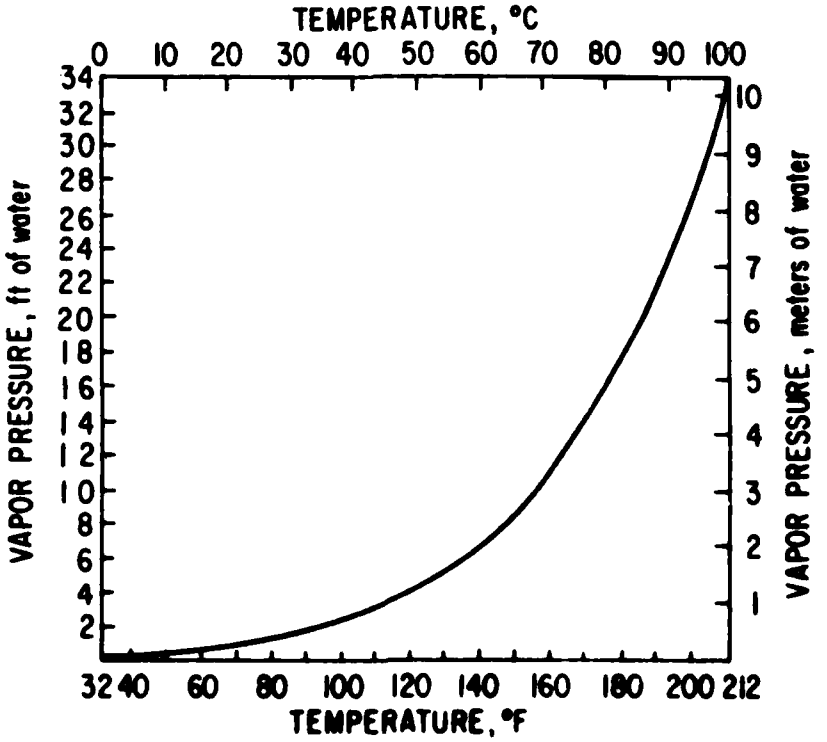


Figure 5-8. Vapor Pressure of Water versus Temperature

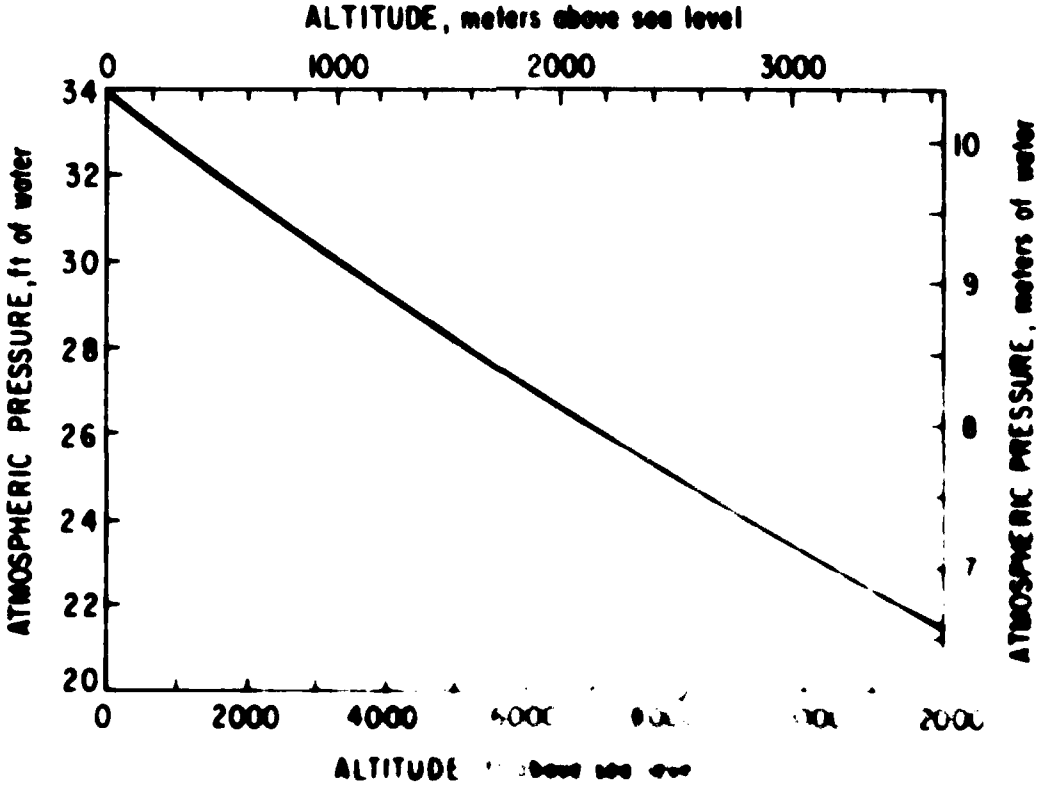


Figure 5-9. Atmospheric Pressure versus Altitude

controls the amount of flow. Globe valves do not permit complete draining of lines when they are placed in a horizontal position, and they impose more resistance to flow than do gate valves. Gate valves (Figure 5-11) are not suitable for controlling the amount of flow but are used to open or close a line. When open, gate valves have only small resistance to flow. They can be used as isolation valves. Most valves used in solar systems are of this type.

Check Valves

Check valves allow fluid to flow in one direction only. Most antifreeze systems require at least one check valve installed between the pump outlet and the collector inlet to prevent thermosiphoning on cold nights. Failure to install the check valve has caused heat exchanger failure because of freezing in several installations.

Two other types of check-valve problems to avoid are installing the check valve backwards and selecting the wrong type of check valve. There are two basic types of check valves, those that are closed by gravity and those that are closed by a spring. The gravity-operated swing-check and and lift-check valves must be installed in the proper orientation with respect to gravity. If they are installed upside down or at 90° to their intended orientation, they will not block a backward flow. The gravity-operated check valves usually have hard seats that allow a small amount of leakage. In some cases, there is enough leakage to cause a thermosiphoning problem.

Spring-operated check valves can be installed in any position with respect to gravity. Resilient seats combined with the spring-operated closure assure complete stoppage of backward flow. Spring-operated check valves are more expensive and have a higher pressure drop than gravity-operated check valves but these disadvantages are outweighed by their greater reliability.

Temperature and Pressure Relief Valves

Any closed subsystem must contain a temperature and pressure (T&P) relief valve to prevent damage to the system from excessive temperature or pressure. T&P valves for domestic hot water tanks usually have 210°F (99°C) temperature and 150 psi (1 MPa) pressure settings. Temperature and pressure settings for other types of tanks will differ from settings for domestic hot water tanks. All T&P valves must meet ASME Boiler and Pressure Vessel Code requirements.

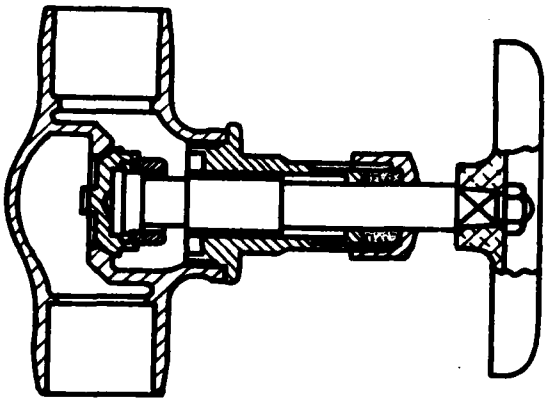


Figure 5-10. Typical Section through a Globe Valve

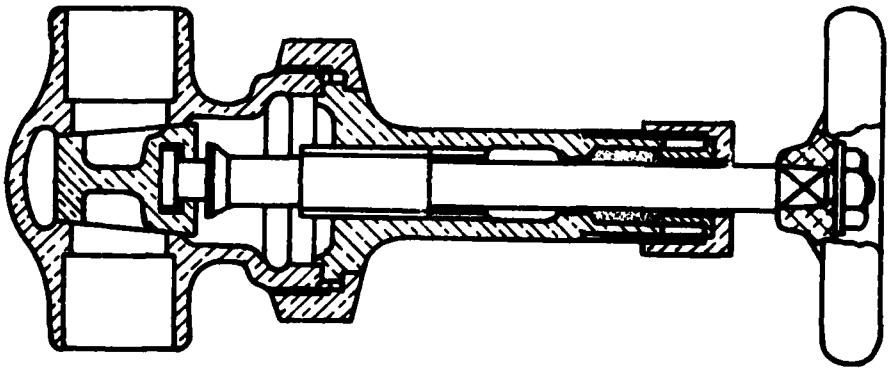


Figure 5-11. Typical Section through a Gate Valve

Expansion Tanks

The liquids in a solar system expand when they are heated. The pipes and other components also expand, but not enough to contain the increased volume of the liquid in a closed system. To allow room for this excess volume in closed systems, an air space must be provided. The air space may be part of the main tank in the system or a separate expansion tank.

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.

- Obtain data on how the fluid's density changes with changes in temperature from the distributor or manufacturer of the fluid.

- 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.

Vacuum Relief Valves

As a solar system cools, the volume of the liquid in the system decreases. Vacuum relief valves must be installed in closed systems to prevent vacuum damage to tanks. For example, if a tank with a 4-foot (1.2-meter) diameter cools until the pressure in the tank is 1 psi (7 kPa) below atmospheric pressure, the force on the top of the tank will be more than 1800 pounds (8000 newtons).

Temperature Sensors

Temperature sensors in storage tanks serve three purposes:

- To tell the controller when to turn the collector pump on. This temperature sensor should be installed in the tank near the bottom connection to the tank. When the collector temperature is warmer than the temperature at the bottom of the tank, the system controller turns on the collector pump. For a well stratified tank, another temperature sensor would be placed at the top of the tank. The pump would not be turned on until the collector output temperature was as high as the temperature at the top of the tank.
- To tell the controller when heat is available for heating. This temperature sensor should be installed in the tank near the top connection to the tank. When the tank is warm enough to supply heat to the load, the controller will take heat from storage to satisfy the load. Otherwise, the controller will turn on the auxiliary heater.
- To limit the temperature in the tank to a safe maximum. Some controllers use the same temperature sensor to indicate when heat is available for the load and to limit the tank temperature, while other controllers use separate temperature sensors for these functions. Unfortunately, many of the controllers that are commercially available do not have provisions for limiting tank temperatures. Limiting the tank temperature is essential if the tank has a rubber or plastic liner or if it is made of fiberglass-reinforced plastic. Even metal tanks can benefit from a temperature limit, for corrosion rates approximately double with each 20°F (10°C) increase in tank temperature.

Thermistors and silicon transistors are the easiest to use, most reliable, and most commonly used types of temperature sensors, although thermocouples, bimetallic elements, and liquid or vapor expansion units have been used. Because its signal is linear with respect to temperature, the silicon transistor is easier to calibrate than the thermistor. The type of temperature sensor selected is less important than its compatibility with the control unit.

The electrical leads of the temperature sensor must be protected from immersion in water. Some temperature sensors are equipped with a protective sheath, while others must be installed in a temperature sensor well (a capped pipe immersed in the water). If a temperature sensor well is used, it should have provisions for good thermal contact between the temperature sensor and the metal.

If the tank is metal, the temperature sensor can be attached to the outside of the tank provided that certain cautions are observed.

- Cover the temperature sensor with insulation so that it reads the tank temperature and not the ambient temperature.
- Clamp or bolt the temperature sensor to the tank so that they are in firm contact. Taping or cementing the sensor to the tank is not adequate.
- Some types of sensors must be electrically insulated from the tank; otherwise, a short circuit can cause false readings.

In addition to the temperature sensors, thermometers should be installed to measure the water temperature at the upper and lower connections to the tank. These thermometers will help the installers start up and adjust the system. After the system is in operation, the thermometers can help detect malfunctions. For example, a failure of the collector circuit would be indicated by an abnormally low tank temperature in the afternoon of a sunny day.

Auxiliary Heating System

Solar space heating systems must have auxiliary heaters that are able to supply 100 percent of the heating load. Liquid-based systems are often combined with forced air heating systems, requiring a liquid-to-air heat exchanger in a forced-air duct to transfer heat to the building air. The heat exchanger can be installed either upstream of the auxiliary heater as shown in Figure 5-12a or downstream of the auxiliary heater as shown in Figure 5-12b.

The configuration shown in Figure 5-12a has a minimum operating temperature of about 70 to 75°F (21 to 24°C). The minimum operating temperature is a few degrees above the return air temperature because of the temperature drop caused by the heat exchanger. Studies have shown that people find 70°F (21°C) circulating air chilling. To ensure that the air will circulate at a comfortable temperature, the auxiliary heater is turned on whenever solar energy is unable to maintain a duct temperature of about 95°F (35°C) or more. While the auxiliary heater is operating, solar energy preheats the air entering the auxiliary heater. This system is usually used with an electric heater. Because the solar-heated air must pass through the auxiliary furnace's heat exchanger, some solar heat may be lost up the flue of a gas or oil furnace. If the auxiliary furnace has an automatic flue damper to prevent this loss, the solar preheat arrangement is feasible with a gas or oil furnace.

The configuration shown in Figure 5-12b is usually used with gas or oil auxiliary heaters without flue dampers or for situations where there is no room to install the heat exchanger between the blower and the heater. As in the previous configuration, the auxiliary heater is turned on whenever solar energy cannot maintain a duct temperature of about 95°F (35°C) or more. In this configuration the pump that circulates water from storage to the heat exchanger must be turned off when the auxiliary heater is turned on so that the auxiliary energy heats the building instead of recharging the storage unit.

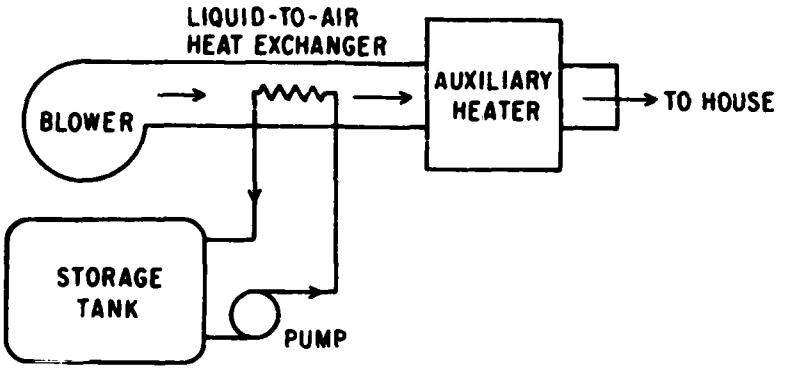


Figure 5-12a. Liquid-to-Air Heat Exchanger Installed Upstream of Auxiliary Heater

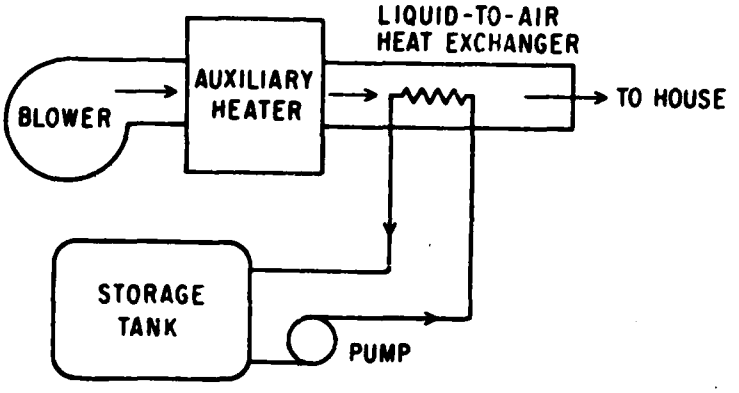


Figure 5-12b. Liquid-to-Air Heat Exchanger Installed Downstream of Auxiliary Heater

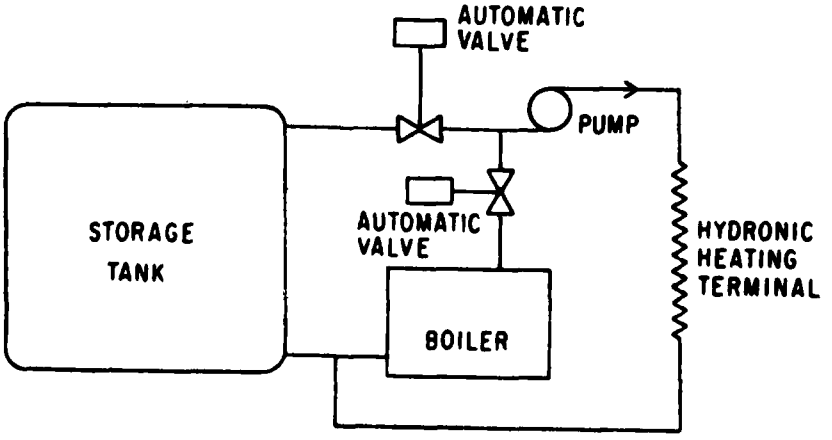


Figure 5-12c. Combined Solar and Hydronic Heating Systems

The solar space heating system can also be combined with hydronic heating systems (Figure 5-12c) using baseboard heaters, fancoil units, ceiling panels, or floor panels. Each type of hydronic system has a minimum operating temperature that varies with the outdoor temperature. If the temperature in storage is less than the minimum operating temperature of the system, the automatic valves must shut off the storage loop, and the boiler must be turned on.

5.2.5 Tank Installation

Proper installation of storage tanks is essential. The foundation under the tank must be strong enough to support the weight of the tank and the water that the tank will hold. Water weighs about 8.3 pounds per gallon (1 kilogram per liter), so the water in a 1000-gallon (3800-liter) tank will weigh 8300 pounds (3800 kilograms) -- more than four tons. If the foundation is not built to hold this weight, the tank may settle and cause leaks in the connected piping or in the tank itself. Have a structural engineer review the foundation design for any storage tank. Local building codes will specify the type of footings required.

Vertical steel tanks that are not buried need a concrete ring-wall foundation. Horizontal steel tanks above ground should be supported on concrete saddles with appropriate foundations. The tank manufacturer will indicate the tank's weight to determine how much support it needs.

Fiberglass tanks are designed to be installed with full bottom support. If an aboveground tank is mounted on a concrete pad, the concrete must be smooth and have enough reinforcement to support the weight of the full tank. Underground tanks are usually supported by the backfill. In both cases, the manufacturer's installation instructions must be followed carefully to avoid damaging the tank and to preserve the warranty.

For tanks installed underground, anchorage must be provided to prevent bouyant uplift when the tank is empty. The tank should be anchored to a concrete pad at least 6 inches (15 centimeters) 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 centimeters) deep and spread evenly over the concrete to separate it from the tank. Fiberglass or steel hold-down straps should be anchored 1 foot (30 centimeters) 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 centimeters) 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 centimeters) above the tank. Provide concrete pads for nozzles and manholes extending to grade. See Figure 5-13 for additional details.

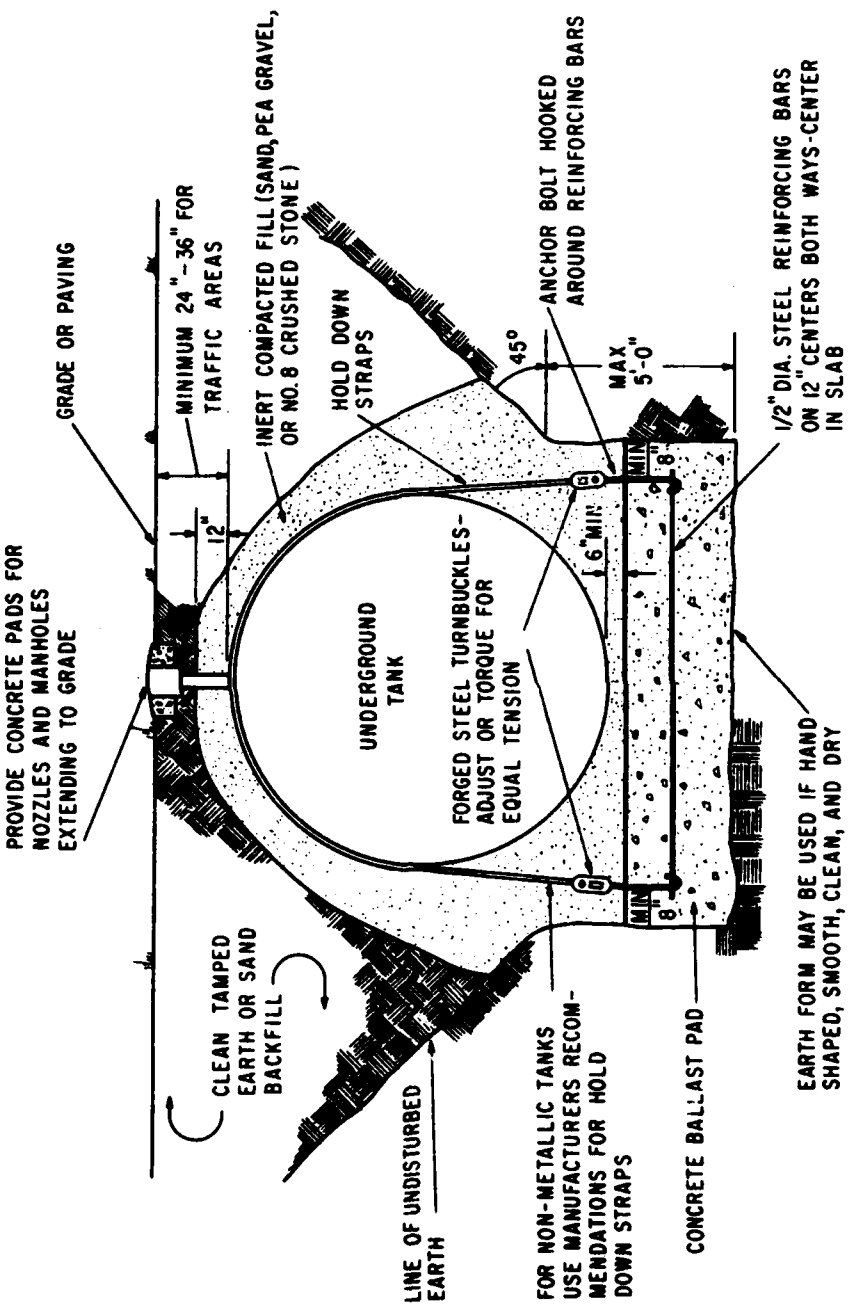


Figure 5-13. Installation of an Underground Tank

- Pumps or valves reversed.
- Inlet and outlet connections to tanks, heat exchangers, or collectors reversed.
- Normally open valves installed in place of normally closed valves, or vice versa.
- Lines in drain-down systems improperly sloped.
- Check valves improperly oriented.

This is a good time to begin labeling the piping, although some parts must be labeled after the insulation is installed. The contents (air, water, ethylene glycol, and so on) should be tagged on each major line, along with the notation for liquids "potable" or "nonpotable." Potable water is fit for human consumption. Toxic substances, such as ethylene glycol and many corrosion inhibitors, are, of course, nonpotable; however, a substance need not be toxic to be nonpotable. Water that resides in a storage tank for several weeks is nonpotable, even if it has no toxic additives. A color code can be used for labeling (see the ASHRAE Handbook of Fundamentals).

Safe fluids should be labeled with green, white, gray, black, or aluminum; dangerous fluids with orange or yellow. Pipes should be labeled with flow direction.

Valves, pumps, heat exchangers, and so on should be tagged to correspond to the identification numbers on the system drawings, and flow directions should be marked on the parts. Automatically controlled two-way valves should be labeled "normally open" or "normally closed." The legs of threeway valves should be labeled "common," "normally open," and "normally closed." Labeling components in this way will often disclose installation errors.

Before installing insulation and later before backfilling an underground storage tank, test the entire system for leaks. If the system can be pressurized, the best way to test for leaks is:

> Close the vent valves if necessary. Attach an air compressor to the system through a valve and fill the system to the normal operating pressure of the weakest component. Turn off the air compressor and close the valve. If the air pressure drops more than 20 percent in twenty-four hours, the system leaks. Test all pipe joints and packings with soapy water. Bubbles will appear at the leak. Do not forget to open the vent valves if you have had to close them before testing.
>
> THIS TEST IS ONLY FOR FINDING LEAKS IN THE SYSTEM; IT IS NOT A REPLACEMENT FOR THE ASME PRESSURE VESSEL CERTIFICATION TEST. PERFORM ASME PRESSURE VESSEL CERTIFICATION TESTS OF COMPONENTS BEFORE YOU PERFORM THIS LEAK TEST TO BE CERTAIN THAT ALL COMPONENTS CAN WITHSTAND THE TEST PRESSURE.

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.

Design for Maintainability

Maintenance must be considered as the system is being designed. Many solar systems fail or perform poorly because they cannot be properly maintained. If a storage tank cannot be easily replaced, it should last the lifetime of the building. Before installing any components the designer should consider how each component will be repaired or replaced and provide for working room around the components. All tanks and heat exchangers must have a means of being drained for inspection and repairs.

Simple systems are usually easier to maintain than complicated systems. The advantages of a simple system over a complicated system are:

- Initial cost is lower.
- Installation errors are less likely.
- There are fewer components to fail.
- Controls and operation are easier to understand.
- Defective components can be more easily found and replaced.

Answering three questions will clarify whether the system is too complicated or too simple.

- If a feature were deleted from the system, how much energy collection would be lost?
- If a feature were deleted from the system, would a mode of failure be introduced?
- If a feature were deleted from the system, would human safety be degraded?

A system analysis method (such as f-Chart, SOLCOST, DOE-1, or TRNSYS) can be used to estimate the extra amount of energy collection attributable to a particular feature. If the value of the extra energy collected over the life of the system is less than the cost of the feature, the feature cannot be justified economically. If, in addition, the answers to the second and third questions are "no," the feature should be deleted from the system.

Startup

Before insulation is applied to the pipes, the system must be carefully inspected and tested. Begin by checking all piping, valving, and wiring against the system drawings. Typical problems that might be encountered include:

If the system is not designed for pressure or if it has already been leak-tested with compressed air, fill it to the proper levels with the appropriate fluids. If the fluids are warm when the system is filled, the inspector will not be confused by moisture condensing on the tank and pipes. Inspect all pipe joints and packings for leaks and remove any trapped air pockets.

Trapped air pockets can be removed in several ways.

- Fill the system at its highest point.
- If the system is designed for pressure, pressurize it and open the air bleed valves at the high points.
- Turn the pumps on and allow the liquid to sweep the air from the system.

Check and refill the system to the proper levels.

Test the system in all operating modes to ensure that it functions as intended. Since systems vary greatly in their operating modes, only general guidelines can be given here. Most controller manufacturers make testing devices and publish data on how to use the testers. A set of jumper wires may be required to operate the system in its various modes.

CAUTION: DANGEROUS VOLTAGE MAY BE PRESENT AT CONTROLLER TERMINALS. Flows in pipes usually can be determined by feeling a temperature change and by observing temperature changes with the thermometers installed in the tank and collectors.

A special test is required for draindown systems. After operating the system in the collecting mode and then in the draindown mode, shut off the system and temporarily disconnect the piping, as shown in Figure 5-14a. Apply compressed air as shown in Figure 5-14b to blow trapped water, if any, into buckets. If more than a few drops of water fall into the buckets, the system does not drain properly and could be damaged by freezing. Do not operate the system until the cause of faulty draining has been corrected and the system has been retested. If the system passes the draindown test, reconnect the pipes.

Several final operations should be performed before the system is put into operation. Inspect the filters; if dirt or grit is found, or if the fluid is discolored, do not start the system until the cause has been found and corrected. Test the fluid pH and measure the antifreeze or corrosion inhibitor concentration. If te results are not within the manufacturer's specifications, do not continue until the cause has been found and corrected. Inspect and repair leaks, if any. Finally, install the insulation on tanks and piping and complete the labeling of components. The system is now ready for operation.

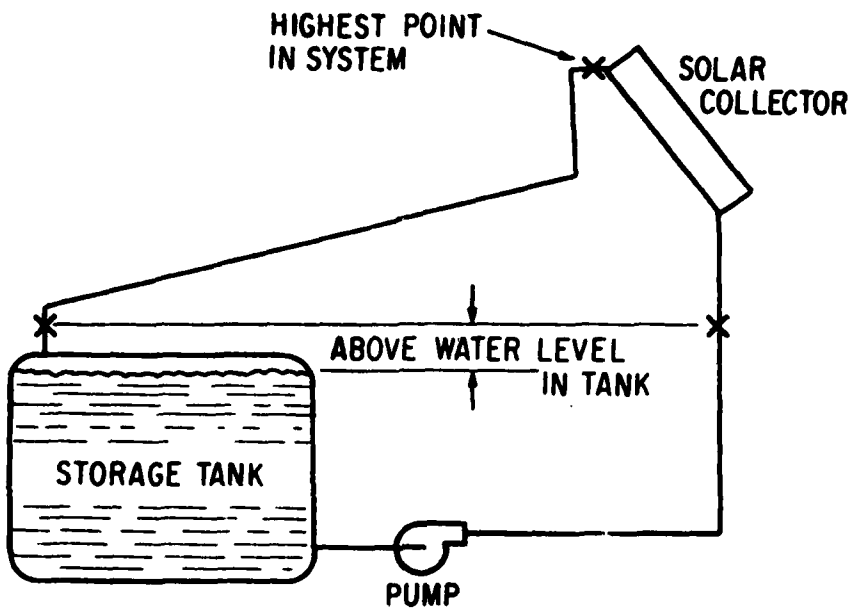


Figure 5-14a. Temporarily disconnect piping at points marked "X" for draindown test.

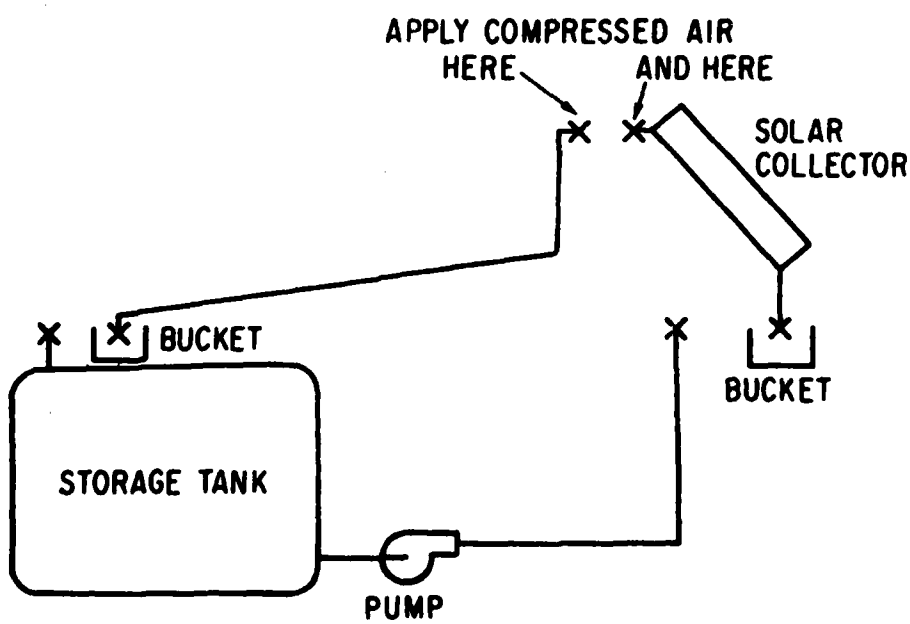


Figure 5-14b. Draindown Test

Periodic Inspection and Maintenance

The following tasks should be performed twice each year, when the system is switched between winter and summer modes.

- Inspect for leaks.
- Check fluid levels in tanks.
- Clean or replace fluid filters.
- Examine fluid samples for grit, sludge, dirt, or discoloration.
- Test the fluid pH.
- Measure the concentration of antifreeze or corrosion inhibitors.
- Check pumps, valves, sensors, and controllers for proper function.

At an interval (usually two to five years) specified by the tank manufacturer, the collector manufacturer, the antifreeze manufacturer, or the corrosion-inhibitor manufacturer, drain and replace the fluids in the system. (For some systems this is unnecessary.)

5.3 Costs

Factors that affect tank installation costs include the tank's size, whether it is being installed in a new building or has to be built into an existing one, location, temperature requirements, insulation requirements, and the materials used.

5.3.1 Size

Tank size is the most important factor affecting cost. Generally, the cost per gallon decreases as the size of the tank increases, as shown in Figure 5-15. Because system performance is not extremely sensitive to tank size (unless the tank is considerably undersized) the best approach is to select a standard size close to the optimum size.

5.3.2 New Building or Existing Building

Whether the tank is to be installed in a new or an existing building limits the choice of tank materials and location. It may be possible to design access to a steel, fiberglass, or precast concrete tank into a new building, but the lack of access in an existing building may require choosing an outdoor location or a different tank material.

Reinforcements to the foundation can be specified before a new building is built, but in an existing building part of the basement floor may have to be removed before a reinforced section of floor can be installed. The greater flexibility in choosing tank materials, tank location, and foundation reinforcement generally gives a solar system in a new building a cost advantage over a solar system in an existing building.

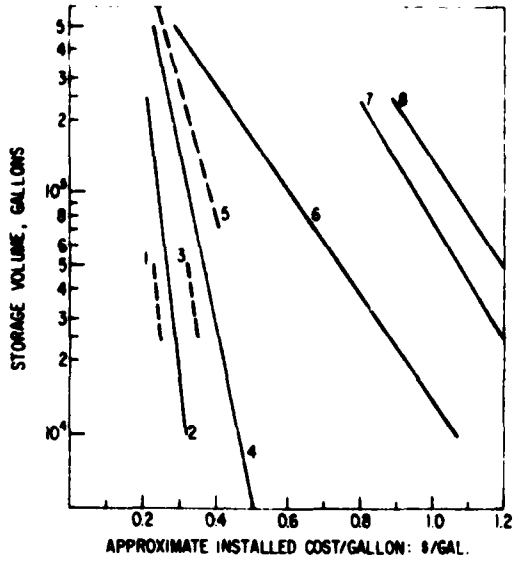


Figure 5-15. Relative Cost of Tanks in the Chicago Area, 1978

Source: B&A Engineers, Ltd. An Assessment of the Technoeconomic Feasibility of Seasonal Thermal Energy Storage Systems

KEY TO FIGURE

Tank Category	Tank Location	Tank Composition
1	Interior below grade tanks integrated with building	All weather wood foundation
2	Interior below grade tanks integrated with building	Built-up concrete masonry units
3	Exterior underground tanks attached to building	All weather wood foundation
4	Exterior underground tanks attached to building	Built-up concrete masonry units
5	Interior tank in building	Cast-in-place concrete
6	Tank-on-Grade	Steel tank
7	Underground tank	Steel
8	Underground tank	Fiberglass

5.3.3 Location

Tank location affects the special requirements for tanks. These requirements include waterproof insulation, extra-thick insulation, freeze protection, protection from groundwater (tie-down straps, exterior corrosion protection, provisions for drainage of groundwater, and so on), long-lifetime components, and limitations on materials that can be used. Each special requirement adds to the system's cost.

Underground tanks generally have the most special requirements. The insulation should be waterproof and should have extra thickness because of the possible presence of groundwater, which reduces its insulating value. Tie-down straps are required to prevent flotation of a partly filled tank. Provisions for draining groundwater and rainwater away from the tank and exterior corrosion protection for steel tanks should also be included. Because access to them is difficult, underground tanks should be designed for a long lifetime. Steel, fiberglass, and concrete can be used for underground tanks, but wood is not recommended because of its short life-time when in contact with earth.

Basement locations generally impose few special requirements on the storage tank. Weatherproof and extra-thick insulation, freeze protection, and protection from groundwater are not needed when the tank is indoors. Steel and fiberglass tanks are not recommended for existing buildings because they ordinarily will not fit through the doors. If a steel or fiberglass tank is to be installed in a new building it should last the lifetime of the building. Both wooden and cast-in-place concrete tanks are suitable for basement installation.

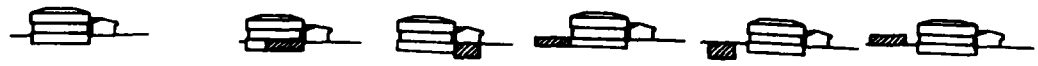
The requirements for tanks in basements also apply to tanks in crawl spaces. In addition, since most crawl spaces are unheated, extra insulation and a means of protecting the tank from freezing are needed.

Garages are excellent locations for steel or fiberglass tanks. The large door allows for easy installation or replacement of the complete tank. Since most garages are unheated, extra insulation and a means of protecting the tank from freezing are needed.

The requirements for outdoor, aboveground storage tanks are the same as for those in garages, except that the tank must also be protected from the weather. Some factory-insulated fiberglass tanks are adequately protected against weathering and do not need additional protection.

Examine Tables 5-2 a-c for summaries of the pros and cons of various locations. Although the locations were evaluated exclusively for seasonal thermal energy storage (STES) most of the major performance attributes remain

Table 5-2a. Selected Seasonal Thermal Energy Storage Location Options for Single-Family Residence



Description of Option		Original Design Non-STES	Partial Basement Storage	Storage Beneath Garage	Below-grade Attached	Under-ground detached	Free-standing detached
major performance attributes		initial cost	1	1	2	3	3
		solar/ electro-mech. interface	2	3	3	4	3
		off-the-shelf procurement	2	3	3	1	1
		durability/ service life	2	3	4	4	2
		maintenance frequency/ cost	2	3	3	2	1
		operating cost	2	3	3	4	3
		repair feasibility	3	2	1	2	1
		replacement feasibility	4	4	2	2	1
		overall rank	B	A	A	B	D

1 or A - excellent
5 or E - poor

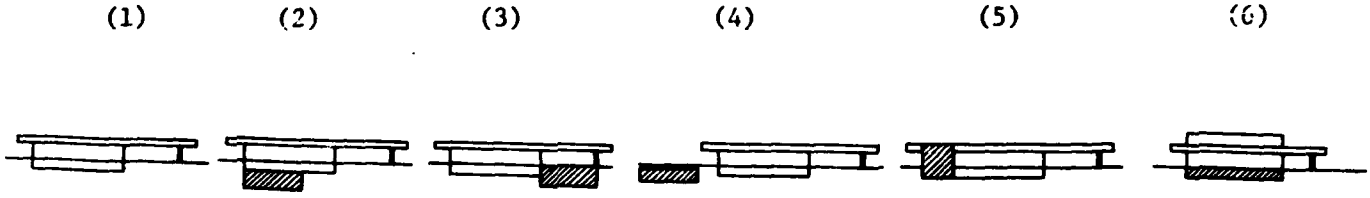
Table 5-2b. Selected Seasonal Thermal Energy Storage Location Options for Multi-Family Residence



Description of Option		Original Design Non-STES	Basement Added for Storage	free-standing detached	below grade attached	under-ground detached	above-ground attached integrated
major performance attributes		initial cost	1	4	2	3	5
		solar/ electro-mech. interface	3	2	4	5	1
		off-the-shelf procurement	2	1	3	1	4
		durability/ service life	2	2	5	4	3
		maintenance frequency/ cost	2	1	5	2	4
		operating cost	3	2	4	5	1
		repair feasibility	5	1	1	2	3
		replacement feasibility	5	1	3	2	4
		overall rank	C	E	A	B	C

1 or A - excellent
5 or E - poor

Table 5-2c. Selected Seasonal Thermal Energy Storage Location Options for Commercial Buildings



Description of Option		Original Design Non-STES (1)	Sub-basement (2)	Storage beneath drive-up facility (3)	Under-ground detached (4)	Above-ground attached integrated (5)	Basement Converted to storage; user space replaced along side (6)
major performance attributes	initial cost		2	1	3	4	2
	solar/ electro-mech. interface		4	4	5	1	2
	off-the-shelf procurement		3	5	1	4	2
	durability/ service life		2	5	4	3	2
	maintenance frequency/ cost		2	4	1	3	2
	operating cost		4	4	5	1	2
	repair feasibility		5	2	1	3	5
	replacement feasibility		5	2	1	3	5
	overall rank		C	A	B	C	E

1 or A - excellent
5 or E - poor

the same for shorter-term storage. The overall ranking for various locations was based on other performance attributes besides those listed*. These are:

- zoning/building code acceptance
- health safety
- land use encroachment
- habitable space encroachment
- architectural/structural interface
- transportability
- contribution to cooling load
- subsidiary benefits

Overall, the location ranking was from best to worst as follows:

(A) single-family homes
(B) multi-family homes
(C) commercial buildings

5.3.4 Temperature Requirements

High storage temperatures are undesirable for the following reasons:

- High temperatures decrease collector efficiency.
- High temperatures increase insulation requirements.
- High temperatures require better quality lining and material for all types of tanks.
- High temperatures increase corrosion rates.

5.3.5 Insulation Requirements

Insulation requirements are primarily determined by the tank's location. Indoor tanks in heated areas require the least insulation, and protection of the insulation can consist of a simple cover. Typical insulation costs for indoor tanks are shown in Figure 5-16. Tanks in unheated indoor locations need extra insulation thickness, but a simple cover is sufficient protection for the insulation. Outdoor tanks have the most severe insulation requirements. Aboveground tanks require extra insulation thickness and protection from weather. Underground tanks require waterproof insulation and extra thickness to compensate for the presence of groundwater. The cost of insulating an outdoor tank is about two to four times the cost of insulating an indoor tank. The cost of materials and labor frequently make a factory-insulated tank less expensive than an on-site-insulated tank.

*For further detail, refer to B&A Engineers, Ltd. An Assessment of the Techno-economic Feasibility of Seasonal Thermal Energy Storage Systems (STES). ANL Report EES-TM-35, Argonne National Laboratory, Argonne, Illinois, December, 1978.

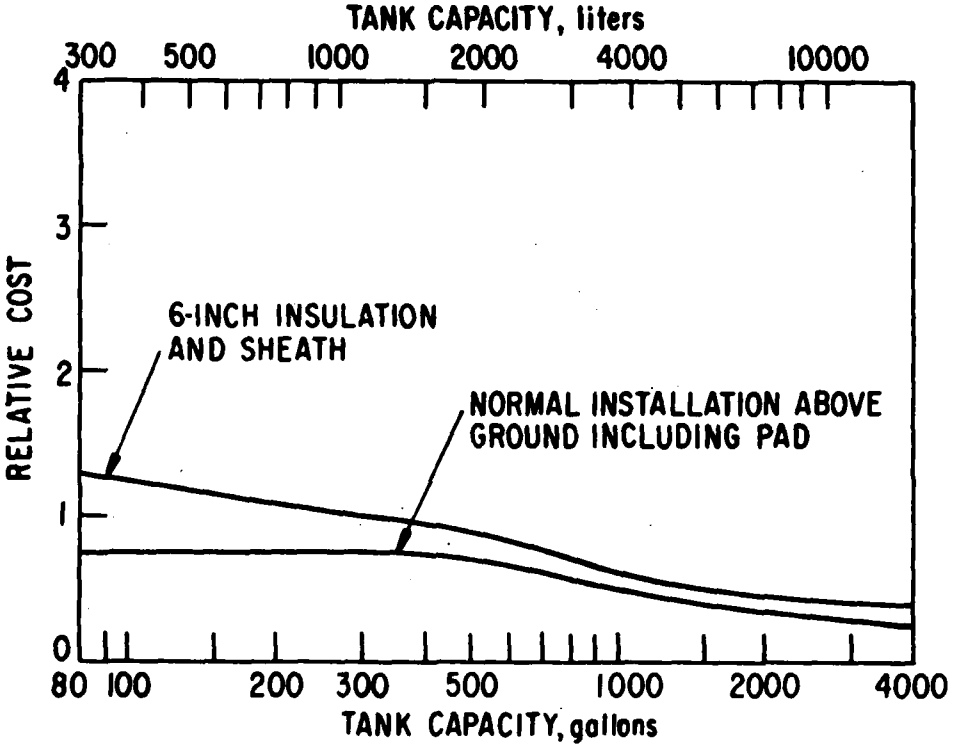


Figure 5-16. Relative Cost of Insulation and Installation for Tanks in the Los Angeles Area, June 1975

Source: E. J. Beck, Jr., and R. L. Field. *Solar Heating of Buildings and Domestic Hot Water*. Civil Engineering Laboratory Technical Report R835, Naval Construction Battalion Center, Port Hueneme, California, April 1976.

5.3.6 Materials

Steel tanks rated for 100 psi (690 kPa) are generally the most expensive type of tank. When the tank, lining, insulation and sheath, and installation costs are added, you can expect the total to be about $2.50 per gallon ($0.66 per liter) for an indoor, 1000- to 2000-gallon (3800- to 7600-liter) tank. This figure should be considered a rough guide and not a precise estimate, since shipping, labor cost and skill, inflation, and availability of tanks influence the total cost. The premium for a pressure-rated tank is generally about $0.50 per gallon ($0.13 per liter), although in some areas the premium can be much less.

The installed cost for a 1000- to 2000-gallon (3800- to 7600-liter) FRP tank without a high-temperature resin for direct space heating is about $1.75 per gallon ($0.53 per liter). On-site-insulated and factory-insulated tanks are similar in cost, since the more difficult and labor-intensive installation procedure for the on-site-insulated tank tends to bring its installed cost up to the installed cost of a factory-insulated tank. As with steel tanks, the choice of manufacturer, shipping, labor cost and skill, inflation, and availability of tanks can significantly affect the installed cost. The cost of high-temperature resin (recommended for direct space heating; unnecessary for a solar-assisted heat pump) is about $0.25 to $1.25 per gallon ($0.07 to $0.33 per liter), varying widely from manufacturer to manufacturer.

The installed cost of a 1000- to 2000-gallon (3800- to 7600-liter) precast concrete tank is about $1.50 per gallon ($0.40 per liter). Shipping costs can add a significant amount if the distance to the manufacturer is more than 100 miles (160 kilometers). Because of the difficulty of shipping these large, heavy objects, manufacturers tend to be local, and prices and availability vary from one part of the country to another. Installed costs of cast-in-place concrete tanks are highly variable, and there is a tendency to neglect cost considerations when designing tanks. With careful design the installed cost of a 1000- to 2000-gallon (3800- to 7600-liter) tank can be less than about $1.50 per gallon ($0.40 per liter). If the tank can be designed into a corner of a building and the tank is poured at the same time the building walls are poured, lower cost is possible. Since this type of construction is labor intensive, costs will vary from one part of the country to another.

Plastic-lined wooden or multicomponent tanks installed in sheltered locations are the least expensive type of tank. The installed cost of a 1000- to 2000-gallon (3800- to 7600-liter) tank of this type can be less than $1.00 per gallon ($0.26 per liter), although some may cost about $1.50 per gallon ($0.40 per liter). Some liner types require periodic replacement, which increases costs.

The above costs (1979-based) are subject to fluctuation and should be used as ballpark figures only. Always contact the manufacturer for specific costs before doing extensive design work.

Paolina Energy Products, Incorporated (West Deptford, New Jersey) manufactures split storage tank kits available in 500, 750, and 1,000 gallon capacities. They are suitable for retrofit installations because the tank sections will fit through most standard doorways. Constructed of fiberglass, they are designed to operate in temperatures up to 250°F. Costs range from near $1,000 to under $2,000 (1981 dollars.)

5.4 List of Symbols

Main Symbols

β	volumetric thermal expansion coefficient, 1/°F(1/°C)
C_p	heat capacity in Btu/lb·°F (kg/m^3)
E	electrical potential in volts
g	gravitational constant, 32 ft/sec^2
H	head (pressure) in feet (m) of water
NPSH	net positive suction head in feet (m) of water
pH	measure of acidity or alkalinity of an aqueous solution
R_i	Richardson number
q	volumetric flow rate in ft^3/hr (m^3/sec)
Q'	heating rate in Btu/hr (W)
u	velocity, ft/sec (m/sec)
Z	difference of elevation in feet (m)
ΔT	temperature difference in °F (°C)
ρ	liquid density in lb/ft^3 (kg/m^3)

Subscripts

a	atmospheric pressure
c	collector
f	friction
p	pressure relative to atmospheric pressure
v	vapor pressure
x	heat exchanger

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6.0 SENSIBLE HEAT STORAGE IN SOLIDS

Sensible heat storage was mentioned briefly in Section 2.3.2. General features of sensible heat storage were discussed in Chapter 4. In this chapter technical details of solid sensible heat storage are explored, with the discussion concentrating on low temperature storage in rock beds and earth beds.

Although the specific heat and thermal conductivity of most inorganic solids are quite low, solids have several advantages as sensible heat storage materials: (1) relative chemical inertness, resulting in low vapor pressure at high temperatures, minimal corrosion, low toxicity, and non-explosiveness, (2) inexpensive cost, and (3) universal availability. In addition, packed bed systems exhibit good heat transfer properties between air and the solids of the bed. Thermal properties of several solid storage materials are given in Appendix F.

6.1 Possible Solid Systems

Although the maximum temperature used with solid storage materials is limited by the melting point, there is usually no lower limit in most applications. However, system design and economic considerations usually require that different materials be used for different storage temperature regimes.

Solid sensible heat storage systems can usually be classified as one of the following types: (1) packed beds, (2) fluidized beds, or (3) solid beds.

Packed Beds: In this type of system, relatively small modular units of the storage material are stacked side by side in a container. The shape of the units is such that a considerable fraction of storage volume is void, allowing the flow of air, or any other fluid, to progress through the bed from one end of the container to the other. Heat is transferred by direct contact between the working fluid and the storage material, eliminating the expense of heat exchange equipment. Packing materials may range from crushed rock at space heating temperatures to relatively expensive refractory bricks, used to supply 2280°F (1250°C) heated air at three atmospheres pressure in blast furnaces. Packed beds can often be used in combination with compressed air storage. Packed bed storage is currently the only solid sensible heat storage system in wide use. Rock bed storage is very popular in air-based solar space heating systems. It optimizes collector efficiency and is often considered the most practical and economical solid storage system. Rock beds are discussed in Section 6.2.

Unpressurized air is commonly used as the heat transfer fluid in packed bed systems. Air is free, nondegradable, and insensitive to leaks, but low density air requires high flow rates and considerable pumping power to over-

come system friction. At lower temperatures, liquids are alternative heat transfer fluids. Water or any of the heat transfer liquids discussed in Chapter 5 or Appendix E are possibilities.

Fluidized Beds: In this system, small solid particles, such as sand, are suspended in a fluid such as air. Heat transfer in and out of storage can be very fast. The large surface to volume ratio of the many small particles enables them to rapidly absorb either sensible heat from gases or reaction heat from chemical processes such as combustion. Further, the heat transfer between the fluidized solids and other large solids immersed in a gas is an order of magnitude higher than the heat transfer between large solids and the gas alone. A further advantage is that fluidized solids can be transported almost as a liquid. Most proposed systems store the solids in an unfluidized state to save pumping costs. The heat loss from the system at this stage is low due to the low thermal conductivity of the settled solid particles. Transportation for either heat absorption or heat transmission can be accomplished by a combination of gravity and pneumatics.

Solid Beds: In this type of system, storage is a container of solid material interspersed with pipes. The pipes carry a heat transfer fluid. Heat is transferred in and out of storage by conduction between the pipes and the rest of the solid. The solid material could consist of steel, concrete, earth, sand, etc. For systems of this type it is necessary to use solid materials with relatively high thermal conductivity, or else too many pipes are required to transfer sufficient heat, making the system uneconomical. The conductivity of earth and sand beds can be significantly raised by adding water. Earth beds are discussed in Section 6.5. Care must be taken in solid bed systems that the solid maintains good thermal contact with the heat transfer pipes.

6.2 Rock Beds

When rocks are used as the heat storage medium, they are placed in a container (a rock bin) that provides enough room for the loose rocks and for air spaces, called plena, that distribute the flow of air through the rocks. When solar-heated air is forced into the rock bin, the plenum allows the air to spread out and pass evenly through the rocks. The hot air gives up its heat to the rocks, and cooled air is returned to the solar collectors. When space heating is needed, the air flow is reversed and warm air is delivered into the building.

6.2.1 Structural Characteristics

Rocks

The ideal rocks to use in a rock bed are the rounded ones typically found in river beds, but crushed rock will work nearly as well. Rocks that conform to ASTM C 33, "Specifications for Concrete Aggregates," are generally acceptable and have been washed to remove dust and dirt.

Several types of rock are unacceptable for use in rock beds. Rocks that crumble, such as schist and softer varieties of limestone and sandstone, should not be used. Carbonate rock minerals such as marble, limestone, and dolomite, are attacked and eroded by carbonic acid formed when carbon dioxide dissolves in water. Since this problem decreases with increasing temperatures, carbonate rocks acceptable for heating systems may cause problems in systems that are used for nighttime cooling in the summer. In addition, some rocks of volcanic origin will smell of sulfur when they are cracked. Test for this before you buy the rocks by breaking several open.

The rocks should be sorted to the same size--within 3/4 to 1-1/2 times the nominal, or average, rock diameter by being passed through a screen mesh. Screen the rocks twice--once to eliminate rocks that are too large and again to eliminate rocks that are too small.

When the rocks are delivered, the smaller ones may have settled to the bottom of the truck, leaving the larger ones on top; mix them before filling the bin. Even if the rocks have been washed they may have picked up dirt and dust during handling; consider washing them again. Some installers dry the rocks in the bin by using the system's fan. Be sure the moisture will not harm the bin components.

Rock Bin

Rock bins, typically constructed of wood, poured concrete, or concrete blocks, must be strong enough to withstand the outward pressure of the rocks. This pressure, already great when the bin is first filled, increases as the rocks expand, contract, and settle with the heating and cooling of the bed. Construction of a poured concrete bin is similar to that of the cast-in-place water tank except that no lining is needed and air inlet and outlet openings must be provided.

Plena

The plena distribute the air uniformly over the top and bottom of the rocks. To do this, the plena must resist air flow much less than the rocks do. We recommend that the cross-sectional area of each plenum, as shown in Figure 6-3, be at least 8 percent of the cross-sectional area of the rock bin. If a plenum is partially obstructed by supports for the rock bed, the plenum area should be increased to 12 percent of the cross-sectional area of the rock bed.

The pressure drop across the rock bed is also important in ensuring proper air distribution. This characteristic will be discussed in section 6.2.2 on performance characteristics.

Air Flow Direction and Rock Bed Shape

In this manual the rock bed is assumed to be vertically oriented-that is, the direction of the air flow through the rocks is from top to bottom when charging (or absorbing heat) and bottom to top when discharging. A vertical bed provides the best thermal characteristics since it utilizes warm air's natural tendency to rise. Vertical rock beds tend to be tall, however, and must be designed carefully to fit into the space available.

Besides the vertical rock bed, two other types have been used: the U-shaped bed and the horizontal bed.

The U-shaped rock bed requires less vertical space and is sometimes preferred to the vertical bed for that reason. Figure 6-1 shows a typical design. Because not much data on the performance of this type of rock bed is available, it should be considered an experimental design. The following potential problems restrict its use:

- More air flows near the center baffle than around the outer walls, resulting in uneven charging.
- Air flow tends to bypass the bottom corners so that they contribute little to storage.
- At the exit side, the tendency for warm air to rise reduces or even reverses the thermal stratification in that section of the rock bed. As a result, the air that is returned to the solar collector will be warmer than it would be in a vertical bin, and the collector will operate less efficiently.

The horizontal rock bed, shown in Figure 6-2a, also requires less vertical space. However, the horizontally oriented rock bed has two major disadvantages caused by its air flow travelling from side to side. First, as the rocks settle, an air space is opened between the top of the rocks and the top of the bin. Air will flow horizontally through this space, bypassing the rocks.

Second, as the warm air flows through the rocks, it will rise and heat the rocks on the top of the bin more than those on the bottom. When the flow is reversed for space heating, part of the air will pass through the warmer rocks at the top while the rest of the air passes through the cooler rocks at the bottom. This nonuniform distribution of heat causes mixed warm and cool air to be delivered to the building at a temperature lower than that of the warmest rocks.

Research on improved designs for horizontal rock beds is being conducted in Japan and the United States. One experimental design is shown in Figure 6-2b.

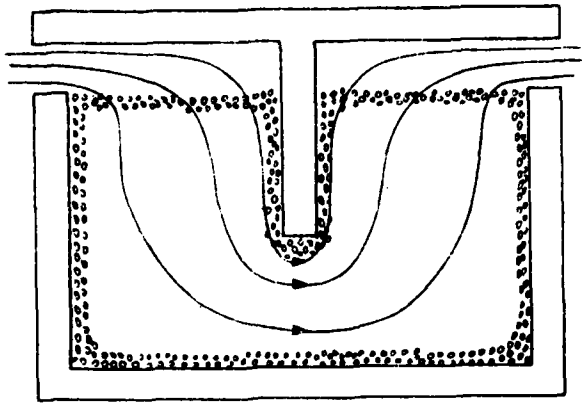


Figure 6-1. U-Shaped Rock Bed.

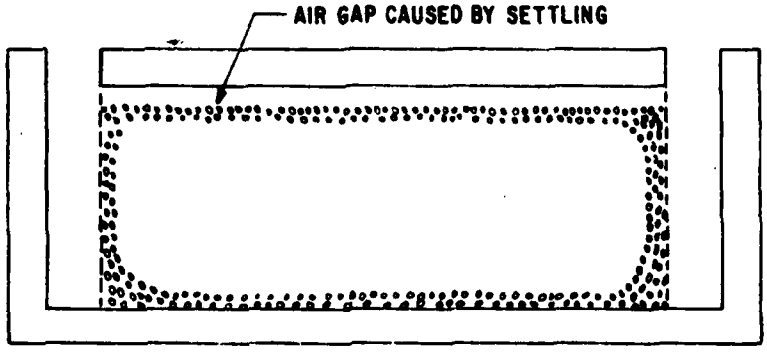


Figure 6-2a. Horizontal Rock Bed

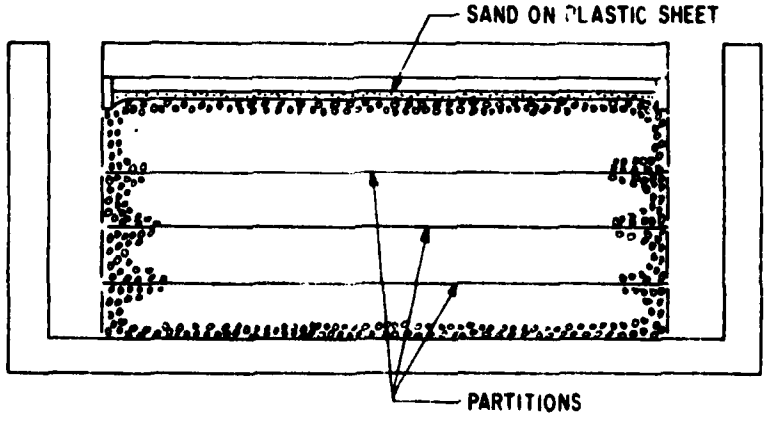


Figure 6-2b. Horizontal Rock Bed with Partitions and Floating Cover

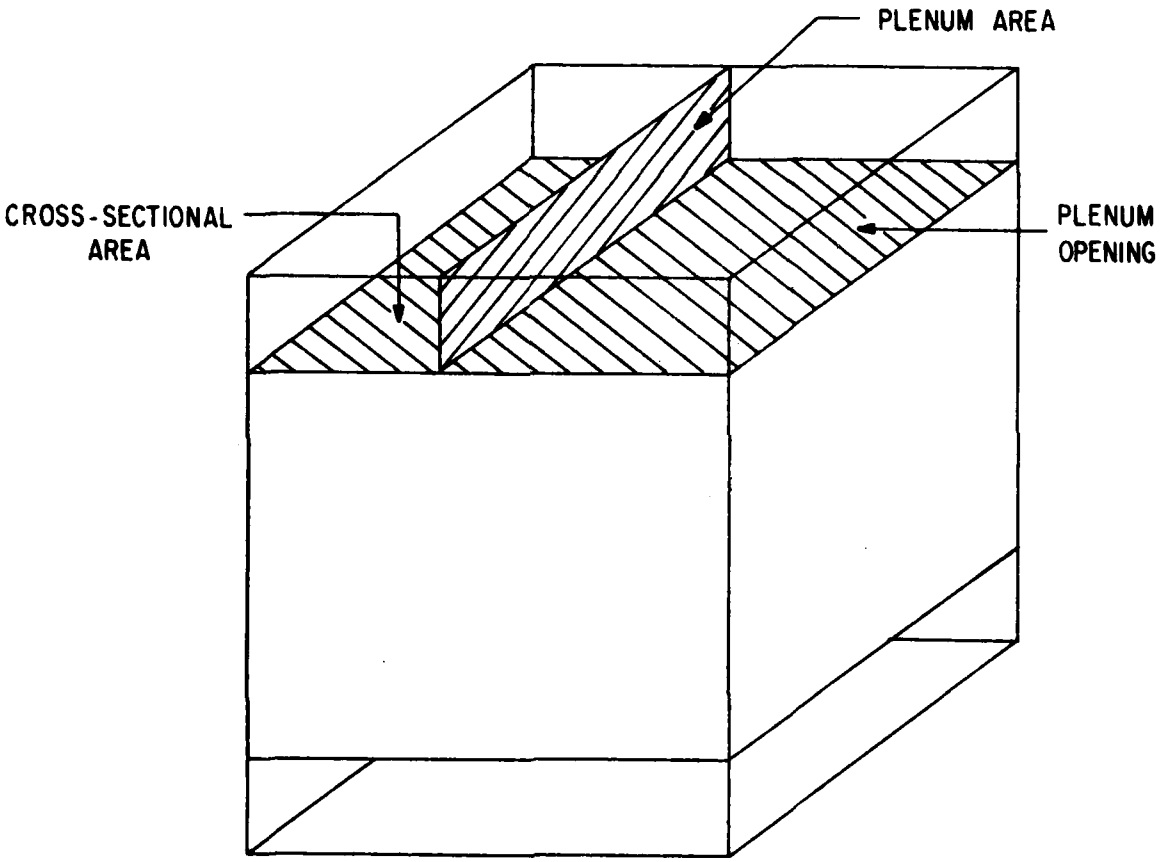


Figure 6-3. Definition of Plenum Area and Cross-Sectional Area in a Rock Bed

6.2.2 Performance Characteristics

Of primary importance to the performance of the rock bed are (1) its volume, (2) the face velocity, (3) the pressure drop across the rock bed, (4) the size of the rocks in relation to the depth of the rock bed, and (5) degree of stratification.

Volume

The method of determining the required rock bed volume was given in Chapter 4. Either the rule of thumb--0.50 to 0.75 cubic feet of rock per square foot (0.15 to 0.20 cubic meters per square meter) of collector--or the calculation method can be used. Having determined the correct volume, determine the dimensions of the rock bed using the procedure outlined in design example 14.2, rockbed heating system.

Face Velocity

The face velocity is defined as the volumetric air flow rate divided by the cross-sectional area of the rock bed. It measures the air's velocity immediately before it reaches the rocks. (Some texts call it the superficial velocity.) The volumetric air flow rate is determined by the collector manufacturer and the size of the collector. Increasing the face velocity by decreasing the cross-sectional area of the rock bed increases the pressure drop across the rock bed.

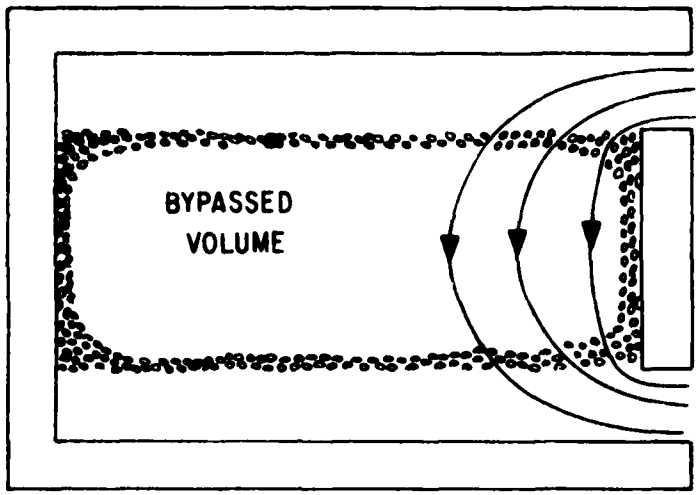
Pressure Drop

Designing the rock bed for a minimum pressure drop of 0.15 inches of water (38 Pascals) will help the plena distribute the air by ensuring that they resist air flow much less than the rock bed does. We also recommend designing the rock bed for a maximum pressure drop of 0.30 inches of water (75 Pascals) so that the fan does not consume an excessive amount of power. Figures 6-4a and 6-4b shows the air flow patterns that occur with proper and improper design.

Dividing the pressure drop by the rock bed depth (for a vertical-flow rock bed) yields the pressure gradient, which will be used in later calculations.

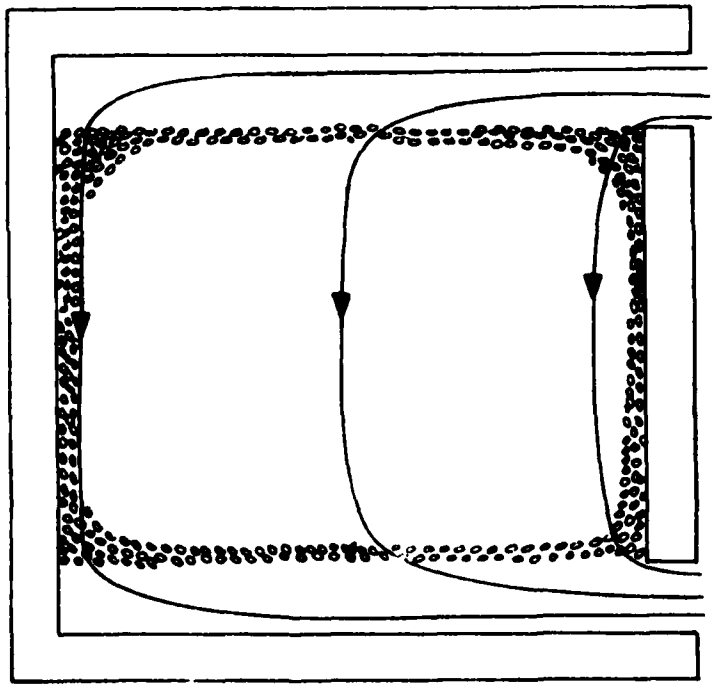
The system pressure drop is the sum of several component pressure drops:

- Rock bed pressure drop.
- Collector pressure drop.
- Filter and damper pressure drops.
- Duct losses, including allowance for bends, branch ducts, and expansions or contractions.



NOT ENOUGH PRESSURE DROP

Figure 6-4a. Air Flow Pattern When the Pressure Drop is Too Low or the Plena Are Too Small



ADEQUATE PRESSURE DROP

Figure 6-4b. Air Flow Pattern When the Pressure Drop and Plenum Size Are Adequate

Rock bed pressure drop is discussed in more detail in Section 14.2. Information about collector, filter, and damper pressure drops should be obtained from the various manufacturers. Detailed procedures for calculating duct losses can be found in the ASHRAE Handbook of Fundamentals, Chapter 25, "Air Duct Design Methods." The pressure drops caused by expansion from the air duct into the plenum of the rock bed and the corresponding contraction at the opposite end of the rock bed should not be overlooked. The method for calculating these pressure drops is also given in the ASHRAE Handbook of Fundamentals.

Rock Diameter

The fourth parameter affecting performance is the rock diameter. Increasing the rock diameter decreases the pressure drop across the rock bed, because the larger spaces between large rocks have less resistance to air flow than the spaces between small rocks.

Another, more subtle, effect of increasing the rock diameter is that it increases the time (equilibration time) required for each rock to reach its equilibrium temperature with the air. If this time is more than the time required for air to traverse the rock bed, the rocks will be unable to absorb all of the heat in the air by the time the air leaves the rock bed. Thus, air will be returned to the collector at a temperature that is warmer than the coolest rocks, and the benefit of thermal stratification will be lost.

If the rocks' equilibration time is much less than the time required for air to traverse the rock bed, however, the air will give up nearly all of its heat to the rocks and will emerge from the rock bed at the temperature of the coolest rocks. The rock bed will remain thermally stratified, with hot rocks at the top and cool rocks at the bottom. As heat is added to the rock bed, the interface between hot and cool rocks moves downward, but air leaves the rock bed at the temperature of the coolest rocks until the rock bed is almost fully charged. To maintain thermal stratification in this manner, make the rock bed depth at least twenty times the rock diameter if the rocks are less than 4 inches (10 centimeters) in diameter or by at least thirty times the rock bed diameter if the rocks are larger than 4 inches in diameter.

Stratification

The performance of typical air space heating systems is markedly affected by the degree of stratification in the sensible heat storage environment. Figure 6-4c illustrates both the extent of this stratification and also that collector heat loss does not become a problem until late afternoon. Stratification occurs because the heat transfer rate at each point in the pebble bed is proportional to $T_{air}-T_{rock}$. During collection, as the value of T_{rock} near the top (see Figure 6-4d) approaches T_{air}, the air slowly propagates its heat in the form of a temperature wavefront. Because this stratified character enhances both charging and discharging efficiencies, the

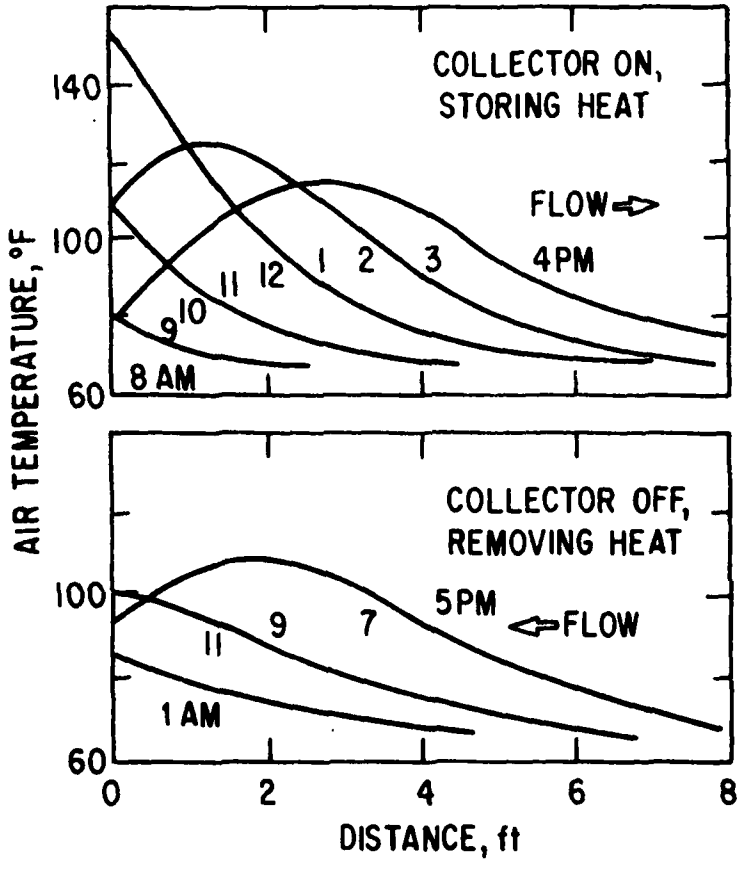


Figure 6-4c. Rockbed Charging and Discharging Curves

(Adapted from Connor in Solar Energy Technology Handbook, Part A- Engineering Fundamentals, 1980.)

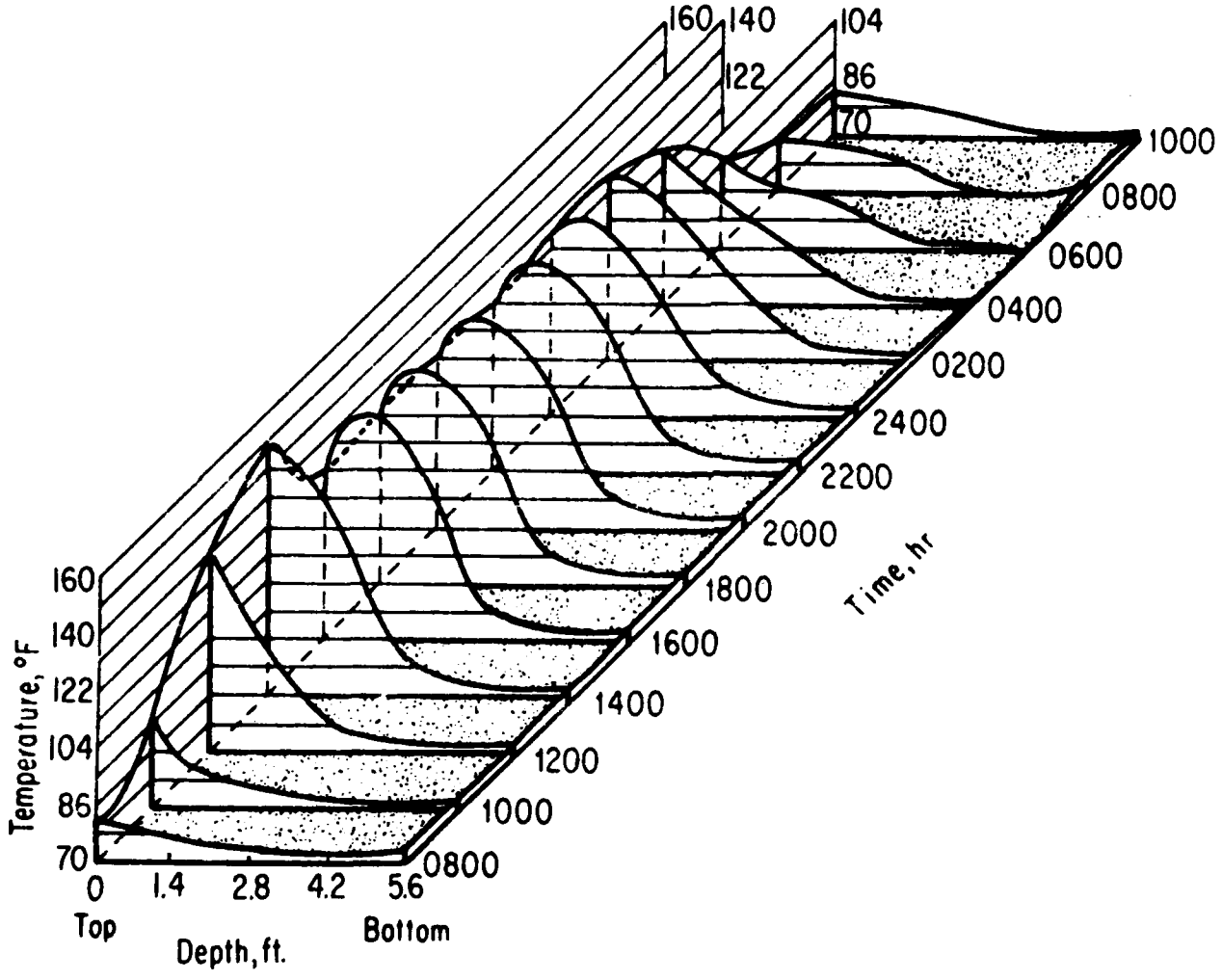


Figure 6-4d. Temperature Profiles in Rockbed During Peak Heating Season (Adapted from Lof in Solar Energy Handbook, Chapter 12, 1980.)

overall efficiency of air space heating systems approaches that of liquid-collector systems. Another feature of air heating systems is that electricity consumption is similar to that for liquid systems. In fact, if the liquid system requires a heat exchanger between collector and storage, its power requirements may surpass those of the air type. Liquid systems that use a coil-in-tank (natural convection) heat exchanger do, however, require less pumping power than air systems.

6.2.3 Fan Selection

When you select a fan for the system, your main concern will be matching the fan's pumping characteristics to the system's pressure drop characteristics. The fan's diameter, type (axial flow or centrifugal), blade angles, and operating speed (RPM) all affect its pumping characteristics. Fan manufacturers publish data giving flow rate versus static pressure for their products. If the fan can operate at more than one speed, data for several speeds will be published. Typical curves for three different fan speeds are shown in Figure 6-5.

The designer must calculate the system's pressure drop at the operating flow rate and select a fan and operating speed that will give a matching static pressure. Figure 6-5 shows the operating flow rate. For the fan data shown in the figure, the fan should operate at 1100 RPM to provide the operating flow rate required by the system.

To illustrate what will happen if the designer makes the wrong choice of fan or fan speed, a curve labeled "system pressure drop" has been drawn on Figure 6-7. If the fan is too large or the fan speed too fast, the system will operate at Point A. Both the air flow rate and the system pressure drop will be greater than planned for, and the fan will consume more electric power than a properly sized fan.*

If the fan is too small or the fan speed too slow, the system will operate at Point B. The system pressure drop will be less than its design point; but, more importantly, the air flow rate will be lower than it should be. The low air flow rate will decrease the amount of heat removed from the collector, thus increasing the collector temperature and degrading the system's performance.

Power Requirements

Having selected the fan, the designer must choose a motor to power it. If the motor is too small, the fan will not be able to pump the necessary amount of air, and frequent motor burnouts will be likely. An oversized motor will draw only slightly more power than a motor of exactly the proper size (unless the motor is grossly oversized). Thus, it is better to select a slightly oversized motor than an undersized one. Belt drives must be rated for one and a half times the motor power and should include an adjustable sheave on the motor.

*Points A and B are in related Figure 6-5.

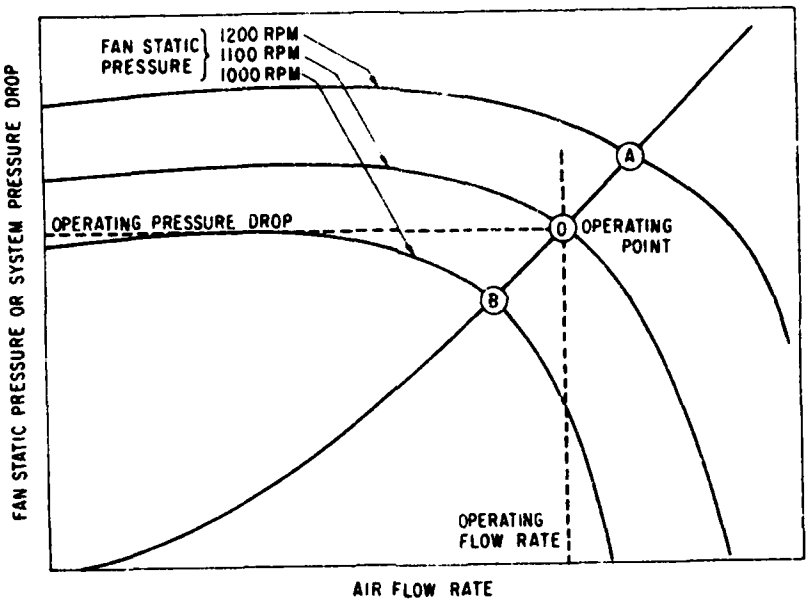


Figure 6-5. Typical Fan and System Characteristics

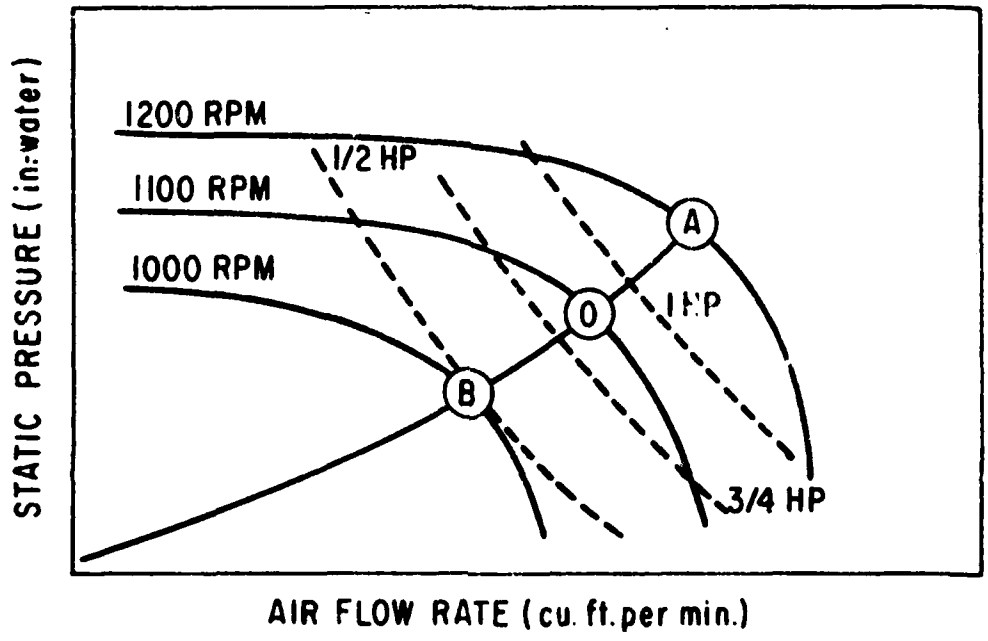


Figure 6-6a. Typical Fan Performance Curves Showing Motor Power Requirements

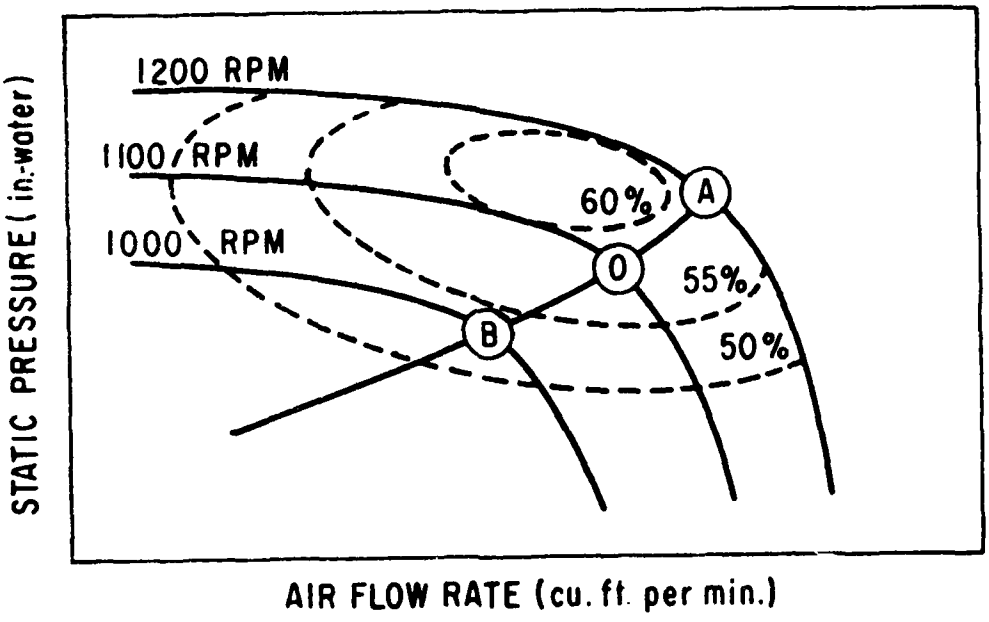


Figure 6-6b. Typical Fan Performance Curves Showing Fan Efficiency

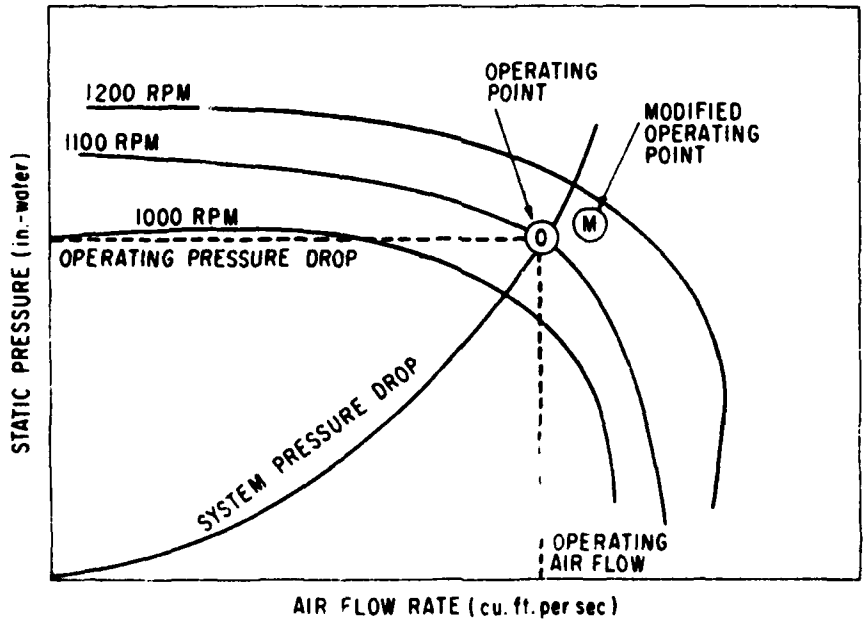


Figure 6-7. Modified Operating Point

Many fan manufacturers publish the motor requirements with the fan performance curves, as shown in Figure 6-6a. If the manufacturer's data is presented in this way, select the larger of the two motors indicated by the dashed lines on either side of the operating point. For example, in Figure 6-6a dashed lines corresponding to 3/4 and 1 horsepower lie on either side of the operating point (Point O). Choose the 1 horsepower motor.

Sometimes the manufacturer presents fan efficiency, as shown in Figure 6-6b, instead of motor horsepower. A short calculation is required to determine the minimum motor power, HP_{min}, in horsepower (kilowatts):

$$HP_{min} = \frac{1.25\ q\ \Delta P_T}{c\eta} \qquad (6\text{-}1)$$

where

q = volumetric air flow rate in cubic feet per minute (cubic meters per second).

Δp_T = system pressure drop in inches of water (Pascals).

η = fan efficiency in percent.

c = a unit conversion constant. Use $c = 63.46$ to convert cubic feet per minute, inches of water, and percent to horsepower; use $c = 10$ to convert cubic meters per second, Pascals, and percent to kilowatts.

Fan Installation

The temperature of the air a fan must handle in a solar system can sometimes present a problem not often encountered in conventional heating systems. Study your system carefully and determine the maximum air temperature the fan will encounter. If that temperature exceeds 100°F (38°C), the fan must meet the following specifications.

- The fan bearings must be able to operate continuously at the maximum air temperature. Special bearings may be required. Alternatively, the bearings can be located outside the stream of hot air, and shaft seals can be specified to minimize leakage.
- The motor and drive belts must be outside the stream of heated air or a Type B motor connected directly to the fan must be used.
- The fan should be selected on the basis of a modified operating point (Point M in Figure 6-7) instead of the previously defined operating point (Point O). To find the modified operating point multiply both the air flow rate and the fan static pressure at the operating point by the factor F.

$$F = \frac{T + T_o}{T_r + T_o} \qquad (6\text{-}2)$$

where

T = air temperature in the duct in degrees Fahrenheit (degrees Celsius).

T_o = conversion to absolute temperature scale. Use $T_o = 460°F$ for Fahrenheit scale or $T_o = 273°C$ for Celsius scale.

T_r = room temperature. Use $T_r = 70°F$ or $T_r = 20°C$.

The modified operating point applies only to fan selection and should not be used for other calculations.

Since the major operating expense of an air-based system is the cost of electricity, it is important to install the fan so that it will operate at its highest efficiency. This is accomplished by connecting the inlet of the fan to the ductwork with a straight section of duct at least five duct diameters long. The duct should match the diameter of the fan inlet so that there will not be a sudden contraction or expansion as the air enters the fan. If a transition from a rectangular duct to a round fan inlet must be made, the transition slope should not exceed 4 in 12 inches (18°). It is especially important to avoid using bends or elbows near the fan inlet, because the turbulence they cause reduces fan efficiency. Use similar care in designing the outlet ductwork.

6.2.4 Other Components

Filters

Filters for the air-based solar system should be located at both the rock bin inlet and outlet where they are easily accessible for service or replacement. Filter mounts must minimize the amount of leakage bypassing the filter and leakage escaping the duct.

The face velocity of the filter (air flow rate divided by filter area) should not exceed 300 feet per minute (1.5 meters per second). If the filter is larger than the cross-section of the duct, a transition to the full filter size, with a slope not exceeding 4 in 12 inches (18°), must be made.

Install a filter replacement indicating gauge at each filter. The gauge can be self-indicating or remote-indicating, but in either case the indicating part of the gauge must be located where it will be easy to see.

Dampers

Since dampers have proven to be the least reliable components in existing air-based systems, it is worthwhile to invest in high-quality dampers for your system. Automatically controlled dampers are essential to control the direction of air flow through the rock bed and to control the collector and space heating loops. Spring-loaded, motor-driven dampers can provide fail-safe operation in case of a power failure. Backdraft dampers should be installed in the ducts between the collectors and the rock bed to prevent thermosiphoning (i.e. natural convection) at night--a major source of heat loss to the system. Choose good backdraft dampers with seals made of felt or other resilient material such as silicone rubber. The backdraft dampers must close by either gravity or springs and must remain tightly closed until the blower opens them.

Air Handlers

Air handlers, including a fan and as many as four motorized dampers in one package, are available. The main advantages of an air handling unit are:

- Air handlers require less installation labor than separate components.
- Air handlers specifically designed for solar applications can be purchased.

Choose an air handler in preference to individual components if an air handler that meets your system's flow rate and control requirements is available.

Temperature Sensors

A low-limit temperature controller in the rock bed can minimize temperature fluctuations in the living space by cutting off air from the rockbed when its temperature is too low. Otherwise, cool air would be delivered from the rockbed until the room thermostat actuates the auxiliary heater.

Use the type of temperature sensor recommended by the controller manufacturer.

Two temperature sensors should be placed in the rock bed, one 6 inches (15 centimeters) below the top of the rocks and the other 6 inches (15 centimeters) above the bottom of the rocks. Low temperature readings by both sensors indicate that little heat remains in the rock bed, and the auxiliary heater must supply heat. High temperature readings by both sensors indicate that the rock bed is fully charged. A high temperature reading at the top and a low temperature reading at the bottom indicate that the rock bed is partially charged.

To avoid damaging temperature sensors and to make replacement simple, we recommend that the sensors be placed inside pipes that extend from outside the rock bin's inner wall to the center of the rock bed. The sensor leads are then run from the pipe out through the insulation to an electrical box for connection to the controller.

Air-to-Water Heat Exchanger

If the solar system is to provide domestic hot water, an air-to-water heat exchanger is usually installed in the duct returning from the collector. Although it is possible to bury a water tank in the rock bed, using an air-to-water heat exchanger in the duct offers the advantages of (1) good heat transfer characteristics and (2) the ability to bypass the rock bed during the summer while providing solar-heated water. (Bypassing the rock bed in summer will reduce the air conditioning load if the rock bed is located in an air conditioned part of the building.)

Install a low-leakage automatic damper between the heat exchanger and the collector to protect the heat exchanger from freezing. The damper should close automatically when the collector is not collecting. In several instances heat exchangers have frozen when cold air from the collector settled around the heat exchanger where there was no damper to separate them, or where the damper leaked. The preferred type of actuator uses a 24-volt motor to open the damper and a spring to close it, so that the heat exchanger will be protected even during a power failure.

Auxiliary Heating System

Every solar-heated building must have an auxiliary heating unit to furnish heat when the sun is not shining or the thermal storage device is depleted. The auxiliary heater must be able to supply 100 percent of the heating load without any assistance from the solar system. Although any type of conventional air heater can be used, a Class B furnace blower motor capable of operation in air at temperatures up to 150°F (66°C) must be used. Such a unit should be located as close as possible to the storage device to minimize the length and cost of the connecting ducts. Almost all air-based solar systems use forced-air auxiliary heaters, which can be placed in the duct following the rock bed, as shown in Figure 6-8a, or in parallel with the rock bed, as shown in Figure 6-8b. However, if a heat pump is used instead, the separate two-part type with outdoor and indoor sections is most practical.

The configuration shown in Figure 6-8a has a minimum rock bed operating temperature of about 65 to 70°F (18 to 21°C). Studies have shown that people find circulating air at 70°F chilling. To ensure that the air will circulate at a comfortable temperature, the auxiliary heater is turned on whenever solar energy is unable to maintain a duct temperature of about 95°F (35°C) or more. While the auxiliary heater is operating, solar energy preheats the air entering the auxiliary heater. This system is usually used with an electric heater. Because the solar-heated air must pass through the aux-

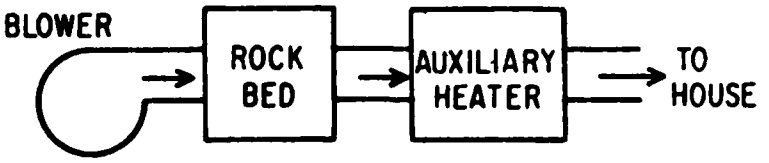


Figure 6-8a. Rock Bed in Series with Auxiliary Heater

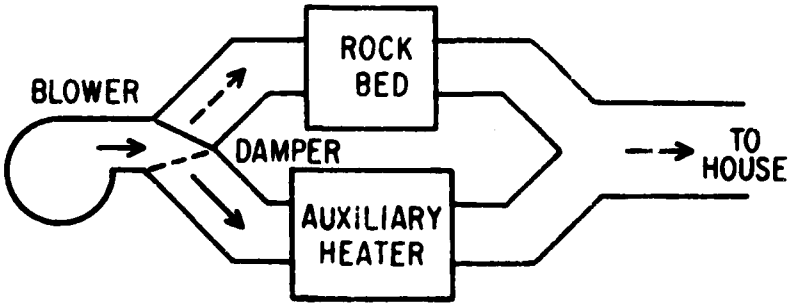


Figure 6-8b. Rock Bed in Parallel with Auxiliary Heater

iliary furnace's heat exchanger, some solar heat may be lost up the flue of a gas or oil furnace. If the auxiliary furnace has an automatic flue damper to prevent this loss, the solar preheat arrangement is feasible with a gas or oil furnace.

The configuration shown in Figure 6-8b is usually used with gas or oil auxiliary heaters without flue dampers or for situations where there is not room to install the rock bed between the blower and the auxiliary heater. The auxiliary heater is turned on whenever solar energy cannot maintain a duct temperature of about 95°F (35°C) or more. If the rock bed cannot maintain a 95°F outlet temperature, a motorized damper switches the air flow to the auxiliary heater, and the auxiliary heater is turned on. Thus, if the rock bed and auxiliary heater are installed in parallel, the minimum usable rock bed temperature is 95°F.

In some solar heating installations the design air flow rate through the collectors does not match the residential space air flow requirements. For example, if the building is to be cooled as well as heated, the air flow requirements will normally be based on the cooling load and will be higher than the air flow required for heating. On the other hand, in areas where the design temperatures and the amount of available sunshine are low, it is possible that the solar collector's flow rate requirements will exceed the conventional heating system's flow specifications.

In those systems where the collector and the conventional heating system air flows are essentially balanced, the solar system blower can provide the total air movement. A second blower is necessary when an air imbalance exists or when constant air circulation through the building is required for ventilation or filtration. This second blower may be a component of a standard furnace, a roof-top unit, or an air handling system. When air movement requirements of the conventional system exceed those of the solar system, a duct must be installed to bypass the rock bed, and air balance between the two systems must be adjusted with a damper.

6.3 Operation and Maintenance

Design for Maintainability

Well-designed rockbeds require no maintenance; only the air handler has parts subject to wear. Although properly installed and adjusted motors, blowers, and motorized dampers will be trouble-free for many years, maintenance must be considered as the system is being designed. Many solar systems fail or perform poorly because they cannot be properly maintained. Before installing any components the designer should consider how each component will be repaired or replaced and provide for working room around the components. A component that cannot be easily replaced should last the lifetime of the system.

Simple systems are usually easier to maintain than complicated systems. The relative advantages of simple systems are:

- Initial cost is lower.
- Installation errors are less likely.
- There are fewer components to fail.
- Controls and operation are easier to understand.
- Defective components can be more easily found and replaced.

Answering three questions will help determine whether the system is too complicated or too simple.

- If a feature were deleted from the system, how much energy collection would be lost?
- If a feature were deleted from the system, would the possibility of failure be increased?
- If a feature were deleted from the system, would human safety be degraded?

A system analysis method (such as f-Chart, SOLCOST, DOE-1, or TRNSYS) is required to estimate the extra amount of energy collection attributable to a particular feature. If the value of the extra energy collected over the life of the system is less than the cost of the feature, the feature cannot be justified economically. If, in addition, the answers to the second and third questions are "no," the feature should be deleted from the system.

Indoor Air Pollution

Indoor generated pollution has become an increasing problem as people respond to fuel prices by making their homes snugger and more energy efficient.

Building and heat storage materials provide a major source of indoor pollution. Carcinogens such as asbestos in fireproofing, formaldehyde from insulation, and radon derived from thermal mass materials (earth, concrete, rocks) are very hazardous with inadequate ventilation.

$Radon^{222}$ is a decay product released by trace amounts of uranium in rock and soil everywhere. Normally the level remains low enough that no health risk is posed. However, radon levels in a number of well-insulated, air-tight, homes that have been measured exceed the maximum permissible occupational values set for uranium mines.

Control methods, however, are numerous and effective.

- Air-to-air heat exchangers should be used to assure proper ventilation but still minimize heat loss.

- If possible, the earth or rock should be checked prior to installation for the extent of radon out-gassing.
- An activated carbon filter system should be used to absorb radon emissions from the earth/rock bed.

Odors

Although the problem has never been encountered, a recurring question that is brought up about rock beds is the possibility of unpleasant odors occuring from mold, mildew, or other microorganisms. Scientists at Pennsylvania State University recently studied this question in a laboratory rockbed under conditions simulating solar heat storage. Nocturnal cooling conditions (where condensation and high relative humidity are expected) were also simulated. They found no evidence of microbial attack after four months and concluded that significant growth was unlikely under normal operating conditions. Apparently, the daily combination of flowing air, heat, and low relative humidity makes the rockbed quite a sterile place.

Startup

Before applying insulation to the ducts and the rock bin, carefully inspect and test the system. Begin by checking all ducts, dampers, and wiring against the system drawings. Typical problems that might be encountered include:

- Inlet and outlet connections to rock bed, fans, heat exchanger, or collector reversed.
- Normally open automatic dampers installed in place of normally closed automatic dampers.
- Fan rotation reversed.

Leak detection is more difficult in air-based systems than in liquid-based systems. Although leakage is not as dramatic, it can result in inflow of cold air or outflow of hot air. Check your system for leaks before installing the insulation by disconnecting the ducts going to the collectors and the building and sealing the ends of these ducts. Connect the blower so that it will pressurize the system and then introduce smoke from a nontoxic smoke candle into the blower inlet. Carefully inspect the rock bin and all ducts and connections for smoke leaks. A slightly negative pressure is preferred and usually practiced.

If you find a leak, try to repair it permanently before applying duct tape and insulation. Silicone caulk or epoxy putty can be used to plug small leaks, but larger leaks require refitting of components. Leaks are most likely to occur at the seams of the rock bin, at joints between duct sections, and at connections between ducts and other equipment. Tight closure of the damper is also critically important.

This is a good time to test the system in all operating modes to ensure that it functions as intended. Since systems vary greatly in their operating modes, only general guidelines can be given here. Most controller manufacturers make testing devices and publish data on how to use the testers. You may need a set of jumper wires to operate the system in its various modes. CAUTION: DANGEROUS VOLTAGE MAY BE PRESENT AT CONTROLLER TERMINALS. Flows in ducts can usually be determined by feeling a temperature change and by observing temperature changes with the temperature sensors installed in the ducts, rock bed, and collectors.

Install the insulation, R-10 preferred, on the ducts, rock bed, and other components. Label the ducts according to air flow direction. Tag the dampers, fans, filters, and so on to correspond with the numbers on the system drawings. Automatically controlled two-way dampers should be labelled "normally open" or "normally closed," and the legs of automatically controlled three-way dampers should be labelled "common," "normally open," and "normally closed."

Operate the system in its heating, noncollecting mode to fully discharge the rock bed. With the rock bed discharged, its temperature sensors should indicate low temperatures. Change to the collecting, nonheating mode to charge the rock bed. Now as it charges, its top temperature sensor should indicate a high temperature, its bottom temperature sensor a low temperature. When the rock bed is fully charged, both temperature sensors should indicate high temperatures. Replace defective sensors.

Inspect the air filters. If they are excessively dirty, find the cause and repair it before you install clean air filters. The system is now ready for operation.

Periodic Inspection and Maintenance

The following tasks should be performed monthly during the heating season or at intervals specified by component manufacturers.

- Replace air filters.
- Inspect fan belts.
- Lubricate motor and fan bearings.

Before each heating season, inspect the system for leaks and check fans, dampers, sensors, and controllers for proper function. Keep a detailed operating/maintenance log to expedite trouble shooting, etc.

Owner's Manual

The contractor should provide the owner with, or the owner should specifically request, a manual that includes the following:

- A summary description of how to operate the controls.
- Instructions on how to do periodic maintenance.

- A detailed description of how the system operates.
- Schematics of ducting and wiring with labels that correspond to the labels attached to the hardware.
- Component and system warranties.

6.4 Rock Bed Costs

Information about solid storage systems is not as extensive as for liquid systems and the cost of rock beds varies widely from region to region.

1978 prices were obtained for the wholesale component prices of a collector and also for the complete cost of building and installing a rock bed. The complete price of the 500-ft^2 collector was about $8500; the cost for a rock bin, sized to handle this collector, about $500-$1000. Thus, in 1978, the rock bed was a minor part of the expense, increasing the net cost about $1 to $2 per square foot of collector.

The cost of rocks at the quarry typically ranges from three to ten dollars per ton, although "ornamental rock" may cost as much as sixty dollars per ton. Carefully specify the type of rocks and inspect them before they are delivered. In locations distant from suitable quarries, delivery is a major expense.

The cost of the rock bin will, of course, depend upon the type built. Wooden containers are the least expensive, followed by cinder block and concrete. Figure 6-9 shows the relative cost per cubic foot of these three types of containers in one area of the United States. The relative costs are for complete rock beds--that is, they do not reflect such possible savings as using a basement corner as two walls of a concrete or cinder block rock bin. The cost per cubic foot for larger containers is less than for smaller ones, because the volume goes up faster than the surface area, and it is the surface area that determines the amount of material and labor involved in constructing the container.

6.5 Earth Beds

Although not yet commercially proven, earth beds should provide low cost alternative heat storage methods. Earth bed thermal storage consists of a volume of earth in which heat exchange pipe is placed. The advantage of such a system is that except for the pipe, the only cost associated with storage is from earth moving. The storage temperatures are relatively low, less than 100°F (38°C), usually requiring a heat pump to deliver heat to the load. The storage can be charged with inexpensive low temperature solar collectors. Earth beds may be used in two different ways in a storage system:

1. The most effective use of earth beds is as a long term or annual storage device. Winter space heating solar collectors always generate excess heat in the summer, even if the collectors are also

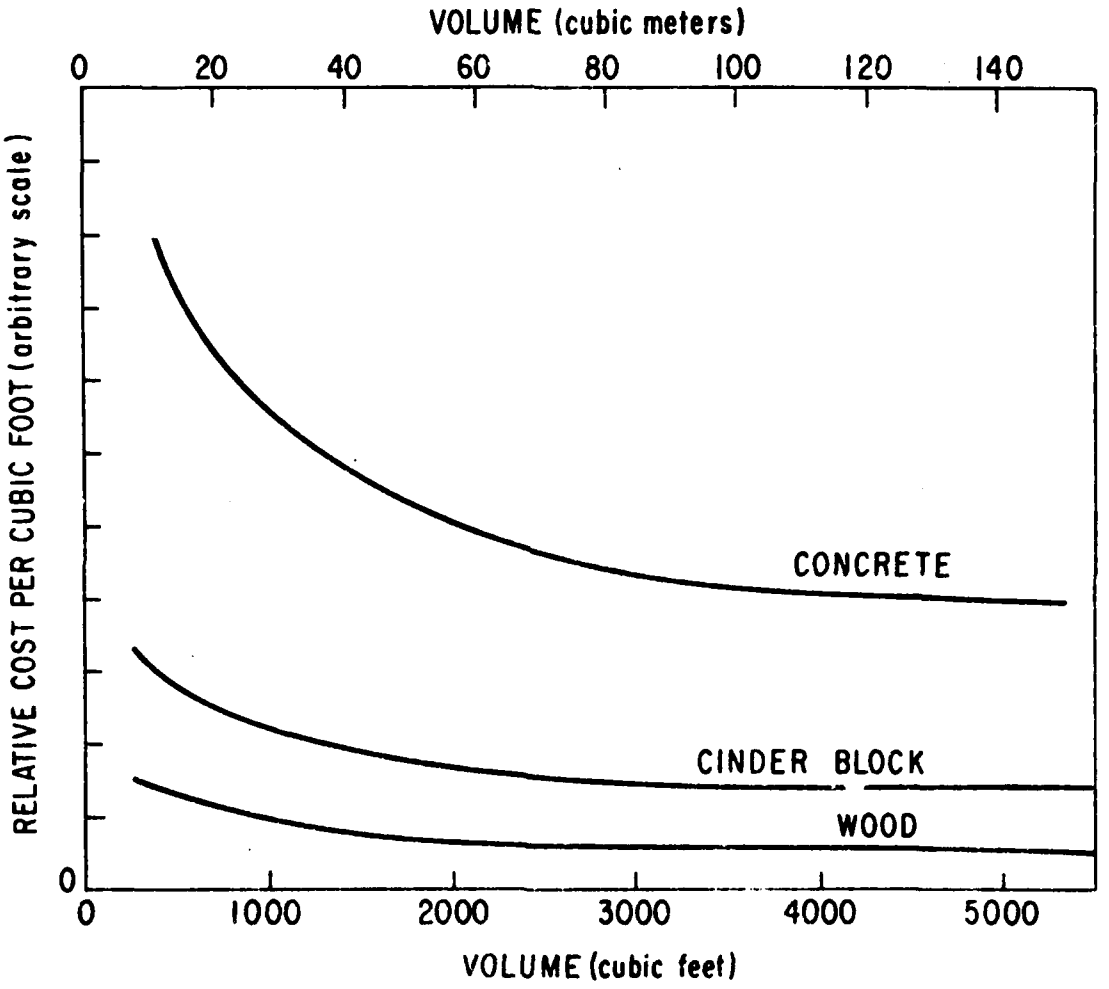


Figure 6-9. Relative Cost of Rock Bed Containers

Source: United States Energy Research and Development Administration, Division of Solar Energy. *Inter-Technology Corporation Technology Summary, Solar Heating and Cooling*. ERDA Report no. COO/2688-76-10, 1976.

used for domestic hot water. Long term storage would allow use of a smaller collector area. Since collectors are the most expensive part of a space heating system, great economic savings are realized.

2. The earth bed may act as ground-coupling in a series solar/heat-pump system. In this case the solar collector supplies heat to a water tank or rockbed, which in turn acts as a source for the heat pump. The earth bed is heated during the summer from the surrounding ground. When the regular storage temperature is below the ground temperature, the ground acts as the source for the heat pump. The ground is coupled to the heat pump primarily during midwinter (December 10 to February 10) when heating demand is highest and the regular collector system cannot meet all of the load. Using the ground, as opposed to cold ambient air, keeps the COP (coefficient of performance) of the heat pump high and eliminates the need for electric resistance heating, somewhat improving the economics of series solar/heat-pumps. An alternative to heat exchange pipes in the ground in this case is to use the air in the crawl space under a house as the source for the heat pump.

Two systems of placing the heat exchange pipes in an earth bed are illustrated in Fig. 6-10. In Fig. 6-10a a serpentine coil is buried in the earth. A typical coil would use 1 1/2 in. to 4 in. (3.8 cm to 10.2 cm) diameter plastic pipe, with pipe lengths spaced at least four feet (1.2 m) apart. The coil should be buried at least four feet (1.2 m) deep; in cold climates it should be several feet below the frost line. This system is recommended when solar collectors are to transfer heat to the earth. The system drawn in Fig. 6-10b is a series of wells placed into the ground. Water is circulated in the shaft of the well. The amount of land area utilized in this configuration is less than in the serpentine coil system, but heat recovery is not as efficient. This system should be used only when natural ground temperatures but not solar collectors are to be utilized. A similar system can be utilized in some retrofit situations if cisterns are available. The water temperature in a cistern tank is usually kept above the winter air temperature by heat transfer from the surrounding earth. Using a cistern as a source for a heat pump provides an inexpensive space heating system. However, the capacity of many cisterns may not be enough to carry a building through the winter unless strong conservation measures are taken first.

Earth beds without solar assist are probably more practical in southern U.S. climates. In northern climates the soil temperature is low during the heating season. Either the pipes must be placed lower in the ground or solar assist is required.

The heat pump that is coupled to the earth bed storage must have high efficiency above 40°F (4°C). Several commercial heat pumps meet this requirement, but most do not have this property, as they have been designed to operate at lower ambient temperatures.

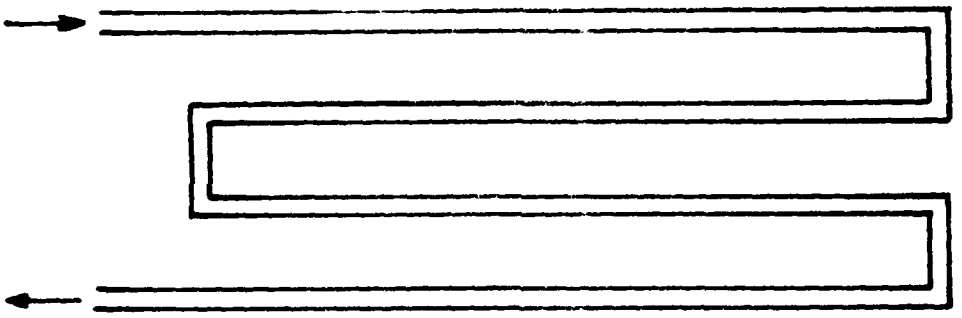


Figure 6-10a. Top view of serpentine coil earth bed heat exchanger.

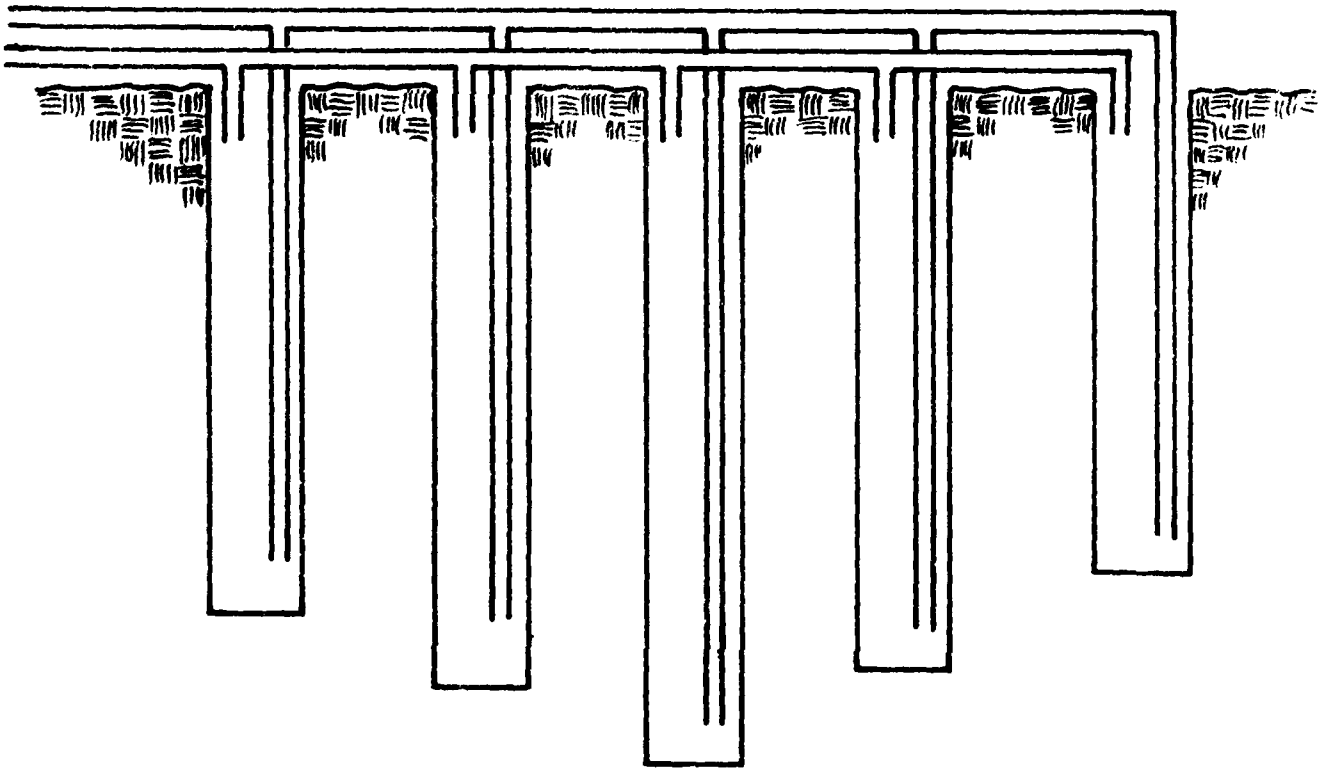


Figure 6-10b. Well/cistern heat exchange system in earthbed.

Earth beds should not be used as storage in systems with ordinary solar collectors, except possibly in very large systems. Heat flow from the typical storage temperatures (100° to 180°F) (38° - 82°C) to the normal ground temperature (55°F) (13°C) is much too high for uninsulated beds. Heavy insulation of earth beds is as troublesome and almost as costly as installing a rockbed or water tank and is not recommended. A moderate amount of self insulation can be obtained with the use of vapor barriers. Moist soil has a high conductivity (0.87 - 1.44 Btu/hr.ft.°F or 1.5-2.5 W/m.°C), while dry soil has a conductivity an order of magnitude lower. The storage part of the earth bed should consist of moist earth surrounded by a vapor barrier. Once the earth bed is charged, the temperature difference between storage and the surrounding ground will dry out the soil immediately surrounding the vapor barrier, thus providing natural insulation. The area must be carefully drained so that rain water does not seep down around the earth bed, rewetting the soil. The vapor barrier must be chosen to last the life of the system and resist perforation by animal and plant activity. Commercial plastic liners, used for industrial waste containment ponds, are available for less than $1 per square foot ($10 per square meter).

The water table can typically be considered as a constant temperature heat sink when the water is essentially free moving through the soil. If the height of the free water table is too close to the earth bed, the site is inappropriate for storing solar energy. On the other hand, a high water table enhances the potential for ground-coupling, as the water table acts as a constant source of 55°F (13°C) heat. The distance criteria for the appropriate type depends on soil conditions. However, if the distance is 30 (9 m) feet or greater, solar-coupling is appropriate. If the distance is 10 (3 m) feet or less, ground-coupling is appropriate.

Because of the small number of earth bed systems studied to date, only rough sizing guidelines can be given. The total length of heat exchange pipe necessary is given by:

$$L = \frac{Q'}{2\pi K_m(\Delta T)} \ln (Rb/Ra) \qquad (6\text{-}3)$$

where L is the length of pipe in ft,
Q' is the heat flow in Btu/hr,
K_m is the thermal conductivity of the moist soil in the earth bed in Btu/hr.ft.°F,
ΔT, in °F, is either

(1) $T_{min}-T_{\rho,\ell}$ if the pipe is delive ng heat to the load. $T_{\rho,\ell}$ is the temperature of the pipe when connected to the load.

(2) $T_{\rho,s}-T_{max}$ if the pipe is delivering heat to the bed. $T_{\rho,s}$ is the temperature of the pipe when connected to the source.

Select the smaller value of ΔT from (1) and (2) so that the calculated value of L is maximized.

Ra is the radius of pipe in inches,
Rb is the average distance heat must travel to or from the pipe as it enters or leaves the reservoir. Rb=24 inches is a conservative estimate.

The volume of earth bed required for a particular storage application depends on the amount of storage needed and heat loss from the bed. A very rough estimate is given by

$$V = \frac{Q - \frac{K_d A(T_{max}-T_{min}.... -2T_e)/t}{2x}}{C_v(T_{max}-T_{min})} \qquad (6-4)$$

where V is the earth bed volume in ft^3,
Q is the amount of stored energy in Btu,
T_{max} is the maximum storage temperture in °F,
T_{min} is the minimum storage temperature in °F,
K_d is the thermal conductivity of the dry soil surrounding the earth bed in Btu/hr.ft.°F,
A is the surface area of the earth bed in ft^2
T_e is the temperature of the heat sink surrounding the earth bed, typically about 50-60°F,
C_v is the heat capacity of moist soil in $Btu/ft^3.°F$,
x is the mean distance from the earth bed to the surrounding heat sink in feet. A conservative estimate is the width of dry soil around the earth bed- about three ft,
t is the total storage time in hours.

Many of the operation and maintenance features discussed for rockbeds also apply for earthbeds. When designing an earthbed, refer back to the appropriate information for components, air pollution, and so forth.

6.6 Symbols Used

Main Symbols

A	area, ft^2 (m^2)
c	unit conversion factor
C_v	volumetric heat capacity, Btu/lb•°F (J/kg•°C)
F	correction factor
HP	power, horse-power (KW)
K	thermal conductivity in Btu/hr•ft•°F (W/m•°C)
η	efficiency, %
ΔP	pressure drop in H_2O (Pa)
Q	energy stored, Btu (J)
$\dot{Q}$	heat flow, Btu/hr (W)
q	volumetric flowrate, $ft^3/hr(m^3/sec)$
Ra	radius, in. (cm)
T	temperature, °F (°C)
t	time, hr (sec)
V	volume, ft^3 (m^3)
x	distance, ft. (m)

Subscript

d	dry
e	earth
m	moist
max	a maximum quantity
min	a minimum quantity
p	pipe
r	room temperature
o	conversion to absolute scale

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7.0 PHASE CHANGE HEAT STORAGE SYSTEMS

Phase change heat storage systems gain or lose heat by changing phase (for example a solid melting to a liquid) without changing temperature. The important feature of phase change materials (PCMs) is that the energy absorbed or released during the phase change is large; in a sensible heat storage medium the same amount of energy would cause a temperature rise of many tens of degrees. Phase change storage systems are desirable when small storage volumes and a limited range of storage temperatures are required. In a heating system, the phase change unit would take the place of either the water tank in Figure 2-2 or the rock bed in Figure 2-3. The volume of the PCM unit would be much smaller, however, making it especially attractive in retrofit situations by allowing greater flexibility in siting the storage location.

Various types of phase transitions, such as melting-freezing, solid-solid, and boiling-condensing, as well as many phase change materials have been investigated. Despite this, only a small number of phase change heat storage systems are commercially available. These are all of the melting-freezing type. Phase change systems can also be used for cold storage, which is discussed in Section 2.6 and Chapter 8. General features of phase change storage systems are discussed in Section 2.5. Although PCMs are also very useful in passive heating and cooling systems, this use of PCMs is not discussed in this handbook.

7.1 Types of Phase Change

7.1.1 Melting-Freezing

Melting-freezing is the only type of phase change heat storage system that is commercially available. For this type of phase change, heat is added to storage when the PCM melts, and heat is removed from storage when the PCM freezes. The melting-freezing phase change is characterized by the latent heat of fusion, which is the amount of heat that can be stored per unit mass of PCM for a complete phase change. This means that materials with a large latent heat will generally make a compact storage unit (see Table 7-1).

Two general classes of common melting-freezing PCMs are: (1) organic compounds, such as paraffins, plastics, and some organic acids (such as stearic acid); and (2) inorganic salt hydrates, such as Glauber's salt, calcium chloride hexahydrate, and sodium thiosulfate pentahydrate. Mixtures of organic compounds are possible, but the phase change is then spread over a larger temperature range. A large number of inexpensive salt hydrates are available, providing a wide range of melting point temperatures to choose from. However, as discussed below, the behavior of salt hydrate systems is complicated by the different types of melting behavior possible. Methods of dealing with these complications are discussed in Section 7.3.

Melting behavior, melting temperature, heat of fusion, and several other properties are listed in Table 7-1 for a few common melting-

Table 7-1. Latent Heat Storage Materials

Material	Melting Point, °F (°C)	Heat of Fusion, Btu/lb (kJ/kg)
Calcium Chloride Hexahydrate[c,d] $CaCl_2 \cdot 6H_2O$	84.9 (29.4)	73.1 (170)
Sodium Carbonate Decahydrate[c,d]	91.0 (33)	108.0 (251)
Disodium Phosphate Dodecahydrate[c,d]	97.0 (36)	114.0 (280)
Sodium Sulfate Decahydrate[c,d] $Na_2SO_4 \cdot 10H_2O$ (Glauber's Salt)	90.3 (32.4)	109.0 (253)
Sodium Thiosulfate Pentahydrate[c,d] $Na_2S_2O_3 \cdot 5H_2O$	120.0 (49)	86.0 (200)
N-Octodecane[d]	82.4 (28.0)	105.0 (243)
N-Eicosane[d]	98.1 (36.7)	106.0 (247)
Polyethylene Glycol 600[d]	68-77 (20-25)	63.0 (146)
Water[d]	32.0 (0.0)	143.1 (333.4)
Tristearin[d]	133.0 (56)	82.1 (190.8)
Sunoco 116 Paraffin Wax[e]	116.0 (47)	90.0 (209.2)
Sodium Sulfate Decahydrate/Sodium Chloride/Ammonium Chloride Eutectic[e]	55.0 (13)	78.0 (181.3)
Barium Hydroxide Octahydrate $Ba(OH)_2 \cdot 8H_2O$	180.0 (82)	114.0 (265)
NaF/MgF_2[j]	1530 (832)	269.0 (625)
NaOH[j]	608.0 (320)	68.0 (159)
Cross-Linked High-Density Polyethylene	270.0 (132)	99.0 (230)

[a] C = congruent, S = semicongruent, I = incongruent, SC = tendency to supercool

[b] liquid

[c] Source: M. Telkes. Solar energy storage. ASHRAE Journal 16:44, September 1974.

[d] Source: United States National Aeronautics and Space Administration, George C. Marshall Space Flight Center, Alabama. Phase Change Materials Handbook. NASA CR-61363. Prepared by D. V. Hale, M. J. Hoover, and M. J. O'Neill, Lockheed Missiles and Space Company, Huntsville, Alabama, September 1971.

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[f] Source: Frank Baylin. Low Temperature Energy Storage: A State-of-the-Art Survey. SERI/RR-54-164D, Solar Energy Research Institute, Golden, CO, March 1979.

Table 7-1. (Continued)

Melting Behavior[a]	Solid Density, lb/ft^3 (kg/m^3)	Heating Capacity--Solid, Btu/lb·°F (kj/kg·°C)	Heating Capacity--Liquid, Btu/lb·°F (kJ/kg·°C)
s[f], sc[g]	102.0 (1630)	0.320 (1340)[i]	0.552 (2310)[i]
I[g], sc[g]	89.9 (1440)	-	-
c[d], s[f], sc[g]	95.0 (1520)	0.404 (1690)	0.464 (1940)
I[f]	91.2 (1460)	0.459 (1920)[h]	0.779 (3260)[h]
sc, s[f]	106.0 (1690)	0.60 (2510)[e] 0.346 (1450)[i]	- 0.570 (2389)[i]
c	50.8 (814)	0.515 (2160)	-
c	53.4 (856)	0.528 (2210)	0.481 (2010)
c	52.9[b] (847)	0.399 (1670)[i]	0.550 (2300)[i]
c, sc	57.24 (916.8)	0.487 (2040)	1.00 (4210)
c	53.8 (862)	-	-
c	49.0 (785)	0.691 (2890)[h]	-
	-	-	-
	-	-	-
	-	-	-
	-	-	-
	60.0 (960)	0.60 (2500)	-

[g]Source: United States Energy Research and Development Administration. Solar Energy Subsystems Employing Isothermal Heat Storage Materials. ERDA 117. Prepared by George A. Lane et al., Dow Chemical Company, Midland, MI, May 1975.

[h]Source: J. J. Jurinak and S. I. Abdel-Khalik. Sizing phase-change energy storage units for air-based heating systems. Solar Energy 22 (4): 355-359, 1979.

[i]Source: Charles D. Hodgeman, Editor. Handbook of Chemistry and Physics. Chemical Rubber Publishing Co., Cleveland, OH, 30th Edition, 1946.

[j]Source: P. G. Grodzka. Phase Change Storage Systems, p. 795-809, in W. C. Dickinson and P. N. Cheremisinoff, Eds., Solar Energy Technology Handbook, Marcel Dekker 1980.

[k]Source: C. J. Swet. Energy Storage for Solar Applications, pg. 6.1-6.16 in J. F. Kreider and F. Kreith, Eds., Solar Energy Handbook, McGraw-Hill, 1980.

freezing PCMs. Phase change behavior of salt hydrates is complicated because hydration-dehydration can occur in addition to freezing-melting. The four types of salt hydrate melting behavior listed in Table 7-1 are congruent melting, semicongruent melting, incongruent melting, and super cooling. These classifications are defined and discussed in the following paragraphs.

Congruent Melting

PCMs that do not change their chemical nature when they melt, such as water or paraffin wax, melt congruently. When a salt hydrate melts, some or all of the water of hydration separates from the salt molecules. If the salt is completely soluble in its water of hydration, it will rehydrate easily when heat is removed from it. In this case, the salt melts and freezes congruently. Congruent melting is desirable in a PCM because the material will readily change from its liquid phase back to its solid phase as heat is withdrawn and will therefore give up all of its heat of fusion, even after many freeze-melt cycles. This is a condition of optimal conversion efficiency where the maximum fraction ($\eta = 1.0$) of stored heat is recovered from the system. Unfortunately, none of the congruently melting salt hydrates make suitable storage materials because of cost or safety considerations.

Incongruent Melting

Incongruent melting occurs when the salt is not completely soluble in its water of hydration. In this case the undissolved salt can settle to the bottom (precipitate) due to its higher density. When the remaining solution freezes, the hydrated salt precipitates on top of the dehydrated salt so that the dehydrated salt and the liquid are not in physical contact. This phenomena is illustrated in Figure 7-1. When the system remelts, only the layer of hydrated salt will melt, and more dehydrated salt will precipitate from it during the next freezing period. If this process is allowed to occur, some heat storage capacity will be lost with each freeze-melt cycle as more and more salt precipitates. Glauber's salt ($Na_2SO_4 \cdot 10H_2O$) is an example of an incongruently melting salt.

Semi-Congruent Melting

Semi-congruent melting occurs when a PCM has two or more hydrate forms, each with a different solid composition and melting temperature. Usually the design temperature of a storage system is that for the hydrate with the lowest melting point. If a hydrate with a higher melting point is formed, this salt settles out in much the same way as the anhydrous salt in the incongruent melting, discussed above. Unlike the incongruently melting system, the energy storage available from the phase change can be restored by raising the system to the higher melting temperature for an extended time period. An example of a semi-congruently melting salt is $CaCl_2 \cdot 4H_2O + \cdot 6H_2O$.

Supercooling

Supercooling is a melting behavior problem that occurs with some PCMs. As heat is taken out of the PCM, its temperature can drop below its

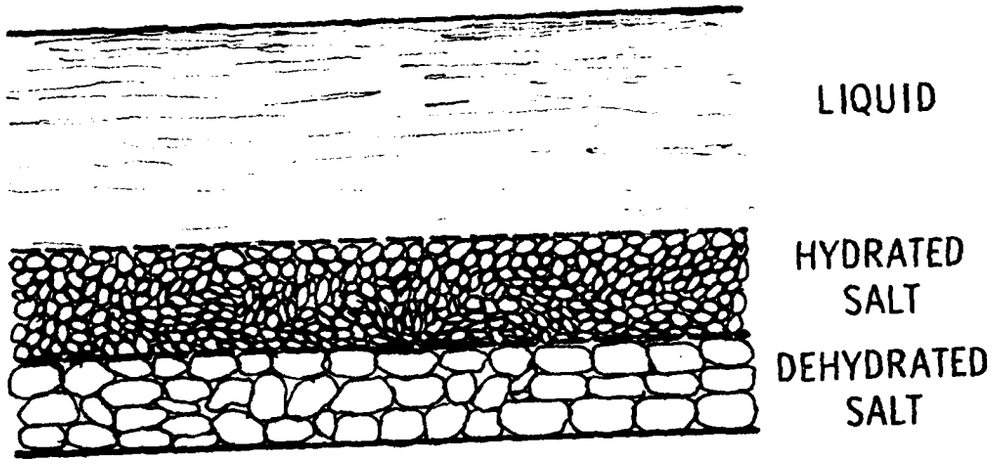


Figure 7-1. Selective precipitation in incongruent melting.

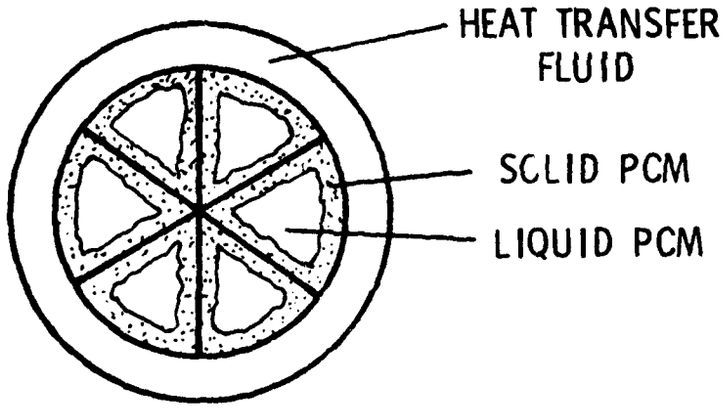


Figure 7-2. Increasing heat exchanger surface area by fins (see Section 7.3.4).

normal melting point without a change in phase. The temperature will continue to drop until a crystal of solid PCM forms. At that point, rapid crystal growth occurs and the temperature increases to the normal melting point. The PCM then reverts to its normal freezing behavior. Two problems arise from supercooling: the system may cool below the point at which useful heat can be extracted, and the supercooled condition may give the controller a false signal that storage is completely discharged when, in fact, it is almost completely charged. An example of a PCM that supercools is $Na_2S_2O_3 \cdot 5H_2O$ (sodium thiosulfate pentahydrate, also known as photographic hypo).

7.1.2 Solid-Solid

Some solid materials undergo an isothermal change in crystalline structure at certain temperatures. Changing from one solid crystalline structure to another requires a heat gain or loss. Examples include anhydrous sodium hydroxide and anhydrous sodium sulfate. These materials have been proposed for high temperature thermal storage.

High density polyethylene (HDPE) has a melting temperature of 266°F (130°C). Cross linked HDPE is under experimental investigation at Argonne National Laboratory as hot side storage for solar air conditioning. At the HDPE melting temperature, cross linked HDPE does undergo a phase change equivalent to melting, with a latent heat of fusion of approximately 80 Btu/lb, but maintains its shape. At the melting point, individual pellets have the consistency of gumdrops. Individual pellets stick together when they touch but do not fuse completely until well above the melting point.

Another solid-solid phase change is found for a group of organo-metallic compounds called layer perovskites. Experimental work indicates that these compounds will change phase repeatedly over a large number of cycles and have a large energy of transition. A large number of these compounds are known, and there is a wide choice of transition temperatures available in the range 40-250°F (4-120°C). (V. Busico et al., Solid-Solid Phase Transition for Thermal Energy Storage. In Thermal Storage of Solar Energy, ed. C den Ouden, pp. 309-324 (1980)).

Other solid-solid transition materials under investigation include pentaerythritol (transition temperature T=370°F (188°C)), pentaglycerine (T= 178°F (81°C)), and neopentyl glycol (T = 109°F (43°C)). Solid-solid transitions have the advantage of minimal volume change on transition. Another advantage is that the material can be used in a packed bed without encapsulating the PCM.

7.1.3 Boiling-Condensing

When a liquid boils, a large amount of energy is needed to make the phase change from liquid to gas. Unfortunately the volume increase from liquid to gas is large and it is expensive to contain the heated vapors. For this reason the boiling-condensing phase change is not used for heat storage.

7.2 Desired Properties of Phase Change Materials

The properties of PCMs discussed in the rest of this chapter are generally those of freezing-melting materials. However, some of the discussion is also applicable to other types of phase change.

The desired properties of phase change materials can be summarized as:

(1) high heat of phase change transition

(2) high thermal conductivity

(3) high heat capacity

(4) appropriate and sharply defined transition temperatures

(5) little or no supercooling

(6) high rate of crystallization

(7) low phase-change volume change

(8) low coefficient of thermal expansion

(9) chemical stability (no loss of storage capacity over time)

(10) low vapor pressure (allowing low-pressure design and better chemical stability)

(11) low corrosiveness

(12) low toxicity

(13) non-flammability

(14) low cost

Heat Capacity

If the volume of the storage container is known, then the amount of PCM needed is calculated from

$$m = \rho V, \qquad (7\text{-}1)$$

where m is the mass of PCM in pounds (kilogram)
ρ is the density of the PCM in pounds per cubic feet (kilograms per cubic meter)
V is the storage container volume in cubic feet (cubic meter)

Since the density of the PCM as a solid may be somewhat different then the density as a liquid, allowance must be made for expansion on phase change. Therefore, the smallest value of density that is to be encountered during the charge-discharge cycle should be used in Equation (7-1).

Besides storing latent heat, some phase change systems are allowed to store sensible heat by cooling below or heating above the melting temperature. For these systems, the heat capacities of the solid and liquid PCM must be known so that the amount of PCM necessary to store a given amount of heat can be calculated. Although the amount of sensible heat stored in these systems is usually small compared to the amount of latent heat stored, it must be considered. Equation 7-2 clarifies this. Sizing of latent heat systems is discussed in section 14-3.

The amount of heat, Q, in Btu (joules), stored is:

$$Q = m\,[C_{ps}\,(T_m - T_{min}) + \eta h_f + C_{p_\ell}\,(T_{max} - T_m)] \qquad (7\text{-}2)$$

where:

m is the amount of PCM in pounds (kilograms).

C_{ps} and C_{p_ℓ} are the heat capacities in Btu per pound per degree Fahrenheit (joules per kilogram per degree Celsius) of the solid and liquid PCM, respectively.

T_m, T_{max}, and T_{min} are the melting point temperature, the maximum storage temperature, and the minimum storage temperature in degrees Fahrenheit (degrees Celsius).

h_f is the latent heat of fusion in Btu per pound (joules per kilogram).

η is the fraction of theoretical latent heat that can be recovered.

Thermal Conductivity

A high thermal conductivity in both the liquid and solid phases of the PCM is desirable to achieve good heat exchange during charging and discharging. The thermal conductivity of the liquid phase is usually high enough that it does not create a problem, but the thermal conductivity of the solid phase tends to limit the rate at which heat can be removed from the system. It is also important that the solid PCM maintain good thermal contact with the walls of any heat exchanger.

7.3 Characteristics of Freezing-Melting Phase Change Heat Storage Systems

In order to design or choose a good PCM system, it is necessary to account for the special problems associated with PCMs. For example, usually

the cost of the container and the heat exchanger in PCM systems are much higher than the cost of the PCMs themselves. The special PCM pros and cons are discussed below with reference to the four most common PCM materials: paraffins, Glauber's Salt, sodium thiosulfate pentahydrate and calcium chloride hexahydrate.

7.3.1 Containers

PCM containers must be able to withstand a large volume change upon phase change and must be able to resist the corrosiveness of the PCM. The container can either be a single tank or many small containers enclosed in a large tank. The size and shape of the container is usually determined by heat transfer considerations.

The inorganic salt hydrates and the organic acids tend to be corrosive. WARNING: several of the salt hydrates react violently with aluminum or magnesium, and calcium chloride is especially corrosive. These substances can be held in plastic or plastic-lined containers however. Waxes are generally inert unless they contain impurities. Generally the manufacturer will have sealed the system but since some materials are rather caustic be cautious while handling the units. Properties of the potentially hazardous contained chemicals can easily be looked up in chemical handbooks.

Latent heat storage containers need less insulation than sensible heat storage containers storing the same amount of energy. This is due to the reduced surface area. Container heat loss is proportional to surface area. Thus, a PCM container with half the surface area of a sensible heat storage container would have half the heat loss, if both containers are at the same temperature and have the same thickness of insulation. If the same amount of heat loss can be tolerated from the PCM system, the insulation need be only half as thick. Since PCM systems operate at lower temperatures than sensible systems, the thickness of the insulation can be further reduced.

7.3.2 Nucleation Enhancement

While little supercooling is observed in paraffins, it is a serious problem for salt hydrates. Supercooling can usually be avoided if one or more nucleation centers are in the system. In large single tank systems, cold fingers are often used. In this method, one section of the tank is always kept well beneath the freezing point so that there are always some solid crystals in the tank. The solid crystals serve as nucleation centers for the rest of the PCM to freeze around.

An alternative method, used especially when the PCM is packaged in many small containers, is to mix nucleating agents with the PCM. Nucleating agents are chemicals with crystal shapes and internal lattice structures similar to the PCM, but with a higher melting point. When the freezing temperature is reached, the PCM starts to crystallize around the nucleating agents. Borax is often used as a nucleating agent for Glauber's salt.

7.3.3 Separation Suppression

Precipitation of salt from the liquid is a problem for incongruent or semi-congruent PCMs. Three methods are commonly used to prevent the separation of salt and water:

(1) the PCM is stirred by rotating the container, bubbling an immiscible fluid through the PCM, or pumping the melted PCM.
(2) Thickeners prevent the settling of salt. Both clay and silica gel derivatives have been used for this purpose with Glauber's salt. Diatomaceous earth has been reported successfully used with calcium chloride hexahydrate (L. E. Bourdeau, Study of Two Passive Solar Systems Containing Phase Change Material for Thermal Storage, Proc. 5th National Passive Conf., pp. 297-301, Amherst (October, 1980)).
(3) An alternative way to suppress the separation of salt and water is to add enough water to the system so that all the salt is soluble. This method decreases the amount of energy stored per cubic foot of container, however, and it does not work for all salts.

Calcium chloride hexahydrate can be made congruent by the addition of a few percent by weight of either strontium chloride hexahydrate or calcium hydroxide. These added chemicals slightly change the solubility of the salts. Strontium chloride additionally serves as a nucleating agent. A PCM of this type is marketed by Dow Chemical Company and is the PCM used in most commercially available calcium chloride PCM systems.

7.3.4 Heat Transfer

To take advantage of the constant temperature storage properties of PCMs, it is desirable to keep the temperature difference between the PCM and working fluid for either the heat source or heat load to a minimum. This is difficult to do, because during the freeze cycle, the PCM first freezes against the cold heat exchanger wall. After an initial period of freezing, it is more difficult for heat to flow from the liquid PCM, which still has latent heat energy, through the solid PCM, which has lost its latent heat energy, to the heat exchanger wall. This is because the solid PCM has a lower thermal conductivity than the liquid PCM. This problem can be further exacerbated if the solid PCM shrinks, leaving an air gap between the PCM and the heat exchanger wall. Air has an even lower thermal conductivity than the solid PCM. This problem demands a large surface area and thin walls for the heat exchanger. Large surface area usually means high cost, and thin walls often do not meet the problems of corrosiveness and volume expansion. Four methods are used to provide large surface areas:

(1) The heat exchanger is made of inexpensive extruded plastic tubing coiled in the tank. The flexibility of the tubing allows the PCM to expand and contract without causing damage.

(2) The PCM is microencapsulated, usually in thin, flexible plastic. The capsules, often held on trays, are then bathed in the heat exchange fluid, which can be either air or liquid.

(3) The container is filled with fins, conducting honeycomb, or metal mesh. The largest distance from PCM to heat exchanger wall is then minimized. The fins could be placed in one side of a shell and tube heat exchanger. An example of this method is illustrated in Fig 7-2.

(4) An immiscible fluid, such as air or oil, is bubbled through the PCM. This direct contact heat exchanger has the additional advantage of stirring the PCM, helping to prevent the phase separation of incongruent melting. A potential problem with this method is that precipitating salt could form in and around the pipes that introduce the immiscible fluid, clogging them.

(5) Direct contact/conventional heat exchanger hybrid. An almost 100% efficient phase change heat storage battery (100,000-Btu) has been developed that can operate without attendant degradation of the PCM heat transfer surface conductance (B. E. Swaiden, Dissolved Salts Thermal Energy Storage, CEN TM No. M-63-80-17, November 1980). Heat transfer rates achieved by exploiting mixing emulsion boundary layer techniques are far higher than for other types of heat exchangers. Other advantages of this novel heat exchanger/storage battery concept include no scaling of critical heat transfer surfaces and constant temperature heat extraction.

7.4 Available Systems

Melting-freezing transitions have been the focus of most PCM commercialization efforts and will be exclusively considered in this section. The latent heat storage system market is rapidly developing; new systems appear frequently. For this reason, both typical systems that are now commercially available and those that appear to be likely candidates in the near future will be discussed. Inclusion of representative different products is for information only; it does not constitute an endorsement or recommendation.

7.4.1 Modular Commercial Systems

The existing modular commercial systems essentially can be found in three different configurations: (1) pipes, (2) flat trays, and (3) round drums.

(1) Pipes:

Pipe Systems Incorporated (Fenton, Missouri) manufacture plastic pipes filled with calcium chloride hexahydrate and stabilizing ingredients, which have a melting temperature of 81°F (27°C). The pipes are six feet (1.8 meters) high with a 3-1/2 inch (8.9 centimeter) outside diameter. They have a four inch (10 centimeter) flange at each end so that when they are stacked together in an insulated bin, there is adequate room for air flow around them. Each pipe contains 30 pounds (14 kilograms) of PCM with a heat of fusion of 82 Btu per pound (190 kilojoules per kilogram) and a sensible heat capacity of 0.53 Btu/lb.°F (2.22 kJ/kg.°C) in the liquid state.

Closely related to these products are the six foot long black polyethylene tubes manufactured by Energy Materials, Incorporated (Englewood, Colorado). These thermal energy storage rods are based on calcium chloride hexahydrate and have ten year warranties.

Energy Materials, Incorporated is now working with a new eutectic blend of magnesium nitrate hexahydrate and magnesium chloride hexahydrate which phase changes at 135°F and has a latent heat of 58 Btu/lb.

Boardman Energy Systems, Incorporated (Wilmington, DE) manufactures 30" (76 cm) long, 4" (10 cm) diameter, plated steel tubes, filled with sodium sulfate. The advantages of this system are built-in spacers and a phase change temperature flexibility (45°, 64°, 74°, 78°, 81°, 89°(F)).

(2) Trays:

At this time we know of no manufacturer still manufacturing these. Historically speaking .. they all leaked.

(3) Round Drums:

A third type of air-based system uses a paraffin wax in canisters. Addison Products Company (Addison, Michigan) uses a paraffin wax that melts at between 115°F (46°C) and 120°F (49°C) and has a heat of fusion of 80 Btu per pound (186 kilojoules per kilogram). In this system the canisters are stacked on shelves that also act as baffles. Air is forced to travel the full length of each shelf before passing to the shelf below (charging) or above (discharging). Each one gallon steel canister weighs 7.25 lb and has a 1 year warranty.

Texxor (Omaha, Nebraska) manufactures Texxor Heat Cell™ drums (containing $CaCl_2 \cdot 6H_2O$ and stabilizing agents) 7" long (17.8 cm) and 4.3" diameter (10.8 cm). These are characterized by thinner walls than the pipe modules-resulting in a higher heat flow.

7.4.2 Space-Saving Systems

The three modular configurations described above need extra space within the storage bin for air flow. There are two different categories on the market that take up less space. In the first class, phase change materials are incorporated into structural materials and form an integral part of

the building. This is an important area of active research. In the second class, phase change materials are used in bulk form rather than in individual containers. These systems are designed for use with liquid-based solar collectors.

Incorporated into Structural Materials

Architectural Research Corporation (New York, NY) manufactures SOL*AR-Tile (Ref. U.S. Pat. #4178727), a polymeric resinous concrete tile, 2 ft x 2 ft, containing a proprietary PCM core. Noted for its extreme durability for ceiling and floor applications and its unique heat storage function. SOL*AR-Tiles are used in MIT's Solar Building 5. Each tile weighs 44 lbs, and at the phase change temperature of 73°F has a heat storage capacity of 1,000 Btu.

Colloidal Materials, Incorporated (Andover, MA) manufactures Heat Pac™, a 3-ply aluminum foil laminate pouch, 3/4 in. by 2 ft. by 2 ft. These pouches containing sodium sulfate decahydrate and chemical enhancers. The pouches store about 200 Btu per square foot at standard phase change temperatures of 88°F, 73°F, and 66°F.

Kalwall Corporation (Manchester, NH) has just developed PCM translucent "window" storage. These thermal storage pods are now fully available for the marketplace.

Still in the developmental stages are: Maria Telke's (American Technological University, Killeen, TX) phase change material - Trombe wall hybrid; Southwall's (Palo Alto, CA) Thermocrete™ concrete-salt hydrate blocks, and S. Mumma's (Arizona State University) concrete-paraffin wax blocks. Within the next few years either these or other similar structurally-incorporated PCM systems should be on the market.

Bulk-form PCM Systems

The Solarmatic™ Division of OEM Products, Inc., (Tampa, Florida) produces a storage device called the Heat Battery™, as shown in Figure 7-3. The Heat Battery™ consists of an insulated tank containing a heat exchanger, Glauber's salt, a heat transfer fluid in which the salt will not dissolve or mix, a floating fluid segregator, and a series of pressure-controlled outlets for the heat transfer fluid.

When the system is being charged, an external pump forces the heat transfer fluid through the top outlets, which are located above the segregator, until the salt has melted down to the next level of outlets, freeing them for fluid flow. There is less pressure drop across the spring-loaded valve in this lower outlet than across the spring-loaded valve in the top outlet, so the fluid now flows out the lower outlet. The process can continue level by level until the entire tank is melted. For heat extraction, the process is reversed.

The movement of the heat transfer fluid keeps the Glauber's salt solution mixed, and because the two fluids are immiscible, the salt solution

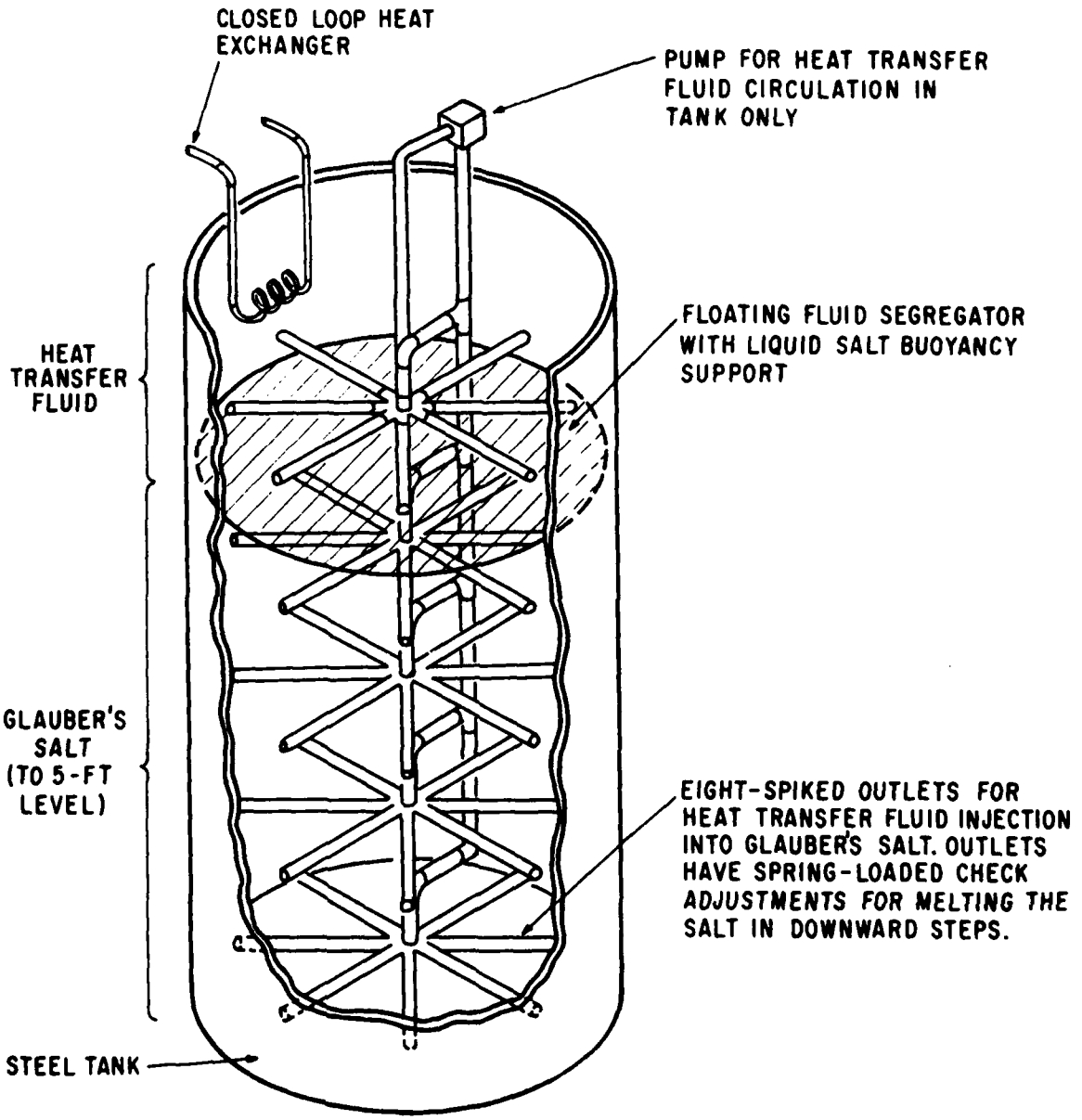


Figure 7-3. Heat Battery™, Produced by Solarmatic™ Division of OEM Products, Inc., Tampa, Florida

remains in small drops. Both of these effects minimize the incongruent-melting problem. The fluid segregator keeps the Glauber's salt solution from being carried along with the heat transfer fluid, where it could foul the pump. The Heat Battery™ is being developed as part of a solar-assisted heat pump system.

Calmac Manufacturing Corporation (Englewood, New Jersey) produces a latent heat storage system in which a tank contains both the PCM, which can be any of several materials, and a plastic tubing heat exchanger. The heat exchanger tubes are placed close enough together so that they will effectively transfer heat into the solidified PCM. The PCM in its liquid state is kept stirred by a pump. In one version, the pump motor stalls without harm when the PCM solidifies around the pump rotor. In another version, a small heater maintains a liquid state around the pump rotor.

Another system, developed by Thermal Energy Storage Systems, Inc., (La Jolla, California), is similar to the Calmac system except that it uses less heat exchanger tubing and uses chemicals to keep the sodium thiosulfate pentahydrate (hypo) from supercooling. An advantage of both systems is that as heat escapes through the insulation some of the PCM will solidify on the walls, in effect creating more insulation and reducing heat losses.

7.5 Cost Of Latent Heat Storage Systems

At present the costs of latent heat storage systems vary considerably. Lee, Taylor, DeVries, and Heibein (1979) list costs that vary from $5,000 to $18,500 per million Btu ($4.75 to $17.55 per million joules). These costs cover a range from partially complete, unassembled units, to complete storage systems. By comparison, sensible heat storage in a water tank costs about $2,000 to $7,500 per million Btu ($1.90 to $7.10 per megajoule)[1]. More recent contact with manufacturers suggests a clustering of several systems at about $10,000 to $12,000 per million Btu ($9.50 to $11.40 per megajoule), but the overall range is still large. Colloid Materials' sodium sulfate ceiling pouch, for example, cost $10,000 per million Btu (1980). As companies gain experience and competition, the range of costs can be expected to decrease.

Latent heat storage systems for heating and cooling applications are unlikely to compete in the near future with sensible heat storage systems solely on the basis of cost. Instead, the compactness of latent heat storage systems and their ability to enhance collector performance will be emphasized. Compactness will allow installation of latent heat storage containers in locations where sensible heat storage containers would be difficult or expensive to install.

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[1] The costs were calculated for a water tank cost of $1.00 to $2.50 per gallon ($0.26 to $0.66 per liter) and an operating temperature range of 40 to 60°F (22 to 33°C).

Enhancing collector performance by maintaining a low average collector temperature may lead to an overall reduction in system cost in some systems. The choice of latent versus sensible heat storage systems can only be made after analysis of the entire system's performance with each type of storage system. Recent versions of simplified analysis methods such as f-Chart are capable of analyzing systems using latent heat storage, as are the more elaborate simulation methods such as TRNSYS.

7.6 Symbols Used

Main Symbols

Cp	heat capacity in Btu/lb·°F (J/kg·°C)
f	fraction of heat lost in time t
h_f	latent heat of fusion in Btu/lb (J/kg)
m	mass of phase change material (kg)
Q	amount of heat stored in Btu (J)
T	temperature in °F (°C)
η	fraction of theoretical latent heat that can be recovered

Subscripts

ℓ	liquid PCM
m	melting point
max	maximum condition
min	minimum condition
s	solid PCM

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8.0 THERMAL STORAGE FOR COOLING

Thermal storage for cooling purposes can use either heat storage or cold storage systems. Heat storage systems were discussed in detail in Chapters 4-7. General features of cold storage systems were discussed in Sections 2.4 and 2.6. While most of the information in this chapter is relevant to thermal storage for any cooling system, the discussion in this chapter is based upon space cooling of buildings.

The type of storage system appropriate in a given situation depends strongly on the characteristics of the cooling system used. The design of the cooling system, in turn, will vary considerably depending on the building load and on whether waste heat, solar energy, or off-peak electricity is used as the energy source. Therefore, the first section of this chapter discusses commonly used space cooling methods. The second section discusses general types of storage appropriate to common cooling situations. The remaining sections discuss cold storage methods.

8.1 Types of Space Cooling Methods

A space cooling system must control both temperature and humidity to achieve a comfortable environment. The additional factor of humidity control places several restrictions on the types of systems used. This section describes general features of several commonly used space cooling methods. A more detailed discussion can be found in most air conditioning textbooks or manuals such as R. C. Gunther, Refrigeration, Air Conditioning, and Cold Storage or the Trane Air Conditioning Manual.

8.1.1 General Space Cooling Processes

There are three major ways in which air is cooled and conditioned:

(1) Sensible cooling occurs when the air temperature is lowered. The moisture content of the air remains the same, but the relative humidity increases.

(2) Cooling and dehumidification is simply sensible cooling continued below the dew point so that water vapor condenses out (like a cold drink sweating on a summer afternoon). Either a cooling coil or a spray of chilled water can be used to refrigerate and dry the air. Dehumidification is necessary in humid climates to keep the conditioned space within the comfort zone. Cooling to 60°F (16°C) is needed to obtain 50% relative humidity for an air temperature of 72°F (22°C). The cooling coil and cold storage must then be less than 60°F (16°C) to attain this condition.

(3) Cooling and humidification is another term for air conditioning by water evaporation. In other words, the heat energy in the air is exploited to change water from a liquid to a gas, thus cooling and moisturizing the air. As the relative humidity increases and the air temperature decreases in this process, the temperature at which the moisture evaporates (wet bulb temperature) remains constant. Evaporative cooling is ideal in situations where the outside air is warm and dry such as the southwestern part of the U.S., but it does not provide the dehumidification necessary in other parts of the country.

8.1.2 Absorption Cooling

Absorption chillers are currently the only commercially available machines that operate directly from a thermal source. This makes them an ideal match for solar collectors or waste heat sources. Absorption machines can be designed to operate at temperatures as low at 170°F (76°C) or as high as 300°F (150°C) depending on the choice of working fluids and other features. After the absorption machine has been designed, it typically operates over a 40°F (20°C) temperature range. This requires careful matching of the machine to the temperature of the heat source.

Systems that operate at lower temperatures do so at considerably lower efficiency. Concentrating collectors are required for the higher temperatures. However, at least one system has been reported to operate successfully with flat plate collectors as the heat source (D. Best, Hot contender in a cool market. Solar Age pp. 63-66, May, 1981).

Principle of Operation

Absorption chillers use absorbent/refrigerant pairs for the working fluid. Many fluid pairs have been investigated, but currently only two now are commercially used. The combination of ammonia as refrigerant and water as absorbent was used in early absorption machines, but now the combination of water as refrigerant and lithium bromide as absorbent is a practical alternative that operates at lower temperatures.

The essential features of the absorbent/refrigerant fluid pair are: (1) at low temperatures the two readily combine together, and (2) at high temperatures the refrigerant readily boils away from the combined liquid. For example, lithium bromide is a solid at room temperature but will readily absorb large amounts of water vapor. The mixture becomes a liquid once the water concentration reaches about 30%. At high temperatures (around 200°F (93°C)) the water readily boils away, leaving behind a solution of concentrated lithium bromide.

A schematic diagram of a typical lithium bromide/water absorption chiller is shown in Figure 8-1. The system consists of four basic stages: generation, condensation, evaporation, and absorption.

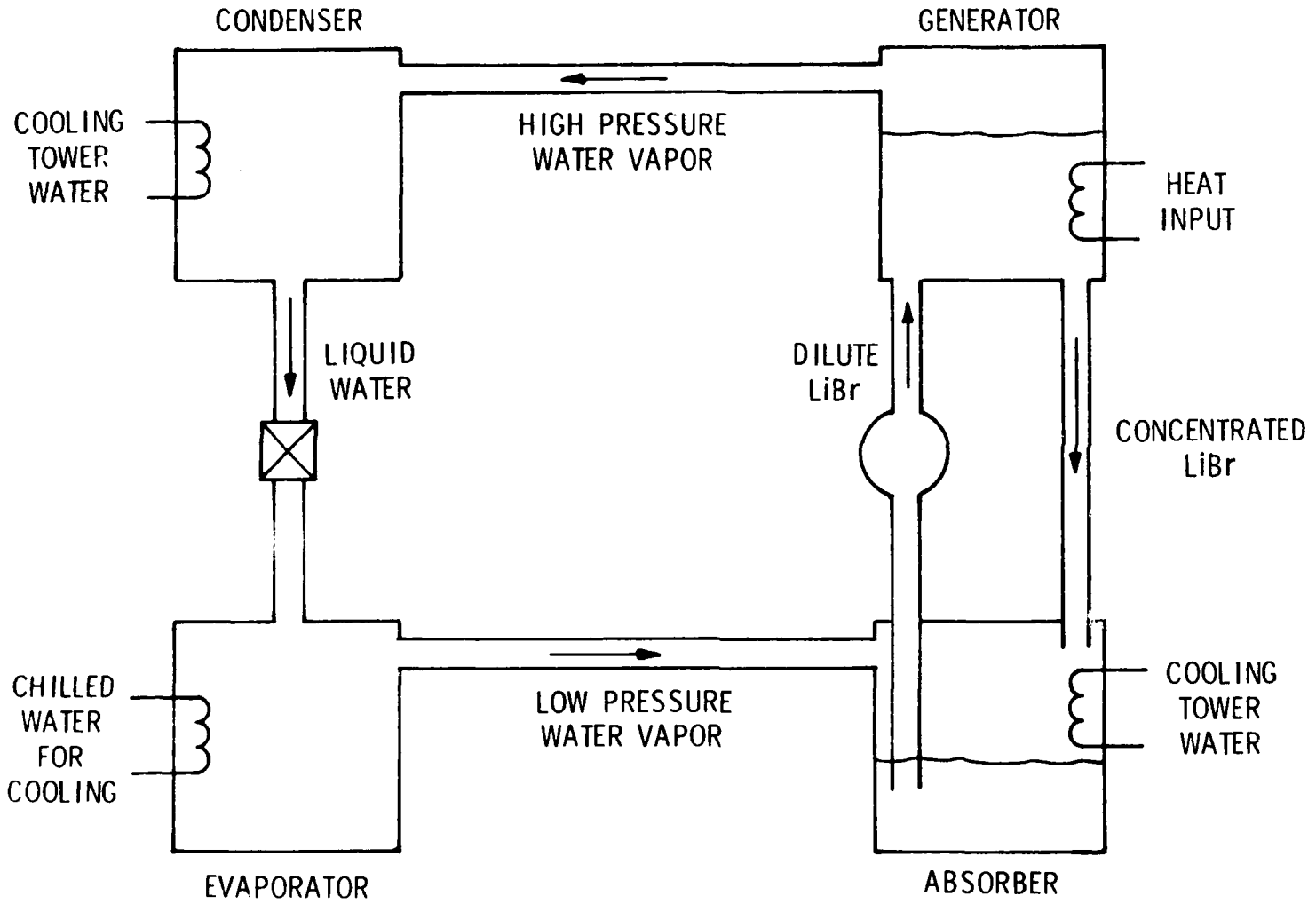
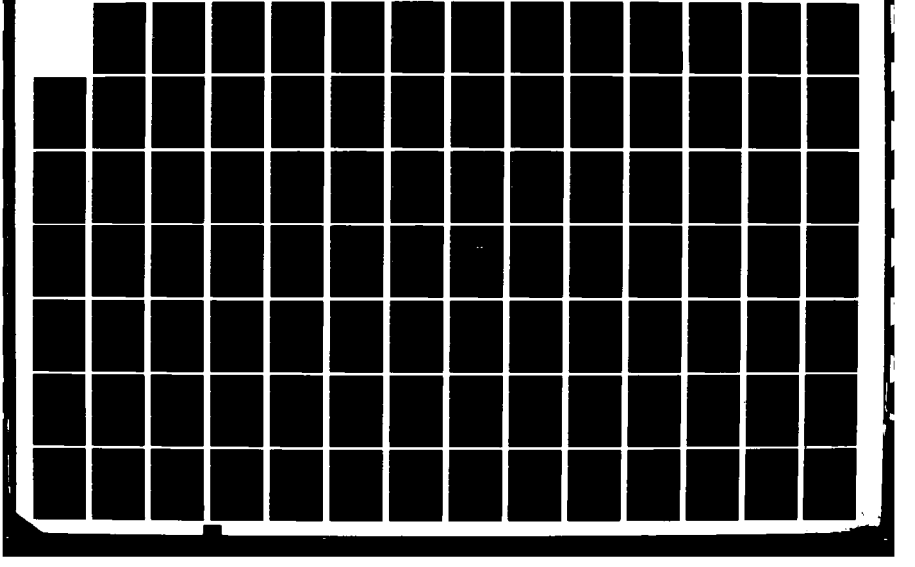
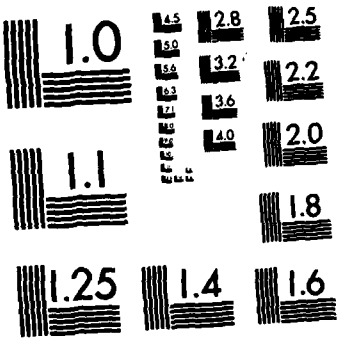


Figure 8-1. Typical Lithium Bromide/Water Absorption Chiller

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MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

(1) Generation: A dilute solution of lithium bromide is pumped to the generator from the absorber. In the generator, heat from storage or a heat source brings the dilute solution to a boil, producing a thick solution of lithium bromide and water vapor. The large amount of high temperature water vapor puts the generator at high pressure.

(2) Condensation: The high pressure - high temperature water vapor from the generator passes into the condensor, where it is cooled to a liquid (condensed) by a cooling water coil. The cooling water is produced in a cooling tower, which is not shown in Figure 8-1.

(3) Evaporation: Liquid water from the condensor passes through an expansion valve into the evaporator. The evaporator is at low pressure (near vacuum), so the water can vaporize (boil) at a low temperature. This process produces chilled water inside the evaporator heat exchange coil by removing latent evaporation heat surrounding the coil. The chilled water in the heat exchange coil then goes to cold storage or to the space to be cooled.

(4) Absorption: Water vapor from the evaporator passes into the absorber. At the same time, thickened lithium bromide solution from the generator is sprayed over the cooling water coil in the absorber. The cooled lithium bromide and the water vapor readily combine to form a dilute lithium bromide solution. The dilute solution is pumped to the generator to begin the cycle over again. Since the water vapor quickly combines with the cooled lithium bromide, vapor pressure in both the absorber and evaporator is low.

Not shown in Figure 8-1, is a heat exchanger between the inlet and the liquid outlet to the generator. The purpose of the heat exchanger, called a regenerator, is to improve the efficiency of the machine by recovering some heat that would otherwise be lost. Water cooling is usually used for cooling the condensor and absorber in lithium bromide/water chillers, requiring a water cooling tower in addition to the chiller. This is necessary because lithium bromide will crystallize at the higher absorber temperatures associated with air cooling. Since only liquid is being pumped, and since pressure differences between high and low pressure sides are small, little energy is needed for pumping, and some systems require no external pumping energy.

Water/ammonia absorption chillers are similar to lithium bromide/water machines, except that a rectifier or reflux system is needed at the top of the generator to prevent water vapor from getting into the condensor. Small amounts of water vapor are given off with the larger amounts of ammonia when the mixture boils. Water/ammonia systems require a higher generating temperature than lithium bromide/water systems, but the absorber can be air cooled.

Other absorber/refrigerant pairs that have been investigated include: sodium thiocyanate/ammonia and lithium bromide/methanol.

Considerations for Storage

An important feature of absorption chillers is that they completely stop functioning below a minimum generator temperature. The minimum temperature depends on individual system design and cooling water temperature. In addition, cooling capacity of a given system usually increases with increased generator temperature.

Lithium bromide/water systems operating off waste heat usually run at a high generator temperature (240-260°F (116-127°C)), with a cooling tower temperature above approximately 80°F (27°C). If solar collectors are the heat source, the system is run at lower generator temperatures (as low as 170°F (77°C)) and the cooling tower is allowed to operate at as low a temperature as possible. The reason for the lower temperatures in this case is that the solar collector efficiency is much higher at lower temperatures, more than offsetting the slight efficiency loss of the chiller.

The heat exchangers in absorption chillers usually take about an hour to completely reach operating temperature, where they run at maximum efficiency. A machine which is cycled on and off regularly will have a greatly reduced average cooling performance than a machine that runs continuously. Thermal energy storage coupled to the chiller can help to eliminate cycling, thus improving performance.

There are two possible sources of intermittent running: (1) intermittent heat source, as in the case of a solar collector, and (2) intermittent cooling load. If waste heat is the source, then storage of chilled water will eliminate most of the problems of an intermittent load. If solar collectors are the source, then heat storage may be necessary as well.

8.1.3 Vapor Compression

Vapor compression cooling machines are commonly used in refrigerators and air conditioners. They have a condensor and an evaporator as in the absorption chillers, but they use a mechanically driven compressor in place of the absorber and generator and use only one refrigerant fluid.

Liquid refrigerant leaves the condensor and flows through an expansion valve to the evaporator. Flow through the expansion valve lowers both the pressure and temperature. Thermal energy from the space to be cooled boils the refrigerant in the evaporator. The compressor then takes the refrigerant vapor and compresses it to high pressure before releasing the vapor back to the condensor. In the condensor the high pressure vapor is condensed back to liquid, releasing heat to a cooling source, usually the outside air. When the evaporator and condensor can be switched to operate from either the living space or outside, the living space can also be heated, and the system is commonly called a heat pump.

The compressor is driven by shaft power, usually supplied by an electric motor. Such systems are good choices when off-peak electricity is economical. Vapor compression machines obtain their best efficiency when the temperature difference between the cooling load and the condensing temperature is low. Most offpeak rates occur at night when outside temperatures are low. Using the vapor compression machine at night to cool cold storage then provides the double benefit of offpeak rates and increased efficiency.

Alternatively, the compressor shaft power could be produced by a heat engine, driven by a solar collector or waste heat source. For heat source temperatures below 600°F (300°C) only Rankine cycle engines have an useful efficiency. However, both Stirling cycle and Brayton cycles are usable at higher temperatures. A number of heat engine concepts are under investigation, but currently no heat-engine-powered vapor compression cooling system appears economically viable.

8.1.4 Adsorption-Desorption (Desiccant)

Another class of cooling systems receiving much research but not yet commercially viable are the adsorption-desorption, or desiccant systems. The adsorption-desorption cycle can also be used in heat storage systems, or combination heating and cooling systems, and is discussed for those applications in Chapter 9.

The basic process in a desiccant cooling system is: (1) dehumidify the incoming air, (2) evaporatively cool the dehumidified air, and (3) regenerate the desiccant (drive off the water) with a heat source. A typical desiccant cooling system is shown in Figure 8-2. Incoming air, passing through the hot desiccant, is dried. The dry air is first cooled by heat exchange with the building exhaust air. The intake air is then further cooled and rehumidified by an evaporative cooler before passing into the building. The exhaust air from the building is evaporatively cooled to improve the performance of the heat exchanger. Once the desiccant can no longer adsorb moisture, it must be switched to regeneration mode. Heat from storage of a source dries either building exhaust air or outside air. This hot dry air passes through the desiccant and desorbs moisture from it. Some systems use a slowly rotating wheel of desiccant to continuously switch from adsorption to desorption. If building exhaust air is used for regeneration, it can be vented back as intake air, thus forming a closed system.

Types of desiccant materials under investigation include silica gel, activated alumina, and zeolites (molecular sieves).

8.1.5 Other Methods

If night air temperatures are cool, night air can be ventilated through the building. This method can often provide many hours of cooling

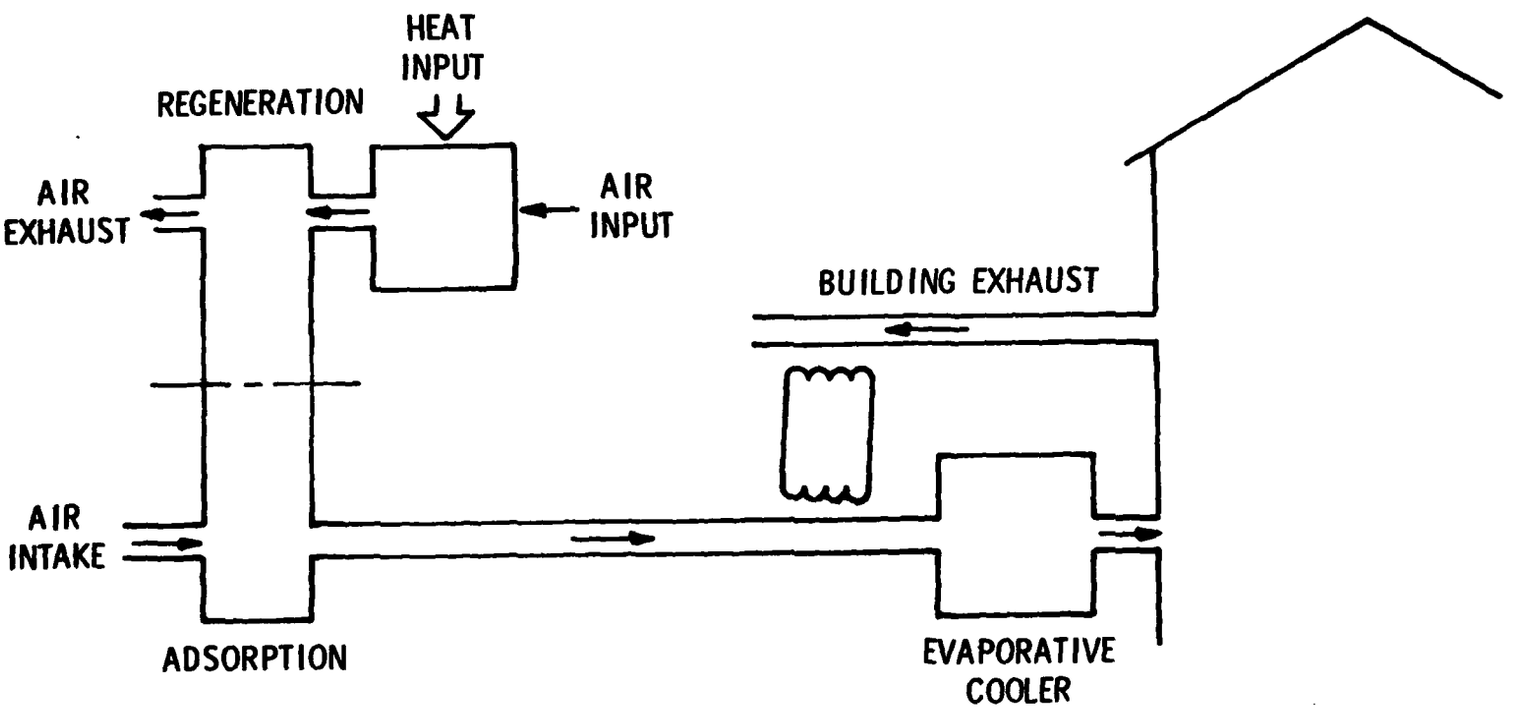


Figure 8-2. Typical Desiccant Cooling System

the next day if the heat capacity of the building is sufficiently large, but can lead to excessive humidity in many climates. Cool night air temperatures can also be used as a source for sensible cold storage. In relatively dry climates, temperatures as much as 20°F (7°C) below night air temperatures can be obtained by letting standard flat plate collectors radiate to the clear night sky. Standard storage methods such as water tanks and rock beds can be used. The advantage of this method is that the same solar equipment can also be used during the heating season.

Natural water sources such as wells, rivers, and lakes can be used for cooling. In some cases the water can be used directly for cooling. For example, in hot dry climates evaporative coolers are an inexpensive, commonly used, method. In other cases the water can be used indirectly, such as for condensor water or as a cool source for heat pumps. In the latter example, the heat pump efficiency is improved, because the water temperature is often lower than the ambient air temperature during times when cooling is needed. In most cases the water must be specially treated, as discussed in Section 8.6.

Aquifers can be used for seasonal cold storage in much the same manner as for seasonal heat storage, discussed in Section 5.3. The cold water to be injected into the aquifer is made by evaporative cooling in the winter. This method is attractive for areas with hot, humid summers, and cool, relatively dry winters; only a few experimental systems are currently installed.

8.2 Comparison of Storage Methods

8.2.1 Hot Side Storage

Hot side storage should be used whenever an intermittent heat source, such as a solar collector, is used to power cooling equipment. The use of thermal storage in cooling applications is not sufficiently developed to definitively compare the desirability of individual systems. Statements, however, relating system efficiency to the general type of storage used can be made using thermodynamic arguments. This section makes this comparison using the example of an absorption chiller coupled to a solar collector.

Figures 8-3 through 8-7 compare different types of hot side storage coupled between a solar collector and an absorption chiller. In the Figures, temperature is plotted against entropy flow for short periods of time in the operating cycle of the system. Arrows on the lines indicate the direction of flow. Note that the area between the solar collector line and the absorption chiller line is energy that is unavailable to the system. The more energy that is unavailable, the higher the collector temperature must be and the less efficient the system. Both cases of the collector transferring heat to storage and storage transferring heat to the chiller are shown in each Figure. A 10°F (6°C) temperature difference is assumed across a given heat exchanger.

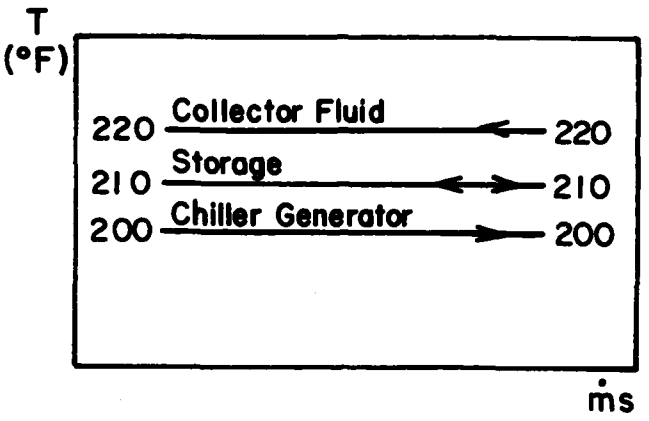


Figure 8-3. Hot-side Storage Between Solar Collector and Absorption Chiller: All-latent heat system

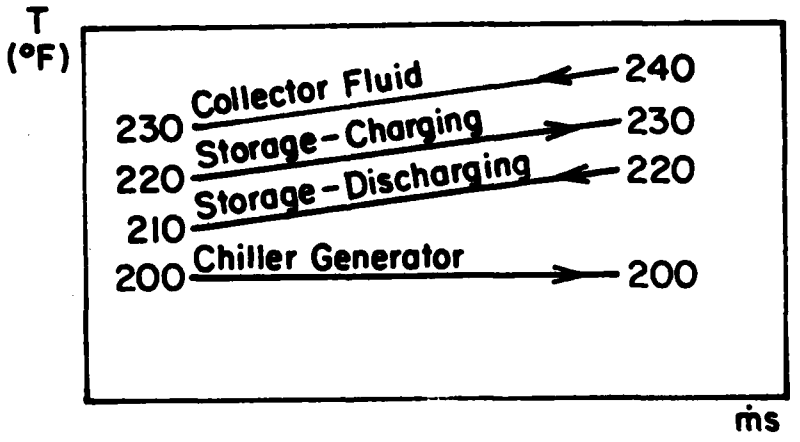


Figure 8-4. Mixed Sensible Heat Storage System: completely discharged

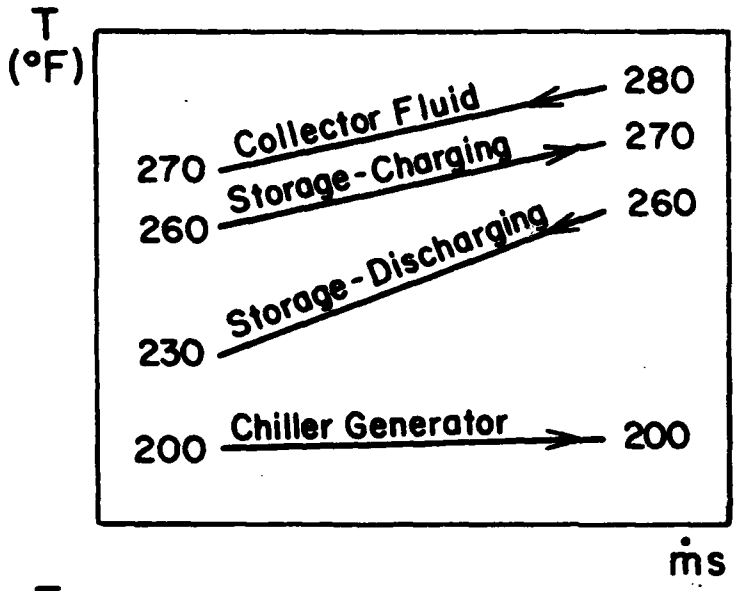


Figure 8-5. Mixed Sensible Storage.. Storage Fully Charged.. Average Collector Temperature = 275°F.

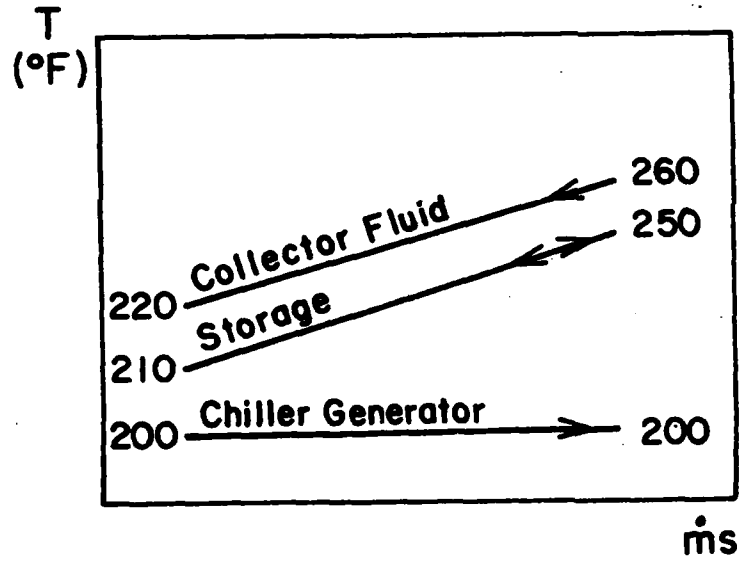


Figure 8-6. Stratified Sensible Storage.. Average Collector Temperature = 240°F.

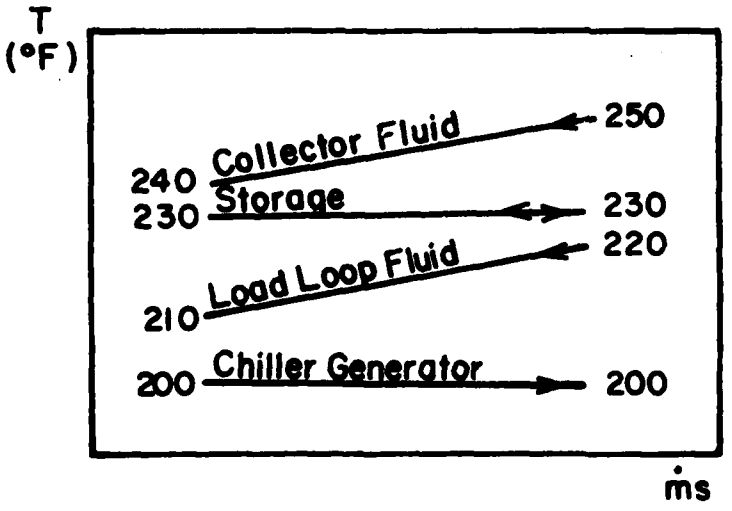


Figure 8-7. Latent Heat Storage with Sensible Heat Transfer.. Collector Temperature = 245°F.

In Figure 8-3, representing an all-latent heat system, the collector is giving up latent heat (e.g. pressurized steam condensing) and losing energy, while the PCM storage gains energy, all at constant temperature. If the storage is transferring energy to the chiller generator, storage loses energy while the generator gains energy, again with each side of the heat exchanger at constant temperature.

Figures 8-4 and 8-5 represent a sensible heat storage system where the storage is completely mixed, i.e. at the same temperature throughout the storage tank. In Figure 8-4, storage is at its lowest possible useful temperature, i.e. completely discharged at 220°F (104°C). The collector can charge the storage system if the average temperature of the collector is 235°F (113°C). Collector fluid enters the heat exchanger at 240°F (116°C) and leaves at 230°F (110°C). On the storage side of the heat exchanger, fluid enters at 220°F (104°C) and leaves at 230°F (110°C). The 230°F fluid leaving the heat exchanger immediately mixes with the rest of the fluid in the tank, leaving the tank slightly hotter than 220°F, due to the extra amount of energy from the collector. On the other hand, if storage is transferring heat to the generator, the fluid enters the heat exchanger at 220°F (104°C) and leaves at 210°F (99°C). The fluid leaving the heat exchanger immediately mixes with the rest of storage, which is now slightly less than 220°F.

Figure 8-5 is the same mixed sensible storage system as in Figure 8-4, but now storage is fully charged. The temperature of the storage tank is 260°F (127°C). It now takes an average collector temperature of 275°F (135°C) to add more energy to the tank. Since there is now a large temperature difference between storage and the generator, a larger temperature drop is experienced by the storage fluid passing through the heat exchanger between storage and the generator. The average collector temperature in the mixed sensible storage system should be the average of the two extreme cases of completely charged and completely discharged - in this case 255°F (124°C).

Figure 8-6 represents a stratified sensible heat storage system. The same storage temperature difference (40°F (22°C)) is used as in the mixed storage example. Storage fluid enters the heat exchanger connected to the collector, from the bottom of the tank at a temperature of 210°F (99°C). It is discharged from the heat exchanger to the top of the tank at a temperature of 250°F (121°C). Similarly, with the heat exchanger connected to the generator, storage fluid enters the heat exchanger from the top of the tank at 250°F and is discharged to the bottom of the tank at 210°F. The average collector temperature in this case is 240°F (116°C).

Figure 8-7 represents a latent heat storage system with sensible heat transfer both from the collector and to the generator. In this case the storage temperature is 230°F (110°C) and the average collector temperature is 245°F (118°C).

The hypothetical systems of Figures 8-3 to 8-7 are summarized in Table 8-1, which lists the average collector temperature in each example. The larger the temperature difference between the collector and the generator, the larger is the amount of collected heat that is lost from the

system. An all-latent heat system would clearly be best, if such a system existed (it currently does not). Next best is a stratified sensible storage system, although a latent heat storage system with sensible heat transfer is almost as good. Fortunately, the worst case - mixed sensible storage - is easy to avoid, as discussed in Chapter 5.

Table 8-1. Summary of average collector temperatures

System	Temperature (°F)	(°C)
All latent heat	220	(104)
Stratified sensible	240	(116)
Latent storage - sensible transfer	245	(118)
Mixed sensible	255	(124)

8.2.2 Cold Side Storage

Cold side storage should be considered whenever an intermittent load would cause an absorption chiller to cycle on and off at short intervals or where favorable off-peak electric rates can be applied to vapor compression refrigeration. An alternative to cold side storage which can be used with some absorption machines is modulation of the air conditioning capacity, that is, reducing the pump speed, etc., to reduce the amount of air conditioning delivered under reduced load.

Statements about the efficiency of general types of cold side storage can be made in an analogous way to the previous section. As an example we consider the condenser of a chiller which removes heat from cold side storage. The storage in turn must be capable of reducing 80°F (27°C) air to 55°F (13°C). Several hypothetical systems are represented in Figures 8-8 through 8-12. In the examples a minimum temperature difference of 10°F (6°C) is allowed for the building-to-storage heat exchanger, while a minimum temperature difference of 5°F (3°C) is allowed for the chiller-to-storage heat exchanger.

Figure 8-8 represents a stratified cold storage system using water as the storage fluid. Water enters the heat exchanger connected to the chiller from the top of the tank at 70°F (21°C) and is returned to the bottom of the tank cooled to 45°F (7°C). Water enters the heat exchanger connected

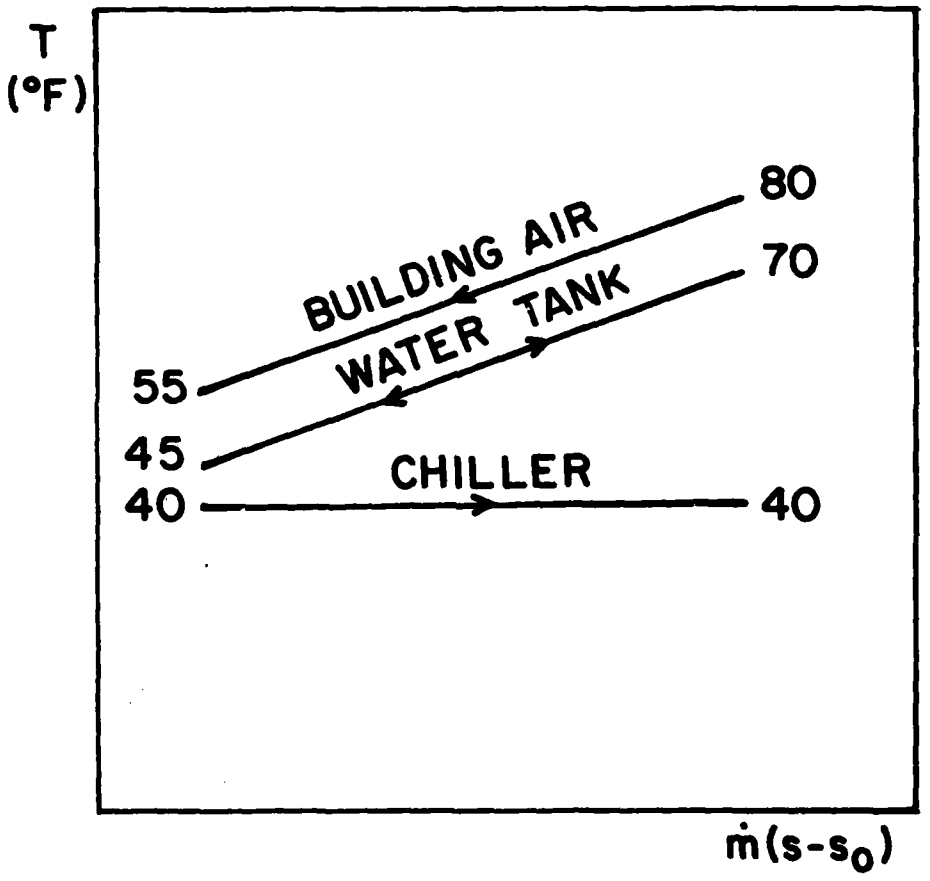


Figure 8-8. Stratified Sensible Heat Tank System

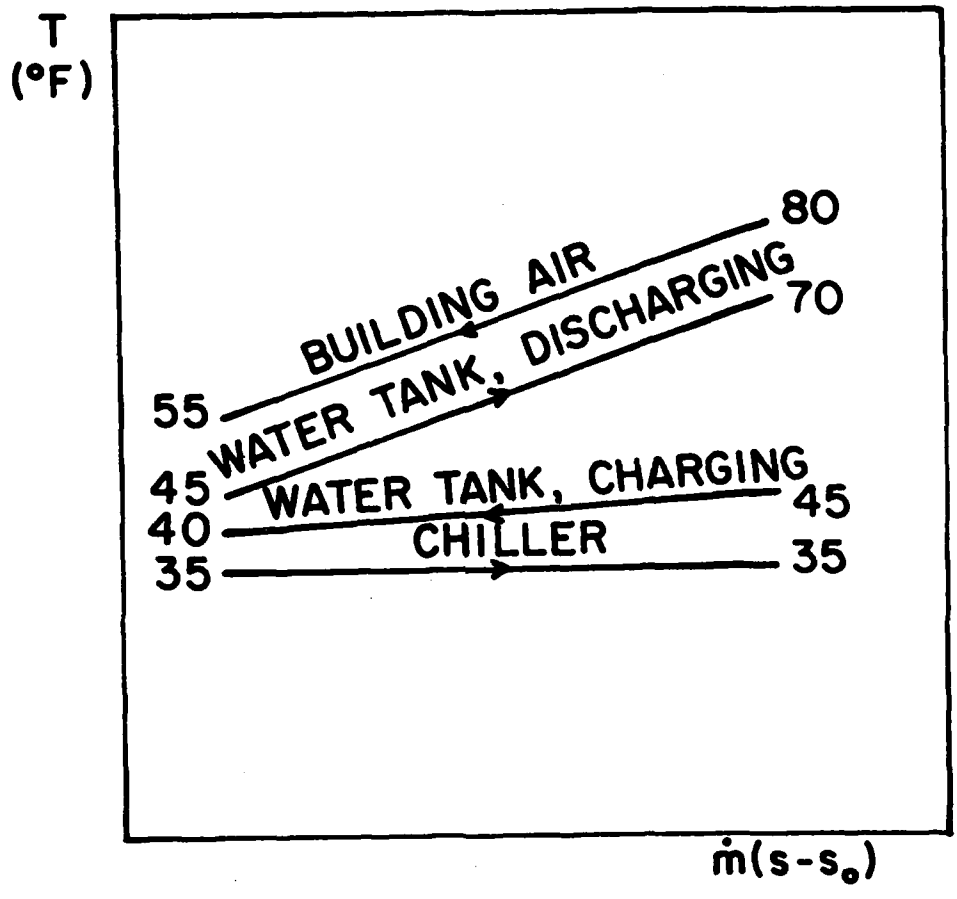


Figure 8-9. Discharged Mixed Sensible Heat Tank System

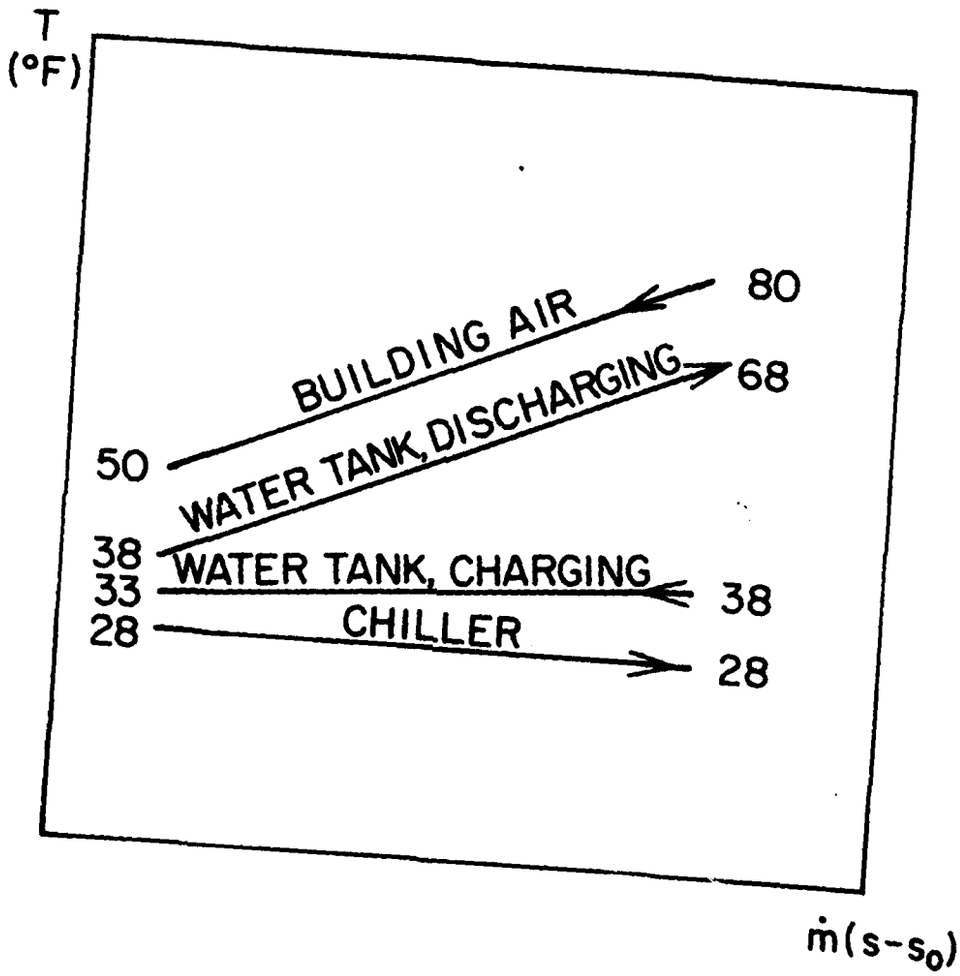


Figure 8-10. Fully charged Mixed Sensible Heat Tank System

Average Chiller Temperature of 8-9 and 8-10 = 31.5°F.

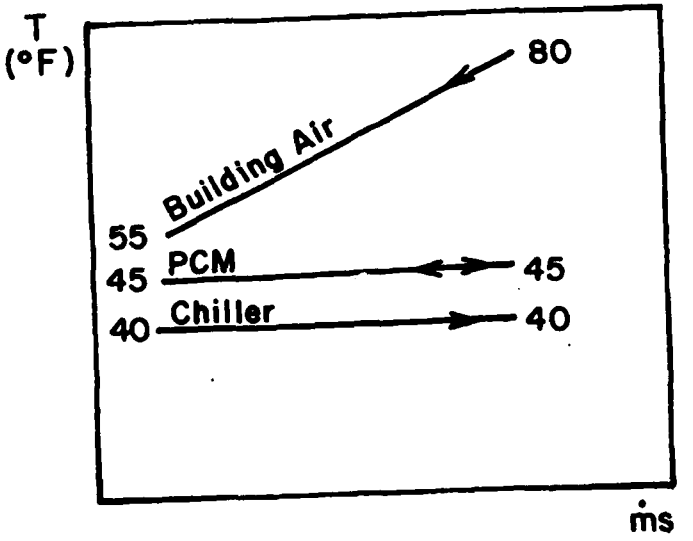


Figure 8-11. Latent Heat Cold-Side Storage with Latent Heat Transfer

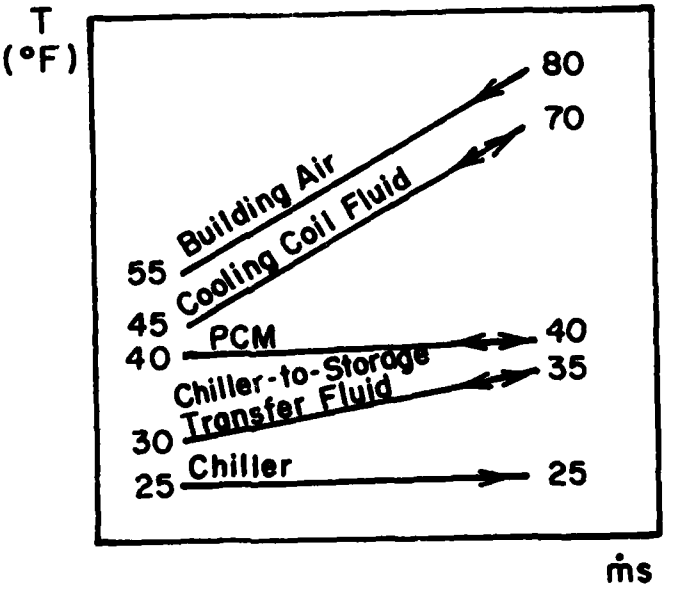


Figure 8-12. Latent Heat Cold-Side Storage with Sensible Heat Transfer

to the building from the bottom of the tank at 45°F and returns to the top of the tank at 70°F. If the system is perfectly stratified, Figure 8-8 represents the system over the complete charge-discharge cycle. Note that the chiller temperature must be 40°F (7°C) to cool the building air from 80°F to 55°F.

Figures 8-9 and 8-10 represent mixed sensible cold storage systems using water. Figure 8-9 shows the system when the storage is completely discharged at a temperature of 45°F. Water enters the heat exchanger to the chiller at 45°F, is cooled, and returns to the tank at 40°F, where it immediately mixes with the storage water, which changes to slightly less than 45°F due to the cooling of the chiller. If the storage is cooling the building air, water enters the heat exchanger to the building at 45°F, is heated to 70°F, and is returned to the tank, where it completely mixes with the rest of the storage water, which rises to slightly hotter than 45°F. The chiller temperature in this case is 35°F (2°C).

Figure 8-10 shows the storage system in a completely charged state, at a temperature of 38°F (3°C). The temperature drop of the storage fluid in each of the heat exchangers is similar to that of Figure 8-9. Here the chiller temperature must be 38°F to remove more heat from storage. The average chiller temperature for mixed sensible storage is the average of the extremes i.e. 31.5°F. In the perfectly stratified case, a usable temperature range of the storage fluid was 25°F (70-45). In the mixed storage case, the usable temperature range was 7°F (45-38). This means that compared to the stratified storage tank, the mixed storage tank must be 3.6 (25/7) times larger.

Figure 8-11 represents a latent cold storage system with latent heat transfer. The temperature of the phase change material (PCM) is 45°F and the chiller temperature is 40°F. For centuries, ice, the best known PCM, has been used for cooling storage. For cooling storage above the ice temperature, low freezing-temperature eutectics of Glauber's salt is the best alternative.

Figure 8-12 shows a latent cold storage system with sensible heat transfer between the PCM and both the building air and the chiller. Here a 5°F temperature difference is assumed between the PCM and the heat transfer fluid. The temperature of the PCM is 40°F and the temperature of the chiller is 25°F. If ice were used instead of a 40°F PCM, the chiller temperature would be reduced to 17°F.

Table 8-2 summarizes the required chiller temperature and relative storage volume of the hypothetical systems considered. The higher the chiller temperature, the higher will be the COP and capacity. The stratified sensible heat system and the all-latent heat system with latent heat transfer both have the highest chiller temperatures. However, stratified sensible storage will be less expensive unless storage volume is a critical issue.

Table 8-2. Comparison of chiller temperature and relative storage volumes for cold side heat storage systems

System	Temperature (°F)	Temperature (°C)	Relative Volume
All-latent heat	40	(4)	1
Stratified sensible heat	40	(4)	0.3-0.6
Mixed sensible heat	31.5	(-0.3)	3.6
Latent storage with sensible heat transfer	25	(-4)	0.3-0.6
Ice storage with sensible heat transfer	17	(-8)	0.2-04.

8.3 Features of Sensible Cold Storage

Many of the features of sensible cold storage systems, such as containers, pumps, insulation, etc. were discussed in Chapters 4-6 for sensible heat storage systems. In this section the discussion is limited to characteristics unique to sensible cold storage.

If the temperature range of cold storage is limited by either the need for dehumidification or for freeze protection, the storage container must be correspondingly larger than that for a heat storage system with an equivalent amount of energy storage. Usually however, the temperature difference between storage and the ambient is less than for heat storage systems, so less insulation is required.

Rock beds may be used for cold storage in much the same way as for heat storage. An important difference is that the storage temperatures are cool. This encourages condensation in the rock bed, with a corresponding encouragement of the growth of mold and other organisms.

8.4 Ice Systems

Ice making is an attractive method of cold storage. Water is plentiful and all of the advantages of phase change storage are present. In regions with sufficiently cold winters, ice can be made in large amounts by natural cooling. In the 1880's the ice business flourished in the northern regions of the U.S., but vitually ceased to exist with the advent of inexpen-

sive mechanical refrigeration in the 1920's. Recently a renewed interest in ice as cold storage has started (First Annual Workshop on Ice Storage for Cooling Applications, ed. A. J. Gorski, June, 1981, Argonne National Laboratory ANL-81-45).

Diurnal storage and seasonal storage comprise the two most feasible ice storage systems. Diurnal ice storage coupled to mechanical refrigeration is commercially available. Application is primarily directed to large commercial (office) buildings. Seasonal ice storage using natural cooling holds the promise of inexpensive cooling but is still under development.

8.4.1 Diurnal Ice Storage

Diurnal ice storage cooling systems are commercially available and are economically attractive for large office buildings. They are currently not economical for individual residences. In an ice storage system a conventional ice-making machine, similar to those found in refrigerators, produces ice in storage tanks.

Diurnal ice storage should be considered whenever cooling loads are high in locations where offpeak electricity rates and/or high electricity demand rates are present. The advantages are the same as for sensible cold storage. In traditional cooling methods, the air conditioning system is turned on whenever cooling demand is present. The cooling demand is usually largest in late afternoon, which is the peak demand period. A large cooling load will require a large amount of power, which may cause the utility to impose a demand charge. Operating the air conditioning system with cold storage then reduces electricity costs in three ways:

- The mechanical equipment is run at night during offpeak hours. The system is run for a longer time period, requiring less power at any given moment, which reduces the demand charge. Rejecting heat to the cooler night air usually results in higher system efficiency.

- The extra cost of storage is partially offset by a lower cost for the lower capacity chiller.

- A further advantage over conventional cooling is that cold storage systems can produce cool air in the building in a matter of minutes. Conventional systems usually must be turned on about an hour before the building is occupied to assure a comfortable environment.

Diurnal ice storage systems have further advantages over conventional chillers (C. MacCracken in First Annual Ice Storage for Cooling Applications Workshop, pp. 13-14).

Cold air from the ice system is colder than air from a conventional system and therefore has greater cooling capacity. This results in less air movement with a corresponding reduction in necessary fan power and duct size.

A conventional system operates at 60% relative humidity and 75°F thermostat setting. Ice systems can operate at 30% relative humidity, which means 80°F (27°C) is within the human comfort range. The 5°F higher thermostat setting significantly reduces the cooling load. A disadvantage of the ice system is that the chiller must operate at a low temperature -- see Table 8.2.

8.4.2 Seasonal Ice Storage

Seasonal ice storage systems are attractive for regions with significant summer cooling needs and winters with a large number of days where the outside air temperatures are below the freezing point of water. So far, no seasonal storage system is commercially available, but this may change in the near future; many of the experimental systems described below have shown encouraging results.

The basic method in most naturally-frozen ice systems is that during the winter, water is frozen, either directly or indirectly, using the naturally available low temperatures and wind. During the air-conditioning season, melt water at about 35°F is pumped from the ice filled reservoir to the space to be cooled. The chilled water can then be passed through a conventional fan coil unit to provide air cooling and dehumidification. The warmed water is then returned to the ice reservoir at about 45°F (9°C).

Ice can be made in several ways:

- Water can be sprayed on a solid surface, such as the bottom of a box or an already frozen slab of ice. Cold air passing over the thin liquid layer freezes it. For air temperatures of 5°F, about 6 inches of ice per day can be made in this fashion. To ensure effective freezing the liquid water layer should never be much thicker than 0.2 inches.

- A larger quantity of ice can be made by spraying fine droplets of water into the air. A group at Princeton University has successfully used a commercial snow machine for this purpose. Unlike most of the snow machines used on ski slopes, this particular machine did not use compressed air and therefore consumed only small amounts of electricity. One of the great advantages of all naturally-frozen ice methods is that auxiliary energy is only required to pump water.

- In climates with large winter snowfalls, ice can simply be collected. Snow removed from the streets, airport runways, etc. can be dumped into a large pit such as a quarry.

- A passively-grown ice technique is under investigation at Argonne National Laboratory. In this system, heat pipes connect an underground water tank to conventional flat plate solar collectors that are above the water tank and exposed to the outside air. The heat

pipes, containing R-12 refrigerant, conduct heat only one way - in this case from the water in the tank to the collectors exposed to the outside air. When the outside air is colder than the water in the tank, the heat pipe thermosiphon effect naturally conducts heat away from the tank, cooling the water. When the outside air is warmer than the water in the tank, the heat transfer stops automatically. When the winter air temperature is below freezing, ice can be formed in the tank. No auxiliary energy is needed for ice production in this method.

Types of containers for seasonally stored ice can vary from tanks and insulated pits for small systems to lined earth excavations or unlined natural pits for very large systems. An insulating blanket must cover an open-topped container in the summer to minimize melting from wind, sun, and warm air temperatures. Heat loss from cold storage to the ground is not as severe as in the case for heat storage. In climates where seasonal ice storage is feasible, the average ground temperature is usually 50-60°F (10-16°C), and unless there is a moving water stream close to the container, the ground itself will insulate. The thickness of the surface insulation blanket, x, can be estimated from Equation 8-1.

$$x = \frac{u \cdot A \cdot \Delta T \cdot t}{Q} \qquad (8\text{-}1)$$

where u is the thermal conductivity of the insulation (Btu·in/hr·ft^2·°F),
A is the surface area (ft^2),
ΔT is the average temperature difference between the storage and the ambient (°F),
t is the storage time (hr),
Q is the maximum amount of energy allowed to be lost (Btu).

Example 8-1. Suppose we have a pit of ice 10 ft deep, with a 30 ft by 30 ft surface. We finish making the ice on March 21 and want to store the ice until September 21 without losing more than 5% of the ice to surface losses. The average ambient temperature for this period is 60°F. How thick must a surface blanket of fiberglass batt be to accomplish this?

First we use Equations 7-1 and 7-2 to calculate the amount of cold energy stored - in this case the energy needed to melt the ice. From Table 7-1, we see that ice has a density of 57.24 lb/ft^3 and a heat of fusion of 143.1 Btu/lb. The mass of the ice is calculated from Equation 7-2. The volume of our container is

$$V = 10 \text{ ft} \times 30 \text{ ft} \times 30 \text{ ft} = 9000 \text{ ft}^3$$

Using Equation 7-1, the mass of the ice is

$$m = \rho V$$

$$= 57.24 \times 9000 = 515{,}200 \text{ lb},$$

a little over 250 tons.

Using Equation 7-2, the amount of energy stored is

$$Q = m \eta h_f$$

$$= 515{,}200 \times 1 \times 143.1 = 73.7 \text{ MMBtu}^*$$

Since we can only lose 5% of the energy stored,

$$Q = 0.05 \times Q$$

$$= 0.05 \times 73.7 = 3.69 \text{ MMBtu}.$$

The temperature difference is 60-32 = 28°F.

The surface area is A = 30 x 30 = 900 ft^2.

The storage time is 182 days, or t = 182 x 24 = 4368 hr.

From Table 4-4, the r-value of the insulation is 3.15.

$$u = 1/r = 1/3.15 = 0.317 \text{ Btu} \cdot \text{in}/\text{ft}^2 \cdot \text{hr} \cdot {}^\circ\text{F}.$$

Using Equation 8-1, the thickness of the insulation is

$$x = 0.317 \times 900 \times 28 \times 4368 \; / \; 3{,}690{,}000 = 9.5 \text{ in.}$$

(1 MM Btu = 1,000,000 Btu)

The seasonal ice storage techniques that use open pit type containers are most suitable to large scale systems. The techniques that use closed tanks, such as the passively-grown ice system, are more applicable to residential uses, but the economics of small scale systems has not yet been thoroughly explored.

8.5 Evaporators

Evaporators are heat exchangers that interface the chiller with either cold storage or the space to be cooled. Heat exchangers were discussed previously in Chapter 4. A direct expansion evaporator coil is usually used to cool air. A shell and tube evaporator is used for chilling water.

8.5.1 Direct Expansion Evaporator

A direct expansion coil consists of a bank of tubes, often connected by fins, containing refrigerant. Heat is removed from warm air flowing across the tubes and fins, which boils the refrigerant inside the tubes. Direct expansion evaporators may be either the dry expansion type or the flooded type.

Dry Expansion Coil

The most common direct expansion evaporator is the dry expansion coil, shown in Figure 8-13. Refrigerant is metered to the coil through an expansion valve. The liquid refrigerant boils in the part of the coil closest to the expansion valve. As the refrigerant vapor passes through the rest of the coil, it continues to pick up heat from the coil walls and becomes slightly superheated. A sensor measures the temperature of the refrigerant gas leaving the coil. The temperature controls the expansion valve to maintain the correct amount of superheat in the refrigerant leaving the coil.

Flooded Type

The flooded type of direct expansion evaporator consists of a bank of straight tubes connected by headers. A float valve keeps the liquid in the coil at a predetermined level. Boiling of refrigerant only occurs where the liquid is in contact with the warm walls of the tubes. Since almost the entire length of the tubes contains liquid, the efficient heat transfer of boiling takes place throughout most of the coil. For this reason the flooded type of coil is more efficient than the dry expansion type, but it requires a larger charge of refrigerant.

Sizing

The performance of an evaporator coil is complicated if it is used both to cool and remove moisture from the air because condensation considerably changes the characteristics of the evaporator. Air entering the coil may be too warm to condense immediately, and condensation may occur on only some of the tubes. We recommend consulting manufacturers' rating tables for evaporator performance characteristics. CAUTION! The performance of evaporators changes greatly if a different refrigerant is used than that used for the rating.

8.5.2 Shell and Tube Evaporators

Shell and tube evaporators are used to provide chilled water and can be either flooded type, semi-flooded type, or dry type. Shell and tube heat exchangers were discussed in Section 4.6 and shown in Figure 4-8 and 4-9.

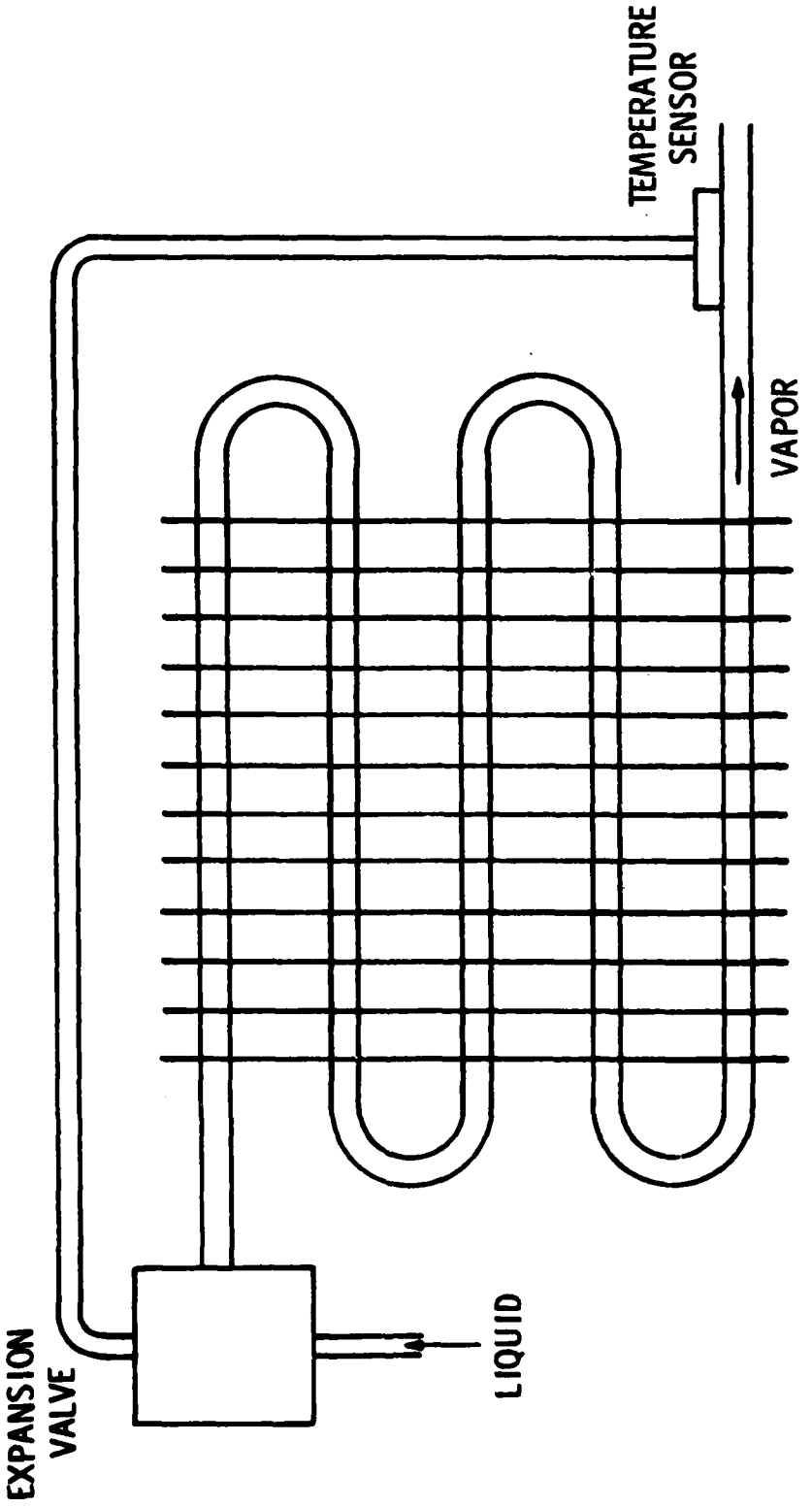


Figure 8-13. Dry Expansion Coil

Flooded Type

Water is pumped through a bundle of tubes which is contained in the shell. Half to three-fourths of the tubes are immersed in liquid refrigerant. Violent boiling around the submerged tubes wets the outside surfaces of all the tubes.

Semi-Flooded Type

This type is similar to the flooded type except that only the bottom row of tubes is submerged in the refrigerant. The semi-flooded type requires less of a charge than the flooded type, but requires careful design to insure that all the tubes are wetted.

Dry Type

In the dry type of shell and tube evaporator, liquid refrigerant boils inside the tubes, and water is circulated around the tube bundle. Compared to the flooded or semi-flooded type, the dry type, while somewhat less efficient, requires less refrigerant; has a smaller pressure drop in the chilled water circuit; and is less likely to be damaged if controls fail and the evaporator freezes.

8.6 Cold Water Sources

Cold water can be used in cooling for direct evaporative space cooling, (condensor cooling), cooling of water used in aquifer storage. This section discusses special considerations that occur when cold water sources are used for these applications.

8.6.1 Water Supply

There are three major classes of water supply: city water, surface water, and ground water. Availability, temperature, and the chemical properties of each source are important concerns in the selection of the water supply.

City Water

City water or municipal water has the advantages of low initial cost, little or no equipment maintenance, and little need for water treatment. The largest disadvantage is that there may be substantial usage and sewage fees if large amounts are used. For condensor cooling or evaporative cooling the supply temperature is an important factor in determining the quantity of water necessary. Cool water can accept a greater amount of heat than warm water.

Surface Water

Surface water consists of water supplied from rivers, lakes, and creeks. Surface water is free; pumping equipment is minimal; and pumping costs are small. Disadvantages of surface water include: cost of filtering equipment, decrease of availability during times of drought, increase of temperature in summer, upstream pollution, and floods. Thermal pollution from the return water may also be a problem. Since the amount of water needed depends on supply temperature, it is important that the water temperature is known, especially during times when availability may be low.

Ground Water

Ground water from wells is used when surface water is not available and city water is too expensive. The temperature of well water is usually equal to the yearly average air temperature for any location. The cost of well drilling and the cost of pumping are usually proportional to the depth of the well. For the most part, the capacity of a given well depends on ground properties in the vicinity of the well water and not on the size of the well; more than one well may be required. Many local codes require that the warm return water be reinjected into the water table, requiring at least one additional well.

8.6.2 Evaporative Cooling

Evaporative coolers for direct space cooling are commercially available from a number of manufacturers. Evaporative coolers consist of boxes containing porous material that is kept saturated with water. A fan draws warm dry outside air through the porous material and discharges the cool moist air into the conditioned space.

Evaporative cooling is also used for condensor water cooling and water cooling for aquifer cold storage. Either spray ponds or cooling towers are used for this purpose. In either case the cooling effect is enhanced by providing a large water surface (e.g. breaking the water into droplets) and a large air flow.

Spray ponds are open tanks which collect water sprayed into the air by nozzles. The air flowing horizontally over the pond evaporates some of the water from each drop, cooling the rest of the drop. Spray ponds are inexpensive to install and have a low operating and maintenance cost. However, compared to cooling towers, they require more area and have more drift loss of the cooled water, even when surrounded by a louvered fence. A cooling tower is essentially a small spray pond surrounded by a very tall fence. The cooling tower may contain slats to help break up the water and may have either a vertical or horizontal air flow. Air flow through the tower is either by forced air from a fan or by natural air currents.

In any evaporation process, makeup water must be supplied to replace the water that is evaporated and drifts away. Depending on air humidity and temperature, approximately 0.8% of the water supplied to a cooling

tower evaporates for every 10°F the water is cooled. The drift from a cooling tower is about 0.1-0.3% of the water flow through the tower. Drift loss from a spray pond is about 5-8% of the supplied water. In addition, as water evaporates, it leaves mineral salts and other particulates behind. The concentration of solids and dissolved salts increases, leading to scale and corrosion. To prevent this from happening, a certain amount of water, called blowdown, is wasted. The blowdown carries away dissolved minerals, keeping the mineral salt buildup to a tolerable level. In many cases the hardness of the makeup water is high enough by itself to cause corrosion, and then the water must be treated before arriving at the cooling tower.

8.7 Available Systems

The number and variety of commercial thermal-powered chillers and ice storage cooling systems are limited by marketplace economics. Currently (1981) there are only about 300 active solar cooling systems installed in the United States. Commercially available ice storage cooling systems are economically attractive for commercial and industrial buildings. Throughout this section all information and price quotations are 1981-based.

8.7.1 Chillers

Of the three types of chiller - absorption, Rankine, and desiccant - absorption chillers are by far the most commonly used in conjunction with solar collectors. Rankine chillers (vapor compression) are still commercially handicapped by a price disadvantage and lack of commercial availability. Desiccant systems are still strictly experimental; no known commercial units yet exist.

Absorption Chillers

There are a number of different systems commercially available: four representative systems are discussed below:

American Yazaki Corporation (Dallas, Texas), is a subsidiary of a very successful Japanese company which is a leader in the manufacture of equipment for solar air conditioning systems. Yazaki manufactures water-fired chillers with cooling capacities of 1.3,2,3,5,7.5 and 10 refrigeration tons*. Larger capacities are obtained by connecting the modular chillers in parallel. The unique design of the absorption chillers allow them to be energized by variable hot water temperatures of 167 to 212°F provided by the high efficiency "Blue Panel" flat plate solar collectors. Other components such as controls, cooling towers, fan-coil units, etc. are manufactured by Yazaki to complement the solar air conditioning system and avoid mismatching. Solar cooling, heating and hot water supply systems are installed for $6000 to

* One refrigeration ton is the amount of cooling equivalent to melting one ton of ice in 24 hours (i.e. 288,000 BTU or 12,000 BTU/hr.)

$12000 per refrigeration ton* depending upon size, location and area of the solar collectors. Generally commercial systems are more cost effective than residential systems especially in areas where fuel prices are high and solar energy is plentiful. The Yazaki absorption chillers have also been used for space or process cooling in cogeneration and waste heat applications.

Arkla Industries (Evansille, Indiana) manufactures the Solaire-36 and Solaire-300 which are respectively 3-ton and 25-ton absorption chiller systems using lithium bromide as the absorbent. These packaged heating and cooling systems are factory assembled and contain all components pre-installed. An old timer in the market, Arkla has been selling systems for industrial waste heat applications since 1949. An installed system cost $7,000-10,000/ton of air conditioning not including the solar collector and is accompanied by a one-year warranty.

York Division, Borg-Warner Corporation (York, Pennsylvania), has been manufacturing steam and liquid absorption chillers for 16 years. These chillers (as with the Arkla chillers) are suitable for cogenerative, industrial waste heat, or solar applications. There are 21 different unit sizes ranging from 100 tons to 1400 tons, making them suitable for heavy industrial use and large office buildings. Air conditioning costs become as low as $190/ton when based on a 1400 ton absorption chiller module. York provides an 18 month warranty and suggests a 20-year life expectancy.

Carrier Corporation (Syracuse, New York) manufactures lithium bromide based water-fired absorption chillers. This unit can be incorporated into a complete solar heating and cooling system.

Rankine Chillers

Barber-Nichols Engineering Company (Arvada, Colorado), although having no catalog items, will custom design and construct Rankine cycle cooling systems with 3 to 100 tons of capacity. A 25-ton integrated system suitable for a small office building would cost about $150,000 or $6,000/ton.

8.7.2 Ice Storage Systems

Although ice storage is still largely in the research and development stages, there are a few diurnal systems commercially available. No seasonal systems are commercially available yet.

Calmac Manufacturing Corporation (Englewood, New Jersey) manufactures Levload™ Ice Banks off-peak thermal storage system. The modular plastic units are available in two sizes: 36 and 54 ton-hours. Designed mainly for off-peak storage in commercial and industrial applications, installed cost is about $60/ton hr. The system is designed so that ice expands upward and not outward; there is no possibility of ice breaking the container. During the freezing (charging) cycle, conventionally chilled antifreeze is circulated to the storage modules where it extracts heat using a coiled mat-type heat exchanger. During the discharge cycle, stored cooling power is drawn from the modules and circulated to the load.

Advance Energy Technologies (Clifton Park, New York) designs and engineers passive ice storage systems in which SO_2-filled heat pipes form ice around a cylindrical evaporation network in an underground water tank. This system is designed primarily for humidity control.

Globe Engineering Company (Chicago, Illinois) also has commercial and industrial experience in designing large ice storage systems for electrical load management.

8.8 Costs

Studies indicate that cool storage in new commercial buildings for electric load leveling can be economically attractive (3-5 year payback time). Partial storage, since it requires less chiller and storage capacity, has an even shorter payback period. Figure 8-14 shows the payback dependence of an actual commercial installation as a function of chiller capacity cost and storage capacity cost.

Although shifting a substantial part of the electrical requirement for building-space-cooling from the utility's peak to off-peak period may result in increased consumption due to the need for more pumping, the demand component of cost will be greatly reduced. Switching the chiller off during the peak period (full storage) minimizes peak period consumption. On the other hand, costs of chiller capacity requirements are minimized by partial storage in which the chiller operates continuously at reduced rate. The cost of the chilled water storage is a composite; reflecting storage tank material, size, location, and construction.

Because ice storage units can ideally offer up to a factor of 10 size reduction over chilled water storage, ice units have an intrinsic economic advantage in retrofit applications. The unit cost of ice storage, including all costs, decreases with increasing size. A 45,000 pound capacity system costs approximately $0.60 per pound of ice capacity.

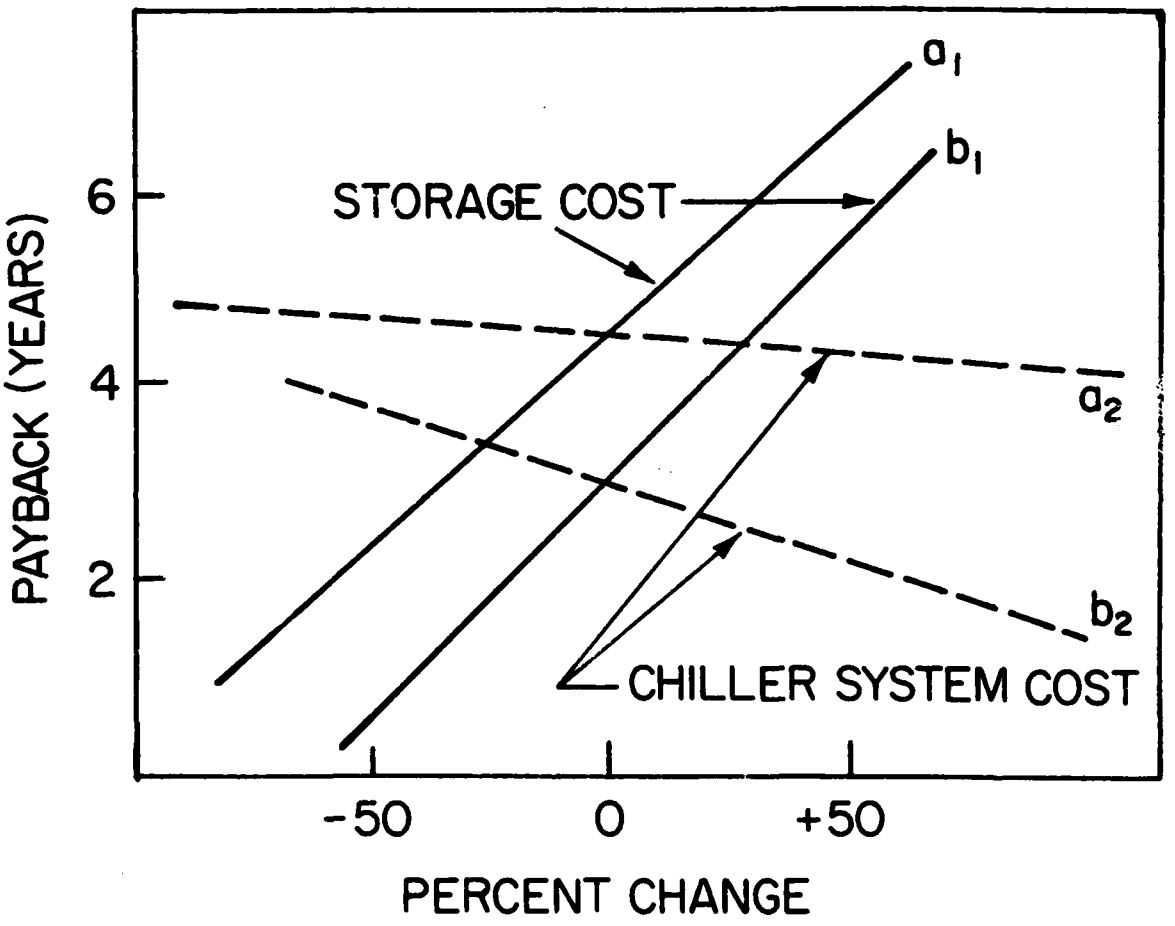


Figure 8-14. Cost of Installation as a Function of Chiller Capacity Cost and Storage Capacity Cost for (a) Full and (b) Partial Storage Systems.

8.9 Symbols Used

A	surface area in ft^2 (m^2)
h_f	latent heat of fusion in Btu/lb (J/kg)
m	mass of ice in lb (kg)
$\dot{m}$	mass flowrate, lb/sec (pg/sec)
s	entropy, Btu/°F·lb. (W/m·°C)
Q	useful energy stored in ice in Btu (J)
r	thermal resistivity in °F·ft^2·hr/Btu·in (°C·m/W)
ΔT	average temperature difference between storage and ambient in °F (°C)
t	amount of time energy is to be stored in hr (sec)
u	thermal conductivity in Btu·in/hr·°F·ft^2 (W/m·°C)
V	volume of storage in ft^3 (m^3)
η	fraction of theoretical latent heat that can be recovered
ρ	solid density of ice in lb/ft^3 (kg/m^3)

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9.0 THERMOCHEMICAL STORAGE

Thermochemical energy storage uses reversible chemical reactions to store and later release thermal energy. General features of thermochemical storage were discussed in Section 2.10. The present chapter discusses thermochemical storage systems in more detail and includes zeolite and metal hydride systems. Electrochemical systems (e.g. batteries) are discussed in Chapter 12.

Thermochemical energy storage is potentially the most attractive of the different thermal storage methods discussed in this handbook. However, few thermochemical energy storage systems are commercially available now or likely to be in the near future. All systems discussed in this chapter are still under development.

9.1 General Features

The advantages and disadvantages of thermochemical storage are described immediately below to give a quick general summary of this energy storage method. The thermodynamics of thermochemical storage are discussed next. There are three major applications of thermochemical storage, each of which requires different considerations. Characteristics which depend on the particular application are discussed in Section 9.2.

9.1.1 Advantages and Disadvantages

9.1.1.1 Disadvantages

(1) Not yet available - Thermochemical storage is still in the development stage. Only one thermochemical system, which is based on zeolites, is currently available.

(2) Complexity - Thermochemical systems will be complex, much more so than indicated in Figure 2-5. It is unlikely that such systems will be economical on a scale suitable for a single-family dwelling.

(3) Dangerous chemicals - Most of the thermochemical systems use dangerous chemicals which must be carefully contained to prevent chemical burns, explosions, or poisoning.

9.1.1.2 Advantages

(1) High energy densities - The energy storage densities of thermochemical systems are higher than those of either sensible or latent heat storage by at least a factor of two.

(2) Storage at ambient temperatures - Both the products and the reactants of the chemical reaction can be cooled and stored at ambient temperatures. This minimizes the need for insulation of the storage containers. The system must be designed carefully however, so that the heat lost in cooling the chemicals is used elsewhere.

(3) Long storage times - At room temperature, it is possible to store the chemicals up to several years without degradation.

(4) Modification of delivery temperature - The temperature of the heat delivered to the load can be either above or below the temperature of the heat received from the source.

(5) Large temperature range - Each particular chemical reaction is best suited for a definite temperature range. However, there are many different reversible chemical reactions available, together spanning a wide temperature range. Potentially, heat sources of any temperature, from high temperature nuclear reactors to low temperature solar collectors, can be coupled to this storage system.

(6) Transportability - Many of the products and reactants are gases or liquids that can be transported in pipelines.

(7) Low Cost - The costs for the storage chemicals and storage containers are generally low. On the other hand, the cost of the chemical reactors is still high.

9.1.2 Thermodynamics

The thermodynamics of thermochemical reactions can be understood in general terms by considering the following idealized and simplified example. As in Figure 2-5 the chemical reaction involves A and B, where both A and B consist of one or more different types of molecules. When A is turned into B, the reaction is endothermic, requiring input heat. When B is turned into A, the reaction is exothermic, releasing heat.

A thermochemical reaction is characterized by an equilibrium temperature T_{eq} as follows: Place chemicals in an idealized container, such that (1) the amount of A and B does not change, and (2) the temperature T of the container does not change. For example, if a little of B is turned into A, more B is added to the container, and some of A is removed. In addition, enough heat is removed from the container to keep the temperature at T. T_{eq} is defined as the temperature for which:

if $T > T_{eq}$, then A $\rightarrow$ B (endothermic reaction), and

if $T > T_{eq}$, then B $\rightarrow$ A (exothermic reaction).

T_{eq} depends on the particular chemical reaction considered and also on the reactor pressure. T_{eq} increases slightly as the pressure increases.

Any particular chemical reaction is also characterized by ΔH, the enthalpy change (equivalent to energy change for our purposes) per unit mass associated with the reaction. ΔH is related to T_{eq} by:

$$\Delta H = T_{eq} \Delta S, \qquad (9\text{-}1)$$

which is approximately valid for most reversible reactions. ΔS is the entropy change associated with the chemical reaction and is approximately constant for any class of reaction (solid to liquid, solid to gas, etc.). ΔH is directly proportional to the storage energy density. The highest storage densities will then occur for those reactions with (1) high equilibrium temperatures and (2) large entropy changes, such as reactions that change solids to gases.

It is necessary to control the rate of the chemical reaction. One method to accomplish this is to choose a reaction that depends on the presence of a specific catalyst. The amount of that catalyst present determines the rate of the reaction. A second method is to choose a reaction that requires two or more components that are easily separable. If one of the components is a gas, the separation is easy. The reaction rate than depends on the amount of gas introduced to the reactor. While the use of a gas makes the separation of the components easy, the energy storge density of a gas is low, and it may be necessary to liquify the gas to attain large storage densities.

9.2 Major Applications

There are three major applications for thermochemical energy storage: thermal storage, thermal energy transport, and chemical heat pumping.

Thermal Storage

The simplest application is that of thermal storage - the same application that was discussed in Chapters 4-8 for other thermal storage methods. This application exploits none of the unique features of thermochemical energy storage. It is unlikely that this application will be cost competitive with other thermal storage methods in the foreseeable future, except for circumstances where storage volume must be kept as small as possible.

Thermal Energy Transport

The ability to store thermal energy at ambient temperatures makes thermochemical storage a natural choice for transport of energy over long distances. No thermal losses would occur in the transportation. The need for thermochemical storage increases as the required temperature of thermal energy increases. Applications include the transport of energy from a field of high temperature solar collectors to centralized storage and also the transport from a central heat source to individual users.

Chemical Heat Pump

Use in chemical heat pump applications is another feature that is unique to thermochemical storage. In its simplest form, the chemical heat pump uses two different reversible reactions in which a gaseous product is common to both reactions. As an illustration, consider the use of a chemical heat pump in a space heating application. The two reactions used are:

$$AG(s) \rightleftharpoons A(s) + G(g) \qquad (9\text{-}2)$$

$$BG(s) \rightleftharpoons B(s) + G(g) \qquad (9\text{-}3)$$

where AG is a solid composed of a single molecule which decomposes into the solid A and the gas G, and
BG is a solid composed of a single molecule which decomposes into the solid B and the gas G.

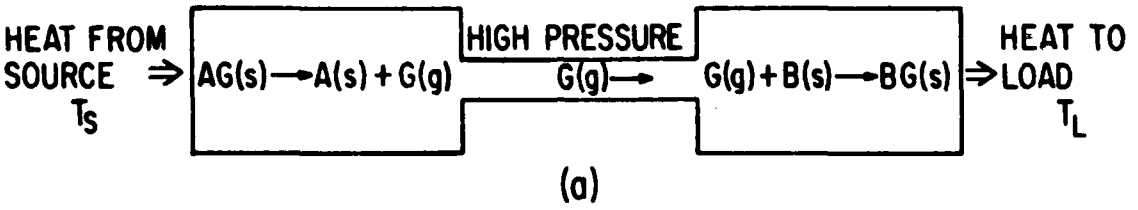
Both reactions (9-2) and (9-3) are reversible. Heat is delivered to the load in two separate stages as shown in Figure 9-1.

In Figure 9-1 (a) heat from the source at temperature T_S is used to drive the gas away from the reactor on the left via reaction (9-2). The gas travels to the reactor on the right, where it combines with B and gives up some of its heat to the load at temperature T_L. Since $T_S > T_L$, ΔH is larger for the reaction in the reactor on the left.

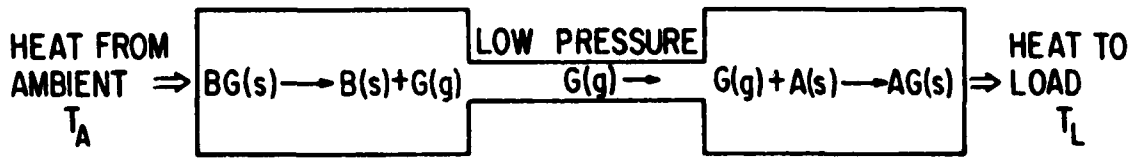
In Figure 9-1 (b), both reactors are at colder temperatures and the reactions are reversed from the previous case. This time, heat is received from the ambient at temperature T_A, and the gas is driven to the reactor containing A, where energy is delivered to the load at temperature T_L. Here T_A can be less than T_L.

The complete cycle is made possible only if the vapor pressure as a function of temperature is considerably different for each reaction, as shown in Figure 9-2. The arrow labeled (a) denotes the process illustrated in Figure 9-1 (a). The arrow labeled (b) denotes the process illustrated in Figure 9-1 (b). The curves for the two reactions must cross for the two-reaction process to be reversed at lower temperature.

The chemical heat pump is similar to the mechanical heat pump, but bypasses the need to turn thermal energy into mechanical energy. Since heat is pumped from the ambient, it is possible to deliver more heat to the load then was available at the source; that is, the coefficient of performance (COP), defined as the ratio of heat delivered to heat available from the



(a)



(b)

Figure 9-1. Schematic Diagram of Chemical Heat Pump System

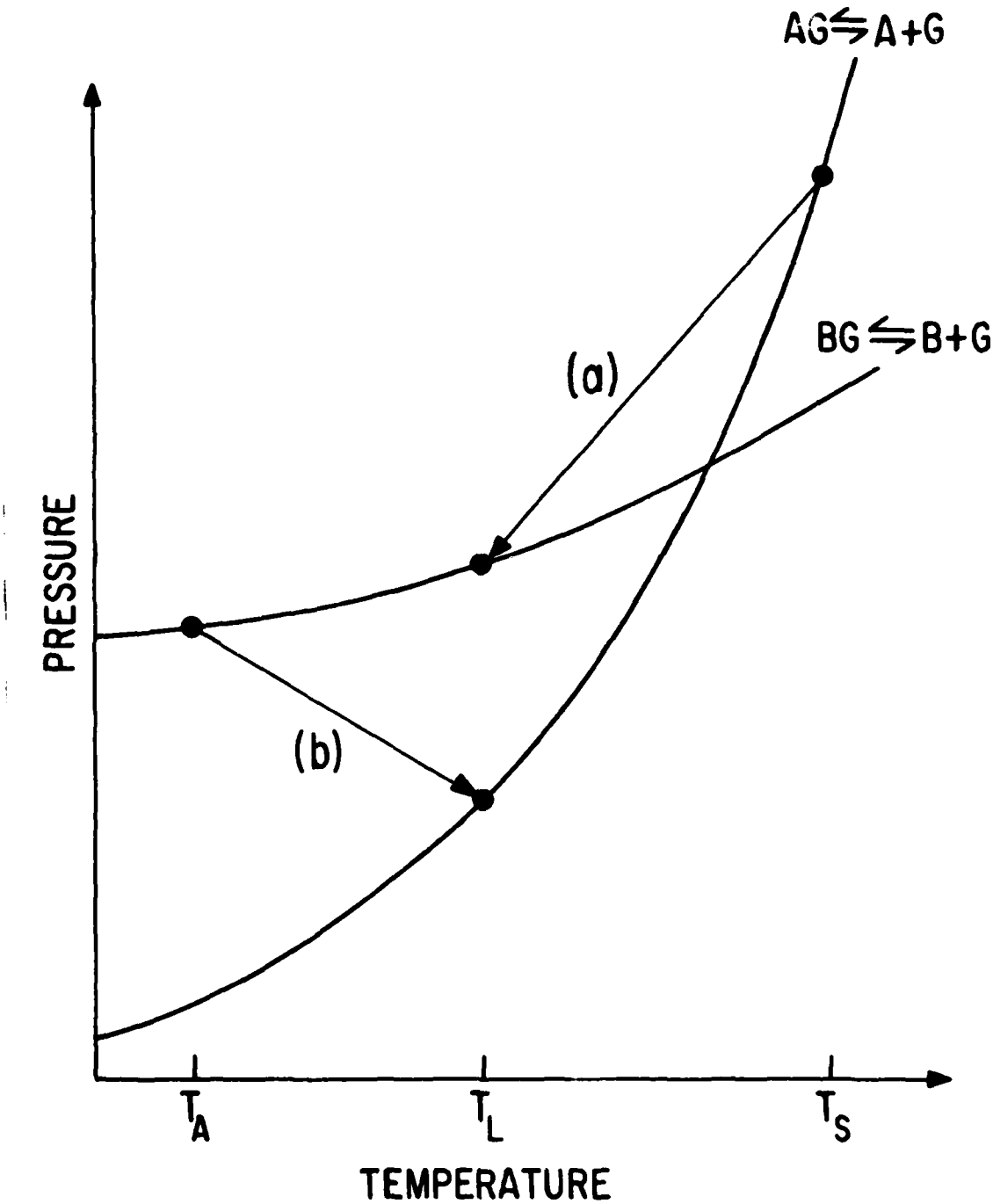


Figure 9-2. Pressure vs Temperature for Typical Pair of Chemical Heat Pump Reactions

source, is greater than one. Likewise, it is possible to produce heat at a temperature greater than that of the source, with a COP less than one. In either case the theoretical maximum amount of available heat that can be pumped is limited by the Second Law of Thermodynamics.

9.3 Possible Systems

There are many possible reactions that have been considered for thermochemical storage. Some of these reactions have been experimentally investigated for suitability in a storage system. A detailed catalog of these reactions can be found by examining the articles referenced in Section 9.5 and the references listed in those articles. In this section several examples of some of the more extensively researched reactions are briefly described.

9.3.1 Storage and Heat Pipe

One of the most advanced thermochemical storage systems is the SOLCHEM process, developed by the Naval Research Laboratory. This process uses the reaction:

$$2\ SO_3 \rightleftharpoons 2\ SO_2 + O_2$$

Sulfur trioxide decomposes at high temperatures (1520°F or 830°C). At room temperature the reverse reaction will not occur unless a catalyst is present. SO_3 solidifies in the temperature range 63-143°F (17-62°C), so pipeline transmission of this chemical is limited to short distances. The process is intended for short term storage of thermal energy from high temperature solar collectors. Long term storage is not feasible, because of the large volumes needed to store the gaseous sulfur dioxide/oxygen mixture.

Another heat pipe process is the EVA-ADAM steam reforming reaction.

$$CH_4 + H_2O \rightleftharpoons CO + 3\ H_2$$

The methane and water combine at a temperature of 1880°F (1030°C). The reverse reaction will not occur at room temperature without a catalyst. Since water is readily available, and the other molecules are gases, this process is suitable for pipeline transmission.

Some reactions involve hydration/dehydration cycles. Both calcium hydroxide $Ca(OH)_2$ and magnesium hydroxide $Mg(OH)_2$ are potential candidate materials for this process and are suitable for lower temperature sources (510°F or 270°C).

9.3.2 Chemical Heat Pump

A chemical heat pump system investigated by the EIC Corporation used the reactions:

$$FeCl_2 \cdot 6NH_3(s) \rightleftharpoons FeCl_2 \cdot 2NH_3(s) + 4\ NH_3(g)$$

$$CaCl_2 \cdot 4NH_3(s) + 4\ NH_3(g) \rightleftharpoons CaCl_2 \cdot 8NH_3(s)$$

One of the most advanced chemical heat pump systems is that of the Rocket Research Corporation of Redmond, Washington. Their system uses a sulfuric acid/water concentration/dilution scheme. This system is similar to an absorption chiller (described in Chapter 8), except that both the concentrated acid and the dilute acid can be stored. The sulfuric acid acts as the absorbent, water as the refrigerant.

9.3.3 Metal Hydrides

It is possible to store hydrogen gas in certain metals, forming a metal hydride. The reaction is such that heat is given off when the hydrogen is added to the metal, and heat must be added to liberate the stored hydrogen from the hydride.

Some metal hydrides, such as $TiFeH_2$ and $LaNi_5H_7$, will release hydrogen gas at or below room temperature. These materials have an energy storage density (as hydrogen fuel) of about 37,000 Btu/cubic foot (1,400 KJ/liter). Other hydrides, such as Mg_2NiH_4 and TiH_2, require temperatures as high as 1100°F (600°C) to liberate hydrogen, but have storage densities as high as 475,000 Btu/cubic foot (18,000 KJ/liter).

9.3.4 Zeolites

Zeolites are molecular sieves that can store and release energy by the adsorption/desorption of water or other refrigerant gases in the tiny pores of the sieve. The system starts off in a dry state. Energy is given off when vapor is adsorbed. Heat energy must be added to drive off the vapor, recharging the system. The fluid can be stored in an adjacent tank, or if water is used, can be derived from moisture in the ambient air. Adsorption/desorption for cooling purposes was discussed in Section 8.1.4.

Zeolites are chemically inert, abundant, and inexpensive. The large surface area offered by the porous structure allows adsorption of up to 30%

by weight of the gas. Since water has the highest heat of vaporization of any of the refrigerants, it is most commonly used. The adsorption process is highly temperature sensitive, and the amount of vapor adsorbed decreases significantly as the temperature increases.

9.4 Symbols Used

Main Symbols

T	temperature in °F
ΔH	change in enthalpy in Btu/lb
ΔS	change in entropy in Btu/°F.lb
(s)	solid
(g)	gas

Subscripts

eq	equilibrium or turning point
A	ambient or outside condition
S	source
L	load

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10.0 MECHANICAL ENERGY STORAGE

Mechanical energy may be stored either as kinetic energy of moving matter (e.g. flywheels) or as potential energy of matter under compression (e.g. compressed air). General features of mechanical energy storage were discussed in Section 2.11.

In both cases mechanical energy storage has several advantages:

- ability to accept and deliver energy at high rates
- energy storage capacity independent of delivery rate
- potential for simplified power conversion
- high recovery efficiency in most cases

Except for traditional flywheel applications in machinery, mechanical energy storage is still in the developmental period.

10.1 Compressed Air

The simplest form of compressed air energy storage (CAS), illustrated in Fig. 10-1a, uses a mechanically driven compressor to force air into a reservoir under pressure. When a source of mechanical energy is available, the reservoir is pressurized. When energy is needed, the pressurized air is available for producing mechanical energy.

Recent studies have indicated that, although environmentally attractive, the simple near-adiabatic method of the above paragraph is not yet economical. However, compressed air storage combined with fossil fuel combustion, as illustrated in Fig. 10-1b, does appear economical.

In this case, during off-peak periods, low-cost electric power is used to compress air into storage in the underground reservoir. During peak-load periods the compressed air is withdrawn from storage, mixed with fuel, burned, and expanded through the turbine to generate peak power. During peak periods, petroleum consumption is 60% less than it is in a conventional peaking plant.

10.1.1 General Features

Compressed air energy storage (CAS) has several advantages:

- Fewer siting and environmental problems than with pumped hydro storage
  - no above ground reservoir is needed
  - more geological sites are usable
  - smaller minimum size is economical.

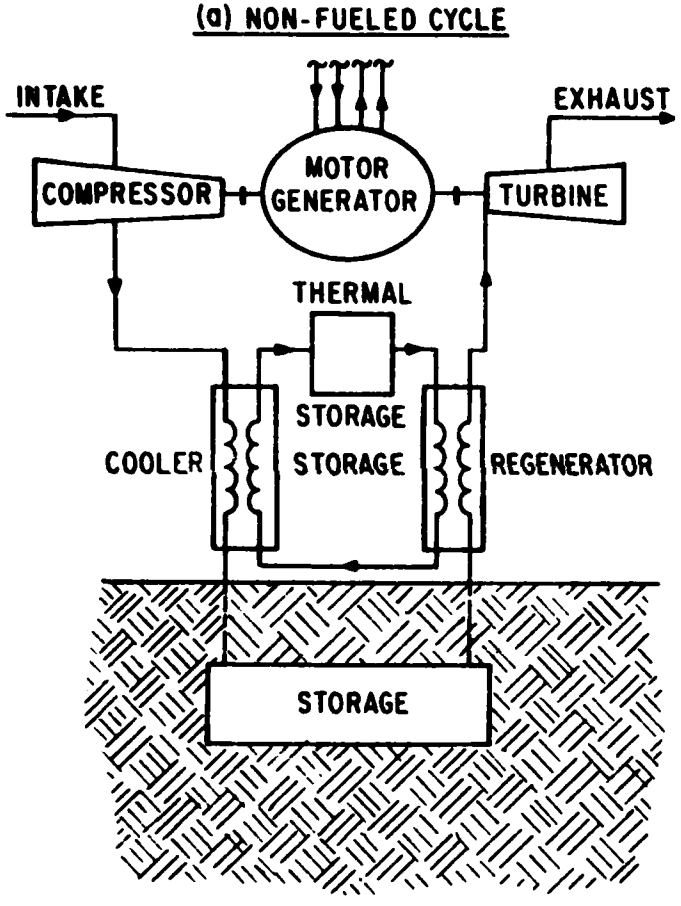


Figure 10-1(a). Adiabatic Compressed Air Energy Storage

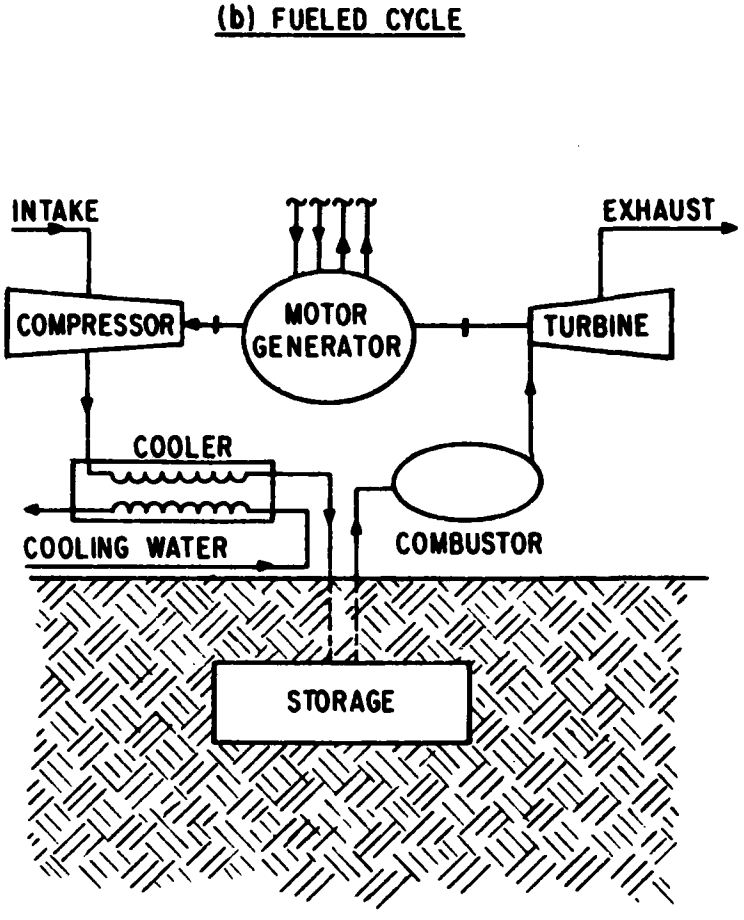


Figure 10-1(b). Conventional Compressed Air Energy Storage

- Cost competitive
  - less expensive than batteries
  - if the load factor of the power plant during off-peak hours is less than about 15%, also less expensive than pumped hydro.

- Applicable to any power system in any climate

On the other hand, CAS also has several disadvantages:

- Still a relatively untried technology - the only commercial CAS now operating is at Huntorf, West Germany.

- Reservoir stability over long time periods is required.

- Cyclic pressure loading is necessary if a large surface water reservoir is not available for pressure compensation.

- In porous reservoirs the champagne effect, in which the compressed air dissolves into the water, will occur during long idling periods. This can result in serious pressure drops and loss of storage efficiency.

- The hot compressed air must be cooled before storage to prevent fracturing of the rock or creeping of the salt. The stored air must then be reheated before it goes into the turbine.

10.1.2 Geological Considerations

Pressurized tanks are still too expensive for CAS applications, so underground reservoirs, such as aquifers, salt cavities, and mined hard rock caverns must be used.

Storage reservoir design and stability are key issues in commercial success. Reservoirs must have lifetimes of over 30 years; thermal and mechanical effects generated by daily reservoir operation could cause severe deterioration during this time. The major types of CAS reservoirs are indicated in Fig. 10-2.

A constant-pressure compensated system, illustrated in Fig. 10-2a, must be located in a mined hard-rock cavern. Rock formations such as homogeneous granite and gneisses are ideal for this, as well as for underground pumped hydro (Chapter 11) applications. Extensive studies have demonstrated that this type of system is quite practical with no problems beyond the scope of a good geotechnical engineer.

Storage associated with dissolved-out salt cavities, illustrated in Fig. 10-2b, exploits the least expensive type of reservoir. The Huntorf

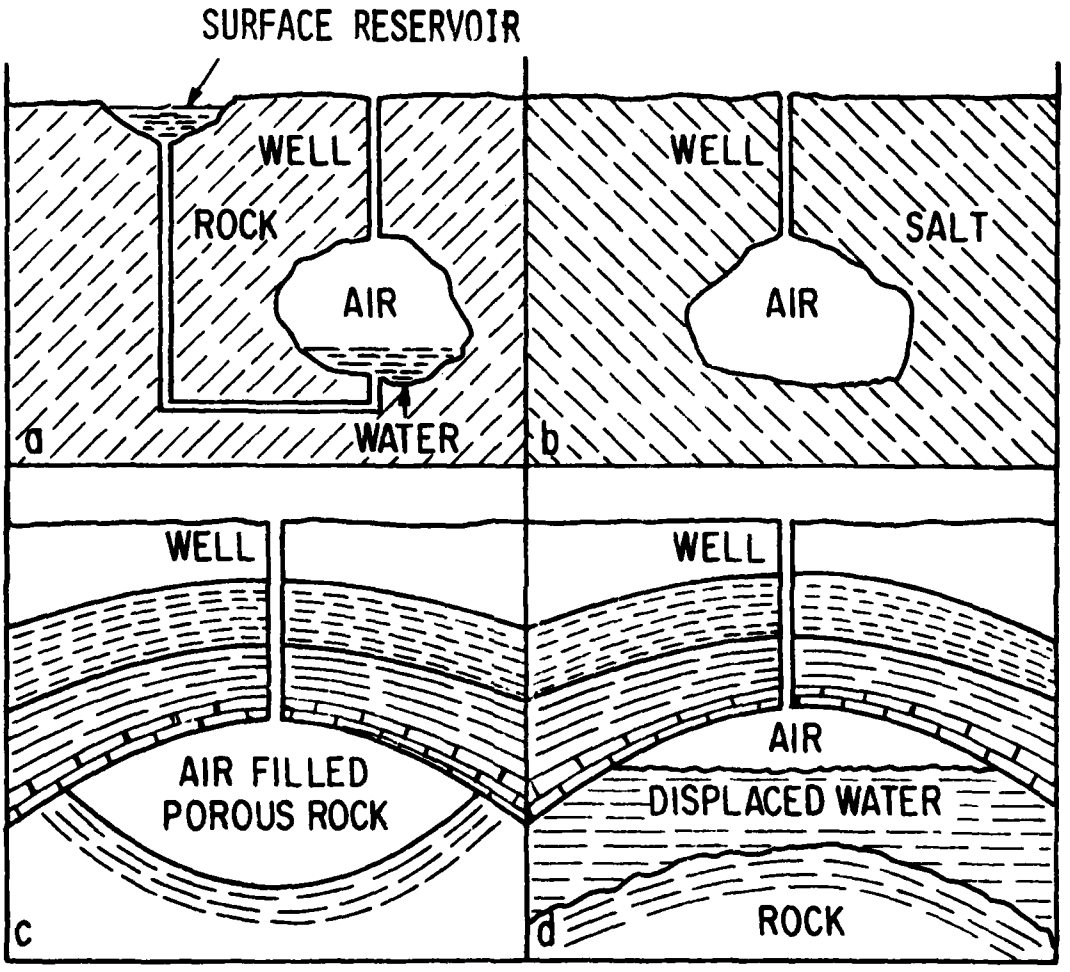


Figure 10-2. Major Types of Subterranean CAS Reservoirs.

a) constant-pressure compensated system
b) uncompensated system in salt cavern
c) storage in abandoned oil field
d) storage in bound aquifer

system uses an uncompensated salt cavity of this type. Water compensation should be avoided in this type of cavity because of incipient problems with surface brine. There are some problems with air leakage also.

Storage in porous media covers variable pressure and volume situations in both abandoned oil fields and bound aquifers. These types of reservoirs are illustrated in Figures 10-2c and d. Some problems exist with increased friability and the possibilities of explosion and contamination. Upper limits on pressure are imposed by structural strength.

10.1.3 Current Systems

Underground Air-Storage Systems:

The first and only commercial CAS went into operation in 1978 at Huntorf, West Germany, and has a generating capacity of 290 MW. It is a constant volume system located in two salt caverns, each 5,300,000 ft^3 (150,000 m^3) in volume and 2,300 ft (700 m) deep. Charging takes place over eight hours for coordination with the nighttime off-peak period; storage pressure runs between 725-1015 psi. About two-thirds of the delivered peak power comes from the compressed air. Huntorf is a fueled-cycle system (see Fig. 10-1b) that is significantly cheaper to run than a conventional gas turbine.

Tank Air-Storage Systems:

Hydro-Quebec Institute of Research built an experimental wind-powered system whose concept is also suitable for photovoltaics. Costs, based on a spherical steel tank, were $240/kWh.

10.2 Flywheels

10.2.1 General Features

Traditionally flywheels have been used for high rate, short term energy storage in applications such as punch presses, piston engines, and grain drying machines. Improved flywheels are now being examined for use in vehicular propulsion, power generation, waste heat recovery, and as battery alternatives in such cases as solar photovoltaic residences. Technical improvements are needed in bearings, low cost rotors, continuously variable transmissions, controls, and power converters.

Flywheels have a number of advantages over competing storage technologies:

- Simplicity, little maintenance, and infinite shelf life
- High power density
- Rapid charge/discharge capability

- Flexibility - input/output can be electrical (AC or DC), hydraulic, mechanical, or any combination
- Space saving - independent from ambient environment - doesn't require reservoir or good bedrock

Serious shortcomings:

- In general, not yet economically competitive with other energy storage alternatives
- Exotic rotor materials must be used for high storage density
- Constant hazard of catastrophic failure
- Limited storage duration

10.2.2 Theory of Operation

The flywheel, shown in Fig. 10-3 is a massive wheel that stores rotational kinetic energy. With proper design, flywheel storage can be used with a reversible electric motor/generator. The fiber composite flywheel shown in Fig. 10-3, designed for electrical peak power applications, consists of the following major components: flywheel and shaft, bearings, support structure, vacuum system, clutch, gearbox, and utility interface equipment.

There is a wide variety of different proposed flywheel configurations and sizes. A cone shaped disc is best for isotropic materials such as steel or aluminum. Composite rotors, a still experimental type made from Kevlar or fiber-embedded epoxy, etc., work best in cylindrical configurations. To minimize windage loss, the flywheel must operate in a vacuum. If the rotor is isotropic, stress can cause complete breakage - the flywheel must be very safely contained. Although flywheel capacities can range from 4 kWh to 200 MWh, only the smaller sizes are currently feasible.

Flywheel configurations and sizes

The flywheel is integrated into a storage system as shown in Fig. 10-4. The motor converts the energy to be stored from electrical to mechanical form, and the generator converts the mechanical energy back into electricity. Inverters and converters interface between the power input and output. Peripherals include clutch, vacuum pump, speed reducers, switches, and cooling fans. Installation requirements resemble those for a turbine generator; containment vaults must be professionally installed.

10.2.3 Current and Potential Uses

Flywheel applications can be subdivided into fixed base and moving base applications. Only fixed base applications are discussed here. Fixed base describes situations in which the flywheel remains stationary. The premium here is on low-maintenance, low-cost components. Storage times in stationary applications can vary from minutes to several days. Similarly, self-discharge requirements can vary from 0.5% to 50% of full storage capacity per hour.

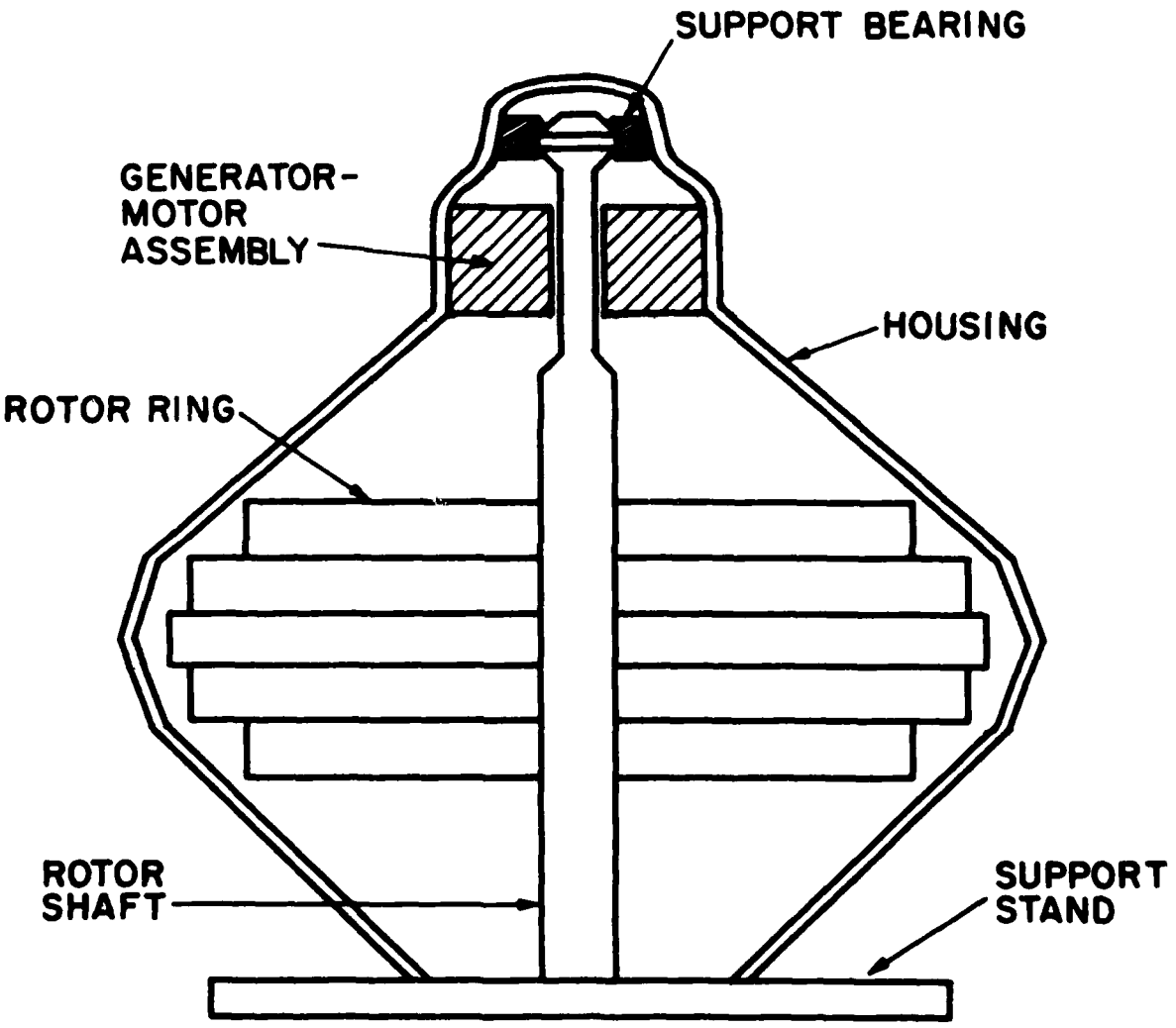


Figure 10-3. Schematic diagram of flywheel

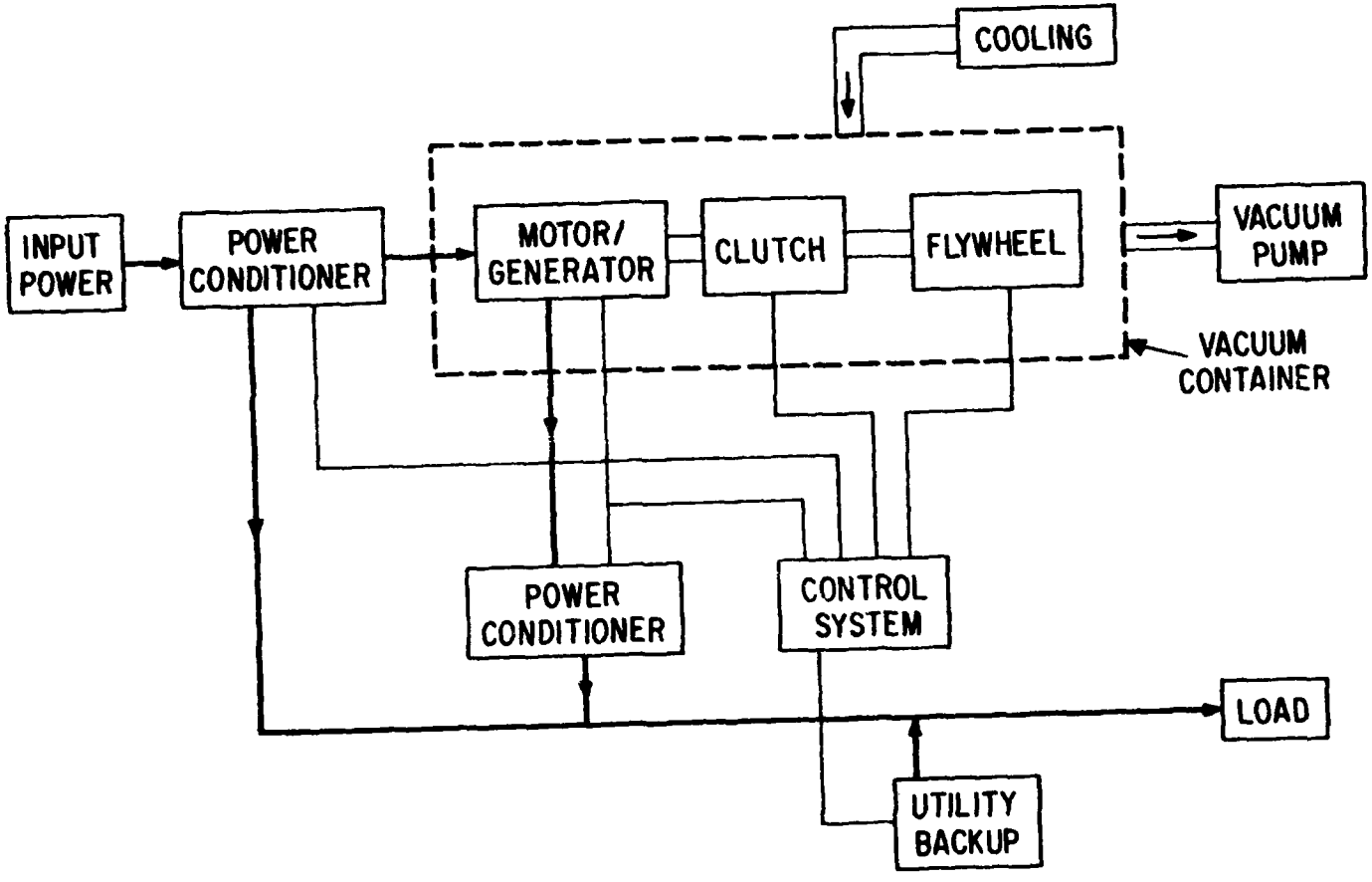


Figure 10-4. Nominal Integration of Flywheel System

Existing fixed base applications have been commonplace in machinery since the industrial revolution. Their primary role has been to mechanically correct the timing mismatch between energy supply and demand. Potentially, flywheels can be used in any application that would otherwise use batteries:

- Many day storage for remote site photovoltaic (PV) systems
- 24-hr storage for utility peaking
- Wayside subway storage
- Storage for a PV buffer in low earth orbit satellites

Cost is a major limitation now handicapping flywheel commercialization. Both the design and the composition of the flywheel rotor are being reexamined to make flywheels more cost competitive.

10.3. Other Mechanical Energy Storage Methods

Elastomeric energy storage is a relatively new option. In an axially-loaded torsion stressing scheme, isolated elastomeric rods are stretched out in the lengthwise direction. Energy is then delivered to or from the elastomer by axial rotation. In studies done on regenerative braking, it has been predicted that 100,000 J can be stored in 1-2 ft^3 of elastomer.

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11.0 PUMPED HYDRO STORAGE

Pumped hydro systems store energy by pumping water to higher elevations. When the water is allowed to flow back to lower elevations, the energy is recovered by conversion to mechanical motion, usually in an electricity-producing turbine. General features of pumped hydro storage were briefly discussed in Section 2.12.

11.1 General Features

Pumped hydro storage is currently the only widely accepted commercial storage method to provide the high-capacity storage requirements of electric utilities. These surface pumped hydro storage (SPHS) systems have demonstrated their value over the past 25 years in installations such as San Luis in California, Ludington on Lake Michigan, and Cabin Creek in Colorado in providing central energy storage to allow low cost off-peak power to meet peak power energy demands. Environmental and economic considerations are spurring research to develop pumped hydro alternatives, such as underground pumped hydro storage (UPHS). Both SPHS and UPHS systems are depicted in Figure 11-1.

Water can be pumped to higher elevations any time an energy source is available that can be readily converted to mechanical energy. Such sources include electricity, wind, and moving water. Low-head hydro systems can be used to pump water in a pumped storage system, if a hydraulic ram is used. A hydraulic ram uses air pressure built up by the momentum of falling water to pump part of the remaining flow upwards. The head of the pumped storage system can be up to 20 times higher than the head of the original hydro system. Water can also be moved to higher elevation by tidal forces.

The energy storage density of water is relatively low. A cubic foot of water raised one foot stores only 0.0802 Btu of energy; a cubic meter of water raised one meter stores only 9,800 J. The total amount of energy stored in an elevated reservoir is

$$Q = c \cdot V \cdot H \qquad (11\text{-}1)$$

where Q is the energy stored in Btu, (kWhr or J)
c is a constant = 0.0802 Btu/ft^4, or 0.0000235 kWhr/ft^4 (9800 J/m^4 or 0.0027 kWhr/m^4)
V is the volume of the reservoir in ft^3,
H is the hydraulic head, or vertical distance between the upper and lower reservoirs in ft.

NOTE: It is the volume of the upper reservoir that determines the amount of energy stored. The volume of the lower reservoir is important only if the water must be recycled.

The frictional resistance in the transportation system determines how far away the lower reservoir can be from the upper reservoir in a horizontal direction. The frictional resistance causes head loss, which is a function of flow velocity and pipe diameter. The head loss is higher for faster flow velocities and smaller pipe diameters. As a rough estimate, under normal circumstances the horizontal run to head ratio should not exceed 10:1.

Tidal power is a specialized case of pumped hydro storage in which the gravitational force between the earth and moon and between the earth and sun, coupled with the earth's rotational energy, provides the energy for pumping. Tidal power is still very much a scheme of the future - there are only two operating tidal power projects in the world; near St. Malo, France and Murmansk, Russia.

11.2 Surface Pumped Hydro Storage (SPHS)

SPHS systems have an excellent track record, despite innate handicaps of high capital costs, long construction periods, and limited site locations. As Table 11-1 shows, one of the virtues of pumped storage is its efficiency. Another advantage is the ease with which the pump/turbine motor/generator is able to add peaking capability to the otherwise inflexible operation of large base-load thermal generator units.

Table 11-1. Pumped Hydro System-Efficiencies(%)

		minimum efficiency
Pumping	• motor and transformer	97.5
	• pump	91.5
	• water passages	96.5
Generating	• water passages	95.5
	• turbine	89.0
	• generator and transformer	97.5
overall cycle efficiency		66%

There are two main types of water turbine wheels: impulse and reaction. Impulse turbines, such as the Michell (Banki) and Pelton types, are vertical wheels that use the momentum (impulse) of jets of water striking buckets that are attached to the wheel. Impulse wheels work best for low-

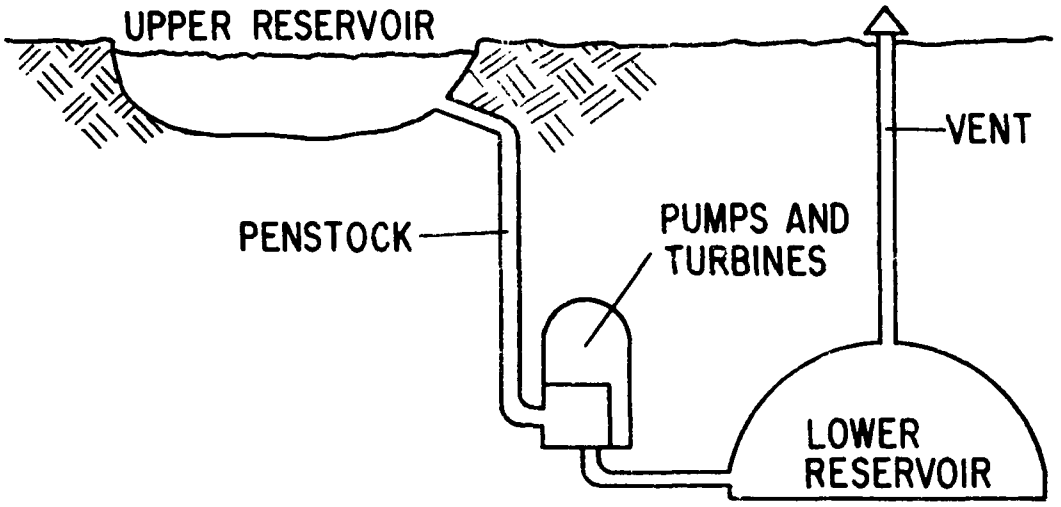


Figure 11-1a. Underground Pumped Hydro Storage (UPHS)

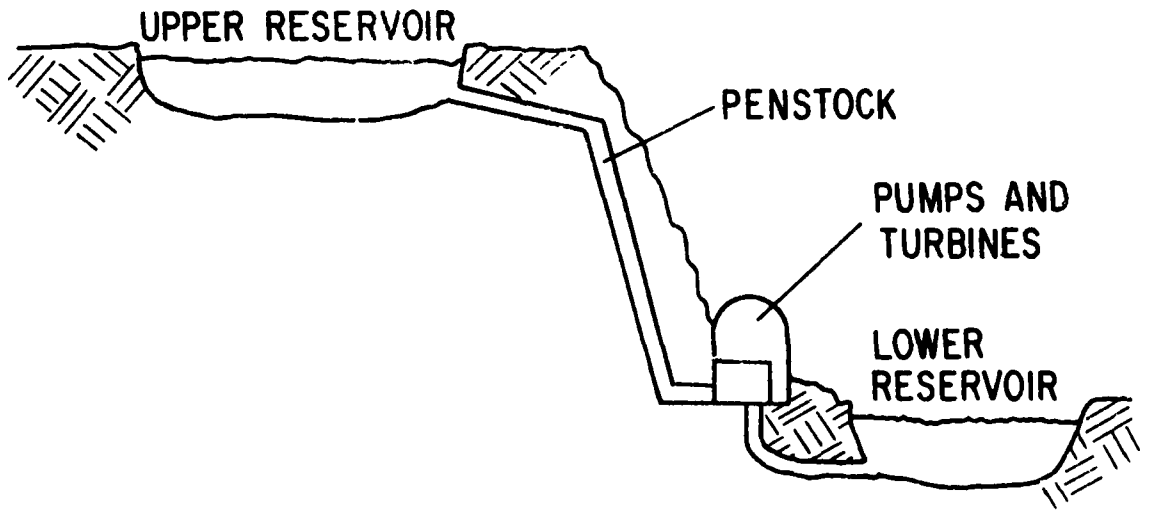


Figure 11-1b. Surface Pumped Hydro Storage (SPHS)

volume, high head systems. Reaction turbines, such as the Francis or Kaplan, are horizontal wheels completely under water. The reaction of water flowing from the dam causes the wheel to rotate. Kaplan turbines work best for high-volume, low-head (14-120 ft) systems. The Francis turbine is the most common type because it can be used for a wide range of heads (100-1500 ft).

Reservoirs for pumped storage operation are selected on the basis of volume, close geographic proximity, and head. Reservoirs can be formed from natural bodies of water such as lakes, or reservoirs of existing water storage systems. Environmentally, it is usually considered preferable to use dedicated man-made surface reservoirs. Locations containing engineering artifacts, such as old abandoned hydrosites with water wheels or irrigation dams, water supply lines, navigation locks remaining from river transportation, and attendant dams can be used to increase the head and thereby increase the plant capacity.

Near Othello, Washington the first of a series of low-head hydroprojects using falling irrigation water has been built. The first irrigation canal hydro project since the 40's, the plant will generate 25 million kilowatt-hours annually during the seven irrigating months. The plant consists of the canal, an intake structure, outlet works, and 12-foot diameter penstock, a pressurized pipe carrying water to the turbine and generator.

Small hydro plants are most useful adjacent to the area of electrical consumption, so that transmission distances are short and the necessary infrastructure is minimal. Small hydropower plants are capable of operating with very little maintenance for over 20 years.

Limitations

SPHS is a very effective energy storage technology, but it has several drawbacks:

- Environmental problems: Systems must be located in an area of great topographical relief (i.e. mountainous and usually scenic). There is often public disapproval to both the initial site and the associated miles of transmission lines. There is sometimes ecological disruption; altered fish migration patterns, excessive lake sedimentation, etc.

- Geological/hydrological problems: It is often difficult siting two adjacent reservoirs that have both an adequate head and close proximity to the energy consumption area.

- Economical problems: Licensing, regulation, and insurance costs can all take their toll. Often construction is delayed due to public opposition which also raises costs.

11.3 Underground Pumped Hydro Storage (UPHS)

UPHS is much like conventional pumped storage: functionally identical, provides peaking power, and has the same components--upper reservoir, waterway passages, powerhouse, and lower reservoir. The difference is the lower reservoir - it is a cavern hundreds of feet below the ground rather than a natural body of water.

The upper reservoir can be much smaller than the one that would be required for a SPHS system of the same capacity. This is because the vertical distance in an UPHS system can be much larger than in a SPHS system, and, as shown in Equation (11-1), energy storage capacity is directly proportional to the height of the head of water. Practical cavern excavation and tunneling methods already exist for construction of the lower reservoir, and high-lift pump-turbine technology is also available.

Underground pumped hydro storage has not yet seen commercial application but may be economical in some situations. The important factors affecting UPHS locations are site geology and the location of major load centers. Favorable locations are characterized by (1) igneous intrusive or crystalline metamorphic bedrock (in other words, solid rock with few discontinuities) and (2) hydrologic stability. Sites to be avoided are previous water-bearing sandstone, deformed rocks, air slaking rocks, and faulted, jointed rock. Typical cost estimates for several UPHS configurations are listed in Table 11-2. The most important cost component is the lower reservoir cost, amounting to 25-30% of the plant total. Recent studies have shown that as the net head increases (all other factors constant), the required underground reservoir volume decreases. The general trend of system cost in relation to net head is graphed in Figure 11-2. The optimum head for a UPHS plant appears to be 3000-5000 ft (1000-1500 m).

Although either UPHS systems or compressed air storage (CAS-discussed in Chapter 10) systems can provide central energy storage for power utilities, the differences between the two make them appropriate for different circumstances. In some cases, combining the two systems would appear to reduce costs due to better integration of equipment and system optimization. A combined UPHS/CAS hybrid is created by pressurizing the air in the underground reservoir of the hydro plant. During the peak hours, water from the upper reservoir flows through the turbines to the underground reservoir, and air is forced up to the surface. During offpeak periods the water is pumped out of the underground reservoir, and air is drawn in.

11.4 Available Systems

Small Systems

There are a number of off-the-shelf turbine units available for small hydro systems.

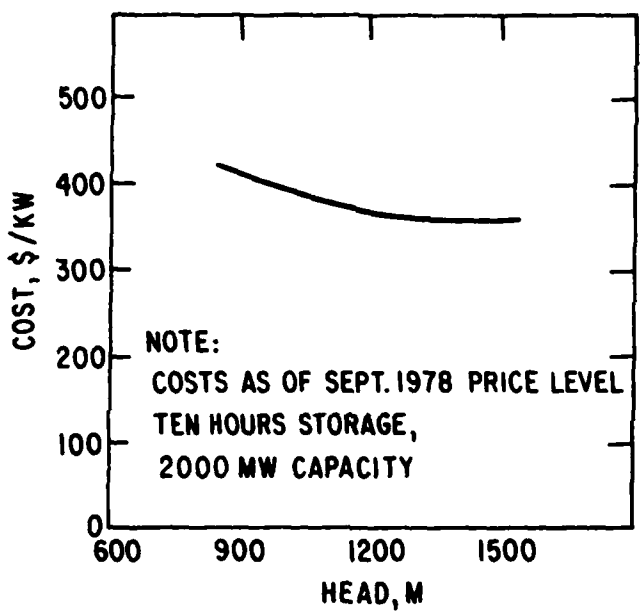


Figure 11-2. Relationship between system cost and net head.

Table 11-2. Cost Estimate Summary on Several UPHS Systems*

Item/System	Multistage RPT	Single Stage RPT (two-drop)	Tandem Unit
Land and Land Rights	2	2	2
Power Plant Structures and Improvements	71	89	75
Reservoirs, Dams, and Waterways	293	320	297
Pump/Turbines	76	44	117
Generators/Motors	52	49	54
Miscellaneous Equipment	51	58	52
Roads and Bridges	6	6	6
Contingencies, Engineering, Supervision and Overhead	183	194	197
Total	734	762	800
$/kW	367	381	400

*1) Adapted from Tam, et al. All three plants are 2000 MW, 1200 m head with 10 hrs storage. RPT is a reversible pump turbine. The two drop scheme uses two RPT at different heights. The upper RPT has a small intermediate reservoir. Tandem unit has several RPT at the same height.

2) Units in 10^6 September 1978 dollars.

3) $/kW in Setember 1978 dollars.

The James Leffel Co. (Springfield, Ohio) has been producing small (and large) hydroelectric units for over 100 years. Their small units, called Hoppes, run on a maximum of 25 ft of head with a maximum output of 10 kW. Cost (1974) was $7,200.

Small Hydroelectric Systems and Equipment (Acme, WA) both manufactures hydro equipment and sells guidebooks.

KacKeller Engineering (Grantown-on-Spey, Scotland) manufactures a range of hydroelectric turbo-generators designed for an output range of 5-30 kW with an 18 ft (6 m) head of water. The manifold is fabricated from steel sections and fitted with a vertical propeller and a belt-driven generator. A solid state control unit maintains a steady output voltage under varying load conditions by dumping excess power into a ballast load. Cost is $11,000-$16,000 (1981). Maintenance is low and turbine lifetime is more than 25 years.

Rife Hydraulic Engine Manufacturing Company (Millburn, NJ) manufactures three different types of hydraulic rams with intake pipe sizes ranging from 1 1/4-8 inches and cost ranging from $300 - $3,000 (1974).

Large Systems

Allis Chalmers Corporation, Hydro Turbine Division (York, PA) was, in 1977, the only U.S. based manufacturer with experience in producing both large scale hydraulic turbines and generators for hydroelectric projects. The Hydro Turbine Division has also worked closely with Argonne National Laboratory to design 500 MW, single-stage reversible-pump/turbine units for operation at heads of up to 3000 ft (1000 m).

There are many other companies that manufacture large turbines, dams, valves, transformers, etc. The trade journal Water Power is a good source of further information.

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12.0 ELECTROCHEMICAL ENERGY STORAGE

Electricity may be stored in the form of chemical energy in electrochemical storge devices. During charging, electrical energy is electrochemically converted to chemical energy; during discharge, the stored chemical energy is electrochemically converted back to electricity. Electrochemical energy storage was discussed briefly in Section 2.13.

Basic characteristics of electrochemical storage are: absence of moving parts, rapid electrical response, compactness, modularity, and for most devices no noise and little odor problem. Because of the modular design, electrochemical storage plants are generally reliable, need minimal maintenance, are easy to expand, and have a short lead time from order to installation. Another feature of electrochemical storage is that only insignificant amounts of heat are given off, even at high discharge rates. Consequently, no special provisions to regulate temperature are required, except at very cold ambient temperatures or for some experimental batteries that operate only at elevated temperatures. In most cases the active ingredients of the electrochemical cells can be stored in the cells for long periods of time.

Batteries are the best known electrochemical storage device. Batteries which are economically rechargable are called secondary batteries. Primary batteries, which are not rechargable, are not discussed in this handbook. Most of the discussion in this chapter is about batteries. Other electrochemical devices, e.g. fuel cells, are discussed in separate sections.

Secondary batteries can be operated over varying loads and are more efficient when used at part load. These batteries have been used for many years in vehicle transportation in the form of auto start batteries and traction batteries. Research in this area continues to play a large role in the further development of battery technology. Secondary batteries have also been used in stationary applications, such as powering telephone switching circuits. They are currently being considered by utilities for peak-load leveling.

Recently batteries have been coupled with renewable energy sources, such as photovoltaic cells and wind generators. The development of low-cost photovoltaic cells will make this application even more attractive in the near future. Currently, coupling batteries with either photovoltaic cells or wind generators is economical only at remote sites away from the utility grid.

12.1 Basic Battery Electrochemistry

Batteries are composed of modular units called cells. During discharge the energy of electrochemical reactions in each cell is converted directly into low voltage direct current electricity. The voltage of an individual cell depends mostly on the chemical makeup of the cell, and to some

extent on the state of discharge and temperature. The voltage of an individual cell rarely exceeds two volts. Individual cells may be connected electrically in series to produce higher voltages.

A typical battery cell is illustrated schematically in Figure 12-1 for the case of a lead-acid battery. Each cell contains a positive electrode, a negative electrode, and a conductive solution called an electrolyte. Both electrodes have facilities for making electrical contact with an outside circuit. The electrolyte provides charged ions to transfer electrons from one electrode to the other inside the cell. In the current example the electrolyte is a sulfuric acid (H_2SO_4) solution. When dissolved in water most of the sulfuric acid molecules split into two ions of positively charged hydrogen (H^+) plus one ion of negatively charged sulfate (SO_4--). Each electrode has its own potential voltage difference with respect to the electrolyte. The voltage of the cell is the sum of these two potential differences. The value of the potential differences is independent of the amount of electrode or electrolyte present, but it does depend somewhat on the temperature and concentration of the sulfuric acid. As temperature decreases, the viscosity of the electrolyte increases; the electrolyte molecules move slower. Of greater importance, the chemical activity of the electrode material in the cell is decreased at lower temperature. Similarly, as the concentration of the electrolyte is reduced, there are fewer ions to carry electrons, and cell activity is reduced. In addition, the electrodes become more nearly chemically similar, which further reduces the cell voltage.

Discharging

When the battery cell is connected to an external electrical load, electric current flows from the positive electrode through the load and into the negative electrode. Physically this is accomplished by the flow of negatively charged electrons from the negative electrode through the load and back to the positive electrode.

Electron transport inside the cell is accomplished by means of a redox reaction. The positive electrode is reduced, and the negative electrode is oxidized. In the example illustrated in Figure 12-1 the positive electrode is composed of lead peroxide (PbO_2), and the negative electrode is composed of sponge lead (Pb). During discharge the reaction at the positive electrode is:

$$PbO_2 + 2H^+ + H_2SO_4 + 2e^- \rightarrow PbSO_4 + 2H_2O \qquad (12\text{-}1)$$

The corresponding reaction at the negative electrode is:

$$Pb + SO_4^{--} \rightarrow PBSO_4 + 2e^- \qquad (12\text{-}2)$$

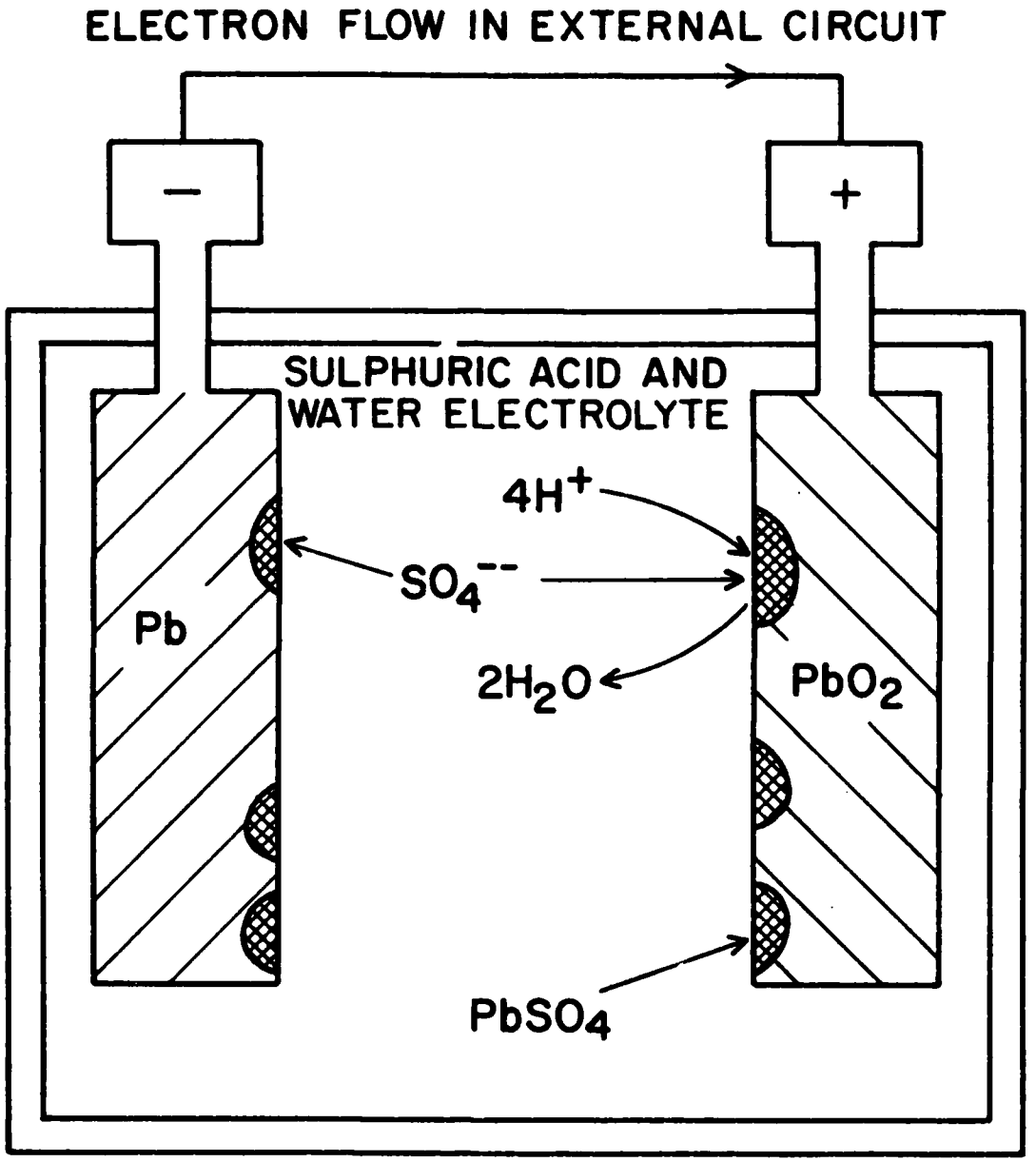


Figure 12-1. Discharge of lead-acid battery cell

Lead sulfate ($PbSO_4$) is produced at both electrodes, and the concentration of sulfuric acid is reduced. Specific gravity measures the sulfuric acid concentration and is often used to measure the state of charge of a lead-acid battery.

When any one of the active components of the electrodes or electrolyte (Pb, PbO_2, or H_2SO_4 in the example) is no longer available, it is impossible to draw electric current from the battery.

Charging

During charging of the battery cell, a counter voltage is imposed upon the cell terminals that forces current through the cell in a direction opposite to the flow during discharge. Electrons enter the cell through the positive electrode and leave the cell through the negative electrode. For the lead-acid battery, the reactions at each electrode proceed in the opposite direction to that indicated in equations (12-1) and (12-2). In that example, when the sulfate (SO_4) is completely removed from either plate, the cell is fully charged.

Most batteries will slowly self-discharge if left idle for extended periods of time. Self-discharge results from slow chemical reactions between the electrodes and the electrolyte. The amount of self-discharge in a given time will vary considerably, depending on battery type. In general, capacity loss increases with age and is accelerated by elevated temperature. In multicell batteries, cumulative effects of small differences in characteristics from cell to cell are a problem when overcharge reactions do not exist. Indication of the state of charge for each cell is necessary to design control circuitry to prevent overcharging.

12.2 Types of Storage Batteries

Storage batteries can be broadly classed by composition into the categories: lead-acid, alkaline, high temperature, metal-air, and others. Only lead-acid and alkaline batteries are commercially available. Table 12-1 compares some features of several batteries that are either commercially available or under development. The number of discharge cycles will vary considerably depending on construction details and application and should only be used as a general comparison.

12.2.1 Lead-Acid

Lead-acid is the most commonly used secondary battery. Its composition and electrochemical reactions were briefly discussed in Section 12.1 and diagrammed in Figure 12-1. The battery is named for its plates, which are lead and lead peroxide, and for its electrolyte, which is sulfuric acid. The cells are usually contained in hard rubber or plastic.

Table 12-1. Comparison of principle types of storage battery

Type	Open Circuit volts/cell	Energy Density Whr/lb.	Whr/kg	Whr/in^3	Whr/dm^3	discharge cycles	initial cost
			existing technology				
Lead-acid	2.0	10-15	22-33	0.6-1.3	37	300-1500	low-moderate
Nickel-iron	1.4	10-14	22-30	0.6-1.0	37-61	2000-5000	moderate
Nickel-cadmium	1.3	8-11	18-24	0.4-0.8	24-49	1000-3000	high
Silver-zinc	1.5	20-100	44-220	3.0	180	100-2000	high
Silver-cadmium	1.2	15-50	33-110	2.5	150	200-300	high
Nickel-hydrogen	1.4					>4000	high
			under development			(currently)	(projected)
Nickel-zinc	1.7	30-35	66-77	2.3	140	350	low
Iron-air	1.3	35	77	1.3	80	200	low
Zinc-air	1.6	35-45	77-100	1.3	80	100	moderate
Zinc-chloride	2.1	55-70	120-150	3.0	180	100	low
Sodium-sulfur	1.8-2.1	55-65	120-140	2.8	170	2000	low
Lithium Aluminum/ Metal Sulfide	1.4-1.9	60-85	130-190	3.5	210	200	low

The lead-acid battery has many advantages. It has the highest volts/cell of any commercially available secondary battery. It also has one of the lowest costs/Whr of stored energy. The electrolyte is unlikely to freeze, except perhaps at the end of a discharge cycle under arctic conditions. While most lead-acid batteries will emit gas during charging and require the addition of distilled water at frequent intervals, batteries are commercially available that require no maintenance and with lifetimes of over five years. The self-discharge rate is relatively low; lead-acid batteries should nominally be recharged every six months if left idle.

The number of discharge cycles that can be sustained depends on the battery construction and on the severity of discharge. The cycle life is shortened by excessive discharge rates, repeated deep discharge, excessive charge rates, and long periods of inactivity. Generally a lead-acid battery will last at least 1000 cycles if cycling is limited to 10-15% of the total depth of discharge. If cycling is 80% of the total depth of discharge, then 300 cycles are more likely. Cells designed for automobile starting have a shorter life when deeply discharged than cells designed for deep discharge.

Types of Plates

There are four general types of plates used for the electrodes of lead-acid batteries: pasted plates, Manchester plates, Gould spun plates, and tubular plates. In general the plates are porous to allow more surface area for electrochemical reaction.

In pasted plates the active ingredient, usually lead oxide (PbO) is pasted into the spacings of a lead alloy grid. The lead oxide is then changed into lead or lead peroxide, depending on whether the plate is to become a cathode or anode. Pasted plates are often used in auto batteries.

Manchester plates use a grid with circular openings into which lead buttons are pressed. The buttons are made by crimping and then rolling up lead tape. Lead peroxide is then chemically formed on the surface of the tape to form positive plates. Batteries using Manchester plates are usually heavy and bulky and used in stationary applications.

Gould spun plates are thick positive plates formed by pressing a heavy sheet of lead between disks, which forms a fin like structure into the lead. Lead peroxide is then formed on the surface of these plates. The positive plates are combined with pasted negative plates and used mostly in stationary applications.

Tubular type positive plates hold the active material in porous walled tubes. A conductor is at the center of each tube. This type of plate is used when many deep discharge cycles are required.

12.2.2 Alkaline

Alkaline batteries derive their name from the electrolyte, which is an alkaline solution. The electrolyte is usually potassium hydroxide (KOH), although sodium hydroxide (NaOH) is used in a few cases. The concentration ranges from 20-45%. For the higher concentration freezing is eliminated even under arctic conditions. Unlike the lead-acid battery, the concentration of electrolyte does not change and cannot be used to indicate the state of charge; however, less electrolyte is needed, and the battery cells can be correspondingly smaller.

Alkaline batteries are generally more expensive than lead-acid batteries, but the different types of alkaline batteries offer several advantages. The major types that are commercially available include: nickel-cadmium, nickel-iron, nickel-zinc, silver-zinc, and silver cadmium.

Nickel-cadmium

Nickel-cadmium batteries are long-lived but are too expensive for large applications. The cost is determined by the price of nickel. The batteries have a very low self-discharge rate and can be charged rapidly. Although the details of the electrochemical reaction are complicated, the discharging reaction can be represented most simply by

$$Cd + 2NiOOH + 2H_2O \longrightarrow Cd(OH)_2 + 2Ni(OH)_2 \qquad (12\text{-}3)$$

where the left hand state represents the charged state with metallic cadmium (Cd) as the negative electrode and nickelic hydroxide (NiOOH) as the positive electrode. The discharged state is represented on the right hand side with cadmium hydroxide ($Cd(OH)_2$) as the negative electrode and nickelous hydroxide ($Ni(OH)_2$) as the positive electrode. KOH, the electrolyte, is not shown in the reaction, since its amount does not change.

Oxygen is generated at the positive electrode, and hydrogen is generated at the negative electrode once the cell is fully charged. Venting of the cells is usually required, although small cells can be sealed.

Increased discharge rate and low temperature both decrease the voltage and capacity of the cells, but this effect is much less severe than for lead-acid batteries.

Nickel-zinc

Nickel-zinc batteries offer a high energy density at a moderate cost. Nickel is the positive electrode and zinc is the negative electrode. The discharging reaction can be written as

$$2NiOOH + Zn + H_2O \longrightarrow 2Ni(OH)_2 + ZnO \qquad (12\text{-}4)$$

The major drawback is the small number of cycles available. Zinc electrodes have a tendency to form zinc dendrites and short the cells out upon the slightest overcharge. Also zinc oxide (ZnO) is partially soluble in the electrolyte. This results in zinc hydroxide forming in the pores of the nickel electrode, reducing the active surface area there. The zinc electrode loses capacity by shape change and passivation, both of which are aggravated by overcharge and fast recharge.

Nickel-iron

Nickel-iron batteries offer an excellent cycle life and rugged construction but at a high initial cost. As in the two batteries mentioned above, the cost is mostly determined by the price of nickel. The battery can be totally discharged in repeated cycles.

Finely divided hydrated nickel peroxide is used for the positive plate. The reaction at the positive nickel electrode is:

$$NiOOH + H_2O + e^- \longrightarrow Ni(OH)_2 + OH^- \qquad (12\text{-}5)$$

There are two reactions possible at the negative iron electrode.

$$Fe + 2(OH^-) \longrightarrow Fe(OH)_2 + 2e^- \qquad (12\text{-}6)$$

$$Fe(OH)_2 + (OH^-) \longrightarrow Fe(OH)_3 + e^- \qquad (12\text{-}7)$$

Usually the batteries are configured to operate using only Equation (12-6).

Since the potential of the negative iron electrode is close to that of hydrogen, the cell is continually forming hydrogen gas at the iron electrode while idle or charging. Thus, the self discharge rate is high, and the cells require constant ventilation. The battery has a poor low temperature performance and consumes more water during charging than do lead-acid batteries.

Silver-zinc

Silver zinc batteries provide high energy densities and minimum weight but are very expensive due to the cost of silver. The overall discharging reaction of the cell can be written.

$$AgO + Zn + H_2O \longrightarrow Ag + Zn(OH)_2 \qquad (12\text{-}8)$$

where silver oxide (AgO) is the positive electrode, and sponge zinc (Zn) is the negative electrode. The cells have a poor cycle life due to deteriora-

tion of the zinc electrode, however they can retain 80-95% of their charge after an idle period of one year.

Silver-cadmium

Silver-cadmium batteries are similar to the silver-zinc batteries discussed above. The difference is the cadmium negative electrode, which gives the cells about five times more deep discharges than the silver-zinc cells. The shelf life is about twice as long.

Others

Alkaline batteries with limited reversibility include manganese-zinc, mercury-zinc, and mercury-cadmium. These batteries are usually used as primary batteries but do have some recharge capability.

12.2.3 High-Temperature

High temperature batteries operate at temperatures significantly above ambient. They offer great potential in providing low cost, high energy density, storage batteries but are still in the developmental stage. The high temperature required for operation means that thermal losses will occur, and the batteries must be insulated. The small size of the batteries should help to make the losses through the insulation low. Also, the reject heat from cycle irreversibilities may provide the necessary heat when the batteries are used regularly.

Lithium-aluminum--iron-sulfide

Lithium-aluminum--iron-sulfide batteries, originally developed at Argonne National Laboratory, use a molten salt electrolyte consisting of LiCl/KCl eutectic, with a melting point of 666°F (352°C). The simplified reaction at the positive electrode is:

$$4\,Li^+ + 4e^- + FeS_2 \longrightarrow 2Li_2\,S + Fe \qquad (12\text{-}9)$$

The reaction at the negative electrode is:

$$LiAl \longrightarrow Li^+ + Al + e^- \qquad (12\text{-}10)$$

Lifetimes of over two years have been accomplished with low performance cells so far, high performance cells have not lasted beyond 200 cycles.

Sodium-sulfur

There are two types of sodium sulfur batteries under development. Both use molten sodium and sulfur as the active materials. The first type uses an ion conducting hollow fine glass fiber to separate the sodium from the sulfur. The molten sodium is inside the fiber, and the fiber is immersed in a sodium sulfide catholyte. To date, breakage of the glass fibers has limited the cycle life.

The second type of sodium sulfur battery, known as the beta battery, uses a ceramic electrolyte of beta alumina ($Na_2O\overline{"}11Al_2O_3$) which is ionically conductive at 570-660°F (300-350°C) with sodium ions as the charge carriers. Sodium, the negative material, and sodium polysulfide, the positive material, are both liquid at the operating temperature, and the electrolyte serves as the separator. During discharge the active materials combine to produce sodium polysulfide. Since beta alumina is not permeable to either sulfur or sodium reactants, self discharge is prevented. The beta battery is still being perfected. One of the largest problems is degradation of the beta alumina, which does not last past 500 cycles in the hot and corrosive environment found inside the battery. Recent laboratory developments indicate that this problem may be solved, but at the time of this writing (December, 1981), this has not yet been validated in long term testing.

12.2.4 Metal-Air

A metal-air battery uses a conventional metal electrode such as zinc or iron combined with a reversibly operating oxygen electrode. Zinc-air has long been used for primary batteries because of its high energy density. The difficulty in obtaining a reversible metal-air battery is that the oxygen electrode is subject to corrosion. In practice this has been recently overcome by using a two-layered electrode. A hydrophobic layer faces the gas - a hydrophilic layer faces the electrolyte.

Another serious lifetime limitation on multiple cycles is the reaction of KOH electrolyte with CO_2 in the air. The cells make excellent scrubbers; however, that is not the intended application here. This problem (electrolyte reaction) may limit metal air cells to a primary battery role only.

An example of a metal-air battery is iron-air. KOH is the electrolyte. During discharge the iron electrode is oxidized to iron oxide, and oxygen from the air is reduced to hydroxide.

12.2.5 Other

Many other types of batteries are under development.

Zinc-chloride

One promising example is the zinc-chloride battery, which potentially is of high density and low cost. One system under development uses $ZnCl_2$ as the electrolyte. Upon charging, zinc is plated out and chlorine gas is liberated. The chlorine gas contacts the cooled electrolyte to form chlorine hydrate which is stored as a solid. Upon discharge, warm electrolyte circulated through the frozen hydrate, liberates chlorine gas which dissolves into the electrolyte. The electrolyte is then circulated to the battery stack to react with the zinc. Zinc-chloride batteries can opeate at ambient temperatures and produce a constant voltage. Their major drawback is the complicated plumbing and the refrigeration system that is required and the possible environmental problem with chlorine gas.

Zinc-bromide

A similar type of battery uses a zinc-bromine reaction. The reactions. take place on opposite sides of a bipolar electrode plate, which is a thin nonporous carbon sheet. The negative electrode reaction during charging is the deposition of zinc onto one side of the sheet. On the other side of the sheet during charging, bromide ions are oxidized to bromine, which dissolves in the aqueous electrolyte. The electrolyte is circulated as in the zinc chloride battery.

Nickel-hydrogen

The nickel-hydrogen battery, originally designed for the space program, is a potentially good candidate for coupling to photovoltaic cells. The battery has deep discharge capability and has a potential lifetime of twenty years. The cells are hermetically sealed and require no maintenance. They can also be repeatedly overcharged or undercharged without damage. The biggest drawback at present is the high cost. The batteries also have a high self-discharge, and the charge efficiency decreases with increased temperature.

12.3 Fuel Cells

Like any electrochemical cell, a fuel cell consists of two electrodes separated by an electrolyte. The fuel cell differs from the battery in that the chemical energy is not stored in the electrodes, but rather in reactants (fuel and oxidizer) which are stored outside the cell. An immediate advantage is that recharging consists simply of refilling the fuel tanks.

One of the advantages of the fuel cell is that the electrochemical reactions can take place at ambient temperature. To accomplish this, the reaction is usually catalyzed with a noble metal, e.g. silver, platinum, palladium, etc.

The reactants can be any pair of materials that are oxidizing and reducing. The principle fuel cell reactions are:

$$2H_2 \longrightarrow 2H_2O \qquad (12\text{-}11)$$

$$C + O_2 \longrightarrow CO_2 \qquad (12\text{-}12)$$

$$2C + O_2 \longrightarrow 2CO \qquad (12\text{-}13)$$

$$2CO + O_2 \longrightarrow 2CO_2 \qquad (12\text{-}14)$$

The latter three reactions are under study by utilities to increase the efficiency of fossil fuel power plants. Part of the electricity is generated in the fuel cell. The exhaust is high enough in temperature to drive a gas and steam turbine bottoming cycle.

The reaction with hydrogen and oxygen is more likely to be useful in an energy storage application. The hydrogen and oxygen can be initially generated by electrolysis of water. The operation of a hydrogen-oxygen fuel cell is illustrated in Figure 12-2. In the pores of the hydrogen electrode the hydrogen molecule is adsorbed on the catalyst and split into positive hydrogen ions (H^+) and electrons (e^-). The electrons are conducted to the external electrical circuit. The hydrogen ions react with hydroxyl ions (OH^-) to form water (H_2O). At the oxygen electrode, oxygen molecules are adsorbed on the catalyst, where each molecule combines with two electrons and a water molecule to form a hydroxyl ion and a perhydroxyl ion (O_2H^-). The perhydroxyl ion is catalytically split into a hydroxyl ion and an oxygen atom. The oxygen atom is free to participate in further reactions or to exit in the exhaust. The reactions proceed as long as the hydrogen fuel is supplied. The oxygen can be derived from the air.

12.4 Applications

The lead-acid battery represents the state of the art for large secondary batteries. To date, these batteries have found application in automobiles, locomotives, forklifts, submarines, and standby power for nuclear reactors, communications equipment, and lighting.

Batteries are also useful in load leveling applications. Utilities are currently interested in batteries for peak-load shaving during the three-five hours of the day when electricity demand is highest. In addition to the lead-acid battery; zinc-chloride, zinc-bromide, and the beta battery (sodium-sulfur) are all being considered for this use in the near future. Large electricity users with high demand charges could also benefit by charging batteries with off-peak power and using the stored energy during highest demand periods.

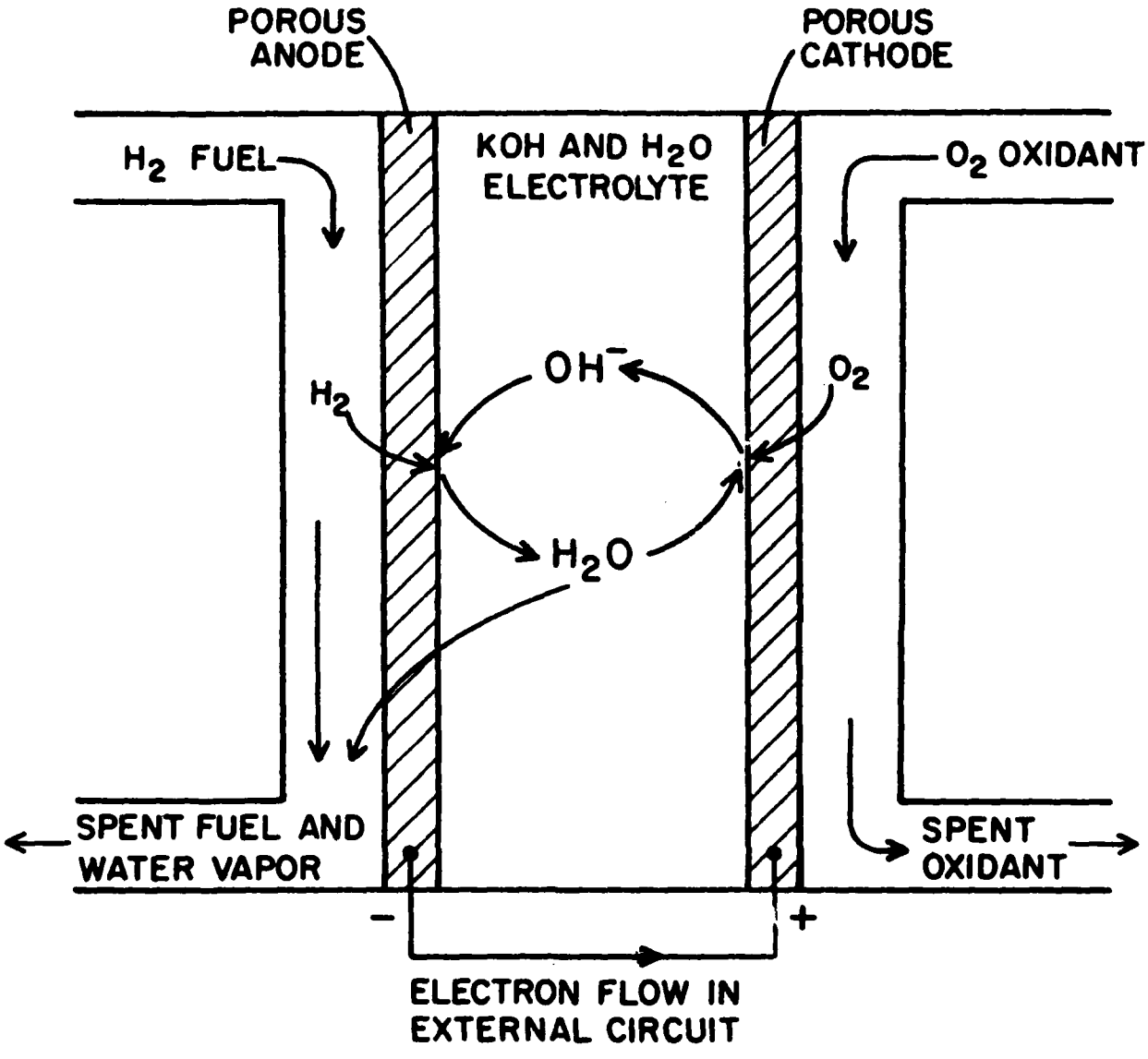


Figure 12-2. Oxidation of hydrogen in fuel cell

Batteries are also useful in combination with photovoltaic cells or wind generators. Currently this application is only economical in locations far from an existing power grid. For such stand-alone systems the following properties would be found in the ideal battery:

(1) low cost,
(2) capable of deep discharge,
(3) more than 4000 cycles,
(4) energy efficiency greater than 65%,
(5) ability to take overcharge,
(6) no maintenance, and
(7) indication of state of charge.

A relatively high self-discharge rate can be tolerated if the batteries are charged daily. Of all the batteries available in the near future, only the nickel-hydrogen battery is closest to meeting all of the requirements. It fails only in its high cost.

Currently, only lead-acid batteries offer a proven low-cost storage technology for coupling with photovoltaics. Lead-acid batteries have been used in the first photovoltaic village power system, installed at Schichuli, Arizona.

One residential system, currently under development by Texas Instruments, immerses small silicon solar cells in the electrolyte of a fuel cell. The current generated separates hydrogen-bromide (HBr). The products of separation, hydrogen gas (H_2) and bromine (Br_2), are stored separately and later recombined in a fuel cell. The HBr product is then returned to the solar cell.

12.5 Charging

Proper selection of the charging procedure and voltage and current of charging is essential in achieving maximum battery performance and life. Excessive overcharging causes increased battery temperature, gassing, and corrosion, which deteriorates cycle life and performance. Consistent overcharging causes a gradual decrease in battery capacity, which is usually irreversible. Because of difference between cells in such characteristics as self-discharge, amount of material, etc., each cell must be balanced on charging over the life of the battery. In series-connected cells, each cell must be charged to full capacity, and no cell should be overcharged. Both the voltage of charging and the current levels must be optimized for most batteries. Manufacturer's specifications should be followed with regard to charging, and the state of charge in the cells should be monitored frequently.

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13.0 DESIGN METHODS

The purpose of a design method is to estimate the performance of a given system in a specific application. For energy storage systems, performance is usually measured in terms of the amount of energy consumed or the cost of providing energy. The application is providing a certain task, e.g. maintaining a comfortable temperature and humidity in a building, and will usually depend on climatic factors such as ambient temperature, amount of sunshine, etc.

In the last few years many design methods have become available that incorporate renewable energy sources such as solar, wind, and photovoltaic. The three SERI (Solar Energy Research Institute) publications listed in Section 13.2 contain detailed descriptions of most of the available design methods. (Updates of three publications may be available from SERI by the time this manuscript is published). This chapter discusses only the advantages and disadvantages of each method in general terms.

13.1 Types of Design Methods

Design methods can be broadly classified into: manual, programmable calculator, small computer, and large computer. The list is in order of increasing cost, but also increasing speed and flexibility.

Manual Methods

Manual design methods include rules of thumb, nomograms, tables, graphs, and worksheets. An example of this latter method is given in Chapters 14 and 15 for several applications.

For technologies in the early stages of development, manual methods are based on intuition and basic equations. For mature technologies the methods are supplemented with correlations from practical experience and from parametric studies of more sophisticated methods.

The major advantages of manual methods are low cost and wide availability. The major disadvantages are that hand methods are limited in the range of problems that can be solved and that some of the methods are tedious to apply. Manual methods usually demand some judgment based on experience in their use on any particular application. For small, low-cost, one-of-a-kind, systems, manual methods are often the only design method used because more sophisticated methods are too expensive. For large high-cost systems, or for systems that will have many similar installations, manual methods are useful in the early schematic and sizing stages of design.

Programmable Calculator Methods

Programmable calculators are available that can run programs of many steps and store these programs on magnetic cards or in electronic memory modules. Programs of varying complexity are available for these calculators. In many cases these programs are calculator versions of manual methods. Small finite difference heat transfer calculations can be performed in a reasonable amount of time, but require considerable judgment on the part of the user.

The advantage of using a programmable calculator is that it is much quicker than manual method. A disadvantage is that it is more expensive. A limitation of present calculator technology is that a detailed analysis of a system will require a considerable length of time. If the designer is going to use a relatively simple manual design method many times, it is probably economical to use a programmable calculator.

Small Computer Methods

Small computers include personal microcomputers and small minicomputers that are finding increasing use in small firms for business purposes. These computers are more expensive than programmable calculators but are approximately 100 times faster. In addition, these computers are usually supplied with a high level language such as BASIC or FORTRAN, enabling interactive programs. In this case the user does not need to know the program details, as the program will prompt the user for information about the system to be modelled.

Modern small computers contain significant amounts of memory, allowing relatively large systems to be modelled using detailed weather data. Although few programs are currently available for small computers, many will be introduced in the near future. Small computers are most useful in the later design stages of a large project, when detailed simulation with a high degree of precision is needed.

Large Computer Methods

Large computers are usually owned by large companies or research institutes and are capable of fast and detailed analysis. It is usually possible to rent time on a large computer. The advantage of using a large computer is that many sophisticated and well documented programs are available through government sources such as SERI or NTIS (National Technical Information Service) for use by the general public. The programs are very flexible, have usually been compared with experimental data to check accuracy, and are ideal for final design calculations of large systems. Many of these programs have been used to calculate the averages and correlations used in manual design procedures.

13.2 Design Methods for Solar Heating and Cooling

Tables 13-1, 13-2, and 13-3 summarize most the design methods currently available for solar heating and cooling applications. Most of these methods allow for incorporation of thermal energy storage in the design. Information in all three tables was taken from SERI/SP-35-232.

Table 13-1 summarizes manual and programmable calculator design methods. Some methods of the latter type are available as prerecorded programs on magnetic cards from the source indicated.

Table 13-2 summarizes computer design methods. Programs listed as suitable for use by builders are interactive programs that prompt the user for information. Programs listed as only suitable for research engineers require a high degree of familiarity with both the operation and assumptios of the program, as well as the details of the system being analyzed.

It must be emphasized that using a program, especially the interactive type, without knowledge of the operation and underlying assumptions thereof can lead to erroneous results. It is important, therefore, for the user to study the program user's manual carefully before performing the analysis. For further details of the programs in Table 13-2, consult SERI/SP-35-232.

Table 13-3 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.

Table 13-1. Manual- and Programmable Calculator Heating and Cooling Design Methods Summary

Description	Author	Availability			Application				Coll. 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 Simple Hand Method for Calculating Solar Collector Area 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	•	•			•	•	•		•	•					•		•	
A Simple Empirical Method for Estimating the Performance of a Passive Solar Heated Building of the Thermal Storage Wall Type	J D Balcomb and R D Farland		1978	Rept No LA-UR-78-1159 Available from NTIS 5285 Port Royal Road Springfield VA 22161	•							•	•	•					•	•		
Optimum 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 S T 35[illegible]	R W Graeff	239 both 142 each 15 weather data	1977	Solarcon Inc 607 Church Street Ann Arbor MI 48104	•	•		•	•	•	•					•		•		•		
Solarcon Program S T 33	R W Graeff	138	1979	Solaron 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	•							•				•	•					•
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 1536 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	R 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 *		+	#	•	•		•	•	•	•		•		•			•		•	•	
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 Swanson and R F Boem		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 Leffar	95	1978	Solar Environmental Engineering Co Inc	•	•	•	•	•	•	•				•			•			•	
SEEC III SEEC II Plus Insulation Optimization	C B Winn D Barley G Johnson J Leffar	125	1978	Solar Environmental Engineering Co Inc	•	•		•	•	•	•				•			•			•	
SEEC VI Passive Solar Heating	C B Winn D Barley G Johnson J Leffar	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	•	•		•	•	•	•	•			•	•	•	•	•	•		•

*E.R.Durlak +1980 #Rept.No. 877
CEL Port Hueneme,CA

Table 13-1. (Continued)

Table 13-2. Computer Methods Summary

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	•	•	•			•	△		•	•		•	CDC UNIVAC	•	HUD
SHASP	1978	Avail. on request				•	•	•	•		•		•			•		UNIVAC	•	DOE
SIMSYAC	1973	300					•	•			•		•			•		CDC	•	NSF
SOLCOST	1978	300	•			•	•	•			•			•	•		•	CDC, IBM UNIVAC	•	DOE
SOLOPT	1978	20				•	•	•			•		•				•	AMDAHL	•	Texas A&M Univ.
SOLTES	1978				Available Argonne Fall 1979	•				•	•		•			•		CDC		Sandia
SUNCAT	1979	Nom.			Limited documentation	•		•				•	•	•		•		Data General Eclipse	•	NCAT
SUNSYM*	1979		•	•		•	•	•	•		•			•		•		IBM	•	Sunworks Comp. Systems
SYRSOL	1978	Nom.			Avail. but not actively marketed		•	•	•			•	•	•			•	IBM		ERDA, NSF, DOE
TRACE*	1979			•		•	•	•	•	•	•	•	•	•				IBM		The Trane Co.
TRNSYS	1978	200	•		Training available	•	•	•	•		•	△	•			•		CDC, IBM UNIVAC		DOE
TWO ZONE	1977			•		•		•	•			•	•			•		CDC	•	LBL
UWENSOL	1978	200				•		•				•	•			•		CDC		State of Wash.
WATSUN	1978	175				•	•	•			•		•			•		IBM	•	Nat'l Research Center of Can.

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

△ *Being added*

Program Name	Sponsor	Author	Contact	Original Release	Current Version	Application				Availability		
						Building Load	HVAC System	Active Solar	Passive Solar	Source Cost ($)	Time Share	Special Arrangements
ECUBE III	American Gas Association (AGA)	Subcontractors	American Gas Association 1515 Wilson Blvd. Arlington, VA 22209 (203) 841-8400 David S. Wood		1979	•	•		•	No info	•	•
ENERGY 1	Gibson-Yackey-Trindade Assoc.	Robert Gibson	Gibson-Yackey-Trindade Associates 311 Fulton Ave. Sacramento, CA 95611 (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-Donaeu 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	•
Westing-house 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

Table 13-3. Heating, Ventilating, and Air Conditioning

13.3 References

- Anonymous
  Analysis Methods for Solar Heating and Cooling Applications.
  SERI/SP-35-232. September, 1979.

- Anonymous
  Analysis Methods for Photovoltaic Applications.
  SERI/SP-35-230.

- Anonymous
  Analysis Methods for Wind Energy Applications.
  SERI/SP-35-231.

- C. J. Michal.
  Desk calculators simplify solar.
  Solar Age, pp. 36-48, April 1979.

- D. Nordham and W. L. Glennie.
  Choosing the right design tools.
  Solar Age, pp. 33-37, November 1980.

- W. A. Wright.
  SOLGRAF/DHW: An easy way to size solar domestic hot water systems.
  Solar Age, pp. 23-30, March 1980.

14.0 DESIGN EXAMPLES

This chapter contains worksheets for four design storage systems:

- water tank heat storage (Section 14.1)
- rockbed heat storage (Section 14.2)
- latent heat storage (Section 14.3)
- cold side storage for cooling (Section 14.4)

The worksheets can be used as guidelines for preliminary design and bidding preparations; they are not meant to take the place of detailed engineering calculations. More sophisticated analysis methods supplant these worksheets and refine the first-approximation calculations provided here.

For some applications, the user may be interested only in certain subsections of each design example. For instance, in Section 14.1 on water tank storage, the user may be interested only in the size of the tank, and want to leave pump and heat exchanger details for an installer or contractor.

In this chapter, symbols for equations are always defined in the worksheets, but may not always be defined in the text. All of the symbols for the chapter are defined in Section 14.5.

Example problems, showing the use of each worksheet for the four design examples, may be found in corresponding sections in Chapter 15.

14.1 Water Heating System

Before constructing a particular liquid sensible heat storage system you must consider a number of variables:

- purpose of storage system such as direct solar space heating, solar domestic hot water, solar-assisted heat pump, or load management.
- size and shape of tank
- temperature range and pressure limitations
- stratification
- type, location and insulation of tank
- heat exchanger
- pump characteristics

This example describes the sizing and design of a water tank storage system for either domestic hot water or space heating.

- Sizing is described in Sections 14.1.1 to 14.1.3.
- A solar collector heat source option is described in Section 14.1.2.
- Water tank details, including insulation, are described in Sections 14.1.4 to 14.1.6.
- Pump characteristics are described in Section 14.1.7.
- Heat exchanger details are described in Section 14.1.8.

The objectives for constructing a liquid-based storage system will determine other characteristics about it. For example, domestic hot water systems usually require temperatures of less than 120°F and a pressurized tank. Potable water must also be carefully guarded against contamination. Toxic substances such as corrosion inhibitors and heat transfer fluids must be in distinctly separate circulation loops.

Using the procedure spelled out by f-Chart* or other appropriate methods, determine the size and type of solar collector which, combined with an auxiliary furnace, will supply the entire heating load at the least possible cost. The amount of collector required for a particular application is dependent on space and water heating loads. Space heating loads can be calculated by the ASHRAE method.** Methods of calculating collector performance and heating loads are beyond the scope of this manual.

If other design methods are not available, make reasonable estimates. The space heating load can usually be estimated by examining old heating bills for the building (or for a reasonably similar building). A very rough estimate of solar collector area can be made by using the procedure in Section 14.1.2.

Water heating loads are easier to determine. The storage system should be large enough to supply approximately 100 percent of the daily hot water load from storage. The HUD Intermediate Minimum Property Standards recommend the following guidelines for determining the consumption:

- For one- and two-family residences and apartments up to twenty units, each unit requires 20 gallons (76 liters) each for the first two persons plus 15 gallons (57 liters) for each additional person.
- For apartments of 20 to 200 units, each unit requires 40 gallons (150 liters).
- For apartments of more than 200 units, each unit requires 35 gallons (130 liters).
- Consumption factor, W, will thus be about 17, 40, or 35.

In most cases there is no conflict between these storage sizes and the rule of thumb of 1.25 to 2 gallons per day per square foot (50 to 75 liters per day per square meter) of collector.

If there is a small conflict between the rule of thumb and the HUD recommendations, choose the larger of the two. In the rare instance that the conflict between the two methods is substantial, only computer methods such as TRNSYS or DOE-1 can resolve the conflict.

*William A. Beckman, Sanford A. Klein, and John A. Duffie. Solar Heating Design by the f-Chart Method. John Wiley & Sons, 1977.

**American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 345 East 47th Street, New York, NY. ASHRAE Handbook of Fundamentals, 1972.

14.1.1 Daily Water Heating Load

Read Section 5.1.

Complete Worksheet A taking into account the information below.

If storage is for:

- Domestic water heating, complete items 1-6, 9.
- Space heating, complete item 7-9.
- Domestic water and space heating, complete items 1-9.

The daily water heating load (L_w) can be estimated from

$$L_w = N \cdot W \cdot (T_{max} - T_{min}) \cdot \rho \cdot C_p = V_d \cdot \Delta T \cdot \rho \cdot C_p \qquad (14\text{-}1)$$

This corresponds to items 1-6 of worksheet A.

The total daily load is L and can be found from

$$L = L_s + L_w \qquad (14\text{-}2)$$

This corresponds to item 9 of Worksheet A.

14.1.2 Tank Sizing

Use f-Chart or some other design tool (Chapter 13) to calculate the optimal storage volume. Lacking this, estimate the number of days of storage needed. For most applications, 1.0 days of storage is approximately the economic optimum.

The size calculation will vary depending on whether the application is only domestic hot water, only space heating, or a combined system. For combined systems, it is recommended that the calculation also be performed for each application separately. These results should be compared with those done for the combined system. See also the discussion for solar collector sizing at the end of this Section.

Complete Worksheet B-1 or B-2 to size the water storage tank.

Sizing Domestic Hot Water Tank from Number of Days of Storage Required

The tank size can be found from:

$$V_{Tw} = N \cdot V_d \tag{14-3}$$

where N is the number of days of storage desired, V_d is the total daily water consumption, and V_{Tw} is the tank volume.

This corresponds to worksheet B-1.

Sizing Space Heating Water Tank

If a solar collector is used for the energy source, and if the collector area has been previously determined, proceed to the paragraph below on Tank Sizing from Solar Collector Area.

Solve, as shown, the following equation:

$$V_{Ts} = \frac{NL_s}{(T_{max} - T_{min})\ \rho\ C_p} = \frac{Q_s}{\Delta T \rho C_p} \tag{14-4}$$

This corresponds to Worksheet B-2.

Combined Tank: Domestic Hot Water and Space Heating

Use the following equation and worksheet B-2 to determine the optimal tank size.

$$V_{Tc} = \frac{NL}{(T_{max} - T_{min})\ \rho\ C_p} = \frac{Q_c}{\Delta T_i \rho C_p} \tag{14-5}$$

Tank Sizing from Solar Collector Area

If a solar collector is used for the energy source, and if the collector area but not the storage requirement was determined using another design method, use Worksheet C, items 1-3 to size space heating or combined system tank size.

We use the rule of thumb: 1.25 to 2.0 gallons of storage per square foot of collector. The storage volume V is given by:

$$V = \gamma A_c \qquad (14-6)$$

where γ is the storage to collector ratio and A_c is the collector area.

Solar Collector Area

Skip this section if a solar collector is not the heat source.

The solar collector area can be calculated using f-Chart or one of the other design methods mentioned in Chapter 13. In this section we use the rule of thumb: 1.25 to 2.0 gallons of storage per square foot of collector.

The collector area A_c is given by:

$$A_c = \frac{V_d}{\gamma} \qquad (14-7)$$

where γ is the storage to collector ratio, and V_d is the total daily water consumption.

It is always permissable to use a smaller collector area than that calculated in this section. The amount of auxiliary fuel, however, will be higher.

14.1.3 Tank Shape

The decision to use vertical cylindrical, horizontal cylindrical, or rectangular storage tanks is arbitrary, but a tall tank is easier to stratify than a shorter one - reread section 5.2.2. If there is enough space available, a tank with a height approximately three times its diameter is a reasonable compromise between thermal losses and good stratification. Tables 4-6, 4-7, 4-8 depict a wide range of possible shapes. Allow for easy access to the storage unit - if access is impossible (buried in ground, for example) purchase a unit with a projected 30-year lifespan.

Cylindrical Tanks

See Worksheet D, items 1-5.

Rectangular Tanks

See Worksheet D, items 6-12.

As indicated in Worksheet D, if $w > w_{max}$, choose a tank with dimensions $L \cdot w_{max} \cdot D$. Otherwise, choose a tank with dimensions $w \cdot w \cdot D$. Although a single tank is usually less expensive and thermally more efficient, multiple storage units can be used to accommodate space restrictions.

14.1.4 Temperature and Pressure Criteria

Storage temperatures depend upon the application, ranging from below 140°F (60°C) for domestic hot water systems to 1400°F (760°C) in the utility industry. Reread section 5.1 to determine storage temperature and liquid.

1. Storage temperature _____ °F
2. Storage liquid _____
3. Maximum tank pressure _____ psi

14.1.5 Stratification in Hot Water Storage Tanks

The procedures given below should not be used for sensible cold-storage water tanks.

Stratification is a simple inexpensive technique to improve storage efficiency and should be employed whenever possible. Reread section 5.2.2 and examine carefully Figures 5-2, a-d. Work through Worksheet E to determine the maximum fluid inlet velocity so that $R_i >> 0.25$, ensuring good stratification.

The following equations and Table 14-1 are the basis of Worksheet E. Symbols are defined there.

$$\text{since } Ri = \frac{g\beta L\Delta T}{u^2} \qquad u = \frac{g\beta L(T_{max}-T_{min})}{R_i} \qquad (14\text{-}8)$$

Table 14-1. Volumetric Thermal Expansion Coefficient for Water

Temperature, °F	(°C)	β, °F^{-1} x 10^{-4}	(°C^{-1} x 10^{-4})
50	10	0.38	0.7
68	20	1.1	2
86	30	1.8	3.3
104	40	2.3	4.2
122	50	2.6	4.6
140	60	3.0	5.4
158	70	3.5	6.3
176	80	3.9	7.0
194	90	4.1	7.3
212	100	4.4	8

14.1.6 Type, Location, and Insulation of Tank

Type and Location

Reread sections 4.4 and 5.3.2 on location, 5.2.1 on type, and complete Worksheet F on tank location and type.

If the tank is to be in the basement or other potential-problem location, determine loading. Worksheet F uses the following equation to determine load:

$$\text{load} = \frac{(V_{TG} \cdot \rho) + m_T}{A_2} \qquad (14\text{-}9)$$

If the load is over 150 lb./ft^2, have a licensed structural engineer determine the load capacity and design whatever reinforcements may be necessary.

Next, on a separate sheet of paper, sketch the proposed system, inserting all heat exchangers, pumps, fluid flow direction, etc.

Insulation Requirements

Read sections 4.5 and 5.2.5.

The determination of insulation requirements can be best approached by using examples 4-4 and 4-5 as a guide. Refer to Tables 4-4, -6, -7, -8 to calculate the minimum R-value and thickness of necessary insulation to meet the SMACNA standard of a 2% heat loss from storage in 12 hrs.

Insulation requirements will be calculated on Worksheet G.

Calculating the minimum necessary R-value and the required thickness of insulation is a five-step process. The insulation on a storage unit consists of two parts, 1) a layer of thickness s that will be adjusted to obtain the required amount of insulation and 2) all other insulation layers. The five steps are:

(1) Determine the required R-value from Tables 4-4, -6, -7, and -8 and knowledge of the average operating temperature and ambient temperature.

$$R = \frac{T_{avg} - T_a}{fQ/At} \qquad (14\text{-}10)$$

(2) Calculate the required resistance R_{req} allowing for fixed parallel heat losses. For example if we want to calculate the insulation thickness of the top and sides of a tank but the thickness of the bottom insulation is to be constant, then the bottom is a fixed parallel heat loss of area A_{par} and resistance R_{par}. Other fixed parallel heat losses are tank support structures and inlet and outlet pipe connections.

$$R_{req} = \frac{A_{total} - A_{par1} - A_{par2} - \ldots.}{\frac{A_{total}}{R} - \frac{A_{par1}}{R_{par1}} - \frac{A_{par2}}{R_{par2}} - \ldots.} \qquad (14\text{-}11)$$

If the value of R_{req} is negative, then the fixed parallel heat loss is excessive, and the tank will lose too much heat. If this is the case, the general design of the tank should be reconsidered.

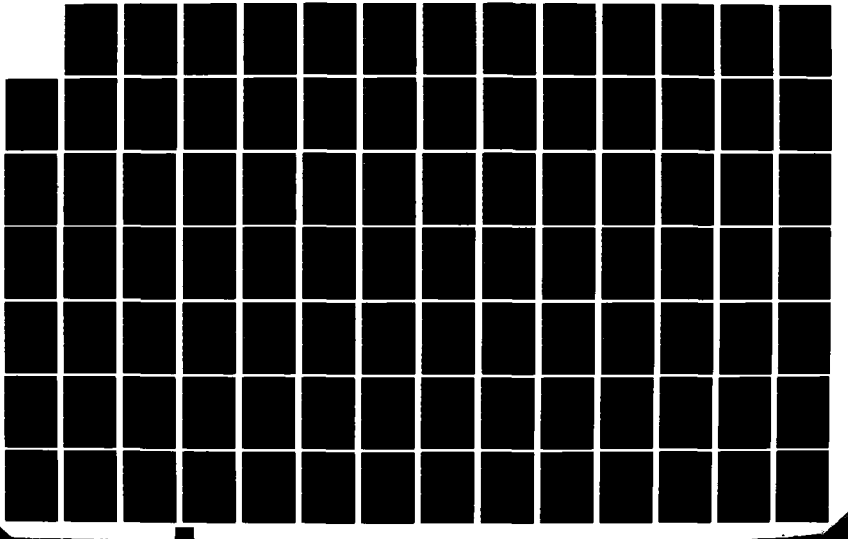
(3) Calculate the total fixed resistance in series with the layer whose thickness is to be determined. In most cases the fixed series resistance will be small and and consists of summing individual components such as surface-to-air resistance, resistance of the cover over the insulation, resistance of tank walls (if not negligible) and so forth.

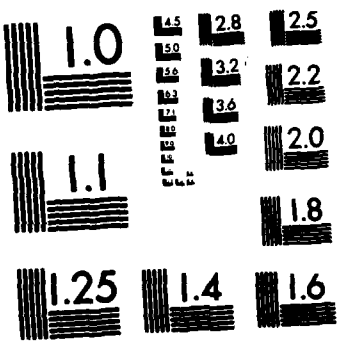
$$R_{ser} = R_1 + R_2 + R_3 + \ldots. \qquad (14\text{-}12)$$

(4) Calculate the resistance per inch of thickness r* of the layer whose thickness is to be varied. If the layer consists of only one component with no parallel heat loss paths, r* can be found in Table 4-4. If parallel heat loss paths are involved (as, for example, with studwall construction of a rock bin or water tank), the following formula must be used to account for the parallel paths.

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MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

$$r^* = \frac{A_1 + A_2 + A_3 + \ldots.}{\frac{A_1}{r_1} + \frac{A_2}{r_2} + \frac{A_3}{r_3} + \ldots.} \qquad (14\text{-}13)$$

(5) Calculate the required thickness s of the insulation layer using the following formula.

$$s = \frac{R_{req} - R_{ser}}{r^*} \qquad (14\text{-}14)$$

Description of Worksheet G

Step (1) of the calculation consists of Worksheet G-1, items 1-7. This step gives a good estimate of R which will be used in equation 14-11.

Step (2) of the calculation consists of Table i and item 8, both in Worksheet G-1. From Table 4-4 and from information about the proposed storage tank, complete Worksheet G-1, Table i on parallel heat paths so that $R_{required}$ (R_{req}) can be calculated in item 8. NOTE: The parallel heat paths include only those heat loss paths that will not be covered with added insulation--the bottom of a concrete tank, vents, heat exchangers, etc. The R value for each path (R_{par}) is found by combining individual resistances in series, eg.

$$R_{par,1} = r_{1a}s_{1a} + r_{1b}s_{1b} + r_{1c}s_{1c} + \ldots.R_{1,sur} \qquad (14\text{-}15)$$

where

(1) the resistance per inch of thickness (dependent on the type of building or type of insulating material used) r_{1a}, r_{1b}, r_{1c}, can be found from Table 4-4.

(2) s_{1a}, s_{1b}, s_{1c}, etc., the thicknesses of the individual resistance layers, are determined from the tank design.

(3) R_{sur}, surface resistance, can be found from Table 4-5.

(4) A_{par} is the area of the parallel heat path--determined from the tank design.

(1), (2), (3) and (4) correspond to the annotation in Worksheet G-1 Table i.

Worksheet G-1, item 8, solves equation (14-11), which can be rewritten as

$$R_{req} = \frac{A_{tot} - \Sigma A_{par}}{\frac{A_{tot}}{R} - \Sigma \frac{A_{par}}{R_{par}}}$$

Step (3) of the calculation consists of Table ii and item 9, both of Worksheet G-3. From Tables 4-4 and 4-5 and from information about the proposed storage tank, complete Table ii in Worksheet G-3, defining contributions to the fixed series resistance.

The fixed series resistance (R_{ser}) is found by summing resistances of building materials that will be covered with insulation (tank walls, cover, etc.) with surface resistance.

$$R_{ser} = \Sigma R_n + \Sigma R_{sur}$$

where

1) ΣR_n is the sum of r x s of the different building materials.

2) ΣR_{sur} is the sum of the individual surface resistances.

3) R_{sur}, r, s are as defined earlier in this section.

Step (4) of the calculation consists of item 10 and Table iii, both of Worksheet G-3. For now neglect the areas over which parallel heat loss occurs; examine the other areas of the projected tank design, the ones that will be covered with insulation.

Under the insulation will be building material discontinuities--a stud will have a different r-value than an adjacent fiberglass batt, for example. All these differences in area and insulating properties are taken into account in equation 14-13 which defines an overall resistance per inch of thickness for the part of the wall containing insulation. If the tank is homogeneous, except for locations of parallel heat loss, then r^* = r-value of the tank material

$$r^* = \frac{A_1 + A_2 + A_3 + \ldots}{\frac{A_1}{r_1} + \frac{A_2}{r_2} + \frac{A_3}{r_3} + \ldots} = \frac{\Sigma A_n}{\Sigma \frac{A_n}{r_n}} \quad \text{and } \Sigma A_n = A_{tot} - A_{par}$$

As shown earlier in (14-13), ΣA_n should equal $A_{tot} - A_{par}$. Table iii in Worksheet G-3 determines r^*.

Step (5) of the calculation consists of item 11, Worksheet G-3. From equation (14-14) the needed thickness of the insulation selected is

$$s = \frac{R_{req} - R_{ser}}{r^*}$$

Further Considerations

If the storage is used for average temperatures much higher than 140°F, a more detailed calculation might be warranted. Appendix G on economic insulation thickness will guide you through each variable to find the optimal insulation thickness for such a case.

Typical domestic hot water tank insulation has a thermal conductance* (U) of 0.10 Btu/hr.ft^2.°F (0.57 W/m^2.°C) or more. By comparison, following the method described in Chapter 4 of this manual would require a thermal conductance of 0.030 to 0.033 Btu/hr.ft^2.°F (0.17 to 0.19 W/m^2.°C) to limit heat loss to 2 percent in 12 hours (assuming a tank temperature of 140°F and an ambient temperature of 70°F). Even if the insulation requirements were lowered to limit losses to 10 percent in 24 hours as specified by the HUD Intermediate Minimum Property Standards, a thermal conductance of 0.071 to 0.083 Btu/hr.ft^2°F (0.40 to 0.47 W/m^2.°C) would be required. Clearly, then, most domestic hot water tanks can benefit from added insulation, and insulation kits are available. The added insulation is very effective on electric- and solar-heated storage tanks, but it cannot reduce the flue losses in gas- or oil-fired tanks. If you install additional insulation, be careful not to obstruct air vents on gas- or oil-fired tanks, controls, pressure and temperature safety valves or water tempering valves. Install an automatic flue damper, also.

---

*When the thermal transmittance (conductance) of the insulation (U) is given, this is the reciprocal of the resistance. Mathematically

$$U = \frac{1}{R}$$

and using dimensional analysis

$$\frac{\text{Btu}}{\text{hr.ft}^2.^\circ\text{F}} = \frac{1}{^\circ\text{F.ft}^2.\text{hr/Btu}}$$

Similarly, the thermal transmittance (conductance) per unit of thickness (u) is the reciprocal of the resistance per unit of thickness.

$$u = \frac{1}{r} \quad \text{and} \quad \frac{\text{Btu.in}}{\text{hr.ft}^2.^\circ\text{F}} = \frac{1}{^\circ\text{F.ft}^2.\text{hr/Btu.in}}$$

14.1.7 Pump Characteristics

Read section 5.2.3.

Examine Figure 5-7 for pump efficiency and required motor horsepower.

Flowrate

First determine the flowrate, q, through the heat exchanger loop using the following equation and Worksheet H-1, items 1-7.

$$q = \frac{Q'}{\rho C_p \Delta T} \qquad (14\text{-}16)$$

Pumping Head

Also on Worksheet H-1, items 8-12, determine the required pumping head using equation (5-3)

$$H_p = Z + H_f + H_x + H_c$$

Pump Parameters

Pump parameter specifications are determined in Worksheet H-2, items 1-5, using Figure 5-7.

- Obtain performance curves from one or more pump manufacturers and plot q against H_p on performance curves such as the upper curves on Figure 5-7 to find a pump that exceeds the pumping head requirement at the required flowrate. The head requirement is from Worksheet H-1, item 12. The pump rate is from Worksheet H-1, item 7.

- Plot Z (from Worksheet H-1, item 8) on upper curves of Figure 5-7; make sure that $Z < H_p$ when $q = 0$.

- Plot impeller diameter d and flow rate q on the lower curve of Figure 5-7 to determine size of motor required by the pump.

Net Positive Suction Head (NPSH)

NPSH criteria are calculated in Worksheet H-2, items 6-14, again using Figure 5-7.

- Plot q on the NPSH curve of Figure 5-7 to determine the pump's NPSH requirement ($NPSH_R$).

- Using Worksheet H-2 determine the available NPSH from equation (5-4)
$$NPSH = Z - H_f - H_v + H_a + H_p$$

If item 14 ($NPSH-NPSH_R$) is a positive number, then a net positive suction head exists and cavitation won't occur.

14.1.8 Heat Exchangers

Read section 4.6 and Appendix E on heat transfer fluids, and fill out Worksheet I, which gives the manufacturer enough information to be able to select the properly sized heat exchanger and minimize the overall cost of the system. Design and Installation Manual for Thermal Energy Storage by Roger Cole, et. al., works out the relatively complex math in detail.

14.1.8.1 Capacity

The following specification is applicable to shell-and-tube and double-walled heat exchangers. Fill in the flow rates and temperatures with numbers that you calculated in designing your system.

You must also specify the fluids to be used in the heat exchanger. In this example, a water solution has been specified, but you may require a different fluid. You must also specify fouling factors appropriate to the fluids used. For the water found in most solar energy systems a fouling factor of 0.001 hr.ft^2.°F/Btu is adequate. For hard water (over 15 grains per gallon) use a fouling factor of 0.003 hour-square foot-°F per Btu.

Sample Written Specification:

The heat exchanger shall have the capacity to heat _____ gpm of water (in the tubes) from _____°F when supplied with _____gpm of propylene glycol solution (50 percent) at _____°F (at the inlet end of the shell). Fouling factors shall be 0.001 hour-square foot-°F per Btu for both tube side and shell side.

14.1.8.2 Heat Exchanger Area

The following type of specification should be used if you specified a U-tube bundle heat exchanger. Since the U-tube bundle uses natural convection, you cannot specify the flow rate on the exterior of the tubes. Instead you must calculate and specify the required heat exchange area.

Sample Written Specification:

The heat exchanger shall have a minimum heat exchange area of _____ square feet. The heat exchanger will be supplied with _____gpm of propylene glycol solution (50 percent) at _____°F and will be used to heat water.

14.1.8.3 Pressure Drops and Velocities

For most systems maximum pressure drops should be specified to ensure that the pumps will be able to maintain the required flow rates and that the cost of energy for the pumps does not become excessive.

Sample Written Specifications:

> The maximum pressure drop on the shell side shall be _____psi with the fluid, flow rate, and temperature specified.
>
> The maximum pressure drop on the tube side shall be _____psi with the fluid, flow rate, and temperature specified.

14.1.8.4 Dimensional Requirements:

It is useful to specify the dimensions of the heat exchanger so that it will fit into the available space.

Sample Written Specifications:

1. Maximum heat exchanger length _____
2. Heat exchanger diameter _____
3. Location of inlet and outlet _____

Worksheet A: Daily Water Heating Load

1.	(a) Number of people in household, or Number of living units in apartment	people or units
	(b) Consumption factor, W, as shown in the introduction to 14.1	
	(c) Multiply lines 1a by 1b to find total daily water consumption V_d	gal
2.	(a) Maximum tank temperature- Minimum acceptable hot water temperature, T_{max}	°F
	(b) Minimum tank temperature - If the tank holds the delivered potable water, this is the measured temperature of main supply water, T_{min}	°F
3.	Subtract 2b from 2a to find ΔT	°F
4.	Water density, ρ	8.33 lb/gal
5.	Heat capacity of water, C_p	1.0 Btu/lb.°F
6.	Multiply lines 1b, 3, 4, and 5 to find L_w the daily domestic water heating load	Btu/day
7.	Monthly space heating load	Btu/month
8.	(a) Daily space heating load, L_{s1}, if known	Btu/day
	(b) Divide line 7 by 30 days, L_{s2}	Btu/day
	(c) Use the greatest value from L_{s1} and L_{s2}	Btu/day
9.	Add lines 6 and 8c to obtain L, the total daily heating load	Btu/day

Worksheet B-1: Sizing Domestic Hot Water Tank

1.	Number of days of storage desired	days
2.	Total daily water consumption V_d (from line 1c of Worksheet A)	$\frac{\text{gal}}{\text{day}}$
3.	(a) Multiply lines 1 and 2 to find total tank volume, V_{TGw}	gal
	(b) Multiply 3a by 0.13368 to obtain tank size V_{TFw} in cubic feet	ft^3

*The subscripts associated with volume have this origin:
- T - total
- G - expressed in gallons or F - expressed in ft^3
- d - daily
- w - domestic water
- s - space heating
- c- combined

Thus V_{TFw} refers to total volume in ft^3 of the domestic hot water tank.

Worksheet B-2: Water Tank Sizing - Space Heating or Combined DHW and Space Heating

1.	Number of days of storage, N	days
2.	Daily space heating load, L (from line 8c of Worksheet A) (line 9 if combined heating)	Btu/day
3.	Multiply lines 1 and 2 to find energy stored, Q_s (Q_c if combined heating)	Btu
4.	(a) Minimum usable water tank temperature, T_{min}	°F
	(b) Maximum water tank temperature, T_{max}	°F
	(c) Subtract line 4a from line 4b to determine ΔT	°F
5.	Water density, ρ	8.33 lb/gal
6.	Heat capacity of water, C_p	1.0 Btu/lb°F
7.	Multiply lines 4c, 5 and 6	Btu/gal
8.	(a) Divide line 3 by line 7 for V_{TGs}	gal
	(b) Multiply 8a by 0.13368 to obtain tank size V_{TFs} in cubic ft	ft^3

Worksheet C: Tank Sizing - Collector Sizing

Tank Sizing from Solar Collector Area

1.	Choose γ (between 1.25 and 2.0)	$\frac{gal}{ft^2}$
2.	Collector area A_c	ft^2
3.	Multiply line 1 by line 2 to find V	gal

Solar Collector Area from Tank Size

4.	Choose γ (between 1.25 and 2.0)	$\frac{gal}{day.ft^2}$
5.	Number of days of storage, N (from Worksheet B-1, B-2)	days
6.	Total tank volume V_{TG} (from Worksheet B-1, B-2)	gal
7.	Divide line 6 by line 5 to find V_d	$\frac{gal}{day}$
8.	Divide line 7 by line 4 to find collector area, A_c	ft^2

Worksheet D: Tank Shape

Cylindrical Tanks

1. Desired height-to-diameter or length-to-diameter ratio L/d

2. Approximate tank diameter $D_{app} = \left(\frac{4V}{L/d}\right)^{1/3}$ ft

   (For V, use item 3b from Worksheet B-1, 8b from B-2, or item 3 from Worksheet C. Use L/d from line 1 above)

3. Choose a standard tank diameter d close to d_{app} ft

4. Tank height or length $\left(L = \frac{4V}{d^2}\right)$ ft

5. Actual dimensions designed _____ft x _____ft x _____ft

Rectangular Tanks

6. Desired tank depth d ft

7. Width of a square tank $(w = [V/d]^{1/2}$ where V is as in item 2 above ft

8. Maximum width permitted, w_{max} ft

9. If $w > w_{max}$, calculate tank length $(\ell = V/dw_{max})$ ft

10. Dimensions of tank _____ft x _____ft x _____ft

11. Relative dimensions of tank (see Table 4-6, -7, or -8) _____x_____x_____

12. Total area of tank, A_{tot} (sum the area from each face) _____ ft^2

Worksheet E: Design for Stratification

1.	Gravitational constant g	32.2 ft/sec^2
2.	Choose the worst-possible-case (lowest volumetric thermal expansion coefficient β (if fluid other than water, obtain β from manufacturer) - for the temperature range of the tank	$^\circ F^{-1}$
3.	Transcribe value L for tank height from Worksheet D: item 3 for horizontal cylindrical tanks, item 4 for vertical cylindrical tanks, or item 9 for rectangular tanks	ft
4.	Maximum water temperature in tank from Worksheet A, item 2a, or item 4b, from Worksheet B-2	$^\circ F$
5.	Minimum water temperature in tank from Worksheet A, item 2b, or item 4a, from Worksheet B-2	$^\circ F$
6.	Subtract item 5 from item 4 for ΔT, temperature difference in tank	$^\circ F$
7.	Value of desired Richardson number $R_i >> 0.25$	
8.	Multiply together items 1,2,3,6	
9.	Divide item 8 by item 7	
10.	Take square root of item 9 to obtain maximum fluid inlet velocity, u	$\frac{ft}{sec}$

11. Design methods to be used to promote thermal stratification:

    a.
    b.
    c.

12. Define Control strategy to be used (see Section 5.2.2 Modifications for Solar Systems)

    a.
    b.
    c.

Worksheet F: Type of Tank and Location

Type of Tank

Type of tank (if pressurized, tank must be steel) _____

Length of warranty if steel: _____ yrs.

Choice of corrosion inhibitor _____

If FRP:

manufacturer's temperature limit _____ °F

If concrete:

cast-in-place or precast _____

If wooden or concrete or multicomponent (indoor use only)

choice of liner (or coating) _____

liner thickness _____

Tank Floor Loading

1.	Transcribe tank capacity V_{TG} from Worksheet B-1, B-2, or C	gal
2.	Density of water ρ	8.33 lb/gal
3.	Obtain mass of tank (m_T) from manufacturer	lb
4.	Multiply items 1 and 2 to obtain weight of water	lb
5.	Sum items 3 and 4 to obtain total weight	lb
6.	Planned surface area, A_2, between storage tank and substructure	ft^2
7.	Divide item 5 by item 6 to obtain load	lb/ft^2

Worksheet G-1: Tank Insulation

1.	Average temperature in storage unit (T_{avg})	°F
2.	Ambient temperature surrounding the unit (T_a)	°F
3.	Subtract line 2 from line 1	°F
4.	Transcribe tank volume V_{TG} (from Worksheet B-1, B-2, or C	gal
5.	Relative dimensions of tank from Worksheet D	
6.	Using lines 4 and 5 and Table 4-6, -7, or -8 determine insulation factor fQ/At	$\frac{Btu}{ft^2.hr}$
7.	Divide line 3 by line 6 to determine R	$\frac{°F.ft^2.hr}{Btu}$

Fill in Worksheet G-2, Table i at this time. Some of the results will be used in item 8 below.

8.	a) A_{tot}, item 12 from Worksheet D	ft^2
	b) Divide line 8a by line 7 to determine $\frac{A_{tot}}{R}$	$\frac{Btu}{°F.hr}$
	c) ΣA_{par} from Table i	ft^2
	d) Subtract item 8c from 8a	ft^2
	e) $\Sigma \frac{A_{par}}{R_{par}}$ from Table i	$\frac{Btu}{°F.hr}$
	f) Subtract 9e from 9d	$\frac{Btu}{°F.hr}$
	g) Divide 9c by 9f to find R_{req}	$\frac{°F.ft^2.hr}{Btu}$

Worksheet G-2: Table 1. Parallel Heat Paths

Heat Loss Path	Parallel Heat Loss Material	(1) r-value of Heat Loss Material $\frac{°F \cdot ft^2 \cdot hr}{Btu \cdot in}$	(2) Thicknesses of Heat Loss Material, s in	Individual Resistance r x s $\frac{°F \cdot ft^2 \cdot hr}{Btu}$	(3) R_{sur} $\frac{°F \cdot ft^2 \cdot hr}{Btu}$	R_{par} $R_{sur} + \Sigma rs$	(4) A_{par} ft^2	$\frac{A_{par}}{R_{par}}$	Comment
1 (a) (b) (c)									
1				$\Sigma rs=$					
2									
2				$\Sigma rs=$					
3									
3				$\Sigma rs=$					
4									
4				$\Sigma rs=$					
							Σ	Σ	

Worksheet G-3: Tank Insulation

Table ii. Fixed Series Resistance					
Fixed Resistance	Source of Resistance	r-value of material -r-	thickness of material (in.) -s-	R_n r.s	R_{sur}
1					
2					
3					
				$\Sigma=$	$\Sigma=$

9. Add $\Sigma R_n + \Sigma R_{sur}$ from Table ii to find R_{ser} — $\frac{°F.ft^2.hr}{Btu}$

10. a) Decide what material will be used for insulation.

    b) Materials r-value from Table 4-4 — $\frac{°F.ft^2.hr}{Btu.in}$

Table iii. r^* Determination			
Material	Area material covers ft^2	r-value of material	$\frac{A}{r}$
	$\Sigma A_n =$		$\Sigma \frac{A}{r} =$

11. Divide ΣA_n by $\Sigma \frac{A}{r}$ from Table iii to find r^* — $\frac{°F.ft^2.hr}{Btu.in}$

12. a) Transcribe 8g -- R_{req} — $°F.ft^2.hr/Btu$

    b) Transcribe 9 -- R_{ser} — $°F.ft^2.hr/Btu$

    c) Subtract 12b from 12a — $°F.ft^2.hr/Btu$

    d) Divide 12c by 11 to get s — in

Worksheet H-1: Pump Characteristics

Flowrate

1.	Heating rate Q' (which is worked out in section 15.1.7)	Btu/hr
2.	Fluid density ρ	lb/gal
3.	Heat capacity of heat transfer fluid C_p	Btu/lb.°F
4.	Temperature change of heat transfer fluid ΔT	°F
5.	Multiply items 2-4	
6.	Divide item 1 by item 5 to obtain flow rate q	gal/hr
7.	Divide item 6 by 60 to obtain q in gpm.	gal/min

Pumping Head

8.	Height difference between collector top and tank liquid level Z (open-drop drain down system only)	_____ ft
9.	Pipe friction head loss H_f (summary of calculations per ASHRAE Handbook of Fundamentals, Chapter 26)	_____ ft
10.	Heat exchanger head loss H_x (from manufacturer)	_____ ft
11.	Collector head loss H_c (from manufacturer)	_____ ft
12.	Add items 8-11 to determine sum of head losses around the loop, H_p	_____ ft

Worksheet H-2: Pump Characteristics

Pump Parameters

1. Impeller diameter d _____ inches

2. Pump motor _____ HP

3. Pump manufacturer _____

4. Pump model _____

5. Size of motor required by pump _____ HP

Net Positive Suction Head (NPSH)

6. Determine $NPSH_R$ from Figure 5-7 _____ ft

7. Height difference Z (from item 8, Worksheet H-1) _____ ft

8. Pipe friction head loss H_f (from item 9, Worksheet H-1) _____ ft

9. Liquid vapor pressure head H_v (from Figure 5-8) _____ ft

10. Atmospheric pressure H_a (from Figure 5-9) _____ ft

11. Minimum tank pressure H_p (from item 12, Worksheet H-1) _____ ft

12. Add items 7, 10, and 11 _____ ft

13. Subtract items 8 and 9 from item 12 to determine available NPSH for system _____ ft

14. Subtract item 6 from item 13 _____ ft

Worksheet I: Heat Exchanger Specification Form

	FLUID A	FLUID B
Description		
Heat Capacity (Btu/lb·°F) (J/kg·°C)		
Density (lb/ft^3) (kg/m^3)		
Thermal Conductivity (Btu/hr·ft·°F) (W/m·°C)		
Viscosity (lb/ft·hr) (Pa·sec)		
Flow Rate (gpm) (liters/sec)		
Inlet Pressure (psi) (MPa)		

Rate of Heat Transfer (Btu/hr) (W) _____

Approach Temperature Difference (°F) (°C)
(Temperature of heating fluid inlet minus temperature of heated fluid outlet) _____

14.2 Rockbed Heating System

Chapter 6 and the references at the end of Chapter 6 provide an excellent background to this section. They should be examined before proceeding any further.

The design example has been constructed to direct the reader through the planning stages as they would actually occur.

- General rockbed design considerations are discussed in Section 14.2.1.
- The heat load and rockbed volume are calculated in Section 14.2.2 and Worksheet A. Coupling to a solar collector is also discussed in this section. The rockbed volume for this case is calculated in Worksheet B.
- In Section 14.2.3 and Worksheet C the size and shape are specified, including depth, cross-sectional area, and the resultant volume.
- Optimal rock size can be determined by following Section 14.2.4 and Worksheet D.
- The system pressure drop, needed to size the blower, is worked through step-by-step in Section 14.2.5 and Worksheet E.
- Type and location of the solid material storage system are discussed in Section 14.2.6 and uses Worksheet F.
- Insulation is discussed in Section 14.2.7 and uses Worksheet G of Section 14.1.
- Section 14.2.8 and Worksheet H selects the correct fan and motor for the system.
- The design example is based on rock bed storage coupled to an air-based solar collector in a space heating application, however, the early sections are more general.

14.2.1 General Rockbed Design Considerations

Most rock bed designs are subject to several constraints. Typical constraints include sizes of rocks that are available and floor-to-ceiling height available for the bin, as well as several rules of thumb, which are summarized below.

Rules of Thumb:

(1) Volume of the Rock Bed
The volume of the rock bed should be between 0.5 and 0.75 cubic feet of rock per square foot of solar collector (between 0.015 and 0.020 cubic meters per square meter of collector).

(2) Pressure Drop across the Rock Bed
The total pressure drop across the rock bed should be between 0.15 and 0.30 inches of water (between 37 and 75 Pascals).

(3) Plenum Size
The area of the plenum perpendicular to the air flow should be at least 8 percent of the cross-sectional area of the rock bed (see Figure 14-1). If a plenum contains rock bed supports that obstruct the air flow (such as bond beam blocks) then its area should be increased to 12 percent of the rock bed cross section.

(4) Rock Size and Rock Bed Depth
The rock bed's depth should be at least twenty times the nominal rock diameter if the rocks are less than 4 inches (10 centimeters) in diameter, or thirty times the nominal rock diameter if the rocks are more than 4 inches in diameter.

Other Considerations

Common sense and building codes dictate a number of considerations that a designer should keep in mind when designing a rock bed.

- Do a thorough job in designing the first few systems. Many of the details of these designs can be incorporated in later systems.
- Be sure the rock bin walls can hold not only the pressure of the rocks when the bin is first filled but also the additional pressure that will develop as the rocks expand, contract, and settle when the bed is in use. Have a licensed structural engineer design a rock bed foundation that will meet building code requirements and ensure the safety of the building.
- Choose materials that can withstand high temperatures for the life of the system. Fire codes generally outlaw placing flammable materials in contact with the hot air system, so sheet metal or fire-retardant gypsum board must be specified as the interior lining of wooden rock bins.

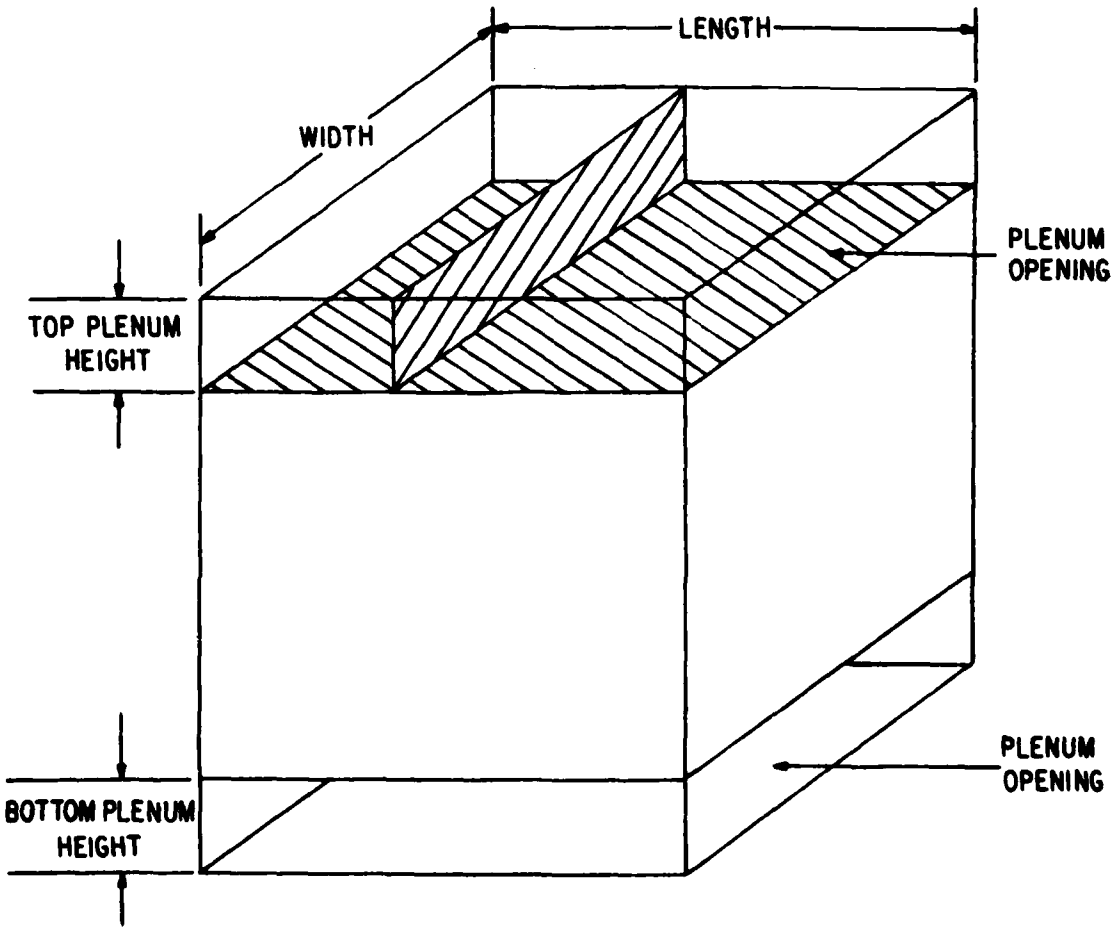


Figure 14-1. Definition of Length and Width for Design Example

- Consider such constraints as space restrictions and available rock sizes.

- Use standard materials and dimensions. This will save both time and labor.

- Use flexible silicone caulk for sealing the rock bin. Silicone caulk will not dry out and crack and will therefore minimize leaks.

- Design for easy maintenance. All components (blowers, filters, and so on) must be accessible for repair and maintenance.

- Install a small (1/4 inch ID is sufficient for most residential systems) drain to remove any moisture that might enter the rockbed. In a typical installation the rockbed drains onto the floor near the floor drain. Do not drain the rockbed directly into a sewer because the heat from the rockbed will dry out the trap and allow sewer gas to enter the heating system.

- Design to reduce summer cooling loads. For rock beds located within the heated area, provide a bypass for summer domestic hot water heating so the rock bed can be kept cool. Install the rockbed in a room that can be vented when heat is not needed during the day but is needed at night.

Analysis of Storage Fundamentals

First determine whether rockbed storage is suitable for the energy system.

1. Energy Source _____

To determine whether or not a solid material storage system best interfaces with this energy source, examine Tables 3-1, 3-2, and 3-3.

If a very rough sizing has already been made and a first-approximation economic analysis is desired at this stage, refer to Rosalie T. Ruegg and G. T. Sav, The micro-economics of solar energy in Solar Energy Handbook, Chapter 28, J. F. Kreider and F. Kreith, eds., McGraw Hill, 1980.

2. Type of solid sensible heat storage system to be used _____

(If the entry for item 2 is not rockbed, do not continue with the worksheets in the rest of this section.)

3. End use of stored energy. _____

14.2.2 Heat Load and Rockbed Volume

The heat load and rockbed volume are calculated in this section. The corresponding worksheets are A and B.

If the rockbed is coupled to a solar collector, skip the next two subsections and proceed to the subsection below on Collector Specifications. Use Worksheet B, instead of Worksheet A to determine rockbed volume.

Heating Load

The space heating load is determined in items 1 and 2 of Worksheet A. The calculation is similar to that in Section 14.1.1.

Space Heating Rock Bed Volume

The rockbed volume can be determined by solving equation (14-17). This corresponds to items 3-10 of Worksheet A.

$$V_{RB} = \frac{NL_s}{(T_{max} - T_{min})\ \rho\ C_p} = \frac{Q_s}{\Delta T\ \rho\ C_p} \qquad (14\text{-}17)$$

where

V_{RB} is the volume of the rockbed,
N is the number of days of storage required,
L_s is the daily heating load,
T_{max} is the maximum rockbed temperature (often determined from the temperature of the source),
T_{min} is the minimum rockbed temperature (usually determined by the characteristics of the space heating distribution system),
ρ is the rock density corrected with a fill factor,
C_p is the heat capacity of the rocks,
Q_s is the energy storage capacity of the rockbed
ΔT is the temperature difference between the rockbed when it is charged and when it is discharged.

Collector Specifications

If a solar collector has been chosen as the energy source for the rockbed, the rockbed can be sized using the first rule of thumb, as stated in Section 14.2.1. This procedure is specified in Worksheet B.

If the rock bed is to be storage for a solar collector, determine the appropriate collector size (refer to f-Chart type programs) and then select the collector to be used. The collector manufacturer will specify a recommended air flow requirement for the collector. See items 1-4 of Worksheet B.

Using the first rule of thumb from Section 14.2.1, equation (14-18) can be used to determine the minimum and maximum rockbed volume.

The minimum rockbed volume V_{min-RB} is

$$V_{min-RB} = A_c \cdot 0.50 \qquad (14-18a)$$

The maximum rockbed volume V_{max-RB} is

$$V_{max-RB} = A_c \cdot 0.75 \qquad (14-18b)$$

where

A_c is the collector area in ft^2.

This corresponds to items 5-6 in Worksheet B.

If f-Chart or some other design tool, (see Chapter 13) is available to you, use it at this point to calculate the optimum storage volume before proceeding to the calculation.in the next section.

14.2.3 Size and Shape

Section 6.2.1 discusses the structural characteristics of rock beds, including the rock bed shape.

Worksheet C will presuppose that the rock bed has a vertical orientation. Although a vertical bed provides the best thermal characteristics, it must be designed carefully to fit into the space available.

Other rock bed shapes are horizontal and U-shaped.

Depth

When the total height of the rock bed is being planned, the size of plena, insulation, cover, and space should be considered.

The total height of the rock bin, D_{RB}, can be calculated as follows:

$$D_{RB} = D_{ci} + D_{cover} + D_{pb} + D_{pt} + D_{bi} + D_r \qquad (14-19)$$

where

D_{ci} is the depth of the cover insulation,
D_{cover} is the depth of the cover,
D_{pb} is the depth of the bottom plenum,
D_{pt} is the depth of the top plenum,
D_{bi} is the depth of the bottom insulation,
D_r is the depth of the part of the bin filled with rocks.

This corresponds to Worksheet C-1.

Cross-sectional Area

Refer to Figure 14-1 and Section 14.2.1 - rule of thumb (3). The cross-sectional area, A_x, of the rock bed is its volume divided by its height.

$$A_x = \frac{V_{RB}}{D_R} = W \cdot L \qquad (14\text{-}20)$$

where

W is the rockbed width,
L is the rockbed length.

According to rule of thumb (3), we have the additional relation for the flow area in the bottom plenum A_{pb}

$$A_{pb} = 0.12 \cdot A_x = D_{pb} \cdot W \qquad (14\text{-}21)$$

where

D_{pb} is the depth of the bottom plenum.

The factor of 0.12 is used because in vertical rockbeds the bottom plenum will always contain support for the rocks.

The cross sectional area calculations correspond to items 1-11 of Worksheet C-2. Section 15.2.3 works through an example of the cross-sectional area in great detail.

The rockbed dimensions, area and volume are summarized in items 12-17 of Worksheet C-2.

14.2.4 Rock Size

Read section 6.2.2 on performance characteristics of the rock bed and the paragraphs under Section 6.2.1 Rocks.

Selecting the rock type and size needs to be done with some care. The best rocks for this application are actually rounded river pebbles. Note: For a sample of rock containing mixed sizes, the smallest rocks should be no smaller than 0.75 x nominal rock size, and the largest rocks should be no larger than 1.5 x nominal rock size.

Increasing the rock diameter has two major effects:

- Decreases the pressure drop across the rock bed.
- Increases the equilibration time.

Face Velocity

Read the paragraph on face velocity in section 6.2.2.

The face velocity (also called the superficial velocity) is the volumetric air flow rate divided by the cross-sectional area of the rockbed.

- Increasing the face velocity will increase the pressure drop across the rock bed.
- Increasing the face velocity will also increase the optimal diameter of pebbles or crushed rock within the bed.

$$u_f = q/A_x \qquad (14\text{-}22)$$

where

u_f is the face velocity,
q is the air flow rate,
A_x is the cross-sectional area.

This is calculated in items 1-3 of Worksheet D.

Pressure Gradient Limits

The pressure gradient is necessary for a later calculation. It is determined by dividing both the maximum and minimum pressure drop by the rock bed depth, D_R. This is only for a vertical-flow rock bed.

$$\Delta P_{RB} = \frac{P_{RB}}{D_R} \qquad (14\text{-}23)$$

where

ΔP_{RB} is the pressure gradient across the rockbed,
P_{RB} is the pressure drop across the rockbed,
D_R is the depth of the rocks

The actual value of the pressure gradient is determined after the rock size is chosen. This is discussed in Section 14.2.5.

The pressure gradient calculation above is performed in items 4-8 of Worksheet D.

Rock Bed Performance Map

Examine the rock bed performance map (Figure 14-2) and use the newly determined values u_f and ΔP_{RB-min}, ΔP_{RB-max} to determine the rock size suitable for the system. The results will be entered in items 9-12 on Worksheet D.

- Draw (in red pencil) a line for each of the pressure gradient limits on Figure 14-2. (part of Worksheet D and E)

- In blue, pencil a vertical line for the value u_f.

- The correct rock size will be shown by a curved black line that intersects the blue line within the red borders. Enter this nominal rock size in item 9 on Worksheet D

Thermal Stratification

Now is a good time to check out 14.2.1 rule of thumb (4) to ensure that thermal stratification is satisfactory. This calculation corresponds to items 13-17 of Worksheet D.

If items 15 or 17 are affirmative, then thermal stratification is ensured.

14.2.5 System Pressure Drop

As mentioned earlier in Section 6.2.2, the system pressure drop is the sum of several component pressure drops:

- Rock bed pressure drop (from rock bed performance map and actual pressure gradient).

- Collector pressure drop (from manufacturer).

- Filter and damper pressure drops (from manufacturer).

- Duct losses, including allowance for bends, branch ducts, and expansions or contractions (from ASHRAE Handbook of Fundamentals).

The rock bed pressure drop P_{RB} will be calculated in Worksheet E. The total system pressure drop, also calculated in Worksheet E, will be used to size the blower.

Actual Pressure Gradient

Use the rock bed performance map (Figure 14-2) to determine the pressure gradient corresponding to the nominal rock size and face velocity selected in Worksheet D. This corresponds to items 1-4 of Worksheet E.

Rock Bed Pressure Drop

Rock bed pressure drop, P_{RB}, is determined by multiplying the rock bed depth, D_R by the pressure gradient $\Delta P_{RB\text{-actual}}$.

$$P_{RB} = \Delta P_{RB\text{-act}} \cdot D_R \qquad (14\text{-}24)$$

This corresponds to items 5-6 of Worksheet E.

Total Pressure Drop

The total pressure drop is the sum of five component pressure drops.

$$P_T = P_{RB} + P_c + P_{fil} + P_{da} + P_{du} \qquad (14\text{-}25)$$

where

P_T is the total pressure drop,
P_{RB} is the pressure drop across the rock bed,
P_c is the pressure drop across the collector,
P_{fil} is the pressure drop across the air filter,
P_{da} is the pressure drop across the damper,
P_{du} is the pressure drop through the ducts.

This calculation corresponds to items 6-11 of Worksheet E.

14.2.6 Type and Location of Unit

The objectives the storage system are to meet will determine the type, location, and insulation of the rock bed. Chapter 6 provides further information. Worksheet F is for type and location of the rockbed.

Type of Bin

Typical materials for rock bins include wood, concrete, and cement blocks. Roger L. Cole, et al, Design and Installation Manual for Thermal Energy Storage, National Technical Information Services, U.S. Department of Commerce, Springfield, VA, 1980, furnishes details for construction of both wooden and poured concrete bins.

Location of Unit

Refer to chapter 5; wherever a water tank can be located so can a rock bed. However, as with water tanks, floor loading may be a constraint.

The mass of the rocks, assuming a 30-percent void fraction is defined by:

$$m_R = V_{RB} \cdot \rho \cdot 0.7 \qquad (14\text{-}26)$$

where

m_R is the mass of the rocks,
V_{RB} is the volume of the rockbed,
ρ is the density of the rocks,

and the factor of 0.7 is the fraction of the volume that is taken up by the rocks, i.e., a void fraction of 0.3.

This calculation coresponds to items 3-6 of Worksheet F.

Load

The load L is then

$$L = \frac{m_R}{A_x} \qquad (14\text{-}27)$$

This corresponds to items 7-8 of Worksheet F.

If the load value calculated in item 6 of Worksheet F is greater than 150 lb/ft^2 have a licensed structural engineer determine whether the floor strength is adequate.

Schematics

Sketch the proposed system on a separate sheet of paper, inserting all important components.

14.2.7 Insulation

Read section 14.1.6, the references therein on insulation, and Worksheet G of Section 14.1. The protocol is very much the same; follow and use the design example there, except substitute Table 4-9 to find the insulation factor $\frac{fQ}{At}$.

Section 4.5 on limiting heat loss is also very useful. When the bin is wooden it is most efficient to follow that example.

14.2.8 Selection of Other System Components

Read section 6.2.3 and review Figures 6-5 and 6-7 for fan and motor sizing. Examine the sketch from 14.2.6 to determine which other components will require sizings. Possibilities include:

- Heat exchangers - follow the guidelines in 14.1.8.
- Pumps - refer to 14.1.7.
- Auxiliary furnace - manufacturer's information.

Many of the references at the end of this chapter are invaluable for sizing and selecting components.

Fan Selection

Examine the sketch of the complete system from Section 14.2.6. Carefully determine the maximum temperature to which the fan will be exposed. Study carefully section 6.2.3. Enter this as item 1 of Worksheet H. Complete items 1-7 of Worksheet H to select the fan

Determination of Minimum Motor Horsepower.

Where the motor power requirements are not indicated (as in Figure 6-6b) the fan efficiency η is given instead.

A short calculation is then required to determine the minimum motor power, HP_{min}, in horsepower:

$$HP_{min} = \frac{1.25\ q_m\ P\ T_m}{C\ \eta} \qquad \text{(from equation 6-1)}$$

This calculation corresponds to items 8-13 of Worksheet H.

Also use item 13 of Worksheet H to determine minimum belt drive capacity.

Worksheet A: Rockbed Volume

Heating Load

1.	Monthly space heating load,	Btu/month
2. (a)	Daily space heating load, L_{s1}, if known	Btu/day
(b)	Divide line 1 by 30 days to get L_{s2}	Btu/day
(c)	Use the greatest value from L_{s1} and L_{s2}	Btu/day

Volume

3.	Number of days of storage, N	days
4.	Daily space heating load, L_s (from line 2c)	Btu/day
5.	Multiply lines 1 and 2 to find energy stored, Q.	Btu
6. (a)	Minimum usable rock bed temperature, T_{min}.	°F
(b)	Maximum rock bed temperature, T_{max}	°F
(c)	Subtract line 6a from line 6b to determine ΔT.	°F
7.	Rock density, ρ. (Multiply actual density by fill factor of 0.7.)	lb/ft^3
8.	Heat capacity of rocks, C_p	Btu/lb°F
9.	Multiply lines 6c, 7 and 8	Btu/ft^3
10.	Divide line 5 by line 9 for V_{RB}	ft^3

Worksheet B: Rockbed Volume with Solar Collector

Collector Specifications

1. Desired size of solar collector, A_c. _____ ft^2

2. Type of solar collector to be used. _____

3. Manufacturer and model of collector. _____

4. Necessary volumetric air flow rate q (from manufacturer) _____ $\frac{ft^3}{min}$

Volume Limits

5. Multiply item 1 by 0.5 to obtain minimum volume V_{min-RB} _____ ft^3

6. Multiply item 1 by 0.75 to obtain maximum volume V_{max-RB} _____ ft^3

Worksheet C-1: Size and Shape

Depth

1. Depth limit D_L (also total height of rock bin) for rock bin -- floor to ceiling height, include depth of recessed floor _____ in

2. Minimum of two inches for cover installation, D_{ci} _____ in

3. Thickness of the cover, D_{cover}. _____ in

4. Height of bottom plenum, D_{pb}. _____ in

5. Height of top plenum, D_{pt}. _____ in

6. Thickness of bottom insulation, D_{bi}. _____ in

7. Add items 2-6. _____ in

8. a) Subtract item 7 from item 1 to obtain rock depth, D_r _____ in

   b) Divide item 8 by 12 to obtain rock depth, D_R in feet. _____ ft

Worksheet C-2: Size and Shape

Cross-sectional Area

1. Minimum volume V_{min} (item 5 of B) _____ ft^3
2. Maximum volume V_{max} (item 6 of B) _____ ft^3
3. Rock bed depth D_R (item 8b of C-1) _____ ft
4. Divide item 1 by item 3 to obtain minimum cross-sectional area A_{min} _____ ft^2
5. Divide item 2 by item 3 to obtain maximum cross-sectional area A_{max} _____ ft^2
6. Add item 4 to item 5 and multiply result by 72 to get A_{avg} _____ in^2
7. Bottom plenum height D_{pb} (Worksheet C-1, item 4) _____ in
8. Divide item 7 by 0.12 to obtain the rock bed length L. _____ in
9. Divide item 6 by item 8 to get rock bed width W _____ in
10. Choose convenient W value close to the value in item 9. _____ in
11. Multiply item 8 by item 10 and divide by 144 to obtain actual cross-sectional area A_x. Make sure that A_x is between the values for items 4 and 5 above. If not, change the rock bed width in item 10. _____ ft^2

Summary of Dimensions

12. Rock bed height D_R (item 3) _____ ft
13. Rock bed length (item 8 divided by 12) _____ ft
14. Rock bed width (item 10 divided by 12) _____ ft
15. Rock bed cross-sectional area, A_x (item 11) _____ ft^2
16. Multiply item 12 by item 15 to find volume V_{RB} _____ ft^3
17. Examine Table 4-9 and lines 12, 13, and 14 above to determine relative dimensions of the rockbed _____

Worksheet D: Rocksize

Face Velocity

1. Air flow rate q (Worksheet B, item 4) _____ ft^3/min
2. Cross-sectional area A_x (from C-2, item 15) _____ ft^2
3. Divide item 1 by item 2 to find u_f _____ ft/min

Pressure Gradient Limits

4. Maximum pressure drop, P_{RB} 0.30 $in.H_2O$
5. Minimum pressure drop P_{RB} 0.15 $in.H_2O$
6. Rock depth D_R (from C-2, item 12) _____ ft
7. Divide item 4 by item 6 for ΔP_{RB-max} _____ $in.H^2O/ft$
8. Divide item 5 by item 6 for ΔP_{RB-min} _____ $in.H^2O/ft$

Rock Bed Performance Map

Examine the rock bed performance map (Figure 14-2) and use the newly determined values u_f and ΔP_{RB-min}, ΔP_{RB-max} to determine the rock diameter suitable for the system.

9. Nominal rock diameter _____ in
10. Smallest rock diameter: (0.75 x item 9) _____ in
11. Largest rock diameter: (1.5 x item 9) _____ in
12. Rock diameter range: (item 10 to item 11) _____ in to _____ in

Stratification

13. Rock bed depth, D_R (item 6 times 12) _____ in
14. If item 9 < 4 inches, multiply item 9 by 20 _____ in
15. Item 14 should be less than item 13. (yes/no) _____
16. If item 9 > 4 inches, multiply item 9 by 30 _____ in
17. Item 16 should be less than item 13 (yes/no) _____

If item 15 or item 17 are affirmative, then thermal stratification is ensured.

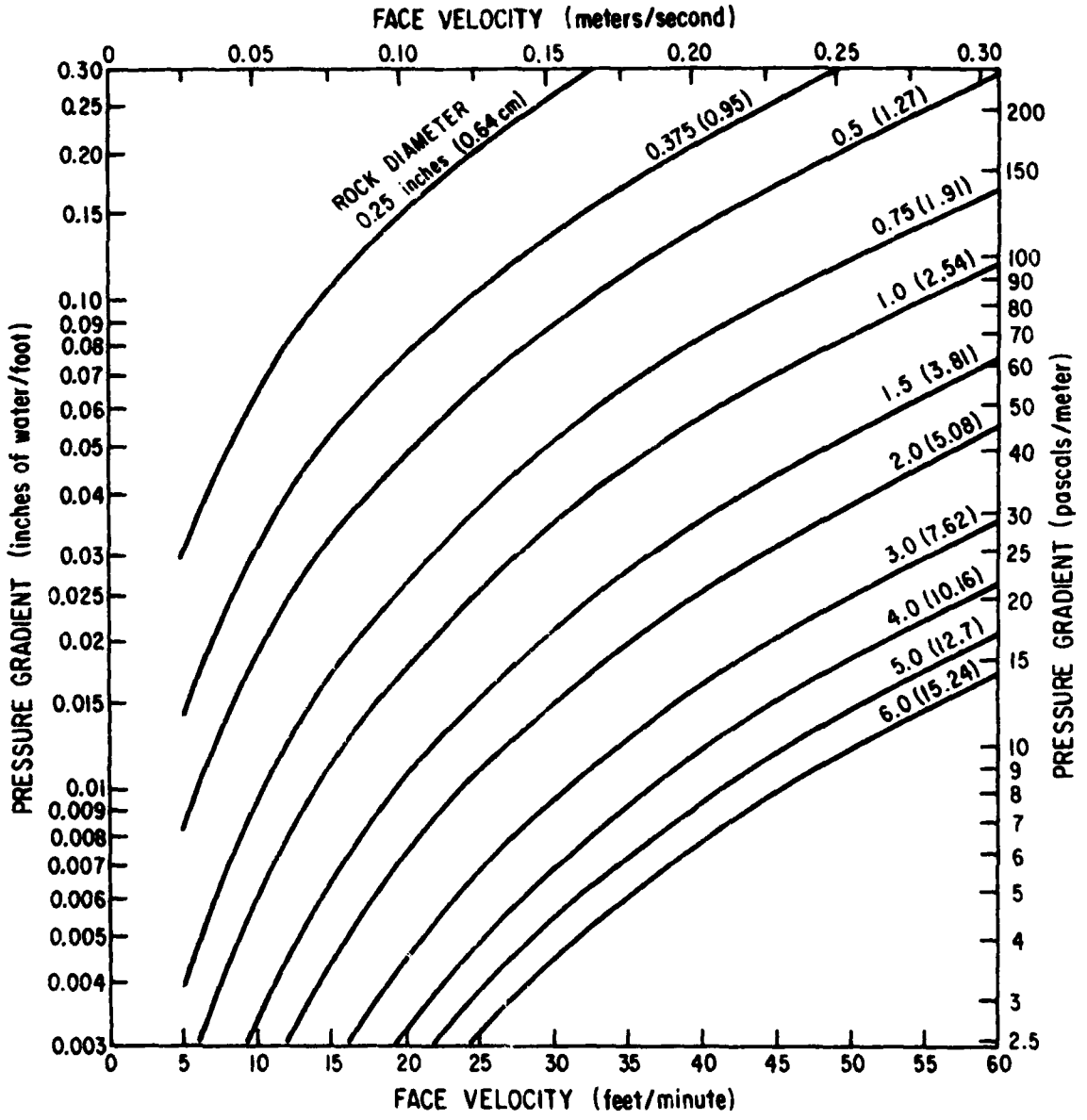


Figure 14-2. Rock Bed Performance Map

Worksheet E: System Pressure Drop

Actual Pressure Gradient

1. Rock diameter (Worksheet D, item 9) _____ in

2. Face velocity u_f (Worksheet D, item 3) _____ ft/min

3. (a) On Figure 14-2, draw a vertical line from the value u_f to intersect the rock diameter.

   (b) From this point, draw a horizontal line to intersect the pressure gradient axis.

4. Pressure gradient value determined from 3b, $\Delta P_{RB\text{-actual}}$ _____ in.H_2O/ft

Rock Bed Pressure Drop

5. Rock bed depth D_R (Worksheet C-2, item 12) _____ ft

6. Multiply item 4 by item 5 to determine rock bed pressure drop, P_{RB} _____ in.H_2O

Total Pressure Drop

7. Collector pressure drop P_c (from manufacturer) _____ in.H_2O

8. Filter pressure drop P_{fil} (from manufacturer) _____ in.H_2O

9. Damper pressure drop P_{da} (from manufacturer) _____ in.H_2O

10. Duct pressure drop P_{du} (calculated using ASHRAE Handbook of Fundamentals, Chapter 25). _____ in.H_2O

11. Sum items 6-10 to find P_T _____ in.H_2O

ROCKBED HEAT STORAGE

Worksheet F: Type and Location

Type

1. Type of bin _____

2. Materials for wall construction and thickness (inside working out)
   i) _____
   ii) _____
   iii) _____

Location of Unit

3. Rockbed volume (from C-2, item 16) V_{RB} _____ ft^3

4. Rock density (from Appendix F) ρ _____ lb/ft^3

5. Rock mass fraction = 0.70 0.70

6. Multiply items 3-5 to find mass, m_R _____ lb

7. Cross-sectional area, (from C-2, item 15) A_x _____ ft^2

8. Divide item 6 by item 7 to find load L. _____ lb/ft^2

If the load value calculated above is greater than 150 lb/ft^2 have a licensed structural engineer determine whether the floor strength is adequate.

Worksheet H: Fan and Moter Selection

Fan Selection

1. Maximum fan temperature T. _____ °F

2. Airflow-rate q (from B, item 4) _____ cfm

3. Static pressure P_T (from E, item 11) _____ in.H_2O

4. If line 1 is less than 100°F, enter 1.0.
   Otherwise, enter factor = (T+460)/530 _____

5. Modified flowrate q_m,
   multiply line 2 by line 4. _____ cfm

6. Modified static pressure P_{Tm},
   multiply line 3 by line 4 _____ in.H_2O

7. Obtain fan performance data from fan manufacturers and select a fan on the basis of lines 5 and 6.

   a) Fan manufacturer _____

   b) Fan type _____

   c) Model number _____

   d) Operating speed _____ rpm

   e) Motor power requirements or _____ HP

      Fan efficiency η at design point _____ %

Minimum motor horsepower

8. Service factor. 1.25

9. Product of lines 5, 6, and 8 _____

10. Fan efficiency, η (%) _____

11. Unit conversion factor, c 63.46

12. Multiply lines 10 and 11 _____

13. Divide item 9 by item 12 to find
    minimum motor size HP_{min}, _____ HP

Also use line 13 for minimum belt drive capacity.

14.3 Phase Change Material Heating System

Thermal energy storage using phase change materials (PCMs) is still a very new technology. Although showing great potential, this technology has been troubled by both technical and economic problems in the marketplace. For this reason, no sophisticated strategies have been developed to size all parameters of PCM energy storage units. This design example attempts to cover the salient features; but for more detail about a specific system contact the manufacturer. Section 7.5 describes a wide range of available systems and lists manufacturers and addresses.

An outstanding feature of latent heat storage systems is the compactness of the storage unit compared with sensible heat storage units. The volume of 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 is not a concern in phase change storage systems, because their temperatures remain nearly constant throughout the charge-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 fully charged sensible heat storage system typically operates at 40° to 60°F (22° to 33°C) above its minimum operating temperature. 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.

Although PCM systems are characterized by the above advantages, the following limitations will appear throughout the rest of the section:

erratic melting behavior
phase separation problems
low thermal conductivity and heat transfer rates
corrosiveness
fledgling technology - marketplace uncertainty

The following, discussed in sections 14.3.1-14.3.7, are important considerations in designing a PCM energy storage system:

- analysis of storage purpose
- choosing appropriate storage material
- sizing the system
- determining type, location, and insulation of container

- heat exchangers and other means of providing heat transfer
- pump and/or fan characteristics
- cost

14.3.1 Analysis of Storage Purpose

Review chapter 7, read section 14.1.1, and for more precise calculations, refer to DOE-1 or TRNSYS.

A simpler method of analyzing system performance, f-Chart, has been adapted for PCM energy storage in air-based solar heating systems* but not liquid-based systems. Some studies, however, have indicated that it is considerably more advantageous to use PCM storage with air-based systems**. Storage volume reduction is more pronounced and there is one less heat-exchange process.

Latent heat storage is appropriate for a wide range of energy sources--ranging from passive solar to active solar systems, off-peak energy for heat pumps, or even waste heat.

As shown in Figures 2-2 and 2-3, the cycle of usage, very standard for solar systems, is as follows:

1. All systems off.
2. Collect and store energy in storage unit when demand is zero (charging mode).
3. When energy is not available for collection, and the demand is not zero, extract and distribute energy from storage (discharging mode).
4. When demand exists, storage is exhausted, and the collector supplies insufficient energy: use energy from an auxiliary source such as a furnace.
5. When storage capacity limit is reached, reject collected energy.
   (a) If demand exists, collected heat goes to load.
   (b) If no demand, reject collected heat to ambient.

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*J. J. Jurinak and S. I. Abdel-Khalik. Sizing phase-change energy storage units for air-based solar heating systems. Solar Energy 22: 355-359, 1979.

**D. J. Morrison and S. I. Abdel-Khalik. Effects of phase-change energy storage on the performance of air-based and liquid-based solar heating systems. Solar Energy 20: 57-67, 1978.

Load Determination

The load can be variable, depending on the need. For space heating and domestic hot water refer to section 14.1.1 and complete Worksheet A in Section 14.1.

First Estimates of Other Storage Parameters

If the energy source is a solar collector, and if the area of the solar collector has already been determined, then the amount of storage area needed can be estimated from

$$Q = 600\ A_c \tag{14-28}$$

where

Q is the amount of storage in Btu,
A_c is the collector area in ft^2

The melting point temperature of the PCM, T_{mp}, can be estimated from

$$T_{mp} = T_{min} + \Delta T \tag{14-29}$$

where

T_{min} is the end use temperature,
ΔT is the temperature difference necessary for the heat exchanger to deliver an adequate amount of heat from the PCM to the load

The estimates made in these two equations are calculated in Worksheet B.

14.3.2 Choosing Appropriate Storage Material

This is probably one of the more difficult decisions to make when designing a latent heat storage system. Refer to Section 7.2 and the following paragraphs on melting point, latent heat of fusion, and heat capacity. Then fill in Worksheet C, listing qualities about PCMs. Values for the various criteria can be found from Table 7-1, the manufacturer, or handbooks such as those cited at the end of Chapter 14. When complete, use the comparisons as a guide in choosing the PCM.

The ΔT values in item 21 of Worksheet C and in item 5 of Worksheet B can be refined when a better estimate of heat exchanger performance is obtained.

Latent Heat of Fusion

The latent heat of fusion given in Table 7-1 tells how much heat can be stored per unit mass of PCM. Materials with a large latent heat will generally make a compact storage unit. Water has the highest latent heat, but its low melting point makes it useful only for cold-side storage. The hydrated inorganic salts generally have a slightly larger latent heat than the organic substances. Both types of materials are used in commercially available systems.

Melting Point

The melting point has a strong influence on the type of system that can be used with a particular PCM. In heating systems, the melting point must be several degrees warmer than the delivered temperature because of the temperature drop in the heat exchanger. Studies have shown that people find 70°F (21°C) air chilling in a forced-air heating system. For traditional ductwork and air velocities, temperature of the air leaving ducts into room space must be about 120°F for occupants to feel comfortable. Lower temperatures (80°-90°F) can be used as in passive systems, but require lower air velocities and correspondingly larger ductwork. Accordingly, about 80°F (27°C) is the lowest practical melting point for use in an air-based direct solar heating system. A liquid-based system, which has one more heat-exchange process than an air-based system, has a minimum practical melting temperature of about 90°F (32°C). Solar-powered cooling systems would require a minimum melting temperature of about 200°F (93°C) for hot-side storage. For cold-side storage, a delivered temperature of less than 55°F (13°C) is necessary to achieve dehumidification. This means that a maximum melting temperature of about 45°-50°F (7°-10°C) would be required for cold-side storage. Solar-assisted heat pumps have considerable flexibility in their storage temperature and can use materials with melting points between 50° and 90°F (10°-32°C).

There are commercially available latent heat storage systems applicable to liquid- and air-based direct solar heating and also solar-assisted heat pump systems. But at present, there are no known commercially available latent heat storage systems applicable to either hot-side or cold-side storage in solar-powered cooling. Ice storage is available for non-solar-powered cooling systems. The PCMs used in the commercially available systems include calcium chloride hexahydrate, sodium sulfate decahydrate, sodium thiosulfate pentahydrate, and several waxes. Although the inorganic salt hydrates have fairly well-defined melting points, there is considerable variation in the melting points of the waxes. Some, being mixtures of several waxes, do not have a unique melting temperature but melt over a small temperature range.

Heat Capacity

Besides storing latent heat, most latent heat storage systems are allowed to store a small amount of sensible heat by cooling below or heating above the melting temperature. Storing the sensible heat allows a temperature sensor to detect when the storage unit is completely charged or discharged. For these systems, you will need to know the heat capacities of the solid and liquid PCM so that you can calculate the amount of PCM required to store a given amount of heat. The amount of sensible heat stored in these systems is usually small compared with the latent heat stored. Determining the size of the latent heat storage system is covered in another section of this chapter.

14.3.3 Sizing Latent Heat Storage Systems

PCM Mass

As illustrated in section 7.2, equation 7-2 defines the amount of heat, Q, stored in terms of the mass of phase change material and latent sensible heat term contributions. Since Q was obtained in Worksheet A, item 3, the equation can be rearranged to determine the quantity of PCM which is:

$$m = \frac{Q}{C_{ps}\,(T_{mp} - T_{min}) + \eta\, h_f + C_{pl}\,(T_{max} - T_{mp})} \qquad (14\text{-}30)$$

where

m	is the mass of the PCM,
Q	is the amount of heat stored,
C_{ps}	is the heat capacity of the solid PCM,
T_{mp}	is the melting point temperature of the PCM
T_{min}	is the minimum allowed storage temperature
η	is the fraction of the theoretical latent heat that can be recovered,
h_f	is the latent heat of fusion,
C_{pl}	is the heat capacity of the liquid PCM,
T_{max}	is the maximum storage temperature.

This is calculated in Worksheet D.

Volume and Dimensions of Storage Unit

The volume of the storage unit can be determined by

$$V_{PCM} = m/\rho \qquad (14\text{-}31)$$

This is calculated in items 1-3 of Worksheet E.

Note that encapsulation, packaging, heat exchangers, and so forth can cause the volume of the storage subsystem to be considerably larger than the volume of the PCM only.

Since latent-heat storage systems are commercially manufactured, the dimensions of the storage unit will be strongly influenced by the choice of manufacturer. The dimensions are influenced by whether prepackaged, encapsulated, storage modules, or a bulk storage system, is selected. The number of modules, N, needed is

$$N = V_{PCM}/V_{unit} \qquad (14\text{-}30)$$

This is calculated in items 4-7 of Worksheet E.

14.3.4 Type, Location, and Insulation of Storage Unit

Latent heat storage systems have both liabilities and advantages. The type of containment is rather limited, but location and insulation are more flexible than with sensible heat storage systems.

Corrosion Potential and Type of Container

If tank and heat exchanger components are plastic, corrosion is eliminated. Use Table 14-2, outlining chemical compatibility of latent heat storage and construction materials, to complete Worksheet F, items 1-5. Avoid any combination where the corrosion rate (item 3) is greater than 4×10^{-3} inches/year.

4×10^{-3} inches/year is the upper limit for uniform attack, but if non-uniform attack, pitting, or crack corrosion are involved, the upper limit should be much less. Also corrosion inhibitors will radically change corrosion rates or type of attack. Beware of galvanic couples or impurities in either the metals or the PCM. These radically alter the rate of corrosion.

Although a corrosion appraisal is most important for bulk storage systems where tank construction must be selected, it also provides a good check for already encapsulated units. Be particularly cautious of leakage around seams and seals.

Location

Normally located in the basement of a building, PCM bulk storage can be installed indoors or outdoors, above, partially above, or below ground.

If in the basement or other location, withstanding only limited structural stress, determine loading.

$$L = \frac{m_{PCM} + m_{misc}}{A_f} \qquad (14\text{-}33)$$

This is calculated in items 6-10 of Worksheet F.

If floor loading is over 150 lb sq.ft., have a licensed structural engineer determine the load capacity and design whatever reinforcements may be necessary.

Schematic

On a separate sheet of paper draw a sketch of the system.

Table 14-2. Corrosion Potential of Different Phase Change Materials

	TYPE AND RATE OF CORROSION ATTACK (in.x 10-4/year)			
	Stainless Steel	Mild Steel	Copper	$AlMg_3$
Lauric acid	uniform 4	uniform 35	uniform 4	uniform 4
Wax-ester	uniform 4	-----	uniform 4	uniform 20
Sodium thiosulfate pentahydrate $Na_2S_2O_3.5H_2O$	uniform 4	uniform 4	uniform 291	uniform 4
Sodium acid phosphate $Na_2HPO_4.12H_2O$	uniform 4	uniform 4	uniform 4	pitting 26
Calcium chloride hexahydrate $CaCl_2.6H_2O$	uniform 4	uniform 8	uniform 4	pitting 4
Glauber's salt $Na_2SO_4.10H_2O$	-----	-----	-----	-----

Adapted from: D. Heine, The chemical compatibility of construction materials with latent heat storage materials, in Proceedings from International Conference on Energy Storage, Brighton, UK, 1981.

Insulation

The compactness of latent heat storage containers means that they need less insulation than sensible heat storage containers of the same capacity. Counting the comparable amount of external insulation and thermal performance, ratios of volume between water tanks or rock beds and bulk storage PCMs can be as high as 4:1.

Having a smaller volume and thus less surface area to insulate also implies that the insulation can be thinner. Suppose, for example, that a latent heat storage (LHS) unit occupies only 60% as much volume as a sensible heat storage (SHS) unit of the same storage capacity. If the two storage units are geometrically similar, the LHS unit will have $0.6^{2/3} = 0.711$ as much surface area to insulate as the SHS unit. Because the LHS unit has only 0.711 as much area as the SHS unit, the insulation required for the LHS unit to maintain the same overall heat loss rate (Btu/hr) is only 0.711 as thick as the insulation required for the SHS unit. Thus, there is a double saving of insulation and the LHS unit requires $0.711 \times 0.711 = 0.506$ (about half) as much total insulation as the SHS unit in this example. In addition to reduced surface area and thickness, the lower average temperature at which the latent heat storage unit operates makes it possible to use less insulation.

While latent heat storage devices require substantially less insulation than sensible heat storage devices, Tables 4-6 through 4-9 cannot be used to calculate the amount of insulation they require. Equation 14.3-1, however, is completely general and can be used to calculate the maximum allowable thermal transmittance, U, in Btu/hr.ft^2.°F (W/m^2.°C), for latent heat storage containers.

$$U = \frac{fQ}{At} \frac{1}{(T_{mp} - T_a)} \qquad (14\text{-}34)$$

This is calculated in items 1-11 of Worksheet G.

The thickness of the insulation can be found from

$$s = \frac{1/U - \Sigma R_{fix}}{r^*} = \frac{R_{req} - R_{fix}}{r^*} \qquad (14\text{-}35)$$

The thickness is calculated in items 12-17 of Worksheet G. See also Section 14.1.6 and Worksheet G of Section 14.1.

14.3.5 Heat Exchangers and Other Means of Providing Heat Transfer

As mentioned in 7.2 and 7.3.4, a high thermal conductivity in both liquid and solid phases facilitates good heat exchange during charging and discharging. The solid phase generally has a lower conductivity that limits the rate at which heat can be removed from storage.

Below is a check list of techniques to increase thermal conductivity and heat transfer of solid phase:

- Package PCM in thin containers with large surface area
- Stir liquid phase
- Include immiscible heat transfer fluid
- Use a specially designed heat exchanger

Encapsulation

Packed beds of encapsulated phase-change materials have good heat transfer characteristics because of large surface areas involved. Heat transfer involves sensible heat during the solid and liquid phase and latent heat during the phase transition.

The surface area per unit volume A/V should be maximized.

Agitation

Stirring is usually done in conjunction with a heat exchanger or heat transfer fluid. By mixing solid PCM with melted material, stirring minimizes the problems of insoluble hydrates separating from an incongruently melting PCM. In addition, the agitation tends to prevent subcooling. The disadvantage of agitation is that it requires energy (usually electrical).

Immiscible Heat Transfer Fluid

Also called direct contact liquid-liquid heat transfer, this technique effectively remedies phase stratification and heat transfer problems.

Bubbling heat transfer liquid directly through the salt hydrate storage medium in the discharge cycle produces relatively fine crystals and a more homogeneous system. However, some problems with solid PCM plugging the nozzles have been experienced.

Heat Exchanger

Generally heat exchangers are the major cost component of PCM thermal energy storage. The low thermal conductivity of organic PCMs and the absence of convection in phase-stabilized salt hydrates results in the need for heat exchangers with large surface areas per unit storage volume.

Two major heat exchanger options for latent heat systems exist:

- inexpensive extruded plastic tubing coiled in the tank
- shell and tube heat exchanger associated with fins, conducting honeycomb, or metal mesh filling the container

Review section 14.1.8 on heat exchangers including Worksheet I of Section 14.1. The heat exchanger for PCM systems may also be determined with the help of Worksheet H.

14.3.6 Sizing Other Components

Pumps are important for a water-based solar system; fans for an air-based system. In the situation where the load includes both domestic hot water and space heating, both may be required.

Refer to section 14.1.6 on pump characteristics and 14.2.7 on selecting a fan and motor. Complete as much of these design examples as necessary. The worksheets in these sections will provide adequate direction to size units for a PCM system.

14.3.7 Cost

No one procedure has really been defined for determining economic competitiveness of PCM heating systems compared to sensible heat systems. There are several--different results will be obtained depending on which economic model is used.

One useful comparative index between various storage technologies is the storage peak capacity capital cost, C_2. This can be determined by equation

$$C_2 = \frac{C}{Q} \qquad (14-36)$$

where

C is the system cost,
Q is the storage capacity

This is calculated in items 1-3 of Worksheet I.

Another approach, more immediately useful to the system designer, is the following.

$$I = \frac{C_L - C_S}{C_T - C_S} \qquad (14\text{-}37)$$

where

I is the cost index,
C_L is the installed cost of the latent heat storage subsystem,
C_S is the installed cost of a sensible heat storage subsystem of equivalent capacity,
C_T is the estimated cost for the total installed system.

If item 9 on Worksheet I is 0.05 or less, consider using latent heat storage. If item 9 is 0.15 or more, latent heat storage is unlikely to be cost effective. If item 9 is between 0.05 and 0.15, a detailed analysis as outlined below is necessary to determine whether latent heat storage is cost effective.

A more detailed approach (which is beyond the scope of this manual) is to analyze and compare several complete alternative systems. These might include a system with latent heat storage and a system with sensible heat storage. An analysis method such as TRNSYS, DOE-1 or f-Chart would be used to estimate the system performance. System performance and cost data would then be used as inputs to the life-cycle cost method.*

Although the calculations required by this detailed approach are lengthy and require a computer, the effect of substituting latent-heat storage for sensible-heat storage upon life-cycle cost is readily apparent. This detailed approach is the only way to determine whether increased collector efficiency due to the lower average storage temperature of latent-heat storage compared to sensible heat storage is enough to offset the higher initial cost of latent-heat storage.

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*Rosalie T. Ruegg. Solar Energy: Methods of Economic Evaluation, National Bureau of Standards, NBSIR 1127, Washington, DC, 1975.

Worksheet B: Estimate of Storage Parameters

Storage Capacity

1. Energy source _____

2. Solar collector area (solar systems only) A_c _____ ft^2

3. Multiply item 2 by 600 to obtain the approximate storage capacity requirement, Q, for solar systems, or enter Q directly for non-solar systems. _____ Btu

Temperature

4. End use temperature T_{min} (typically 90°F for space heating or 120-140°F for domestic water heating). _____ °F

5. Approximate load heat exchanger ΔT (at least 20°F for each air-to-air, liquid-to-air, or PCM-to-air heat exchange separating the PCM from the end use). _____ °F

6. Add items 4 and 5 to obtain the minimum melting point for the PCM, T_{mp} (Subtract item 5 from item 4 to obtain the maximum melting point for a cold storage PCM.) _____ °F

Worksheet C-1: Selection of Phase Change Material

	Parameter	Preferred Value	Phase Change Material		
			Option 1	Option 2	Option 3
1.	Minimum (maximum for cold storage) melting point temperature (Worksheet B, item 6)°F				
2.	Phase change compound considered				
3.	Heat of fusion h_f (Btu/lb) (from Table 7-1)	High			
4.	Solid heat capacity (Btu/lb°F) (from Table 7-1)	High			
5.	Liquid heat capacity (Btu/lb°F) (from Table 7-1)	High			
6.	Melting point (°F) (< item 1 for cold storage)	close to but > item 1			
7.	Proposed application temperature °F				
8.	Thermal conductivity (Btu/ft^2.hr.°F)	High			
9.	Vapor pressure (psi)	Low			
10.	Coefficient of thermal expansion (1/°F)	Low			
11.	Phase-change volume change (ft^3)	Low			
12.	Type of melting behavior	Congruent			
13.	Number of projected thermal cycles without significant phase separation	High			
14.	Charge power density (kW/ft^3) from manufacturer)	High			
15.	Rate nucleation efficiency (read section 7.3.2 for enhancement techniques)	High			

Worksheet C-2: Selection of Phase Change Material

	Parameter	Preferred Value	Phase Change Material		
			Option 1	Option 2	Option 3
16.	Rate corrosiveness	Low			
17.	Rate toxicity	Low			
18.	Rate fire-resistance	High			
19.	Cost	Low			

Selection

20. Using items 1-19 above, select the PCM. _____ Material

    a. melting point T_{mp} _____ °F

    b. latent heat of fusion h_f _____ Btu/lb.

    c. heat capacity of solid PCM C_{ps} _____ Btu/lb°F

    d. heat capacity of melted PCM C_{pl} _____ Btu/lb°F

21. Approximate source heat exchanger ΔT (at least 20°F for each air-to-air, air-to-liquid, or air-to PCM heat exchange or 10°F for each liquid-to-liquid or PCM-to-liquid heat exchange separating the PCM from the heat source or solar collector.) _____ °F

22. Add items 20a and 21 to obtain the minimum required source or solar collector temperature. (subtract item 21 from item 20a to obtain the maximum allowable source temperature for cold storage.) _____ °F

The ΔT values in item 21 and in Worksheet B, item 5, can be refined when a better estimate of heat exchanger performance is obtained.

LATENT HEAT STORAGE

Worksheet D: PCM Mass

1.	Amount of heat stored, Q, from Worksheet A, item 3.	Btu
2.	Heat capacity of solid PCM, C_{ps}	$\frac{Btu}{lb°F}$
3.	Melting point temperature T_{mp}	°F
4.	Minimum storage temperature T_{min}	°F
5.	Subtract item 4 from item 3	°F
6.	Multiply item 5 by item 2	$\frac{Btu}{lb}$
7.	Fraction of theoretical latent heat that can be recovered, η for congruent melting = 1	
8.	Latent heat of fusion h_f	$\frac{Btu}{lb}$
9.	Multiply item 7 by item 8	$\frac{Btu}{lb}$
10.	Heat capacity of liquid PCM, C_{pl}	$\frac{Btu}{lb°F}$
11.	Maximum storage temperature, T_{max}	°F
12.	Subtract item 3 from item 11	°F
13.	Multiply item 12 by item 10	$\frac{Btu}{lb}$
14.	Add together items 6, 9, 13	$\frac{Btu}{lb}$
15.	Divide 1 by item 14 to obtain m, amount of PCM	lb

Values for items 2, 3, 8, 10 can be determined from Table 7-1.

Worksheet E: Volume and Dimensions of Storage Unit

Volume

1. From Table 7-1, density of chosen storage material, ρ _____ lb/ft^3

2. Total mass required from Worksheet D, item 15 _____ lb

3. Divide item 2 by item 1 to determine volume, V_{PCM} of phase change material only. _____ ft^3

Number of Modules

4. Module height _____ ft.

5. Module floor area _____ ft^2

6. Volume of each unit _____ ft^3

7. Divide item 3 by item 6 to determine minimum number of modules needed for specified storage capacity, N _____ modules

Dimensions

8. The manufacturer will suggest preferred packing configurations. _____ packing

9. Height of assembly _____ ft

10. Floor area of assembly _____ ft^2

Worksheet F: Type and Location of Storage Unit

Corrosion Potential and Type of Container

1. Proposed PCM material _____
2. Construction material _____
3. Corrosion rate (COR) _____ in/year
4. Corrosion inhibitor _____
5. Length of manufacturer's warranty _____

Location

6. Required weight of PCM, m_{PCM} (from Worksheet D, item 15) _____ lb.
7. Weight of containers, supports, etc m_{misc} (from manufacturer) _____ lb.
8. Total weight, add items 1 and 2 _____ lb.
9. Floor area occupied by the storage assembly (Worksheet E, item 10) _____ ft^2
10. Floor loading, L, divide item 3 by item 4 _____ lb/ft^2

If floor loading is over 150 lb/sq.ft., have a licensed structural engineer determine the load capacity and design whatever reinforcements may be necessary.

Worksheet G: Insulation Requirements

1. Fraction of heat lost in time t, f _____

2. Amount of heat stored, Q (from Worksheet B, item 3) _____ Btu

3. Multiply item 1 by item 2 _____

4. Surface area of storage unit, A _____ ft^2

5. Length of time used for specifying the maximum heat loss from storage, t _____ hr

6. Multiply item 4 by item 5 _____

7. Melting point temperature T_{mp} _____ °F

8. Average ambient temperature, T_a _____ °F

9. Subtract item 8 from item 7 _____ °F

10. Multiply item 6 by item 9 _____

11. Divide item 3 by item 10 to obtain maximum allowable thermal transmittance, U _____ $\frac{Btu}{hr.ft^2.°F}$

12. Required R-value = 1/U Divide 1.0 by item 11 _____ $\frac{°F.ft.hr}{Btu}$

13. Fixed insulation layer resistances
    a. _____
    b. _____
    c. _____

14. Total fixed layer resistance R_{fix} - add lines 13 a, b, and c _____

15. Subtract line 14 from line 12. _____

16. Assume that the insulation layer of unknown thickness has no parallel heat loss paths. Thus, find r^* in Table 4-4. _____

17. Thickness s of layer, divide line 15 by r^* (line 16) _____ in

Worksheet H: Heat Exchanger for PCM Systems

1. Length _____ ft.

2. Diameter _____ ft.

3. Heat exchanger area _____ ft^2

4. Type of heat exchanger _____

5. Alterations made to improve surface area flow rate a._____

   b._____

6. Flow rate _____ gpm

7. Location of outlet and inlet _____

8. Heat exchanger area/salt volume ratio _____ 1/ft

   should be >15ft^2/ft^3.

Worksheet I: Cost Analysis of PCM System

1. Optimal storage capacity Q, from Worksheet B, item 3 _____ Btu

2. Projected cost of system, C $_____

3. Divide item 2 by item 1 to obtain storage peak capacity capital cost C_2. Compare this figure to the storage peak capacity capital cost for a sensible heat storage system. $_____/Btu

4. Estimated cost for total installed system, C_T $_____

5. Estimated cost of installed sensible heat storage subsystem C_S $_____

6. Subtract item 5 from item 4. $(C_T - C_S)$ $_____

7. Estimated cost of installed latent heat storage subsystem of equivalent capacity C_L $_____

8. Subtract item 5 from item 7. $(C_L - C_S)$ $_____

9. Divide item 8 by item 6 to obtain the cost index I. $_____

14.4 Water Tank Cold Storage

This example describes the sizing and design of a water tank cold storage system for space cooling. Chapter 8 discusses many technical details of thermal cold storage. Chapter 4 discusses general features of sensible thermal storage, and Chapter 5 discusses features of liquid sensible thermal storage. These chapters should be read first before designing a prototype system.

This design example emphasizes the following:

- Calculation of the cooling load is discussed in Section 14.4.1.
- Tank sizing is described in Section 14.4.2.
- Stratification is discussed in Section 14.4.3.
- Other system details, such as tank insulation, pump characteristics and heat exchanger details are similar to water tank heat storage. These features were covered earlier in Sections 14.1.6, 14.1.7, and 14.1.8 and will be mentioned only briefly in this design sample.

14.4.1 Cooling Load

The cooling load consists of two parts, sensible cooling and latent cooling, and is highly dependent on the lighting, use of machinery, and activity of the occupants within the space to be cooled, as well as the outside climate.

A detailed cooling load can be calculated using the techniques described in the ASHRAE Handbook of Fundamentals. The total cooling load for residential applications can usually be approximated as 1.3 times the sensible load.

If electric powered air conditioners are used, the average cooling load can be estimated from the metered electricity consumption of the air conditioner and a careful estimate of its EER rating. (EER is the energy efficiency ratio in Btu of cooling per W.hr of electricity consumed.) Values listed by manufacturers are usually for machines at the beginning of their operating history. EER may drop substantially as the machine is operated, especially in corrosive environments typical of urban locations.

As is shown in items 1-3 of Worksheet A, three equations can be manipulated to provide information about the load.

$$L_T = \frac{\text{EER} \times \text{Whr}}{\text{time (days)}} \quad \text{(Btu/day)} \qquad (14\text{-}38)$$

$$L_T = 1.3\, L_S \qquad (14\text{-}39)$$

$$L_T = L_S + L_L \qquad (14\text{-}40)$$

where

L_T is the total cooling load,
L_S is the sensible cooling load,
L_L is the latent cooling load,
Whr is the amount of electricity consumed in Watt hours.

The EER method of eqn (14-38) will predict L_T without needing to know either L_S or L_L. Note that most electricity bills are in units of kWhr, not Whr. Multiply kWhr by 1000 to get Whr.

14.4.2 Size and Shape

Tank Temperatures

It is necesary to determine three temperatures:

- Minimum tank temperature, T_{min}
- Maximum tank temperature, T_{max}
- Average tank temperature, T_{avg}

The minimum tank temperature is the temperature of the water returning from the chiller evaporator as specified by the chiller manufacturer. To prevent possible freezing problems, this temperature should be 35°F or higher if water is the storage tank fluid. Lower chiller temperatures require an alternate liquid or the addition of antifreeze to the water in the tank to prevent the possibility of freezing.

The maximum tank temperature is the highest temperature that can effectively remove heat from the space to be cooled. With typical heat exchanger efficiencies, this is 10°F below the room air temperature for sensible cooling. The latent cooling load requires a lower maximum temperature of 55°F to condense water vapor from the air. If the tank is stratified, and if the latent load is not too high, then the higher maximum temperature associated with sensible cooling can be used. Cold fluid will be available from the tank until it is completely discharged. Room air moisture will condense out on the part of the heat exchange coil that is below 55°F.

The average tank temperature (T_{avg}) is the average of the maximum and minimum. It is necessary for insulation calculations.

T_{max}, T_{min}, T_{avg} and ΔT, a derive quantity (and the temperature range of the tank) are related to each other by the equations:

$$T_{avg} = \frac{T_{max} + T_{min}}{2} \qquad (14\text{-}41)$$

$$\Delta T = T_{max} - T_{min} \qquad (14\text{-}42)$$

This corresponds to items 4-7 of Worksheet A.

Tank Size

The theoretical sizing procedure is virtually identical to that of Section 14.1.2. The storage volume, V, is given by

$$V = \frac{Q}{\rho \ Cp \ \Delta T} \qquad (14\text{-}43)$$

Complete Worksheet B and see Section 14.1.2 for further details.

Tank Shape

The calculation of tank shape is identical to that for water tank heat storage. Use the procedure described in Section 14.1.3 and Worksheet D of Section 14.1.

14.4.3 Stratification

Stratification of cold water in tanks is not a well understood process. For temperatures less than 45°F, the thermal expansion coefficient is small and changes sign at 39°F. The small thermal expansion means that the Richardson number would be smaller for the same velocities found in a hot water tank making stratification less likely to occur.

The design of cold water storage tanks should incorporate as many of the stratification techniques discussed in Section 5.2.2 as possible. However, the sizing calculation and system design should not depend on stratification.

14.4.4 Type, Location, and Insulation of Tank

There are really no new problems to be confronted here; all sizing calculations can be handled by the pertinent worksheets in the other three sections.

14.4.5 Sizing and Selecting Components

System Sketch

On a separate sheet of paper draw a sketch of the proposed system inserting all anticipated components.

Components

Examine your completed sketch and refer to the appropriate sections on pumps, fans, heat exchangers, and other items. Section 14.1.7 deals with pumps; 14.2.8 covers fans; 14.1.8 describes and sizes heat exchangers.

14.4.6 Economic Analysis of Cooling Systems

Cooling systems coupled to solar collectors, standing on their own, are still far from being economically attractive.

The coefficient of performance, COP, is an indicator of operational efficiency. The coefficient of performance of a conventional cooling system COP_c is generally much higher than the coefficient of performance for a solar cooling system COP_{sol}. In fact, generally their values are approximately:

$$COP_c \simeq 2.5$$

$$COP_{sol} \simeq 0.5 - 0.7$$

Of course, the energy source is different for the two cases. Energy required by the conventional system is given by

$$Q_c = \frac{L}{COP_c} \qquad (14\text{-}44)$$

Energy required by the solar system is given by:

$$Q_{sol} = \frac{L \times (1.0 - f)}{COP_{backup}} \qquad (14\text{-}45)$$

where

L is the cooling load,
f is the solar fraction.

The above equations are used in Worksheet C.

The information from Worksheet C can be used to make a rough estimate of costs and determine life-cycle economics.*

*This is beyond the scope of the outline. Refer to Rosalie T. Ruegg. Solar Energy: Methods of Economic Evaluation, National Bureau of Standards, NBSIR 1127, Washington, DC, 1975.

SENSIBLE COLD STORAGE

Worksheet A: Cooling Load and Tank Temperature

Cooling Load

1. Sensible cooling load L_S (from ASHRAE Method) _____ Btu/day

2. Latent cooling load L_L (from ASHRAE) _____ Btu/day

3. Add line 1 and line 2 to get total cooling load L_T (or from eqn (14-36)) _____ Btu/day

Tank Temperature

4. Maximum tank temperature, T_{max} _____ °F

5. Minimum tank temperature, T_{min} _____ °F

6. Add lines 1 and 2 and divide the result by 2 to get T_{avg}. (This value will be used later in determining insulation requirements _____ °F

7. Subtract line 2 from line 1 to get the temperature range of the tank, ΔT _____ °F

SENSIBLE COLD STORAGE

Worksheet B: Tank Size

1.	Daily cooling load L from Worksheet A, item 1 or item 3	_____ Btu/day
2.	Number of days storage desired, N	_____ days
3.	Thermal storage capacity of tank, Q Multiply item 1 by item 2	_____ Btu
4.	Liquid density, ρ (ρ for water = 8.33)	_____ lb/gal
5.	Heat capacity of water (Heat capacity for water = 1.00)	_____ Btu/lb.°F
6.	ΔT from Worksheet A, item 7	_____ °F
7.	Multiply lines 4, 5, 6	_____ Btu/gal
8.	Divide line 3 by line 7 to get tank size V	_____ gal
9.	Multiply line 8 by 0.13368 to get tank size V in cubic feet	_____ ft^3

SENSIBLE COLD STORAGE

Worksheet C: Economic Analysis of Cooling System

1. Cooling load demanded of storage system L _____ Btu

2. Coefficient of performance of the conventional system, COP_c _____

3. Divide item 1 by item 2 to determine the energy required by the conventional system, Q _____ Btu

4. Solar fraction, f, $\leq$ 60% usually. _____

5. Subtract item 4 from 1.0. _____

6. Multiply item 5 by item 1. _____ Btu

7. Coefficient of performance of backup system COP_{backup} _____

8. Divide item 6 by item 7 for energy required. _____ Btu

Compare item 3 with item 8 to determine the difference in energy requirements between solar cooling and a conventional cooling system.

14.5 Symbols Used

Main Symbols

A	area (ft^2)
A_c	solar collector area (ft^2)
A_x	cross sectional area (ft^2)
c	cost
C_p	heat capacity (Btu/lb.°F)
COP	coefficient of performance
COR	corrosion rate
D	depth (ft)
f	solar fraction
fQ/At	insulation factor (see Chapter 4)
g	acceleration of gravity = 32.2 ft/sec^2
H	pump head (ft)
HP	motor horse power
L	heating or cooling load (Btu/day)
	height or length (ft)
	mechanical load (lb/ft^2)
m	weight (or mass) (lb)
N	number of days of storage
	number of modules
NPSH	net positive section head (ft)
P	pressure (in. H_2O)
Q	energy stored (Btu)
Q'	heating rate (Btu/hr)
q	flow rate (lb/hr or gal/min)
R	thermal resistance (°F.ft^2.hr/Btu)
Ri	Richardson number
r*	thermal resistance per thickness (°F.ft^2.hr/Btu.in)
s	thickness (in)
T	temperature (°F)
ΔT	temperature difference (°F)
U	thermal conductance (Btu/hr.°F.ft^2)
u	velocity (ft/sec)
V	volume (gal or ft^3)
W	consumption factor
	width (ft)
Z	height difference (ft)
β	thermal expansion coefficient ($°F^{-1}$)
γ	storage to collector area ratio (Btu/ft^2)
η	efficiency
ρ	density (lb/ft^3)
Σ	sum

Subscripts

a	ambient or atmospheric
avg	average
bi	bottom insulation
c	combined space and water heating
	collector (as in H_c)
	conventional (as in COP_c)
ci	cover insulation
d	daily
da	damper
du	duct
f	friction (as in H_f)
	face (as in U_f)
	floor
fil	filter
fix	fixed
G	gallons
L	latent
l	liquid
m	modified
max	maximum
min	minimum
misc	miscellaneous
mp	melting point
p	constant pressure (as in C_p)
	pump (as in H_p)
par	parallel
pb	bottom plenum
PCM	phase change material
pt	top plenum
R	required
	rockbed
r	rocks
RB	rockbed
req	required
S	sensible
s	space heating
	solid
ser	series
sol	solar
sur	surface
T	tank
	total
v	vapor
w	water
x	heat exchanger
1,2,3,n	first, second, third, nth

14.6 References

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15.0 EXAMPLE PROBLEMS

In Chapter 14, design examples were presented for sizing each of the following:

1. Sensible heat storage utilizing water as the storage medium.
2. Sensible heat storage utilizing rocks as the storage medium.
3. Latent heat storage utilizing a eutectic salt as the storage medium.
4. Sensible cold storage for cooling purposes.

Worksheets then summarized the information in each case.

In this chapter, plausible real-life problems will be defined (pertinent to specific navy applications) and a solution arrived at, by working through the design example. This then will provide the technical reader with a "how-to" demonstration of each example. Table 15-1 below gives some indication of the variety of problems tackled.

Table 15-1. Summary of Prototype Designs

Design Example	Storage Medium	Energy Source	Facility	Energy Use	Size of Unit (Btu)	Average Storage Temperature (°F)
15.1	Water	Solar	Naval Barracks	Space & Water Heating	2,000,000	135
15.2	Rock	Solar	Single-family residence	Space Heating	262,000	115
15.3	Phase Change Material	Off-peak Electricity	Large Research Center	Water Heating	35,000,000	270
15.4	Water	Waste Heat	Administrative Headquarters	Space Cooling	30,240,000	45

15.1 Water Tank Heating System

*In this application a liquid-based storage system will be installed in Naval barracks housing forty people to provide both domestic hot water and space heating. The input heat is supplied by a solar collector. Backup is by electric resistance heating.

15.1.1 Heating Load

The design month was selected as the average of the four coldest months. Considerable insulating and weather stripping was done on the building two years ago. By analyzing the average monthly fuel consumption for space heating purposes over the past two years, it was determined that the average monthly space heating load is 50,000,000 Btu. Worksheet A is completed as follows:

1.	(a) Number of people in household, or number of living units in apartment, N	~~units~~/people 40	
	(b) Consumption factor as shown above W	15	
	(c) Multiply lines 1a by 1b to find total daily water consumption V_d		600 gal
2.	(a) Minimum acceptable hot water temperature, T_{max}	120°F	
	(b) Measured temperature of main supply water, T_{min}.	50°F	
3.	Subtract 2b from 2a to find ΔT_e		70°F

The *minimum acceptable temperature* will vary depending on the end use. If dishwashing or clothes washing are done, the temperature must be about 160°F. At this elevated temperature, it is probably most economical to use an electric or gas booster heater to increase the temperature upon demands for this special purpose. However, in this case, the only hot water needs are for personal cleaning; 120°F is adequate.

The water temperature is determined by letting the cold water faucet run for several minutes - three different times of the day and averaging the temperatures.

4.	Water density, ρ	8.33 lb/gal	
5.	Heat capacity of water, C_p	1 Btu/lb°F	
6.	Multiply lines 1b, 3,4, and 5 to find L_w (600 gal x 70°F x 8.33 lb/gal x 1 Btu/lb°F = 350,000 Btu)		350,000 Btu/day

*In each section, the worksheet format will not necessarily be an exact copy of that found in the corresponding section of chapter 14.

The total daily load is L and can be found from:

$$L = L_s + L_w \qquad (14\text{-}2)$$

7. Monthly space heating load,		50,000,000 Btu/month
8. (a) Daily space heating load, L_{s1} if known		Btu/day
(b) Divide line 7 by 30 days, L_{s2}	1,670,000 Btu/day	
(c) Use the greatest value from L_{s1} and L_{s2}		1,670,000 Btu/day
9. Add lines 6 and 8c to obtain L		2,020,000 Btu/day

15.1.2 Size and Shape

No other design tool is available, so we choose 1.0 days of storage as our criteria for both domestic hot water and space heating. At this point, the size of the solar collector is not known.

Domestic Hot Water Tank

Worksheet B-1 is used.

1. Number of days of storage desired, N	1.0 days	
2. Total daily water consumption V_d (from item 1c of Worksheet A)	600 gal/day	
3. Multiply lines 1 and 2 to find total tank volume, V_{TGw}		600 gal

Space Heating Water Tank

The existing heating system requires air at a minimum temperature of 100°F to heat the space. Allowing for a 10°F drop across the water to air heat exchanger, the minimum tank temperature is 110°F. The space heating tank will be concrete. 160°F is chosen as the maximum temperature. Higher temperatures are possible, but the tank will be designed for good stratification, and higher temperatures might cause undue thermal stress on the tank walls.

Worksheet B-2 is completed as follows:

1.	Number of days of storage, N	1.0 days	
2.	Daily space heating load, L_s (from line 8c of Worksheet A)	1,670,000 Btu/day	
3.	Multiply lines 1 and 2 to find energy stored, Q.		1,670,000 Btu
4.	(a) Minimum usable water tank temperature, T_{min}.	110°F	
	(b) Maximum water tank temperature, T_{max}	160°F	
	(c) Subtract line 4a from line 4b to determine ΔT	50°F	
5.	Water density, ρ	8.33 lb/gal	
6.	Heat capacity of water, C_p	1.0 Btu/lb°F	
7.	Multiply lines 4c, 5, and 6	416.5 Btu/gal	
8.	Divide line 3 by line 7 for V_{TGS}		4010 gal
9.	Multiply line 8 by 0.13368 to obtain tank size V_{TFS} in cubic feet		536 ft^3

Tank Shape

This section corresponds to Section 14.1.3.

Since the two water tanks will operate at different temperatures, two separate tanks will be specified:

- a standard pressurized steel tank for DHW per ASME Pressure Vessel Code, Section VIII.
- a six-inch-thick walled concrete tank for space heating water storage.

As an illustration of Worksheet D, only the details of the concrete tank are worked out.

The tank must fit in a relatively small rectangular room. A rectangular tank will fill the space better.

The room in which the tank is to be installed is 14 ft by 20 ft and has a 12 ft ceiling. We have to pour a 6 in thick slab on top of the existing floor for the tank bottom, but the top will be metal, covered with

insulation. We leave 2 1/2 ft at the top for crawl space and for room to maneuver the tank top. This makes the desired depth of the tank 9 ft. Two walls of the tank can be shoved into the corner, but a 4 ft access must be maintained on at least two sides. We have two 6-in thick walls, each of which will probably have at most a foot of insulation on them. The maximum allowed width is then:

$$14 - 2\ (1/2) - 2(1) - 4 = 7 \text{ ft.}$$

For Rectangular Tanks:

6.	Desired tank depth D	9.0 ft
7.	Width of a square tank ($w = [V/D]^{1/2}$	7.7 ft
8.	Maximum width permitted w_{max}	7.0 ft
9.	If $w > w_{max}$, calculate tank length ($L = V/Dw_{max}$)	8.5 ft

If $w > w_{max}$, choose a tank with dimensions $L \cdot w_{max} \cdot D$. Otherwise, choose a tank with dimensions $w \cdot w \cdot D$. Although a single tank is usually less expensive and thermally more efficient, multiple storage units can be used to accommodate space restrictions.

10.	Dimensions of tank	9 ft x 7 ft x 8.5 ft
11.	Relative dimensions of tank (see Tables 4-6, -7, or -8)	1 x 1 x 1
12.	Total area of tank, A_{tot} (sum the area from each face)	398 ft2

$$A_{tot} = 2 \times (9 \times 7) + 2 \times (9 \times 8.5) + 2 \times (7 \times 8.5) = 398$$

15.1.3 Solar Collector Area

Since the two separate tanks are at different temperatures, the area for the collectors for each tank is calculated separately. We choose the storage to collector ratio as 2.0 since we will be using good stratification technique, and will therefore need a smaller collector.

This is only a rough approximation to the collector area, but it is useful for the earliest stage of the design. Later stages of the design require f-Chart or other design methods for refining the collector area estimate. Examples of Worksheet C are as follows:

Domestic Hot Water

4.	Choose γ (between 1.25 and 2.0)	2.0 gal/day.ft^2	
5.	Number of days of storage, N (from Worksheet B-1)	1.0 days	
6.	Total tank volume V_{TG} (from Worksheet B-1)	600 gal	
7.	Divide line 6 by line 5 to find V_d		600 gal/day
8.	Divide line 7 by line 4 to find collector area, A_c		300 ft^2

Space Heating

4.	Choose γ (between 1.25 and 2.0)	2.0 gal/day ft^2	
5.	Number of days of storage, N (from Worksheet B-2)	1.0 days	
6.	Total tank volume V_{TG} (from Worksheet B-2)	4010 gal	
7.	Divide line 6 by line 5 to find V_d		4010 gal/day
5.	Divide line 7 by line 4 to find collector area, A_c		2005 ft^2

Since the collector is sized for an average design month, it will not be able to deliver 100% of the space heating needs during the coldest month . Trying to meet 100% of the space heating needs over a complete heating system with an active solar collector is seldom economically justified.

15.1.4 Temperature and Pressure Criteria

As shown in section 5.1, for space heating systems the following specification is typical.

1.	Storage temperature	110-160°F
2.	Storage liquid	water
3	Will it be pressurized	No

15.1.5 Stratification

The worst possible case Richardson number occurs at 110°F. This is not on the Table. We use a conservative number by choosing the entry for 104°F: $2.3 \times 10^{-4}°F^{-1}$. The desired Richardson number is arbitrary. Again for conservation we choose 2.5, which is 10 times the critical value.

Worksheet E is filled in as follows:

1.	Gravitational constant g	32.2 ft/sec^2
2.	Choose the worst-possible-case (lowest) volumetric thermal expansion coefficient β (if fluid other than water, obtain β from manufacturer) - for the temperature range of the tank	$2.3 \times 10^{-4}°F^{-1}$
3.	Transcribe value L for tank height from Worksheet D: item 3 for horizontal cylindrical tanks, item 4 for vertical cylindrical tanks, or item 9 for rectangular tanks	9.0 ft
4.	Maximum water temperature in tank from Worksheet A, item 2a, or item 4b, from Worksheet B-2	160°F
5.	Minimum water temperature in tank from Worksheet A, item 2b, or item 4a, from Worksheet B-2	110°F
6.	Subtract item 5 from item 4 for ΔT, temperature difference in tank	50°F

7. Value of desired Richardson number
   $R_i > 0.25$ — 2.5

8. Multiply together items 1,2,3,6 — 3.33

9. Divide item 8 by item 7 — 1.33

10. Take square root of item 9 to obtain maximum fluid inlet velocity, u — 1.15 ft/sec

11. Design methods to be used to promote thermal stratification

    a. Use inlet diffuser near bottom of tank

    b. Use outlet diffuser near top of tank

    c.

12. Control strategy used (see Section 5.2.2 Modifications for Solar Systems)

    a) Allow fluid from collector into tank only when the temperature of the fluid is 160°F or higher.

15.1.6 Type, Location, and Insulation of Tank

After reading section 5.2.1, it was decided that a concrete cast-in-place tank would be built. Concrete tanks are very inexpensive, corrosion resistant, and well understood. The tank will be 9 ft high - 7.0 ft wide - 8.5 ft long and located in a mechanical room (section 5.3.2).

Type of tank

Worksheet F is filled in as follows:

Type of tank (if pressurized, tank must be steel)	concrete
Length of warranty if steel:	yrs.
Choice of corrosion inhibitor	
If FRP: manufacturer's temperature limit	°F
If concrete: cast-in-place or precast	cast-in-place
If wooden or concrete or multi-component (indoor use only)	
choice of liner	spray-on butyl rubber coating
liner thickness	50 mil

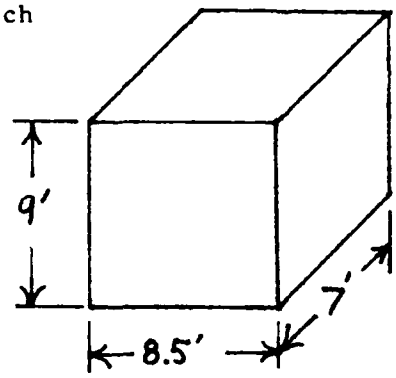
Location of tank

The load is an incipient problem here. Using Worksheet F below, we decided that a licensed structural engineer must participate in this phase of the design work.

1. Transcribe tank capacity V_{TG} from Worksheet B-2	4010 gal.
2. Density of water	8.33 lb/gal.

Since the tank is cast-in-place, a very rough calculation can be made of the tank weight as follows:

3. a) Draw a sketch



6" thick walls

b) Sum the volume of the four sides plus bottom

$$\begin{aligned} 2\ (9 \times 8.5 \times 0.5) &= 76.5 \\ 2\ (9 \times 7 \times 0.5) &= 63 \\ (7 \times 8.5 \times 0.5) &= \underline{29.8} \\ & \quad 169.3\ ft^3 \end{aligned}$$

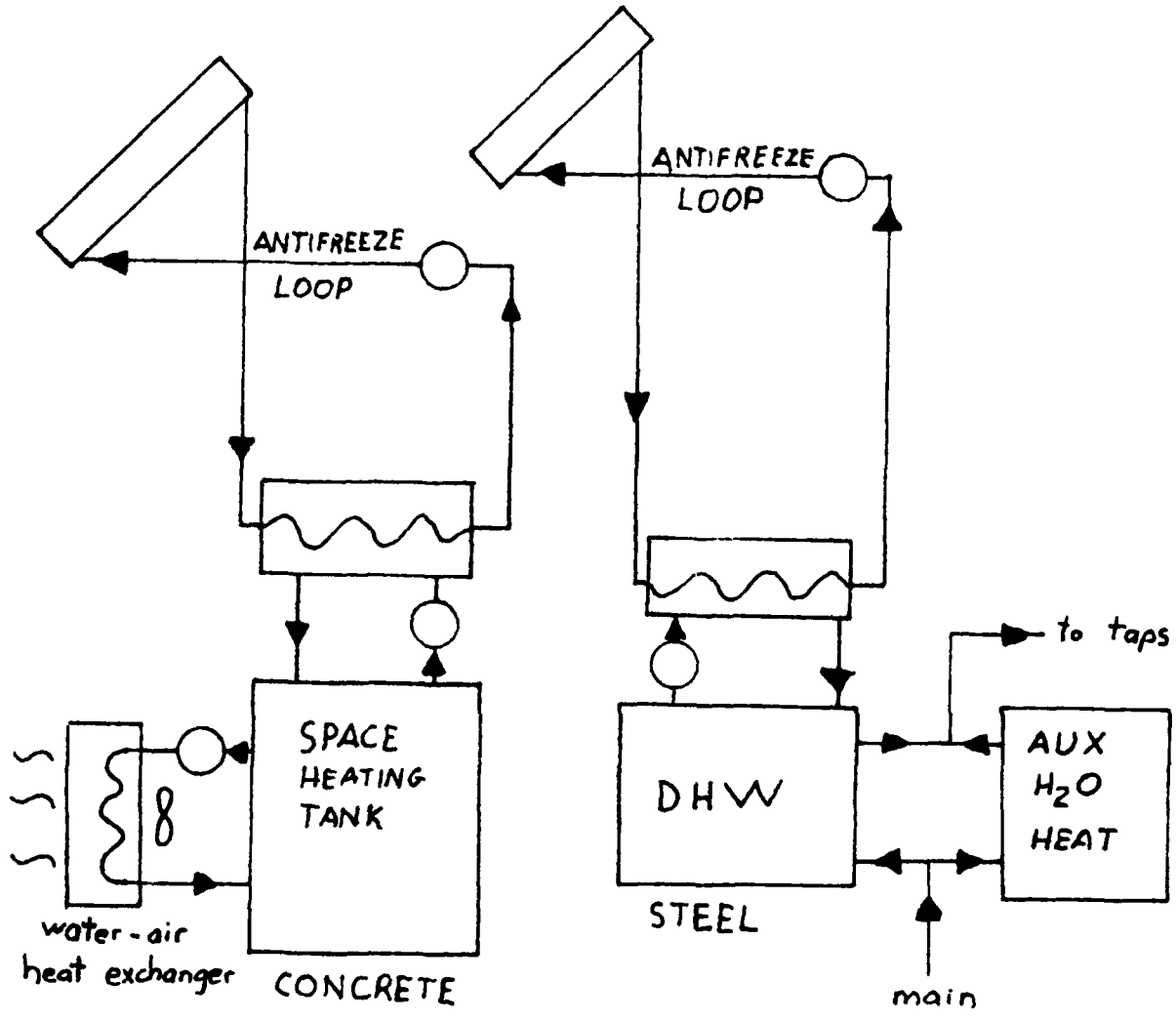
c) Approximate weight (density from Table 4-4)

$$80\ lb/ft^3 \times 169.3\ ft^3 = 13{,}500\ lb.$$

or

d)	Obtain weight of tank (m_T) from manufacturer	13,500 lb.
4.	Multiply items 1 and 2 to obtain weight of water	33,400 lb.
5.	Sum items 3 and 4 to obtain total weight	46,900 lb.
6.	Planned surface area between storage tank and substructure	$7 \times 8.5 = 59.5\ ft^2$
7.	Divide item 5 by item 6 to obtain load	$788\ lb/ft^2$

System Sketch



Use a 600 gal steel pressurized tank per ASME Pressure Vessel Code, Section VIII for domestic hot water.

Use a concrete tank for space heating. If water, as opposed to antifreeze were used in the space heating loop, the heat exchanger between the collector and the space heating tank would not be needed. The cost of antifreeze in this case overviews the slight loss in efficiency from using the heat exchanger.

Worksheet G summarizes the calculation and gives a good estimate of R which will be used in equation 14-11.

1.	Average temperature in storage unit T_{avg}	135°F	
2.	Ambient temperature surrounding the unit T_a	40°F	
3.	Subtract line 2 from line 1		95°F
4.	Transcribe tank volume V_{TGS} from Worksheet B-2	4010 gal	
5.	Relative dimensions of tank from Worksheet D		1 x 1 x 1
6.	Using lines 4 and 5 and Table 4-6, -7 or -8 determine insulation factor fQ/At	7.24 $\frac{Btu}{ft^2 \cdot hr}$	
7.	Divide line 3 by line 6 to determine R	13.12	$\frac{°F \cdot ft^2 \cdot hr}{Btu}$

We assume that pipes and fittings will be well insulated and not contribute enough heat loss to worry about. The only part of the tank that we can not insulate is the bottom. We assume that there is an additional 6 in. of concrete under the tank that conducts heat to the ground outside of the building. The ground temperature is probably not too far from 40°F, so the worksheet assumptions should be valid here.

Worksheet G-2: Table 1. Parallel Heat Paths

Heat Loss Path	Parallel Heat Loss Material	(1) r-value of Heat Loss Material $\frac{°F \cdot ft^2 \cdot hr}{Btu \cdot in}$	(2) Thicknesses of Heat Loss Material, s in	Individual Resistance r x s $\frac{°F \cdot ft^2 \cdot hr}{Btu}$	(3) R_{sur} $\frac{°F \cdot ft^2 \cdot hr}{Btu}$	R_{par} $R_{sur} + \Sigma rs$	(4) A_{par} ft^2	$\frac{A_{par}}{R_{par}}$	Comment
1 (a)	concrete	0.4	6.0	2.4					tank bottom
(b)	concrete	0.4	6.0	2.4					room floor
(c)									
1				Σrs= 4.8	0	4.8	59.5	12.4	
2									
2				Σrs=					
3									
3				Σrs=					
4									
4				Σrs=					
							Σ 59.5	Σ 12.4	

8. a. Transcribe A_{tot} from Worksheet D — 398 ft^2

b. Divide 8a by 7 to get A_{tot}/R — 30.5 $\frac{Btu}{°F.hr}$

c. Transcribe ΣA_{par} from Table i — 59.5 ft^2

d. Subtract item 8c from 8a — 338.5 ft^2

e. Transcribe $\Sigma \frac{A_{par}}{R_{par}}$ from Table i — 12.4 $\frac{Btu}{°F.hr}$

f. Subtract 9e from 9d — 17.9 $\frac{Btu}{°F.hr}$

g. Divide 9c by 9f to find R_{req} — 18.9 $\frac{°F.ft^2.hr}{Btu}$

For the four walls and the top, the only fixed resistance is the concrete wall and the wall to air surface resistance. There is no concrete on top of the tank. To make the calculation simpler, we pretend that there is. We will add an extra inch of insulation to the top afterwards.

Table ii. Fixed Series Resistance

fixed resistance	resistance	r-value of material -r-	thickness of material (in.) -s-	R_n -r.s-	R_{sur}
1	concrete wall	0.4	6	2.4	
2	wall to air				0.68
3			ΣR_n=2.4		ΣR_{sur}=0.68

9. Add $\Sigma R_n + \Sigma R_{sur}$ from Table ii to find R_{ser} — 3.08 $\frac{°F.ft^2.hr}{Btu}$

Look at Table 4-4

10. (a) Decide what material will be used for insulation — fiberglass batt on exterior of tank

    (b) Transcribe its r-value — 3.15 $\frac{°F.ft^2.hr}{Btu\ in}$

Since there are no studs, etc. we skip Table iii...

11. Divide ΣA_n by $\Sigma A/r$ from Table 14-3 to find r* — 3.15 $\frac{°F.ft^2.hr}{Btu.in}$

    Finally calculate the thickness needed of the insulation selected in 11(a) from $s = \frac{R_{req} - R_{ser}}{r^*}$

12. (a) Transcribe 8g - R_{req} — 18.9
    (b) Transcribe 9 - R_{ser} — 3.08
    (c) Subtract 12b from 12a — 15.8
    (d) Divide 12c by 11 — 5.02 in

The side wall insulation is five in., top insulation six in. However, because of the possibility of leakage, compression, etc. we add three extra inches to be on the safe side.

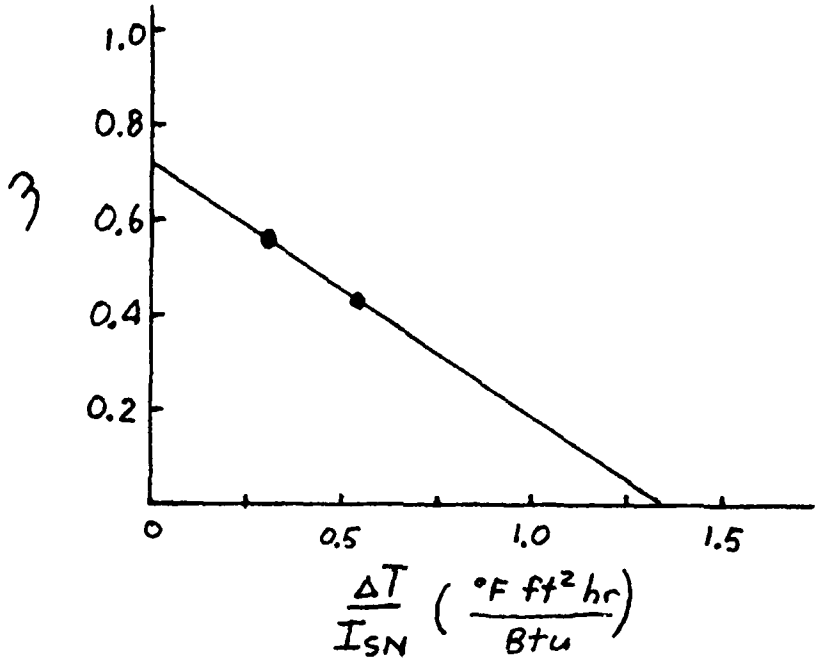
15.1.7 Pump Characteristics

We will only work out the details of the pump between the concrete tank and the tank/collector heat exchanger.

Flowrate

1. Heating rate Q'

Refer to the solar collector manufacturer's information-specifically a graph that looks like this:



This is for a well made, double glazed, flat plate collector with a selective surface. By finding the value of $\Delta T/I_{sn}$, the collector efficiency η can be determined, and from the equation $Q' = I_{sn}\ A_c\ h$ the heating rate can be found.

Since there is a 10°F ΔT across the heat exchanger, the maximum collector temperature must be 170°F. Likewise the minimum collector temperature is 120°F. The average is 145°F. On a relatively cold day the ambient temperature might be 10°F.

$$\Delta T = 195°F\ (T_c) - 10°F\ (T_a) = 135°F.$$

I_{sn} = insolation solar noon ≈ 250 Btu/hr ft^2

$$\frac{\Delta T}{I_{sn}} = 0.54 \text{ and } \eta = 0.43$$

$$Q' = 250 \frac{\text{Btu}}{\text{hr ft}^2} \times 2005 \text{ ft}^2 \times 0.43 = 215{,}500 \frac{\text{Btu}}{\text{hr}}$$

2. The fluid on the tank side of the heat exchanger is water: ρ = 8.33 lb/gal.

3. C_p = 1 Btu/lb°F

4. The temperature change of the heat transfer fluid is the difference between the maximum and minimum tank temperatures

   ΔT = 160 - 110 = 50°F

5. $8.33 \frac{\text{lb}}{\text{gal}} \times 1 \frac{\text{Btu}}{\text{hr°F}} \times 50°F =$ 416 Btu/gal

6. $\frac{215{,}500 \frac{\text{Btu}}{\text{hr}}}{416 \text{ Btu/gal}} =$ 518 gal/hr

7. $q = \frac{518}{60} =$ 8.6 gpm

If the ambient temperature is higher:

$$\Delta T = 145 - 70 = 75°F$$

$$\frac{\Delta T}{I_{sn}} = \frac{75}{250} = 0.3 \text{ and } \eta = 0.56$$

1. $Q' = 250 \times 2005 \times 0.56 = 280{,}700$ Btu/hr

6. $\frac{280{,}700}{416} = 675$ gal/hr

7. $q = \frac{657}{60} = 11.2$ gpm

The higher of the two pump rates (11.2 gpm) should be used. In general it is only necessary to calculate the pump rate using a high ambient temperature.

If the tank was not stratified, we would use a flow rate of 0.022 gpm per ft^2 of collector. Item 7 would be:

$(0.022)\ (2005) = 44.1$ gpm

Pumping Head

8. Height difference between collector top and tank liquid level Z (open-drop drain down system only).

   Note that the heat storage system uses an antifreeze loop — Z = 0 ft.

9. Pipe friction head loss H_f (summary of calculations per ASHRAE Handbook of Fundamentals, Chapter 26) — 10 ft.

10. Heat exchanger heat loss H_x (from manufacturer) — 2 ft.

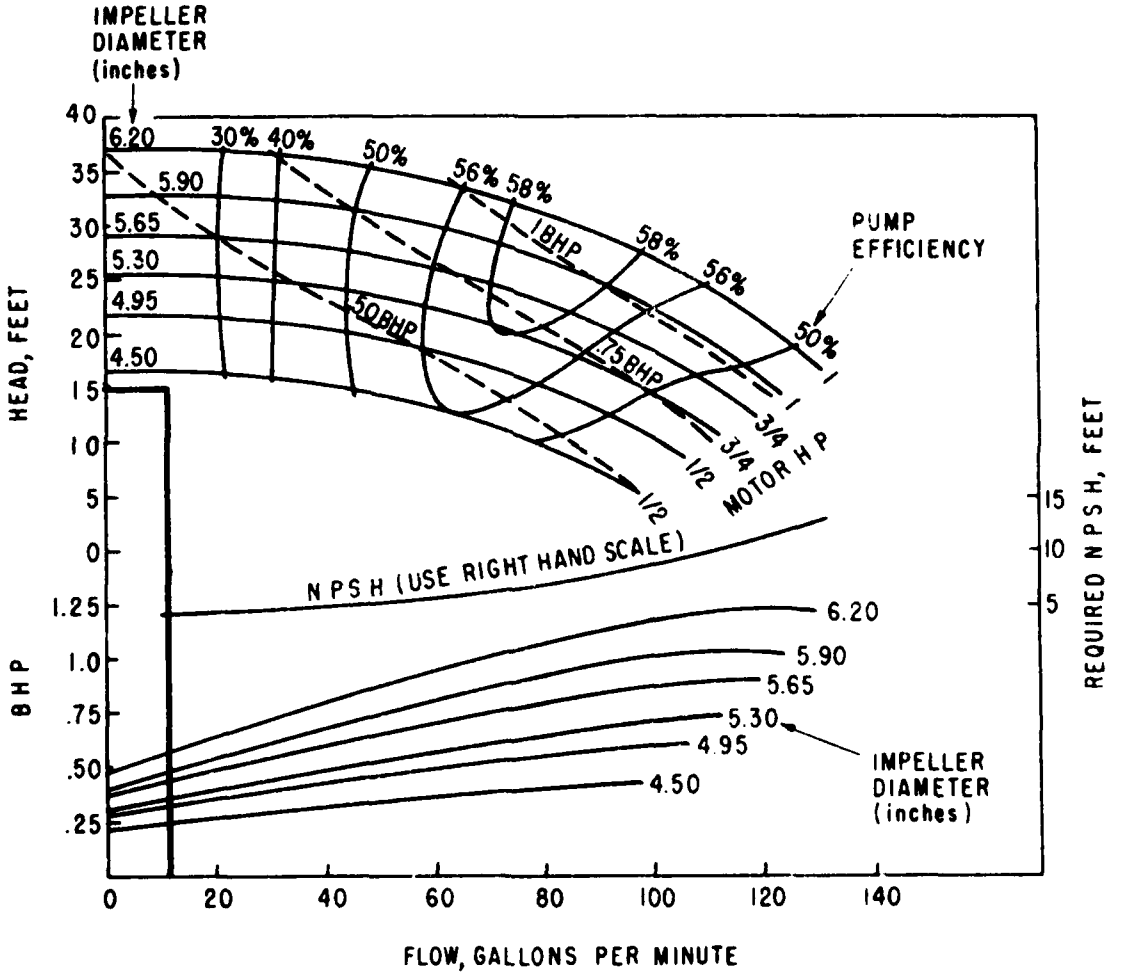
11. Collector head loss (from manufacturer) — 3 ft.

12. Add items 8-11 to determine sum of head losses around the loop H_p which is also minimum tank pressure — 15 ft.

Pump Parameters

Although as the sketch illustrates, in section 15.1.6 there are actually four pumps - we will discuss only the pump in the space heating collector loop.

1. Obtain performance curves from one or more pump manufacturers and plot q against H_p on performance curves such as the upper curves on Figure 5-7 to find a pump that exceeds the pumping head requirement at the required flowrate.



2. Plot Z on upper curve of Figure 5-7; make sure that $Z < H_p$ when $q = 0$. Since $Z = 0$, this is no problem.

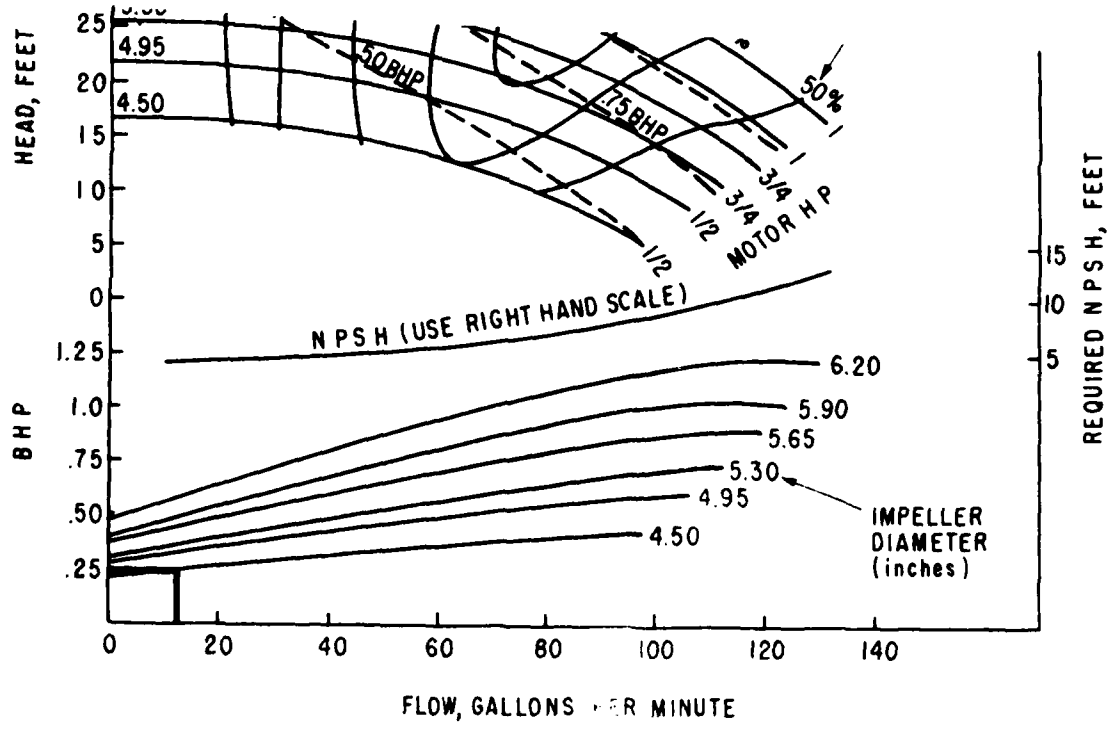
3.

i.	impeller diameter d	4.50 inches
ii.	pump motor	1/2 HP
iii.	pump manufacturer	xyz
iv.	pump model	ABC

Motor

Plot d and q on the lower curve of Figure 5-7 to determine size of motor required by the pump.

Motor size 0.25 BHP

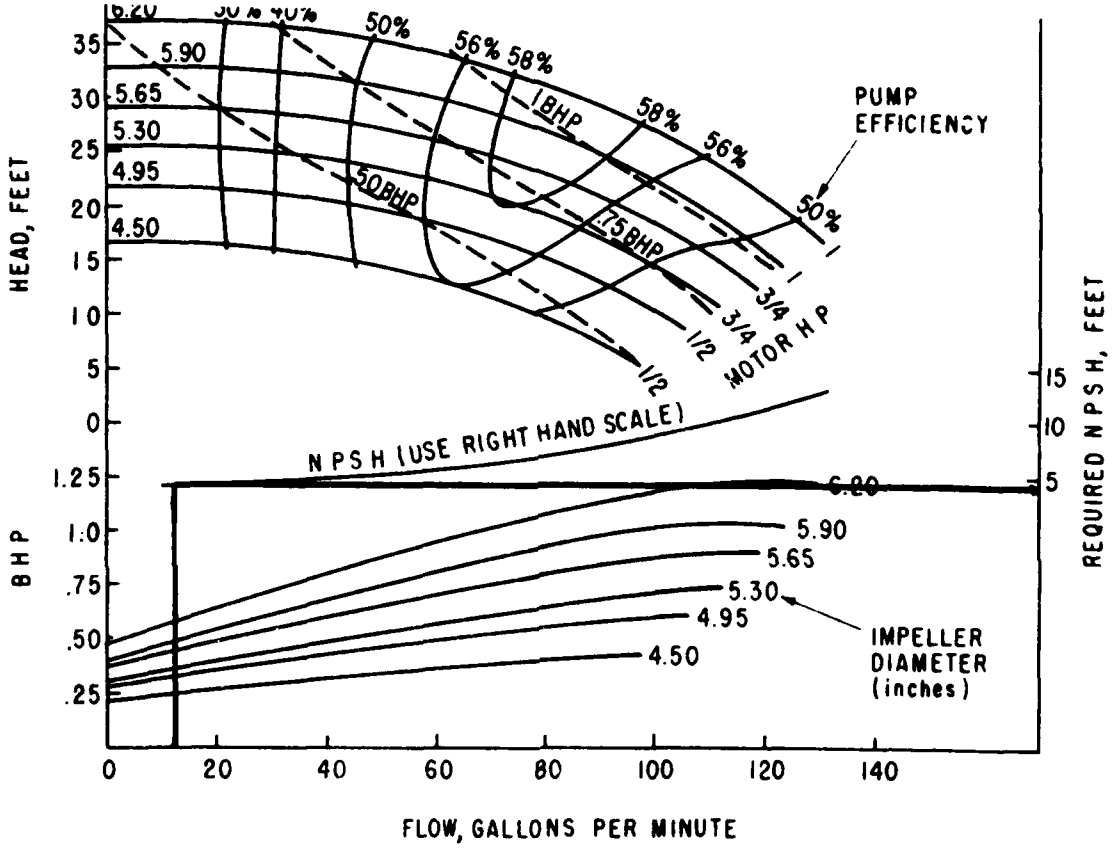


Net Positive Suction Head (NPSH)

NPSH is determined in Worksheet H-2.

6. Plot q on the NPSH curve to determine the pump's NPSH requirement ($NPSH_R$)

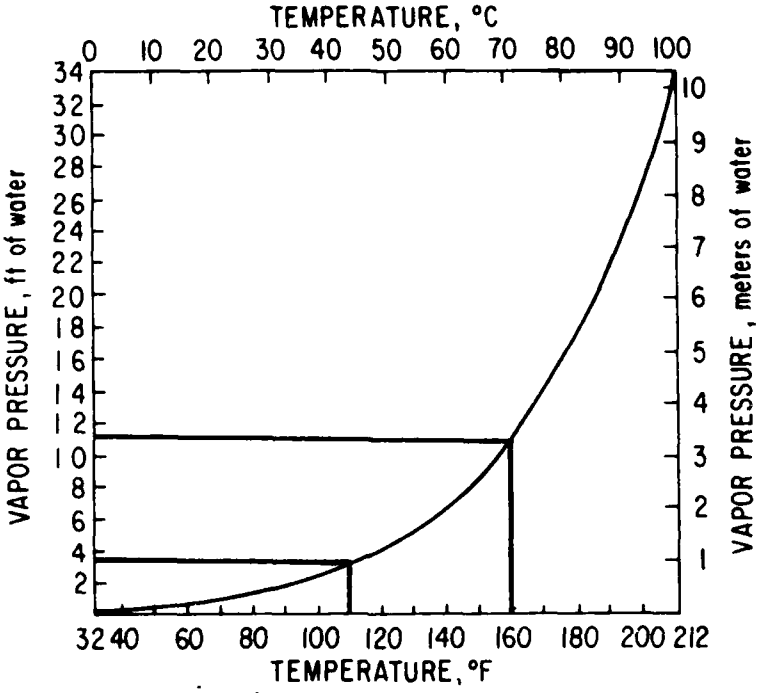
$NPSH_R$ 10 ft



Determine the available NPSH from equation

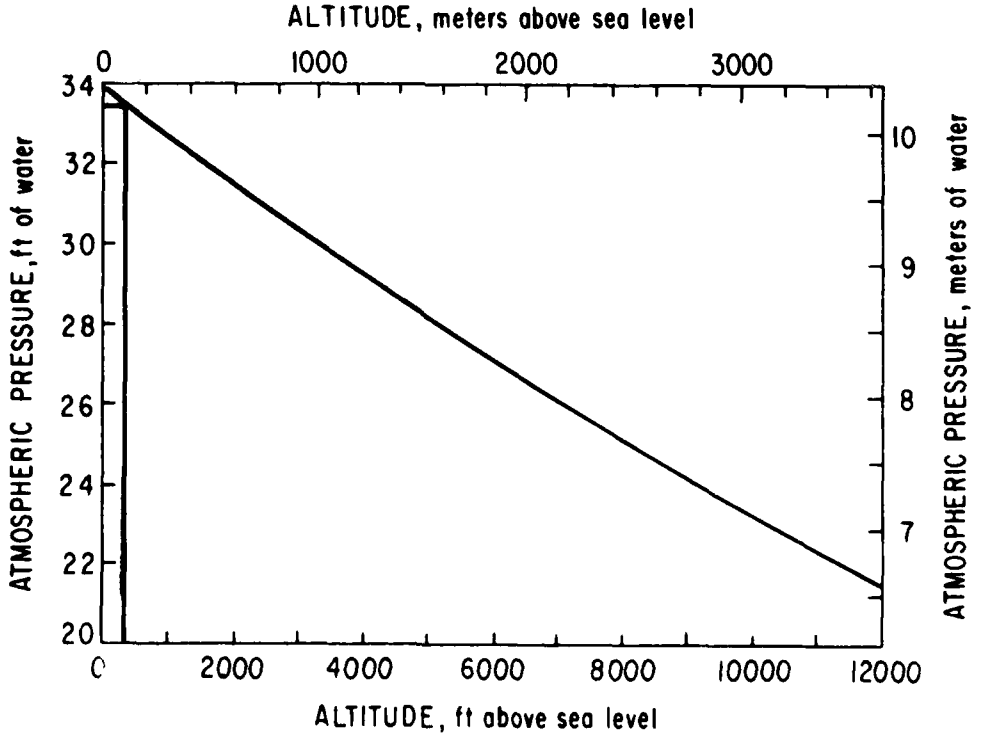
$$NPSH = Z - H_f - H_v + H_a + H_p \qquad (5\text{-}4)$$

7.	Height difference Z (from Worksheet H-1, item 8)	0 ft.
8.	Pipe friction head loss H_f (from Worksheet H-1, item 9)	10 ft.
9.	Liquid vapor pressure head H_v (from Figure 5-8).	11 ft.
10	Atmospheric pressure head H_a (from Figure 5-9)	33.8 ft.



H_v = 3.5 ft to 11 ft

Choose the larger of the two.



At the altitude of the naval base, 250 ft. above sea level, atmospheric pressure = 33.8 ft.

11. Minimum tank pressure H_p (from item 12, Worksheet H-1) 15 ft.

12. Add items 7, 10, and 11

0 ft + 33.8 ft + 15 ft = 48.8 ft.

13. Subtract items 8 and 9 from item 12 to determine available NPSH for system

48.8 ft - 10 ft - 11 ft = 27.8 ft

14.. Subract $NPSH_R$ from item 13

27.8 ft - 4 ft = 23.4 ft

This is a very positive number -- cavitation won't occur.

15.1.8 Heat Exchangers

As the sketch indicates in section 15.1.6 there are three different heat exchangers.

Only the water-antifreeze exchanger adjacent to the concrete tank will be discussed here. This is a shell and tube unit.

Capacity

The water delivered to the concrete tank should be 160°F. We have allowed a 10°F ΔT across the walls of the heat exchanger. The water from the heat exchanger flows directly into the tank.

The heat exchanger shall have the capacity to heat 11 gpm of water (in the tubes) from 110°F when supplied with 20 gpm of propylene glycol solution (50 percent) at 170°F (at the inlet end of the shell). Fouling factors shall be 0.001 hour-square foot-°F per Btu for both tube side and shell side.

The 20 gpm was from the collector manufacturer.

Pressure Drops and Velocities

[2 ft = 0.87 psi]

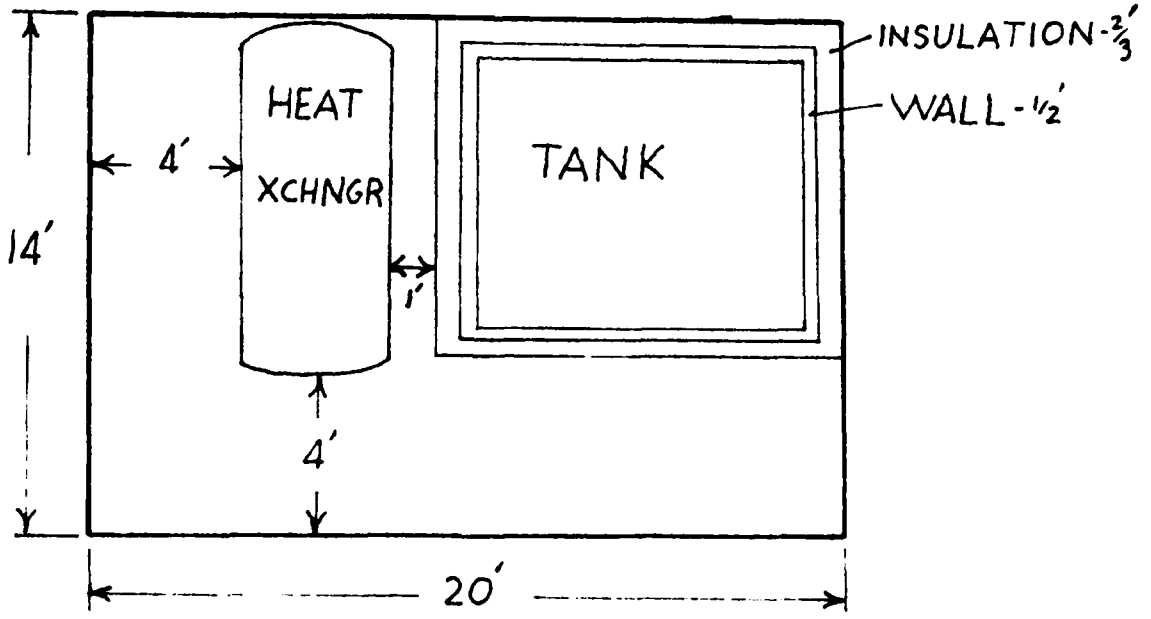
The maximum pressure drop on the shell side shall be 0.9 psi with the fluid, flow rate, and temperature specified.

The maximum pressure drop on the tube side shall be 0.9 psi with the fluid, flow rate, and temperature specified.

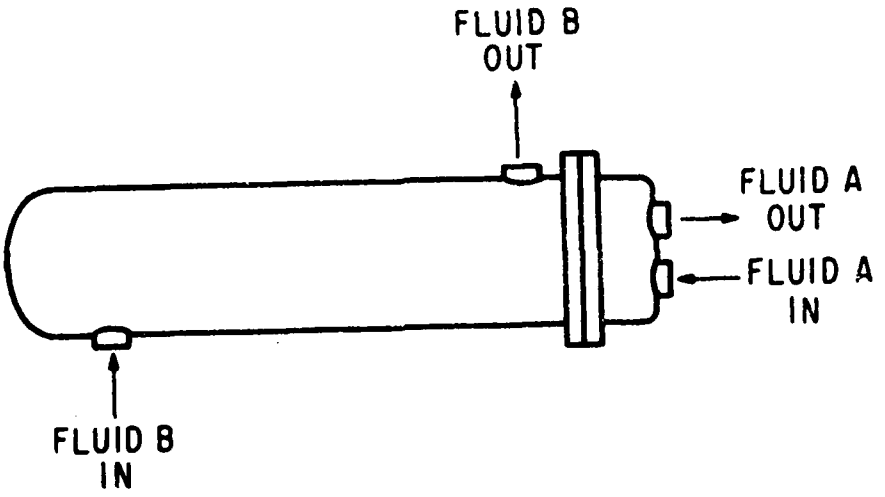
Dimensional Requirements

We want the heat exchanger to fit lengthways beside the storage tank. We want to maintain four feet on two sides and the top for access to pipes etc. We then have eight ft. vertically, and 14 - 4 = 10 ft. by 20 - 4 -8.5 - 2(1/2) - 2(2/3) = 5.2 ft horizontally. In addition, we should probably leave one ft. between the tank wall insulation and the heat exchanger. We decide to lay the heat exchanger on its side with the cylinder axis horizontal and parallel to the tank wall.

maximum heat exchanger length	10 ft.
maximum heat exchanger diameter	4 ft.
location of inlet and outlet	near to front



Worksheet I: Heat Exchanger Specification Form



Look up Properties of Heat Transfer Fluids from Appendix E to help fill in this table.

	FLUID A	FLUID B
Description	water	50% propylene glycol
Heat Capacity (Btu/lb·°F) (J/kg·°C)	1.00 Btu/lb°F	0.9 Btu/lb°F
Density (lb/ft^3) (kg/m^3)	60 lb/ft3	62 lb/ft3
Thermal Conductivity (Btu/hr·ft·°F) (W/m·°C)	0.38 Btu/hr ft °F	0.23 Btu/hr ft °F
Viscosity (lb/ft·hr) (Pa·sec)	0.9 - 1.5 lb/ft"hr	3.5 lb/ft"hr
Flow Rate (gpm) (liters/sec)	11 gpm	20 gpm
Inlet Pressure (psi) (MPa)	atmospheric 15 psi	15 ft = 7 psi 35 psi

Rate of Heat Transfer (Btu/hr) (W) 280,000 Btu/hr

Approach Temperature Difference (°F) (°C)
(Temperature of heating fluid inlet minus temperature of heated fluid outlet) 170 - 110 = 60°F

15.2 Rock Bed Heating System

15.2.1 Design Situation

In this example, we are going to install an air-based solar space heating system in a new house designed for a Navy officer and his family in a coastal climate. We have already determined the following:

- We need 400 square feet (37 square meters) of solar collector.
- The collector manufacturer recommends an air flow (q) rate of 2 cubic feet per minute per square foot (0.01 cubic meters per second per square meter) of collector area, for a total air flow rate of 800 cubic feet per minute (0.38 cubic meters per second).
- We are going to install the rock bed in the basement of the house, which has a floor-to-joist height of 7.5 feet (229 centimeters).
- We can pour a recessed floor for the rock bed no lower than 1 foot (30 centimeters) below the nominal basement floor level in order for the rock bed drain to go to the sump well.
- We will line the wooden rock bin with sheet metal for fire protection.

Analysis of Storage Fundamentals (condensed from information above)

1.	Energy source	collector
2.	Type of solid sensible heat storage desired	rock bed
3.	End use of stored energy	space heating

15.2.2 Rockbed Volume

Since we are using a solar collector, we use Worksheet B.

Collector Specifications

1.	Size of solar collector, A_c	400 ft2
2.	Type of solar collector	flat plate
3.		
4.	Necessary volumetric air flow rate, q	800 ft^3/min

Volume Limits

Using the rule of thumb (1) for the volume of the rock bed, we find that for 400 square feet of collector the minimum rock volume is:

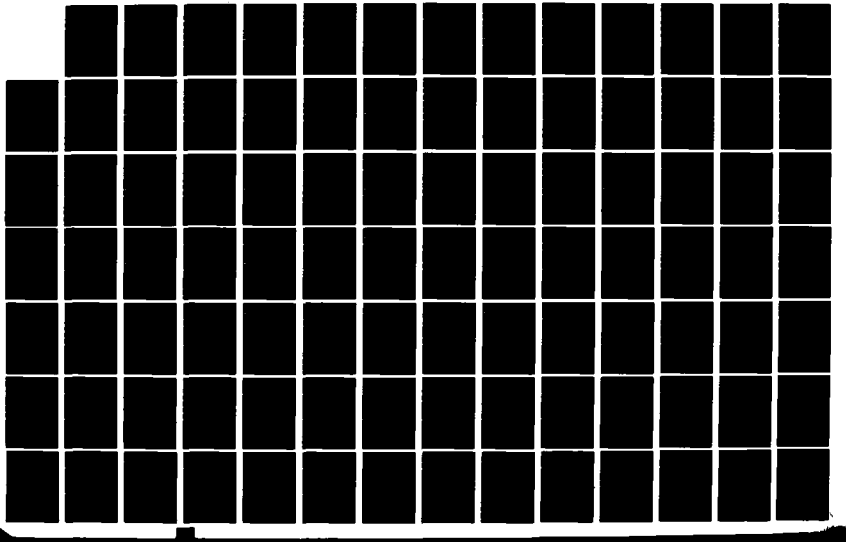
5. V_{min-RB} $400 \times 0.5 = 200\ ft^3\ (5.66\ m^3)$

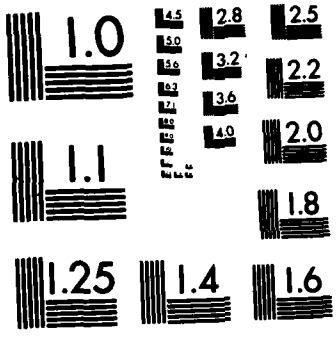
6. V_{max-RB} $400 \times 0.75 = 300\ ft^3\ (8.50\ m^3)$.

Therefore, any volume between 200 and 300 cubic feet will satisfy the rule of thumb requirement.

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15.2.3 Determining Size and Shape

- Rock bed shape — vertical

Depth

The total height of the rock bed, including insulation and plena, is limited to 102 inches (259 centimeters). This was calculated by adding 12 inches for the recessed floor to the 7.5 ft. (90 inches) floor-to-joist height. Because we will need room to put on the cover, we will allow 5 inches (3 centimeters) of working space limiting the height to 97 inches (238.8 centimeters).

We plan to make the cover of 2 x 6 studs (to allow room for insulation) with 1/4-inch (0.65-centimeter) plywood sheathing, so the cover will be 6 inches (15.2 centimeters) thick.

Because the bond beam blocks we will use to support the rocks come in a standard 7-5/8-inch (19.3-centimeter) size, we will assume that the bottom plenum, which will contain the bond beam blocks, will be 7-5/8 inches high. Alternatively, we could use size 8J2 open-web steel joist in place of the bond beam blocks. The top plenum will be 5-1/8 inches (13.0 centimeters) high.

We plan to place the bond beam blocks on a piece of 4-inch (10.2 centimeters) rigid fiberglass insulation.

The total height of all of these components is 22.75 inches (57.8 centimeters), which leaves 97 inches - 22.75 inches = 74.25 inches (188.6 centimeters) for the rocks. For convenience we will assume that we will use 72 inches, or 6 feet (182.9 centimeters), of rocks.

Transcribing all this into the format provided in Worksheet C-1 gives the following:

1.	Depth limit D_L for rockbed -- floor to ceiling height, include depth of recessed floor.	97 in.
2.	Minimum of two inches for cover installation (accounted for in 1)	____ in.
3.	Thickness of the cover, D_{cover}	6 in.
4.	Height of bottom plenum, D_{pb}	7 5/8 in.
5.	Height of top plenum, D_{pt}	5 1/8 in.
6.	Thickness of rigid insulation, D_{bi}	4 in.

7. Add items 2-6. 22.75 in.

8. a) Subtract item 7 from item 1 to obtain rock depth, D_r 74.25 in.
   b) Divide item 8 by 12 to obtain rock depth D_R and round to convenient number.

This leaves approximately 7 inches for cover installation.

Cross-sectional Area

The cross-sectional area of the rock bed is its volume divided by its height. Using the volume limits from 15.2.2.2 and the height from Worksheet C-1 we find that the minimum cross-sectional area is:

$$\frac{200 \text{ ft}^3}{6 \text{ ft}} = 33.3 \text{ ft}^2 \ (3.09 \text{ m}^2)$$

and the maximum cross-sectional area is:

$$\frac{300 \text{ ft}^3}{6 \text{ ft}} = 50 \text{ ft}^2 \ (4.65 \text{ m}^2).$$

To get one of the dimensions (the length as defined in Figure 14-2), we will use the rule of thumb (3) regarding plenum sizing. The rule of thumb for the bottom plenum, which will contain bond beam blocks, says that the plenum area must equal at least 12 percent of the cross-sectional area. The minimum plenum area then, is:

Minimum Bottom Plenum Area = 0.12 x Cross-sectional Area,

which can be written as:

Bottom Plenum Height x Width = 0.12 x Maximum Length x Width.

Since "Width" is common to both sides of the equation, it can be factored out, leaving:

Bottom Plenum Height = 0.12 x Maximum Length.

Therefore, since in Worksheet C-1 we have already specified a bottom plenum height of 7-5/8, or 7.62, inches (19.4 centimeters), we can calculate thus:

$$\text{Maximum Length} = \frac{7.62}{0.12}$$

$$= 63.5 \text{ in. } (161.3 \text{ cm})$$

We will therefore choose a length of 63 inches (160.0 centimeters), which is less than the maximum length permitted by the rule of thumb. The top plenum is correctly sized, since we chose its height to be about two-thirds (8 percent/12 percent) of the size of the bottom plenum to satisfy the rule of thumb.

Anticipating that the walls will be made from 2 x 6 studs with 1-inch (2.5-centimeter) plywood inside and half-inch (1.3-centimeter) gypsum board outside, we choose the other width dimension to be 82 inches (208.3 centimeters) to simplify construction.

The cross-sectional area is then:

$$63 \text{ inches} \times 82 \text{ inches} \times \frac{1 \text{ sq. ft.}}{144 \text{ sq. in.}} = 35.9 \text{ ft}^2 \ (3.33 \text{ m}^2)$$

which is within the limits of 33.3 to 50.0 square feet as calculated above. Transcribing this information into Worksheet C-2 gives the following:

1.	Minimum volume V_{min} (from Worksheet B)	200 ft^3	
2.	Maximum volume V_{max} (from Worksheet B)	300 ft^3	
3.	Rock bed depth D_R (from item 8 of C-1)	6 ft.	
4.	Divide item 1 by item 3 to obtain minimum cross-sectional area A_{min}		33.3 ft^2
5.	Divide item 2 by item 3 to obtain maximum cross-sectional area A_{max}		50 ft^2
6.	Add item 4 to item 5 and multiply the result by 72 to find A_{avg}	5998 in^2	
7.	Bottom plenum height D_{pb} (from Worksheet C-1, item 4)	7.62 in.	
8.	Divide item 7 by 0.12 to obtain the rock bed length L		63 in.
9.	Divide item 6 by item 8 to get rock bed width W		95.2 in.
10.	Choose convenient W value close to the value in item 9.		82 in.
11.	Multiply item 8 by item 10 and divide by 144 to obtain A_x		35.9 ft^2

Summary of Dimensions

Note that these values refer to the rock bed--the actual rocks --and not the surrounding rock bin which will be larger. The completed information from Worksheet C-2:

12.	Rock bed height D_R	6	ft
13.	Rock bed length	5.25	ft
14.	Rock bed width	6.83	ft
15.	Rock bed cross-sectional area, A_x	35.9	ft^2
16.	Multiply item 12 by item 15 to find volume V_{RB}	215	ft^3
17.	Examine Table 4-9 and lines 1, 2, and 3 to determine relative dimensions of the rock bed	1 x 1 x 1	

15.2.4 Rock Size

- Type of rocks available — rounded pebbles - all sizes

Face Velocity

The face velocity is the velocity of the air immediately before it enters the rocks. It is calculated by dividing the air flow rate by the cross-sectional area of the rock bin. In this case the face velocity will be item 3 of Worksheet D.

$$u_f = \frac{800 \text{ ft3/min}}{35.9 \text{ ft}^2} = 22.3 \text{ ft/min (0.11 m/s)}.$$

We will later use this number to help determine the rock size and the pressure drop in the rock bed.

Pressure Gradient Limits

The pressure gradient is the pressure drop per unit length through the rock bin. It is calculated by dividing the total pressure drop by the depth of the rocks. Using rule of thumb (2) in section 14.2.1, we find that the minimum pressure gradient is:

8. $\Delta P_{RB\text{-}min} = \frac{0.15 \text{ inches of water}}{6 \text{ ft.}} = 0.025$ inches of water/ft (20.4 Pa/m)

and the maximum pressure gradient is:

7. $\Delta P_{RB\text{-}max} = \frac{0.30 \text{ inches of water}}{6 \text{ ft.}} = 0.050$ inches of water/ft (40.8 Pa/m).

These limits will now be used to determine the rock size and the pressure drop in the rock bed.

Rock Bed Performance Map

We will use the rock bed performance map in Figure 14-2 and the values u_f and ΔP_{RB-min} and ΔP_{RB-max} to determine the rock size suitable for our system.

First, as illustrated on the next page, we draw a vertical line at the face velocity of 22.3 feet per minute (0.11 meters per second). Next, we draw two horizontal lines at the pressure gradient limits of 0.025 and 0.050 inches of water per foot (20.4 to 40.8 pascals per meter).

Any of the curved lines corresponding to various rock sizes that cross the face velocity line between the pressure gradient limit lines is a suitable choice. In this case, only the curve that corresponds to 3/4-inch-diameter (1.9 centimeter diameter) rock is indicated, so that is the rock size we will use. We will accept washed rocks whose average diameter varies from 3/4 to 1-1/2 times the nominal diameter, or from 5/8 inch to 1-1/8 inches (1.6 to 2.8 centimeters).

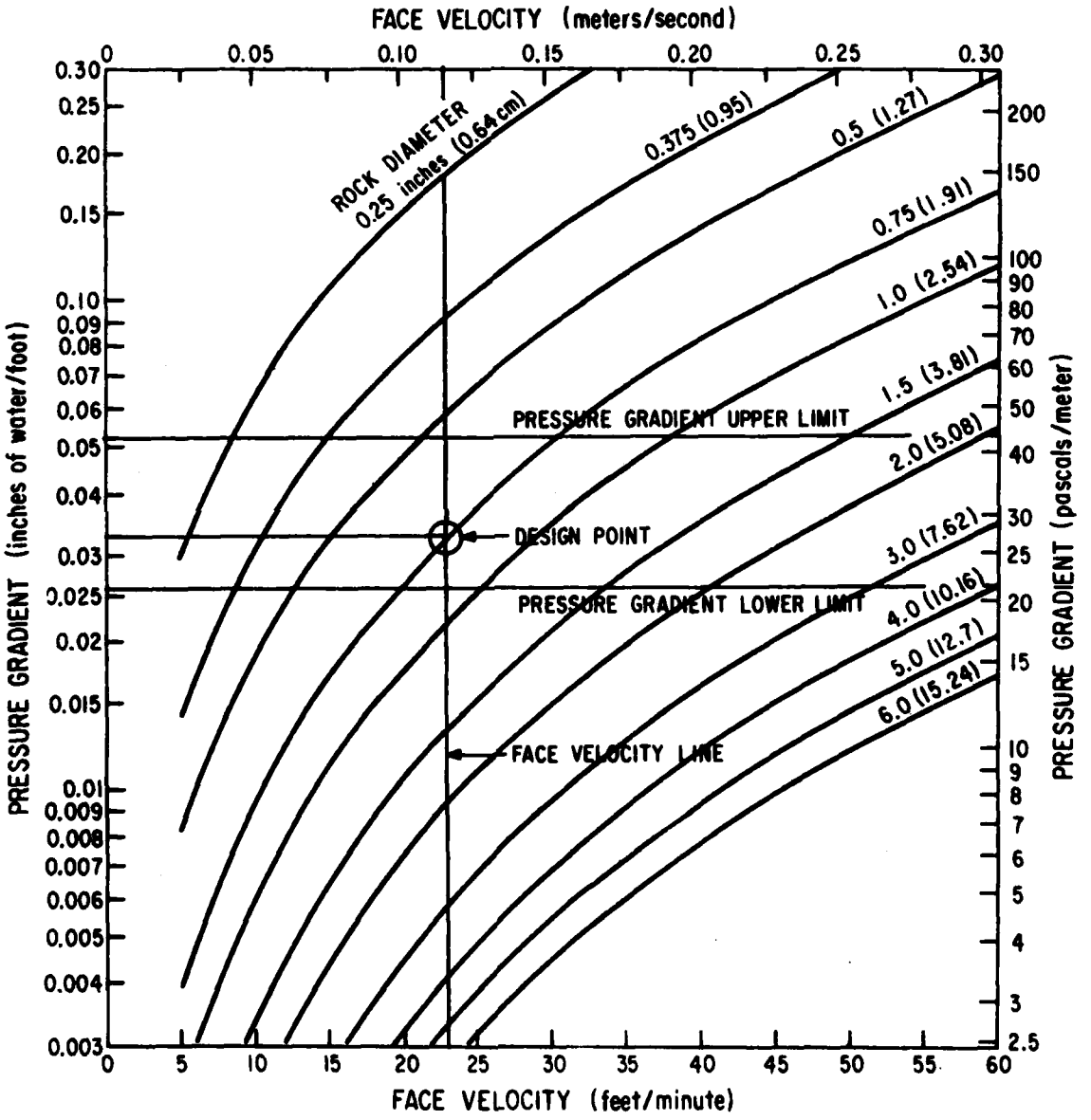
To check rule of thumb (4) in Section 14.2.1, we note that 20 x 3/4 inch = 15 inches (38.1 centimeters), which is much less than the rockbed depth. That means thermal stratification in the rock bed will be good.

The above information is used to fill out the rest of Worksheet D.

9.	Nominal rock size	0.75 in.
10.	Smallest rock size: (0.75 x item 1)	0.625 in.
11.	Largest rock size: (1.5 x item 1)	1.125 in.
12.	Rock size range: (item 10 to item 11)	0.625 to 1.125 in.

Stratification

13.	Rock bed depth, D_r	72 in.
14.	If item 9 < 4", multiply item 9 by 20	15 in.
15.	Item 14 should be less than item 13 (yes/no).	yes



Determination of rockbed design point from face velocity and pressure gradient limits.

15.2.5 System Pressure Drop

Rockbed pressure drops are relatively easy to ascertain using the calculated pressure gradient limits and referring to Figure 14-2 (see illustration on previous page). The design point of our rockbed is the point on the rockbed performance map where the line representing the pressure gradient crosses the curve representing the rock size. In our case, the design point is at a pressure gradient of 0.032 inches of water per foot (26.1 pascals per meter). Multiplying this value by the rock bed depth gives a pressure drop, P_{RB}, of:

$$0.032 \times 6 = 0.19 \text{ inches of water (47.3 Pa).}$$

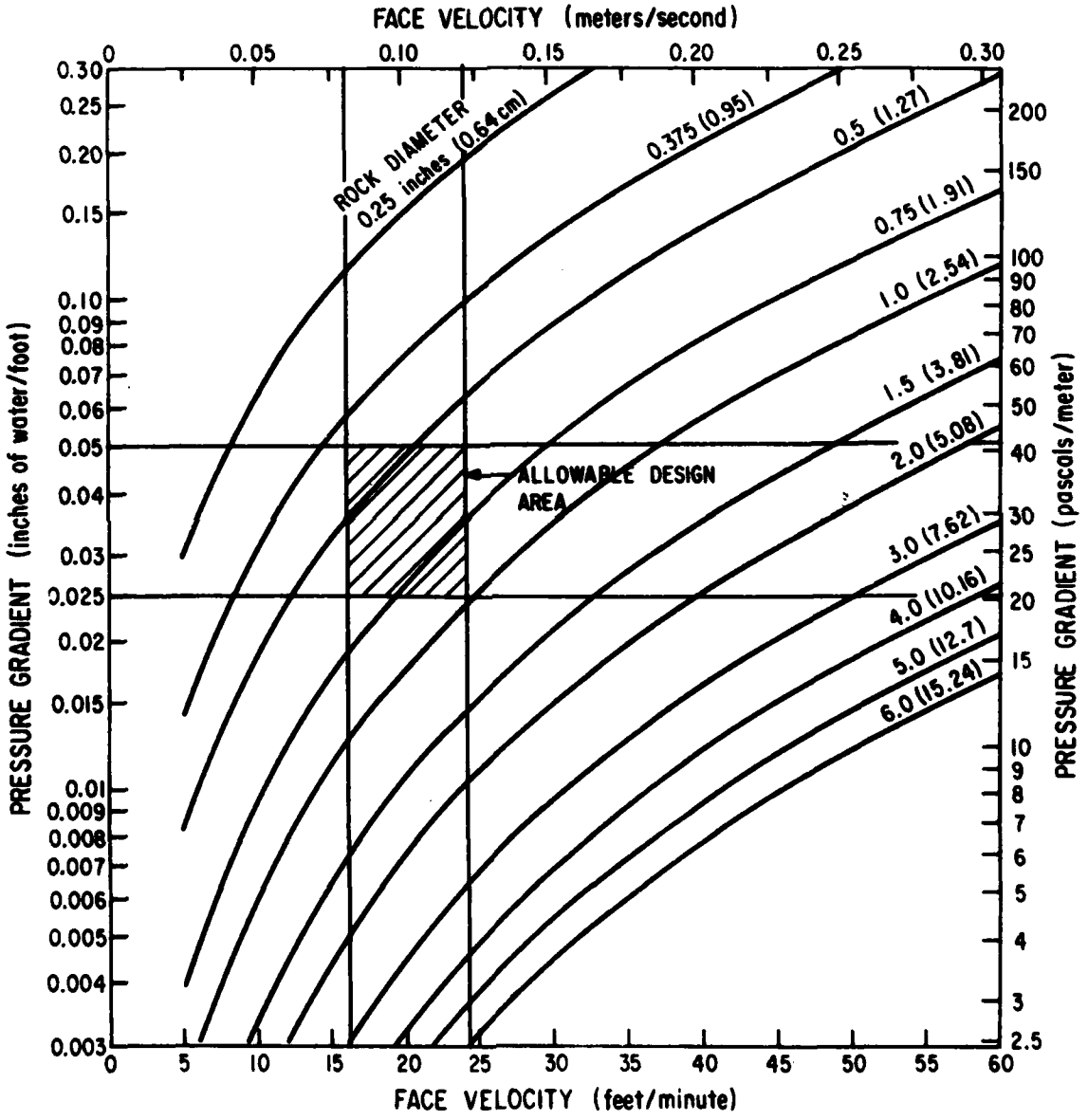
which corresponds to item 6 of Worksheet E.

Note that the design point we have selected is not unique. If we calculate the maximum and minimum values of the face velocity by using the minimum and maximum values of the pressure gradient in Worksheet D (items 7 and 8), we find that the face velocity can be anywhere between 16.0 and 24.0 feet per minute (0.081 to 0.122 meters per second). Lines corresponding to these values as well as the pressure gradient limits are shown in Figure 14-2 redrawn on the next page. Any point within the allowable design area is a valid design point in that it falls within the limits set by the rule of thumb for volume and pressure drop corresponding to the rock bed depth we initially chose. We could change the allowable design area by using a different rock bed depth.

Total Pressure Drop

A typical example from Worksheet E follows.

6.	Rock bed pressure drop P_{RB} (multiply items 4 and 5 above)	0.19 in.H_2O	
7.	Collector pressure drop P_c (from manufacturer)	0.42 in.H_2O	
8.	Filter pressure drop P_{fil} (from manufacturer)	0.15 in.H_2O	
9.	Damper pressure drop P_{da} (from manufacturer)	0.16 in.H_2O	
10.	Duct pressure drop P_{du} (calculated using ASHRAE Handbook of Fundamentals, Chapter 25).	0.18 in.H_2O	
11.	Sum items 6-10 to find P_T		1.10 in.H_2O



Allowable Design Area on the Rock Bed Performance Map

15.2.6 Type and Location of Unit

The objectives the storage system are to meet will determine the type, location, and insulation of the rock bed. Rock beds can be used over a wide temperature range.

Type of Bin

1. Type of bin

   Wooden lined with sheet metal

2. Wall materials and their thicknesses

	sheet metal	1/4 inch
	plywood	1 inch
	rigid insulation	
or	pine studs	5 1/2 inches
	gypsum board	1/2 inch

Location of Unit

The loading on the floor will be:

$$L = \frac{V_{RB} \cdot \rho \cdot f_r}{A_x} \qquad (15\text{-}1)$$

3. Total volume of rocks, V_{RB} — 215 ft^3

4. Rock density, ρ — 167 lb/ft^3

5. Rockmass fraction -- assume a 30% void fraction — 0.7

6. Multiply items 3, 4, 5 together — 25,234 lb.

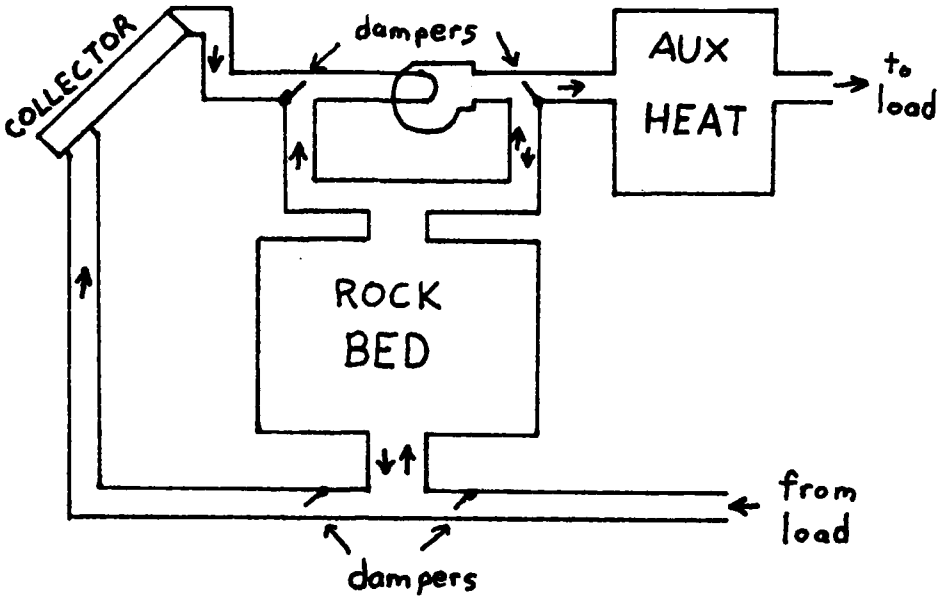
7. Cross-sectional area, A_x — 35.9 ft^2

8. Divide item 6 by item 7 to obtain the loading on the floor, L — 700 lb/ft^2 = 5 lb/in^2

Since most basement floors can support only 150 to 400 pounds per square foot (734 to 1,957 kilograms per square meter), we must have a licensed structural engineer determine the load capacity and design reinforcements if necessary. Failure to do this may lead to a building code violation and structural damage to the building.

Schematics

Draw your system with components.



15.2.7 Insulation

Work through Section 14.1.6, Worksheet G of Section 14.1, and illustrations in Chapter 4 as follows:

1.	Average temperature in storage unit (T_{avg})	115°F
2.	Ambient temperature surrounding the unit (T_a)	68°F
3.	Subtract line 2 from line 1	47°F
4.	Transcribe volume of rockbed	215 ft^3
5.	Relative dimensions of rockbed	1 x 1 x 1
6.	Using lines [illegible] and 5 [illegible], Table 4-9, determine insulation [illegible] r At	1.60 Btu/ft^2.hr

7. Divide line 3 by line 6 to determine R. 29.38 °F.hr/Btu

8. (a) Calculate total surface area 217 ft^2

A_{tot} = 2 x (6 x 5.25) + 2 x (6 x 6.83) + 2 x (5.25 x 6.83) = 217 ft^2

(b) Divide line 8 by line 7 to determine A_{tot}/R 7.39 Btu/°F.hr

The sides and top will be insulated uniformly. However, the bottom is a source of parallel heat loss. The floor of the basement is 6 inch thick heavy concrete. The outside ground temperature is probably colder than 68°F. But the bottom of the rock bed is usually colder than the rest, so the temperature difference assumptions are valid. We ignore the contribution of the bond beam blocks. An abbreviated Table ii looks like:

material	r	s	R	
concrete	0.4	6.0	2.4	
rigid fiberglass	4.0	4.0	16.0	
		ΣR_{par}=	18.4	$\frac{°F.ft^2.hr}{Btu}$

(c) A_{par} = 5.25 x 6.83 = 35.9 ft^2

(e) $A_{par}/R_{par} = \frac{35.9}{18.4}$ = 1.95 Btu/°F.hr

(d) $A_{tot} - A_{par}$ = 181 ft^2

(g) $R_{req} = \frac{181}{7.39 - 1.95}$ = 33.27 °F.ft^2.hr/Btu

Since the planned vertical wall of the rock bin is constructed with the following layers:

- o 1/4 inch of sheet metal (r = 0.00318)
- o 1 inch of plywood (r = 1.25)
- o 5 1/2 inches of polystyrene foam (r = 5.00)
- o 1/2 inch gypsum board (r = 0.90)
- o 2 by 6 pine studs (where the insulation isn't as in Figure 4-1) located on 16-inch centers.

There will be a problem with para'lel heat loss through the studs.

The combined R-value through the insulation is (including surface resistance of R = 0.68):

$$R_{insulation} = (1/4 \times 0.00318) + (1 \times 1.25) + (5 \times 5.5) + (1/2 \times 0.90) + 0.68$$

$$= 29.88°F.ft^2.hr/Btu$$

$$R_{stud} = (1/4 \times 0.00318) + (1 \times 1.25) + (1/2 \times 0.90) + (5\ 1/2 \times 1.25) + 0.68$$

$$= 9.26°F.ft^2.hr/Btu$$

As shown in example 4-4, the thermal transmittance (U) can be calculated from equation 4-6.

Since the (nominal 2 by 6-actual 1 1/2 by 5 1/2) pine studs are located on 16-inch centers a safe, although rough, estimate of the area through the studs is 10% of total rockbin area.

- Stud area is 10% of $A_{tot} - A_{par}$ (from line 8d) $\underline{181}$ ft^2
  10% of 181 = 18.1

$$U = \frac{1}{181}\left(\frac{163}{29.88} + \frac{18.1}{9.26}\right) = 0.0409$$

$$R = 1/U = 24.44$$

This is considerably less than $R_{req} = 33.27$. We decide to wrap the walls and cover with 3 1/2 inch fiberglass batts. (r = 3.15 from Table 4-4).

$$R = 24.44 + (3.5)(3.15) = 35.46$$

which would provide sufficent insulation. This still leaves about 3 inches clearance for cover installation.

15.2.7 Fan and Motor Selection

Fan Selection

The fan will also be used for handling air from both the collector and the auxiliary furnace (see schematics). The maximum fan temperature is 140°F. An example of Worksheet H is given below.

1. Maximum fan temperature T — 140°F

2. Airflow-rate q (from 14.2.1-4) — 800 cfm

3. Static total pressure ΔP_T (from 14.2.5.3-6) — 1.10 inches H_2O

4. If line 1 is less than 100°F, enter 1.0.

   Otherwise enter factor = $\frac{T + 460}{530}$ — $\frac{140 + 460}{530}$ = 1.13

5. Modified flowrate q_m, multiply line 2 by line 4 — 904 cfm

6. Modified static pressure ΔP_{Tm}, multiply line 3 by line 4 — 1.24 in.H_2O

7. Obtain fan performance data from fan manufacturers and select a fan on the basis of lines 5 and 6

   a) Fan manufacturer — ABC

   b) Fan type — Centrifugal

   c) Model number — 001

   d) Operating speed — 1100 rpm

   e) Motor power requirements — ____ HP
   or
   Fan efficiency η at design point — 55%

Use the manufacturer's charts (similar to those in Figures 6-5 through 6-7) to determine the operating speed and the motor requirements.

Where the motor requirements are not indicated (as in Figure 6-6b), the fan efficiency is given instead and can thus be used to calculate the minimum motor power as shown in equation 6-1 and below.

8. Service factor — 1.25

9.	Multiply together lines 1, 2, and 3		1437
10.	Fan efficiency (%)	55	
11.	Unit conversion factor	63.46	
12.	Product of lines 10 and 11		3490
13.	Divide line 9 by line 12 to determine the HP_{min}.		0.41 HP

Also use line 13 for minimum belt drive capacity.

15.3 Latent Heat Storage Systems

In this application a storage system based on phase change materials will be installed to exploit the lower cost of off-peak electricity. We will benefit financially by buying the off-peak electricity and storing the energy in a PCM system to be used during the peak demand period during the next day.

The system will be installed in a new Naval research and development center of $10^6 ft^2$. Located in a geographic setting where electricity is the main source of space heating energy, the stored energy will be used for water heating, only. Backup is by electric resistance heating. Space heating will be accomplished largely by the excellent insulation and passive solar design of the building.

There are 1,000 employees in the building; 200 laboratories and 50 process simulation areas. A study of needs indicated that 40,000 gallons of hot water/day are necessary. Because of the laboratory usage, temperatures must reach about 160°F for general use.

The PCM's examined for this example are currently not commercially available. Costs and any physical parameter not found in Table 7-1 are fictitious.

15.3.1 Analysis of Storage Purpose

Hot water suitable for most laboratory purposes will be supplied from storage. If water temperatures above 160°F are needed, the hot water will be heated further locally.

Load Determination

Examine Worksheet A of Section 14.1 and specifically lines 1(c) through 6. Below are those lines completed for our application.

1.	(c)	Multiply lines 1a by 1b to find total daily water consumption V_d		40,000 gal.
2.	(a)	Maximum tank temperature-Minimum acceptable hot water temperature, T_{max}	160°F	
	(b)	Minimum tank temperature. If the tank holds the delivered potable water, this is the measured temperature of main supply water, T_{min}.	55°F	
3.		Subtract 2b from 2a to find ΔT_e		105°F
4.		Water density, ρ	8.33 lb/gal	
5.		Heat capacity of water, C_p	1 Btu/lb°F	
6.		Multiply lines 1c, 3, 4, and 5 to find L_w		35,000,000 Btu/day

First Estimates of Other Storage Parameters

The first estimates of storage capacity and melting point temperature are made in Worksheet B.

Storage Capacity

1.	Energy source	off-peak electricity
2.	Solar collector area (solar systems only)	_____
3.	Multiply item 2 by 600 to obtain the approximate storage capacity requirement Q for solar systems, or enter Q directly for non-solar systems.	35,000,000 Btu

Temperature

4.	End use temperature (typically 70°F for space heating or 120-140°F for domestic water heating).	160°F
5.	Approximate load heat exchanger ΔT (at least 20°F for each air-to-air, liquid-to-air, or PCM-to-air heat exchange separating the PCM from the end use.)	10-20°F
6.	Add items 4 and 5 to obtain the minimum melting point for PCM.	170-180°F
	(Subtract item 5 from item 4 to obtain the maximum melting point for a cold storage PCM.)	

15.3.2 Choosing Appropriate Storage Material

Worksheet C is used to help select the PCM.

	Parameter	Preferred Value	Phase Change Material Option 1	Phase Change Material Option 2
1.	Minimum allowed (maximum for cold storage) melting point temperature (from Worksheet B, item 6) °F		170-180	170-180
2.	Phase change compound considered		Barium hydroxide	Cross-linked poly
3.	Heat of fusion h_f (Btu/lb)-- (from Table 7-1)	High	114	99
4.	Solid heat capacity (Btu/lb°F) (from Table 7-1)	High	—	0.6
5.	Liquid heat capacity (Btu/lb°F (from Table 7-1)	High	--	0.6
6.	Melting point (°F) (less than item 1 for cold storage)	Close to but greater than item 1 above	180	270
7.	Proposed application temperature (°F)			
8.	Thermal conductivity (Btu/ft^2.hr.°F)	High		
9.	Vapor pressure (psi)	Low		
10.	Coefficient of thermal expansion (1/°F)	Low		
11.	Phase-change volume change (ft^3)	Low		
12.	Type of melting behavior	Congruent		
13.	Number of projected thermal cycles without significant phase separation	High	1000	4000

14.	Charge power density (kW/ft^3) (from manufacturer)	High		
15.	Rate nucleation efficiency (read section 7.3.2 for enhancement techniques)	High	0.6	0.7
16.	Rate corrosiveness	Low		
17.	Rate toxicity	Low		
18.	Rate fire-resistance	High		
19.	Cost	Low	$\$4,000/10^6$Btu	$\$3000/10^6$Btu

We choose the cross-linked polyethylene. The temperature is higher than desired, but the manufacturer gives a much better warranty for a slightly lower cost.

20.	Using items 1 through 19 above, select the PCM.	cross-linked poly	Material
	a. melting point T_m	270°F	
	b. latent heat of fusion h_f	99	Btu/lb.
	c. heat capacity of solid PCM Cps	0.6	Btu/lb.°F
	d. heat capacity of melted PCM Cp_l	0.6	Btu/lb.°F
21.	Approximate source heat exchanger ΔT (at least 20°F for each air-to-air, air-to-liquid, or air-to PCM heat exchange or 10°F for each liquid-to-liquid or PCM-to-liquid heat exchange separating the PCM from the heat source or solar collector.)	20°F	
22.	Add items 20a and 21 to obtain the minimum required source or solar collector temperature. (subtract item 21 from item 20a to obtain the maximum allowable source temperature for cold storage.)	290°F	

The ΔT values in item 21 and in Worksheet B, item 5, can be refined when a better estimate of heat exchanger performance is obtained.

15.3.3 Sizing Latent Heat Storage Systems

PCM Mass

The PCM mass is determined from Worksheet D.

1.	Amount of heat stored, Q, from Worksheet A, item 3.	35,000,000 Btu	
2.	Heat capacity of solid PCM, C_{ps}	0.6 Btu/lb.°F	
3.	Melting point temperature T_{mp}	270°F	
4.	Minimum storage temperature T_{min}	180°F	
5.	Subtract item 4 from item 3		90°F
6.	Multiply item 5 by item 2		54 Btu/lb
7.	Fraction of theoretical latent heat that can be recovered, η for congruent melting = 1	0.7	
8.	Latent heat of fusion h_f	99 Btu/lb	
9.	Multiply item 7 by item 8		69 Btu/lb
10.	Heat capacity of liquid PCM, C_{pl}	0.6 Btu/lb.°F	
11.	Maximum storage temperature, T_{max}	270°F	
12.	Subtract item 3 from item 11		0°F
13.	Multiply item 12 by item 10		0 Btu/lb
14.	Add together items 6, 9, 13		123 Btu/lb
15.	Divide 1 by item 14 to obtain m, amount of PCM		284,600 lb

Values for items 2, 3, 8, 10 can be determined from Table 7-1.

The PCM is in the form of small beads, approximately 1/2 inch in diameter. The beads are placed in a large container, which is provided by the manufacturer. The container volume is approximately 20% larger than the PCM volume.

Volume and Dimensions of Storage Unit

Worksheet E is used for these calculations.

Volume can be determined as follows:

1. From Table 7-1, density of chosen storage material, ρ — 60 lb/ft^3
2. Total mass required - from Worksheet D, item 15 — 284,600 lb
3. Divide item 2 by item 1 to determine volume, V_{PCM}, of phase change material only. — 4740 ft^3

Note that encapsulation, packaging, heat exchangers, and so forth can cause the volume of the storage subsystem to be considerably larger than the volume of the PCM only.

Since latent-heat storage systems are commercially manufactured, the dimensions of the storage unit will be strongly influenced by the choice of manufacturer. The dimensions are influenced by whether prepackaged encapsulated, storage modules, or a bulk storage system, is selected.

4. Module height — _____ ft
5. Module floor area — _____ ft^2
6. Volume of unit — _____ ft^3
7. Divide item 3 by item 6 to determine minimum number of modules needed for specified storage capacity — _____ modules
8. The manufacturer will suggest preferred packing configurations. — single container packing
9. Height of assembly — 6 ft
10. Floor area of assembly — 960 ft^2

15.3.4 Type, Location and Insulation of Storage Unit

Corrosion Potential and Type of Container

The PCM pellets are surrounded by an oil bath inside of a specially constructed steel box. The manufacturer claims there will be no corrosion and warranties the product for 10 years.

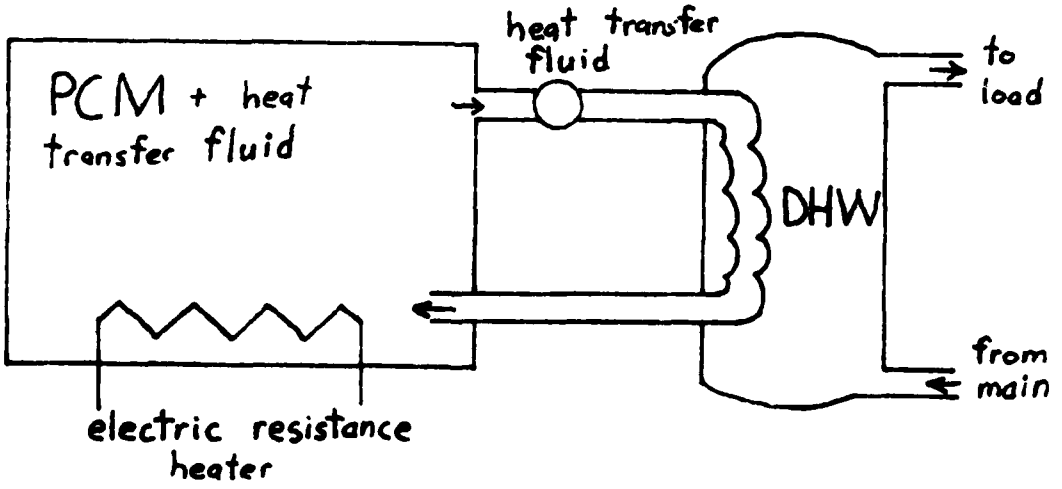
Location

The second part of Worksheet F is filled in as follows:

6.	Required weight of PCM (from Worksheet D, item 15)	284,600 lb
7.	Weight of containers, supports, etc. (from manufacturer)	99,600 lb
8.	Total weight, add items 1 and 2	384,200 lb
9.	Floor area occupied by the storage assembly (from Worksheet E, item 10)	960 ft^2
10.	Floor loading, divide item 8 by item 9	400 lb/ft^2

Since the floor loading is greater than 150 lb/ft^2, we will require the services of a structural engineer.

Sketch of System



Insulation

We wish to lose no more than 10% of the energy stored over the maximum possible storage time of seven days. Examining items 6 and 9 from Worksheet D, it is apparent that more than half of the stored energy is due to latent heat, which is at 270°F. To be conservative we choose this temperature as the average storage temperature.and complete Worksheet G.

The storage unit is 6 feet high, 30 feet long, and 32 feet wide.

$$\text{Area} = 2 \times (6 \times 30) + 2 \times (6 \times 32) + 2 \times (30 \times 32) = 2664 \text{ ft}^2$$

1.	Fraction of heat lost in time t, f	0.10
2.	Amount of heat stored, Q (from Worksheet B, item 3)	35,000,000 Btu
3.	Multiply item 1 by item 2	3,500,000 Btu
4.	Surface area of storage unit, A	2664 ft^2
5.	Length of time used for specifying the maximum heat loss from storage, t	168 hr.
6.	Multiply item 4 by item 5	448,000 ft^2.hr
7.	Melting point temperature T_m	270°F
8.	Average ambient temperature, T_a	70°F
9.	Subtract item 8 from item 7	200°F
10.	Multiply item 6 by item 9	89,500,000°F.ft^2.hr
11.	Divide item 3 by item 10 to obtain maximum allowable thermal transmittance, U	0.039 Btu/hr.ft^2.°F

The container will be insulated by placing rigid fiberglass batts on all sides of the tank.

Using section 14.1.6 as a guide complete the following:

12. Required R-value = $\frac{1}{U}$ — 25.6 °F.ft.hr/Btu

13. a. Fixed insulation layer resistances — _____ °F.ft^2.hr/Btu.in
    b. — °F.ft^2.hr/Btu.in
    c. — _____ °F.ft^2.hr/Btu.in

14. Total fixed layer resistance
    R fixed-add lines 13 a, b, and c — 0 °F.ft^2.hr / Btu.in

15. Subtract line 14 from line 12. — 25.6

16. Assume that the insulation layer of unknown thickness has no parallel heat loss paths.
    - thus find r* in Table 4-4. — 4.0

17. Thickness s of layer, divide line 15 by r* — 6.4 in.

Although the manufacturer of this particular (fictitious for this example) insulation claims it will not compress under the calculated load, long-term examples are not available, and we are concerned about deterioration. We specify 9 inches for the bottom of the container.

15.3.5 Heat Exchangers

The PCM pellets are immersed in a heat transfer fluid. The heat transfer fluid is heated by electric resistance heaters at several places at the bottom of the container. The heat transfer fluid circulates through the container by natural convection

The laboratory water is heating by pumping heat transfer fluid from the top of the PCM container through a heat exchanger in a conventional domestic hot water tank.

15.3.6 Pumps

The only pump in the system is the one that pumps heat transfer fluid from the PCM container through the DHW tank heat exchanger. The specification of this high-temperature, non-water pump is best left to the PCM manufacturer or contractor.

15.4 Water Tank Cold Storage

In this example we use waste heat thermal energy to power an absorption chiller. The absorption chiller supplies chilled water to cool an administrative building. The chilled water is stored in a large concrete tank.

The building is always ain use, requires almost constant cooling, and has a significant latent load. The waste heat source runs intermittantly on a cylcle of several days on and several days off. The longest possible period in the off state is 7 days.

15.4.1 Cooling Load

The design cooling load has been calculated using ASHRAE methods as 15 tons of cooling.

$$L_T = 15 \text{ tons} \times 12{,}000 \ \frac{\text{Btu}}{\text{hr}}/\text{ton} = 180{,}000 \text{ Btu/hr}$$

$$180{,}000 \text{ Btu/hr} \times 24 \text{ hr/day} = 4{,}320{,}000 \text{ Btu/day}$$

This corresponds to item 3 of Worksheet A.

15.4.2 Tank Size

Temperatures

We assume the tank to be well mixed. To control the humidity, the maximum tank temperature is 55°F. The chiller supplies water at 35°F.

4.	Maximum tank temperature T_{max}	55 °F
5.	Minimum tank temperature T_{min}	35 °F
6.	Add lines 4 and 5 and divide the result by 2 to get T_{avg}	45 °F
7.	Subtract line 5 from line 4 to get the temperature range of the tank, ΔT	20 °F

Tank Size

Worksheet B is completed to determine tank size.

1.	Daily cooling load L from Worksheet A, item 3	4,320,000 Btu/day
2.	Number of days storage desired, N	7 days
3.	Thermal storage capacity of tank, Q Multiply item 1 by item 2	30,240,000 Btu
4.	Liquid density, ρ, (ρ for water = 8.33)	8.33 lb/gal
5.	Heat capacity of water (Heat capacity for water = 1.00).	1 Btu/lb.°F
6.	ΔT from 14.4.2.1 line 4.	20 °F
7.	Multiply lines 4, 5, 6.	166.6 Btu/gal
8.	Divide line 3 by line 7 to get tank size V.	182,000 gal
9.	Multiply line 8 by 0.13368 to get tank size V in cubic feet.	24,300 ft^3

15.4.3 Stratification

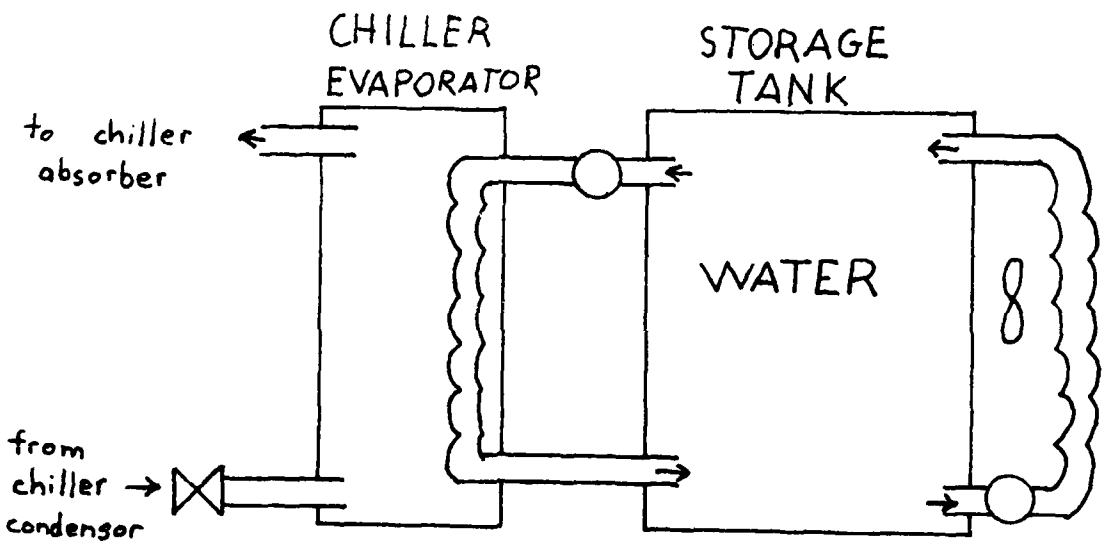
Since the sizing calculation and system design should not depend on stratification it will not be discussed here.

Remember, however, to include as many of the stratification techniques and control strategies from 5.2.2 as possible.

15.4.4 Type, Location, and Insulation of Tank

These characteristics will not be resolved because the procedure has been outlined in great detail in other sections.

Sketch of System



15.4.5 Cost Analysis

Since the energy source is not solar, the design example in 14.4.6 is not pertinent to this application.

APPENDIX A: UNITS AND CONVERSION FACTORS

This appendix contains brief descriptions of the International System of Units (SI) and the English system of units, along with tables for converting from one set of units to another. Discussions of some problem units are also included.

THE METRIC SYSTEM OF UNITS

The SI system is based on seven base units: a unit of length, the meter; a unit of mass, the kilogram (kg); a unit of time, the second (s); a unit of temperature, the kelvin (K); a unit of electric current, the ampere (A); and two units not used in this manual, the mole and the candela.

Many other units are used with these basic ones. The degree Celsius (°C) is more commonly used than the kelvin to measure temperature; it is frequently used in this manual. A temperature interval of one degree Celsius is exactly equal to one kelvin. The difference in the two units is that zero kelvins is absolute zero, while zero degrees Celsius is the freezing point of water which is equal to 273.15 kelvins.

The base units can be combined to form derived units. Density, for example, is given in kilograms per cubic meter (kg/m^3). Some derived units have their own names; those of importance in this manual are listed in Table A-1. Prefixes representing multipliers can be attached to SI units to provide convenient-sized units. Table A-2 lists the most important prefixes, their symbols, and the multiplication factor they represent.

A unit of liquid volume often used with SI units but not defined in the SI system is the liter (ℓ), which is equal to one thousandth of a cubic meter.

THE ENGLISH SYSTEM OF UNITS

The base units of the English system are the foot (ft), a unit of length; the second (sec), a unit of time; the pound-force (lbf), a unit of force; and the degree Fahrenheit (°F), a unit of temperature. The base unit of electrical current is the ampere, which is taken from the SI system. Volts, watts, and ohms are also taken from the SI system.

Some derived units in the English system are single combinations of the base units--for example, pressure is given in pound-force per square foot (lbf/ft^2) --but others are more complicated. A commonly used unit of mass is the pound-mass (lb), which is defined as the amount of mass which exerts a force of one pound-force in the earth's gravitational field.

Table A-1. Some Derived SI Units

Quality	Unit	In Terms of Other Units
Frequency	hertz (Hz)	1/s
Force	newton (N)	$kg \cdot m/s^2$
Pressure	pascal (Pa)	N/m^2
Energy (also heat and work)	joule (J)	N·m
Power	watt (W)	J/s
Electric Potential	volt (V)	W/A
Electrical Resistance	ohm (Ω)	V/A

It is worth noting that all of the derived units in the SI system are derived with a multiplier of exactly one. Thus, one joule is exactly one newton meter. The SI system is said to be coherent for this reason.

Table A-2. Some SI Prefixes

Prefix	Symbol	Multiplication Factor
mega	M	1 000 000. = 10^6
kilo	k	1 000. = 10^3
centi	c	0.01 = 10^{-2}
milli	m	0.001 = 10^{-3}
micro	μ	0.000 0001 = 10^{-6}

The unit of energy is the foot-pound-force (ft·lbf), but the British thermal unit (Btu) is more commonly used in thermal calculations. One Btu is defined as the amount of energy needed to raise the temperature of one pound-mass of water by one degree Fahrenheit. It is approximately equal to 778.1693 ft·lbf.[1] The unit of power is the foot-pound-force per second, but, again, another unit, the horsepower (hp), is more commonly used. One horsepower is equal to 550 ft·lbf/sec.

A commonly used unit of volume is the U.S. liquid gallon (gal), which is approximately equal to 0.1337 cubic feet. Other units of time include the hour (hr), which is convenient for heat transfer calculations, and the minute (min), which is often used in flow rate calculations.

PROBLEM UNITS

The Pound

The term pound can cause confusion because the pound is a unit of both force and mass, and its symbol, lb, is often used to stand for both types of pound. In this manual we use lbf to stand for pound-force and lb to stand for pound-mass.

Confusion can also arise when the symbol g is encountered in equations. In some equations it represents the standard acceleration of gravity, and in others it is a unit conversion factor that converts lb to lbf or vice versa. g has the same numerical value in both uses, but its meaning is different.

In this manual, we have defined g wherever it appears in an equation.

Viscosity

The term viscosity can refer to two kinds of viscosity--absolute, or dynamic, viscosity (used in this manual) and kinematic viscosity, which is defined as the absolute viscosity divided by the density of the fluid.

The units of absolute viscosity are pound-mass per hour per foot (lb/hr·ft) in the English system and pascal second (Pa·s) in the SI system. The units of kinematic viscosity are square foot per hour (ft^2/hr) in the English system and square meter per second (m^2/s) in the SI system.

---

[1]These numbers are not exact because the English units are defined in terms of the metric SI units. For instance, one Btu is exactly equal to 1055.05585262 Joules.

Two other units, the poise for absolute viscosity and the stoke for kinematic viscosity, are the most commonly used. These metric units are not part of either the English or the SI system but are used with both. Because poise and stoke are fairly large compared with the viscosities of most liquids, centipoise and centistoke are often used. These units must be converted to the appropriate units in the English or SI system when they are to be used in equations. One centipoise is equal to one thousandth of a pascal second and one centistoke is equal to one millionth of a square meter per second.

You will often have to check the units in order to determine which type of viscosity data is being presented. The factors for converting from one set of units to another are given in Table A-4.

Pressure

Several different units are used to describe pressure. In the English system, its units are pound-force per square foot (lbf/ft^2); in the SI system, its units are pascals (Pa).

Pressure is often given in terms of pound-force per square inch (psi). Two other units which you may encounter are psia and psig. The "a" in psia stands for absolute. A pressure given in psia is the pressure above an absolute vacuum. The "g" in psig stands for gauge. A pressure given in psig is the pressure relative to the atmospheric pressure on the gauge which is measuring the pressure. Gauge pressure can be either positive or negative but absolute pressure can only be positive.

Other units of pressure are inches of water, feet of water, and meters of water. These units are defined as the pressure at the base of a column of water whose height is measured in inches, feet, or meters. To convert these units, multiply them by the density of water. In the SI system, the result, which is in kg/m^2, must be multiplied by the acceleration of gravity, $g = 9.8\ m/s^2$ to get pressure in pascals. In the English system, the result of multiplying the height of water by the density is pound-mass per square foot (or square inch). But, because one pound-mass exerts a force of one pound-force, the pressure at the base of the column is an equal number of pounds-force per square foot.

Head

Head is a term used to describe the output of centrifugal pumps. It is given in units of length or height without a liquid being specified. The operating properties of centrifugal pumps ensure that at a given flow rate they will exert enough pressure at their output to raise a column of the liquid being pumped to a fixed height no matter what liquid is being used. In other words, centrifugal pumps have the same head at a given flow rate regardless of what type of liquid they are pumping. To calcuate the pressure at the pump output, you must multiply the head by the density of the liquid.

Again, for metric units you must multiply this result by the acceleration of gravity.

Quite often, as in Chapter 3 of this manual, all of the pressure drops in a system are described in terms of head losses. If you wish to convert pressure drops to head losses, you must divide the pressure by the density of the liquid. For metric units, you must also divide the result by the acceleration of gravity.

UNIT CONVERSION

Temperature Conversion Table

Converting temperatures from one set of units to another is somewhat different from other conversions; so temperature conversion equations appear in a separate table, Table A-3.

It is important that you know whether you are working with an actual temperature or with a temperature difference before using these equations, since actual temperature and temperature difference are not converted in the same way. In this manual when temperature units are presented within a set of units such as $^{\circ}F \cdot ft^2 \cdot hr/Btu$, the temperature unit always represents temperature difference.

Unit Conversion Table

Table A-4 contains numerical factors needed to convert quantities expressed in one set of units to quantities expressed in another set of units. Where these factors are very large or very small, scientific notation is used.

The units in Table A-4 are grouped according to type. Conversion factors that are followed by an asterisk (*) are exact. The other factors have been rounded off to five decimal places.

To use Table A-4, find the units that you wish to convert from in the left column opposite the units that you wish to convert to in the right column. Multiply the quantity of those units that you have by the conversion factor in the center column to convert the quantity to the units in the right column.

Example A-1:

Suppose you want to find the number of square meters in an area of 144 square inches. Under the heading Area in Table A-4 find the line:

square inches (in^2)	6.4516×10^{-4}*	square meters (m^2)

The conversion is:

$$144\ in^2 \times 6.4516 \times 0.000\ 1 = 0.092903\ m^2.$$

Example A-2:

Suppose you have some isulation with an R-value of 19°F·ft^2·hr/Btu and you wish to know its value in the SI units of °C·m^2/W. To perform the conversion use conversion factors from Tables A-3 and A-4:

The temperature unit, °F, represents a temperature interval in this value; so from Table A-3 you find that:

$$1°C = 1.8°F \tag{A-1}$$

The other units can be found in Table A-4:

$$1\ m^2 = 10.764\ ft^2.$$

$$1\ s = 2.7778 \times 10^{-4}\ hr = \frac{1}{3600}\ hr. \tag{A-2}$$

$$1\ Btu = 1055.1\ J.$$

These conversions can be written in another way:

$$1 = \frac{1°C}{1.8°F}\ .$$

$$1 = \frac{1\ m^2}{10.764\ ft^2}\ . \tag{A-3}$$

$$1 = \frac{3600\ s}{1\ hr}\ .$$

$$1 = \frac{1\ Btu}{1055.1\ J}\ .$$

The conversion is made by multiplying the original value, 19°F·ft^2·hr/Btu, by the various factors of 1 given in Equation A-3.

$$19\ \frac{°F·ft^2·hr}{Btu} \times \frac{1°C}{1.8°F} \times \frac{1\ m^2}{10.764\ ft^2} \times \frac{1\ Btu}{1055.1\ J} \times \frac{3600\ s}{hr}\ .$$

Now the various units which appear both on the top and bottom of this expression can be cancelled:

$$\frac{19 \times 3600}{1.8 \times 10.764 \times 1055.1} \times \frac{\cancel{°F} \times °C}{\cancel{°F}} \times \frac{\cancel{ft^2}\ m^2}{\cancel{ft^2}} \times \frac{\cancel{hr}\ s}{\cancel{hr}} \times \frac{\cancel{Btu}}{\cancel{Btu}·J}$$

giving: $3.3\ °C·m^2·s/J,$

which, by definition of the watt, is equivalent to 3.3 °C·m^2/W.

Table A-3. Temperature Conversion Equations[a]

TEMPERATURE

$$T_c = \frac{5}{9}(T_f - 32)$$

$$T_f = \frac{9}{5} T_c + 32$$

$$T_c = T_k - 273.15$$

$$T_f = \frac{9}{5} T_k - 459.67$$

TEMPERATURE DIFFERENCE[b]

$$\Delta T_c = \frac{5}{9} \Delta T_f$$

$$\Delta T_f = \frac{9}{5} \Delta T_c$$

$$\Delta T_c = \Delta T_k$$

$$\Delta T_f = \frac{9}{5} \Delta T_k$$

[a] T_c, T_f, and T_k stand for temperature in degrees Celsius, degrees Fahrenheit, and kelvins.

[b] The symbol Δ represents temperature difference in the following terms.

Table A-4. Conversion Factors

Multiply	By	To Obtain
	Area	
cm^2	0.0010764	ft^2
cm^2	1.0×10^{-4}*	m^2
ft^2	0.092903	m^2
in^2	0.0069444	ft^2
in^2	6.4516×10^{-4}*	m^2
m^2	10.74	ft^2
	Density	
kg/ℓ	1000.0 *	kg/m^3
kg/ℓ	62.428	lb/ft^3
kg/m^3	.062438	lb/ft^3
lb/ft^3	16.018	kg/m^3
lb/gal	119.83	kg/m^3
lb/gal	7.4805	lb/ft^3
	Energy, Heat, and Work	
Btu	1055.1	J
ft·lbf	0.077817	Btu
ft·lbf	1.3558	J
J	9.4782×10^{-4}	Btu
kWh	3412.1	Btu
kWh	3.6×10^{6}*	J

Table A-4. (continued)

Multiply	By	To Obtain
	Force	
N	0.22481	lbf
lbf	4.4482	N
	Heat Capacity	
J/kg·°C	2.3885×10^{-4}	Btu/lb°F
Btu/lb·°F	4186.8*	J/kg·°C
	Heat Capacity, Volumetric	
Btu/ft^3.°F	6.706×10^4	J/m^3.°C
Btu/gal·°F	7.4805	Btu/ft^3.°F
Btu/gal·°F	5.0169×10^5	J/m^3.°C
J/ℓ·°C	0.014911	Btu/ft^3.°F
J/ℓ·°C	1000.0*	J/m^3.°C
J/m^3.°C	1.4911×10^{-5}	Btu/ft^3.°F
	Length	
cm	0.032808	ft
cm	0.01*	m
ft	0.3048*	m
in	0.083333	ft
in	0.0254*	m

Table A-4. (continued)

Multiply	By	To Obtain
	Length (cont.)	
m	3.2808	ft
mil	8.3333×10^{-5}	ft
mil	2.54×10^{-5}*	m
mm	010032808	ft
mm	0.001*	m
yd	3.0*	m
yd	0.9144*	m
	Mass	
kg	2.2046	lb
lb	0.45359	kg
	Mass Flow Rate	
lb/h	1.2600×10^{-4}	kg/s
lb/min	60.0*	lb/h
lb/min	0.0075599	kg/s
lb/s	3600.0*	lb/h
lb/s	0.45359	kg/s
kg/h	2.2046	lb/hr
kg/h	2.7778×10^{-4}	kg/s
kg/min	132.28	lb/h
kg/min	0.01667	kg/s
kg/s	7936.6	lb/h

Table A-4. (continued)

Multiply	By	To Obtain
	Power	
Btu/h	778.17	ft·lbf/hr
Btu/h	0.29307	W
ft·lbf/h	0.0012851	Btu/h
ft·lbf/h	$3\ 7662 \times 10^{-4}$	W
ft·lbf/s	4.6262	Btu/h
ft·lbf/s	3600.0*	ft·lbf/h
ft·lbf/s	1.3558	W
hp	2544.4	Btu/h
hp	1.98×10^{6}*	ft·lbf/h
hp	745.70	W
W	3.4121	Btu/h
W	2655.2	ft·lbf/h
	Pressure	
cm of water	2.0481	lbf/ft^2
cm of water	98.064	Pa
ft of water	62.426	lbf/ft^2
ft of water	2989.0	Pa
in of water	5.2022	lbf/ft^2
in of water	249.08	Pa
lbf/ft^2	47.880	Pa

Table A-4. (continued)

Multiply	By	To Obtain
	Pressure (cont.)	
psi (lbf/in^2)	144.0*	lbf/ft^2
psi	6894.8	Pa
m of water	204.81	lbf/ft^2
m of water	9806.4	Pa
Pa	0.020885	lbf/ft^2
	Thermal Conductance (U-Value)	
$Btu/ft^2 \cdot h \cdot °F$	5.6783	$W/m^2 °C$
$W/m^2 °C$	0.17611	$Btu/ft^2 \cdot h \cdot °F$
	Thermal Resistance (R-Value)	
$ft^2 \cdot h \cdot °F/Btu$	0.17611	$m^2 °C/W$
$m^2 \cdot °C/W$	5.6782	$ft^2 \cdot h \cdot °F/W$
	Thermal Resistance per Unit Thickness (r-Value)	
$ft^2 \cdot h \cdot °F/Btu \cdot in$	$0.069335\ 10^{-2}$	$m^2 \cdot °C/W \cdot cm$
$m^2 \cdot °C./W \cdot cm$	14.423	$ft^2 \cdot h \cdot °F/Btu \cdot in$
	Time	
day	24.0*	h
day	8.64×10^{4}*	s

Table A-4. (continued)

Multiply	By	To Obtain
	Time (cont.)	
h	3600.0*	s
min	0.016667	h
min	60.0*	s
s	2.7778×10^{-4}	h
y	8760.0*	h
y	3.1536×10^{7}*	s
	Velocity	
ft/h	8.4667×10^{-5}	m/s
ft/min	60.0*	ft/h
ft/min	0.00508*	m/s
ft/s	3600*	ft/h
ft/s	0.3048*	m/s
m/s	1.1811×10^{4}	ft/h
	Viscosity (Absolute or Dynamic)	
lb/ft·h	4.1338×10^{-4}	Pa·s
centipoise	2.4191	lb/ft·h
centipoise	0.001*	Pa·s
Pa·s	2419.1	lb/ft·h

Table A-4. (continued)

Multiply	By	To Obtain
	Viscosity (Kinematic)	
ft^2/h	2.5806×10^{-5}	m^2/s
centistokes	0.038750	ft^2/h
centistokes	1.0×10^{-6}	m^2/s
m^2/s	3.8750×10^{4}	ft^2/h
	Volume	
cm^3	3.5315×10^{-5}	ft^3
cm^3	1.0×10^{-6}*	m^3
ft^3	0.028317	m^3
gal	0.13368	ft^3
gal	0.0037854	m^3
in^3	5.7870×10^{-4}	ft^3
in^3	1.6387×10^{-5}	m^3
ℓ	0.035315	ft^3
ℓ	0.001*	m^3
m^3	35.315	ft^3
yd^3	27.0*	ft^3
yd^3	0.76455	m^3

Table A-4. (continued)

Multiply	By	To Obtain
	Volume Flow Rate	
ft^3/h	7.8658×10^{-6}	m^3/s
ft^3/min (cfm)	60.0*	ft^3/h
ft^3/min	4.7195×10^{-4}	m^3/s
ft^3/s	3600.0*	ft^3/h
ft^3/s	283.17	m^3/s
gal/min (gpm)	8.0208	ft^3/hr
gal/min	6.3090×10^{-5}	m^3s
ℓ/min	2.1189	ft^3/h
ℓ/min	1.6667×10^{-5}	m^3/s
ℓ/s	127.13	ft^3/h
ℓ/s	0.001*	m^3/s
m^3/s	1.2713×10^5	ft^3/h

ABBREVIATIONS USED IN TABLES

Btu	British thermal unit (international table)
°C	degree Celsius (sometimes incorrectly called degree centigrade)
cm	centimeter
°F	degree Fahrenheit
ft	foot
gal	U.S. liquid gallon
h	hour (often abbreviated hr in English system)
hp	horsepower (550 ft·lbf/s)
in	inch
J	joule
K	kelvin
kg	kilogram
kWh	kilowatt-hour
ℓ	liter (also spelled litre)
lb	pound-mass
lbf	pound-force
m	meter (also spelled metre)
mil	one thousandth of an inch
min	minute
mm	millimeter
N	newton
Pa	pascal
psi	pound-force per square inch
s	second (often abbreviated sec. in English system)
W	watt
y	year (365 days)
yd	yard

APPENDIX B: DIRECTORIES OF MANUFACTURERS

The following is an annotated list of directories that contain product information from various manufacturers:

- ASHRAE Handbook and Product Directory
  American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 345 East 47th Street, New York, NY 10017.
  (The directory includes manufacturers' addresses, product sources, and manufacturers' catalog data).

- Yvonne Howell and Justin A. Bereny.
  Engineer's Guide to Solar Energy
  Solar Energy Information Services (SEIS), P.O. Box 204, San Mateo, CA 94401, 1979.
  (An eleven-page product directory lists manufacturer addresses and type of product).

- Yen-Hsiung Kiang.
  Waste Energy Utilization Technology
  Marcel Dekker, Incorporated, 270 Madison Avenue, New York, NY 10016, 1981.
  (Provides a complete review of proven heat transfer equipment).

- C. W. Martz (ed).
  Solar Energy Sourcebook
  Solar Energy Institute of America, Box 9352, Washington, DC 20005, 1977, 712 pp.
  (Organized compilation of solar energy related products and services in loose leaf binder format - continual updates).

- Master Catalog and Buyers Guide.
  Solar Engineering Magazine, 23636 Walnut Hill Lane, Suite 257, Dallas, TX 75229. 1981.
  (More than 250 different solar products from over 1,000 manufacturers).

- Richard W. Merril and Thomas Gage.
  Energy Primer: Solar, Water, Wind, and Biofuels. Portola Institute Incorporated, Menlo Park, CA 94025, 1978.
  (More than 25% of the book is devoted to reviews of books and hardware sources).

- The Power Company - Midwest, Incorporated
  The Wind Cyclopedia.
  Windcyclopedia, P.O. Box 221, Genesee Depot, WI 53127, 1981.
  (One of the most comprehensive directories of wind energy equipment and manufacturers).

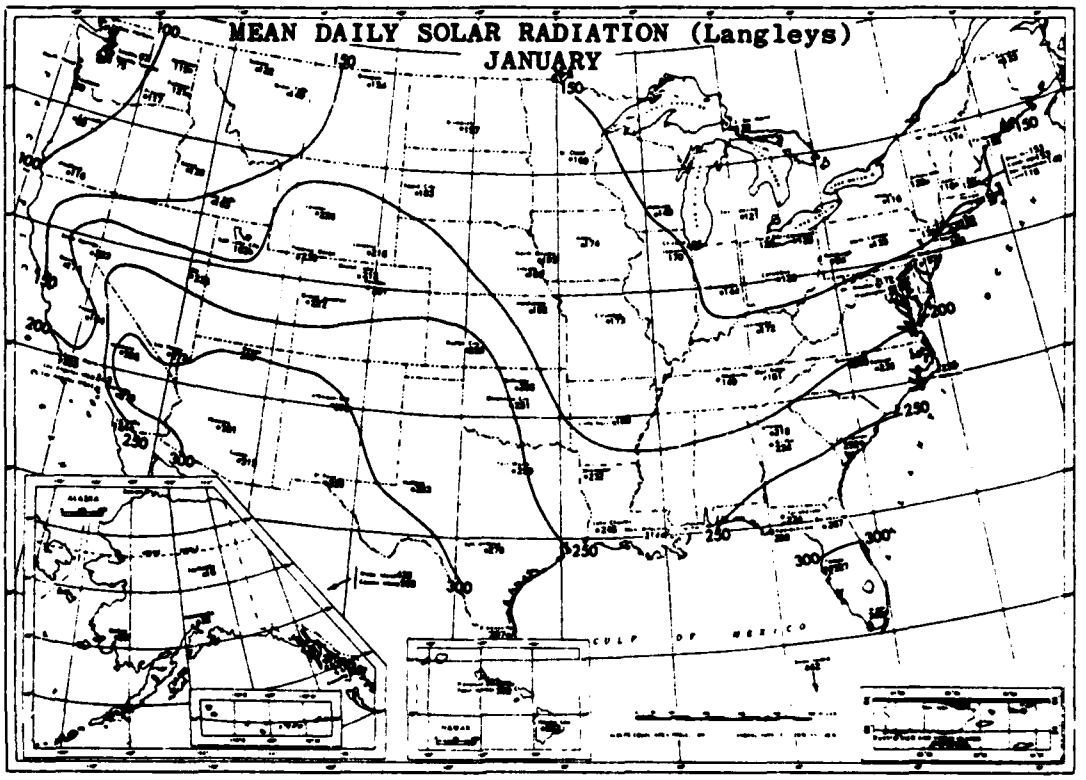
- Solar Age Magazine
  Solar Products Specifications Guide
  Department SA, Solar Age Magazine, Church Hill, Harrisville, NH 03450. (Updated bimonthly. A looseleaf collection of solar product data).

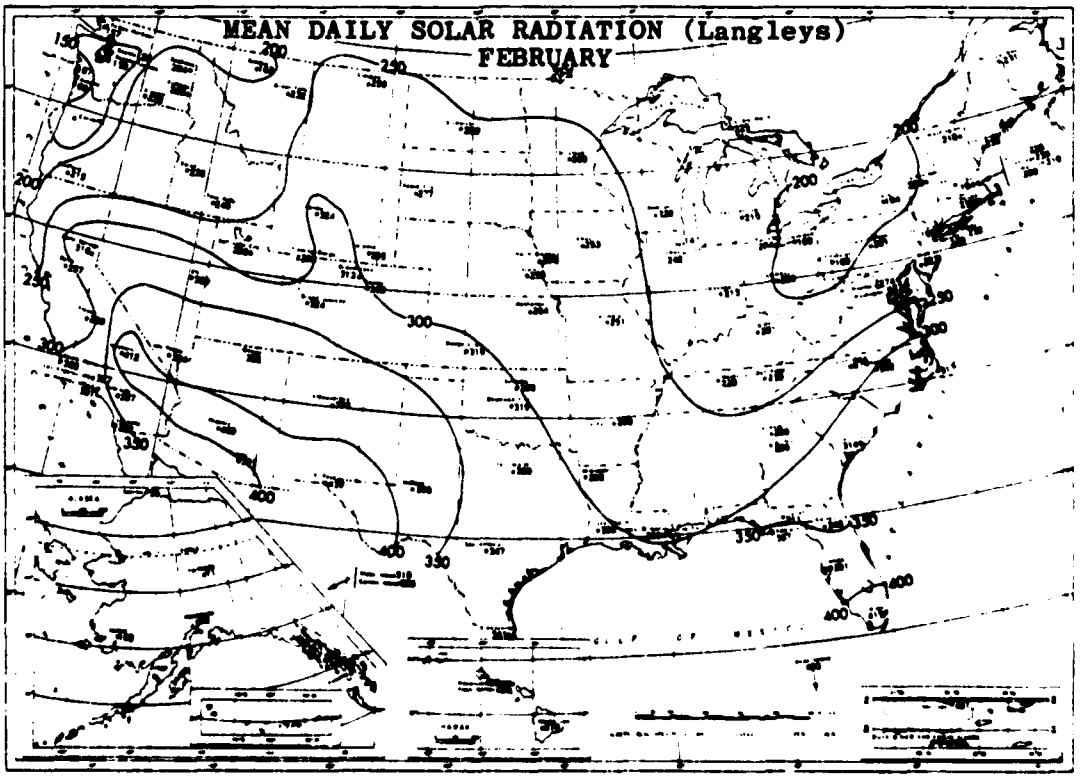
- Solar Energy Industries Association, 1001 Connecticut Avenue, NW Suite 632, Washington, DC 20036.
  Solar Industry Index
  (Published annually. A comprehensive guide to active manufacturers and service organizations).

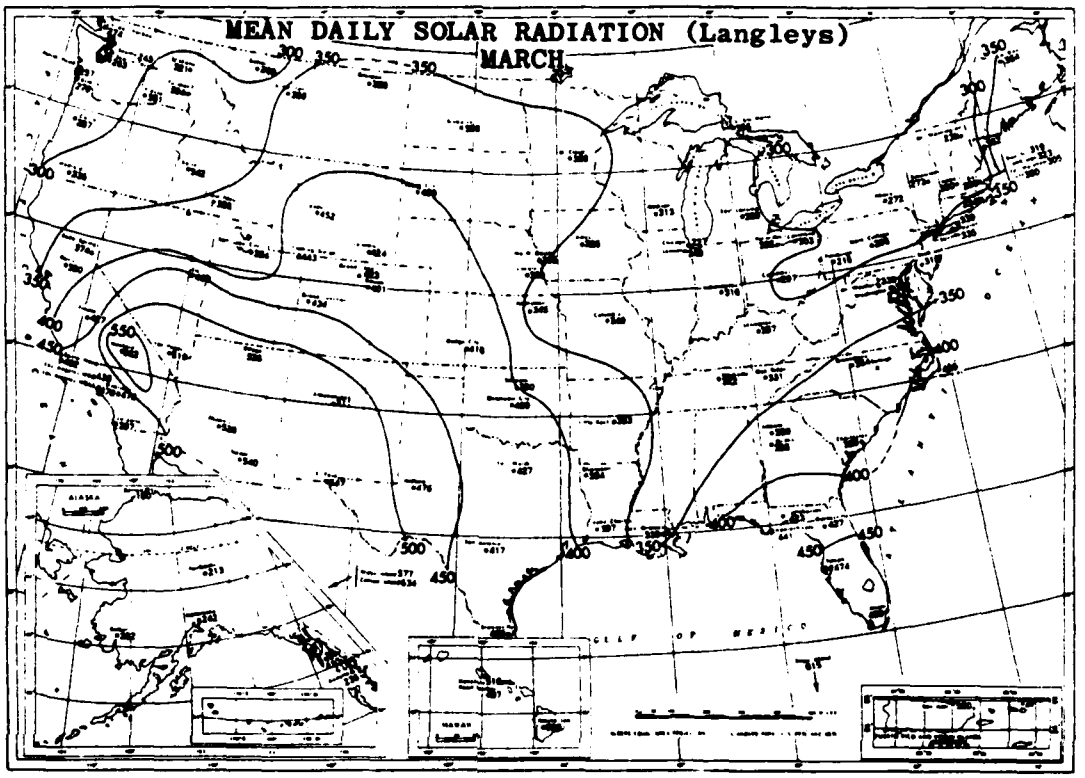
- Solar Vision, Incorporated, 200 East Main Street, Port Jervis, NY 12771.
  Solar Age Catalog, September 1977.
  (Listings of products and manufacturers).

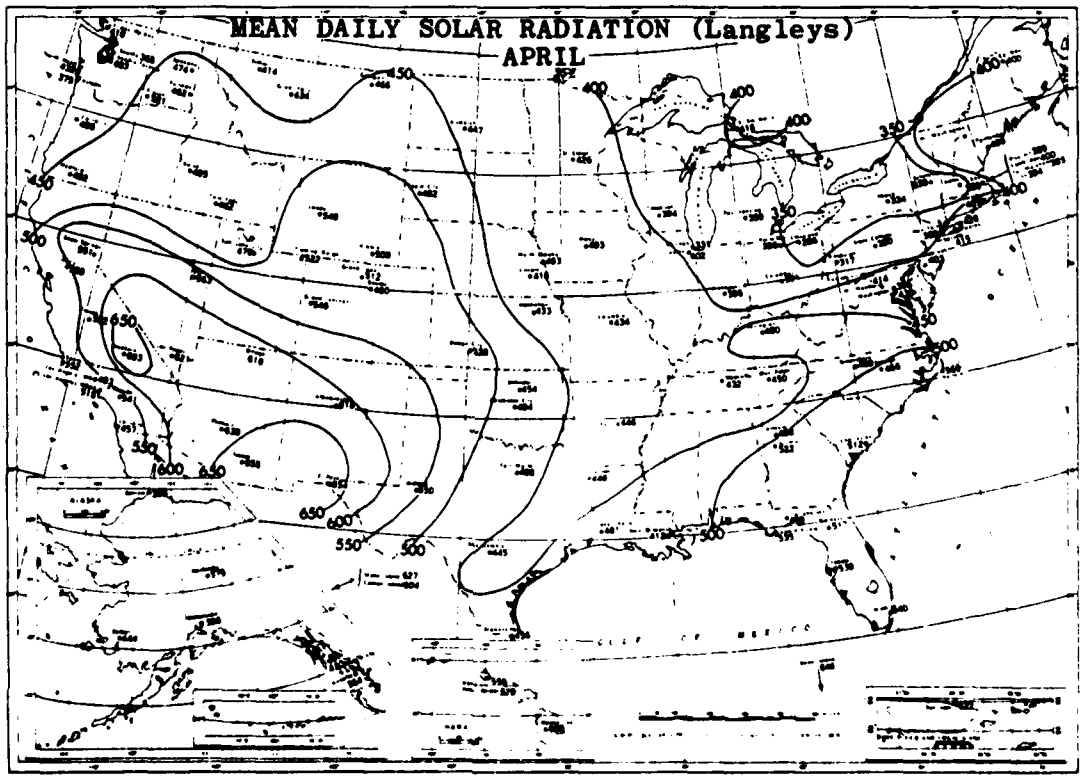
- Thomas Publishing Company, One Penn Plaza, New York, NY 10001.
  Thomas Register of American Manufacturers and Thomas Register Catalog File. (Published annually - The ultimate product directory of American companies).

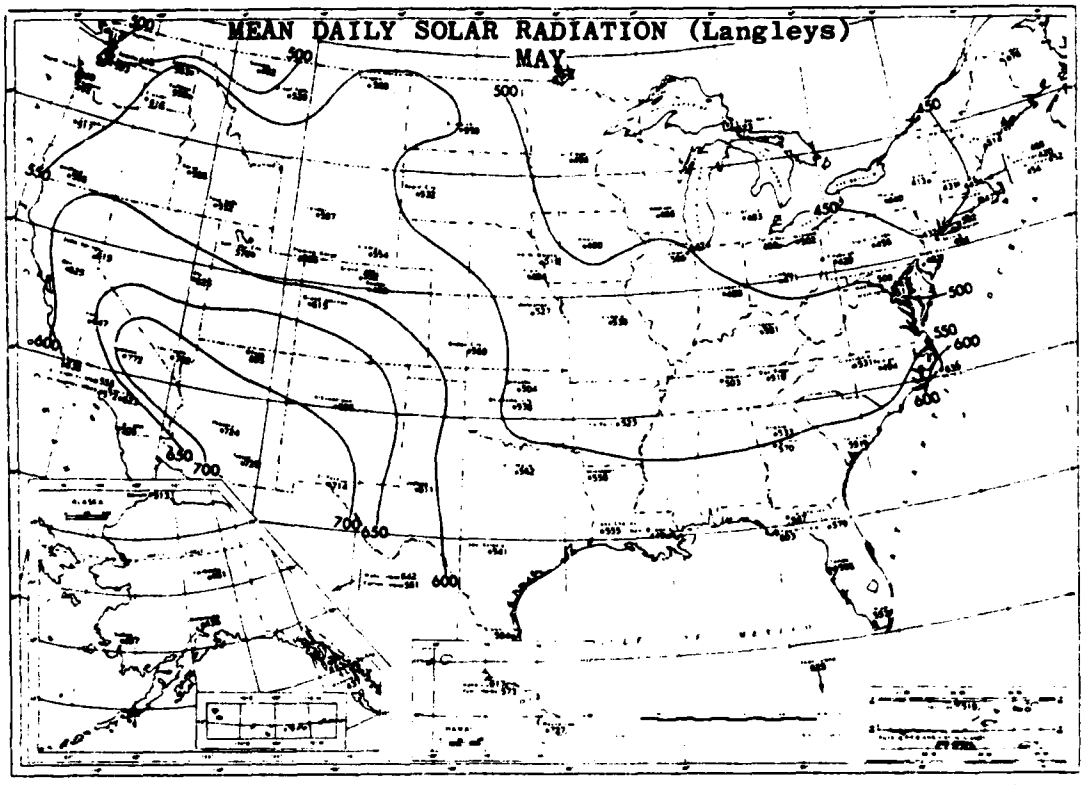
APPENDIX C: SOLAR ENERGY AVAILABILITY MAP
(from Climatic Atlas of the United States, U.S. Government Printing Office, 1968)

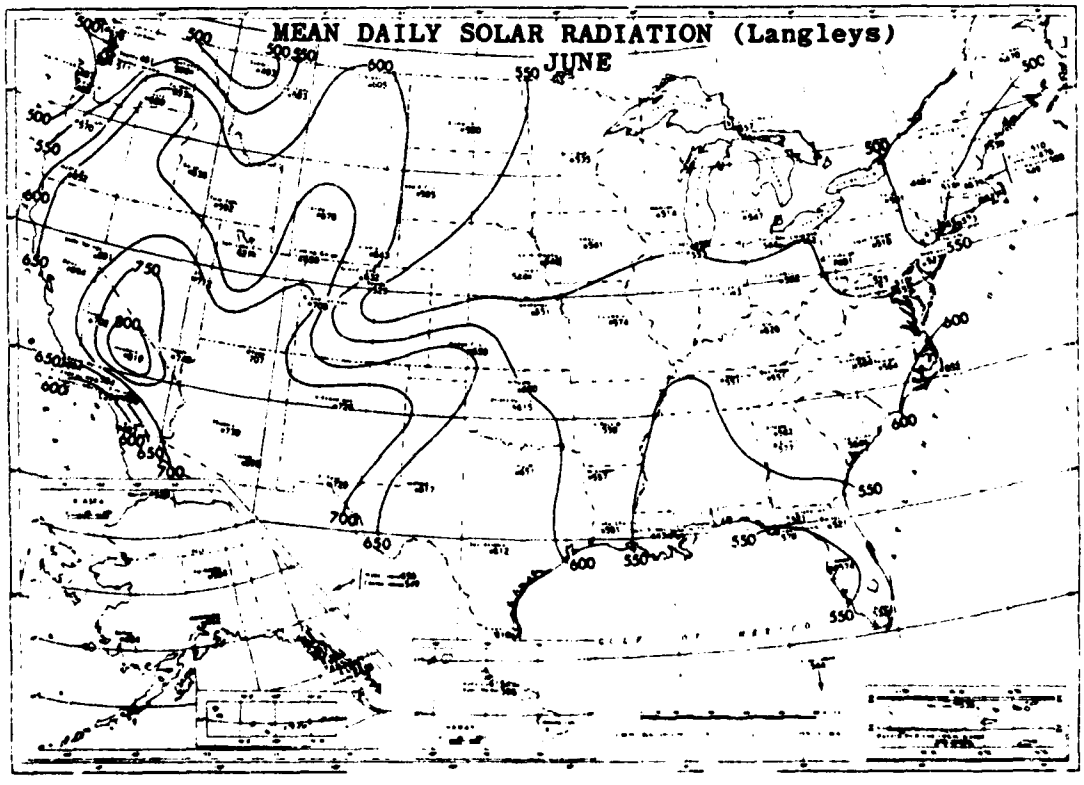


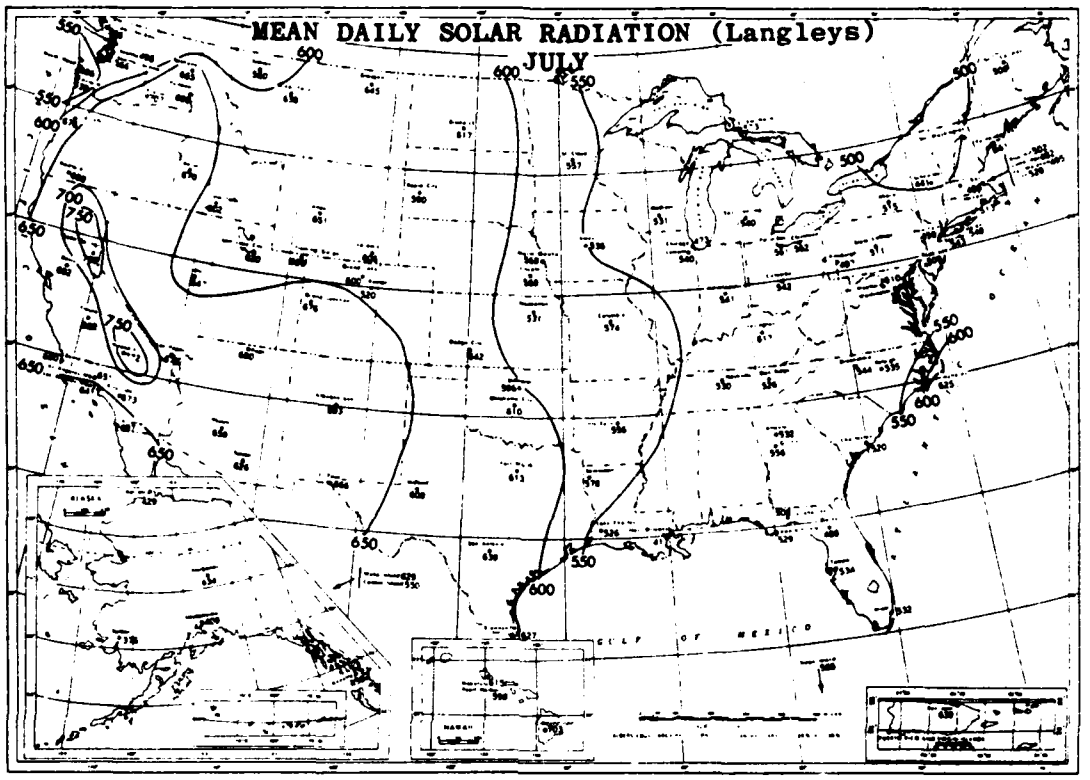


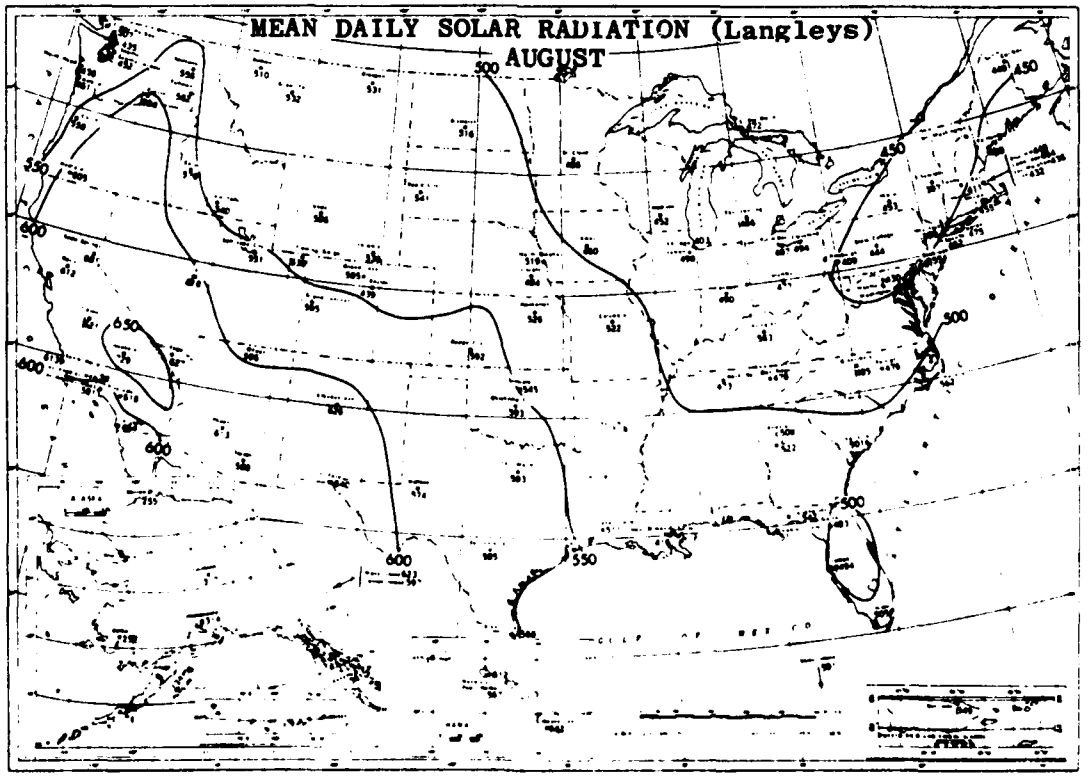

MEAN DAILY SOLAR RADIATION (Langleys)
MARCH
GULF OF MEXICO

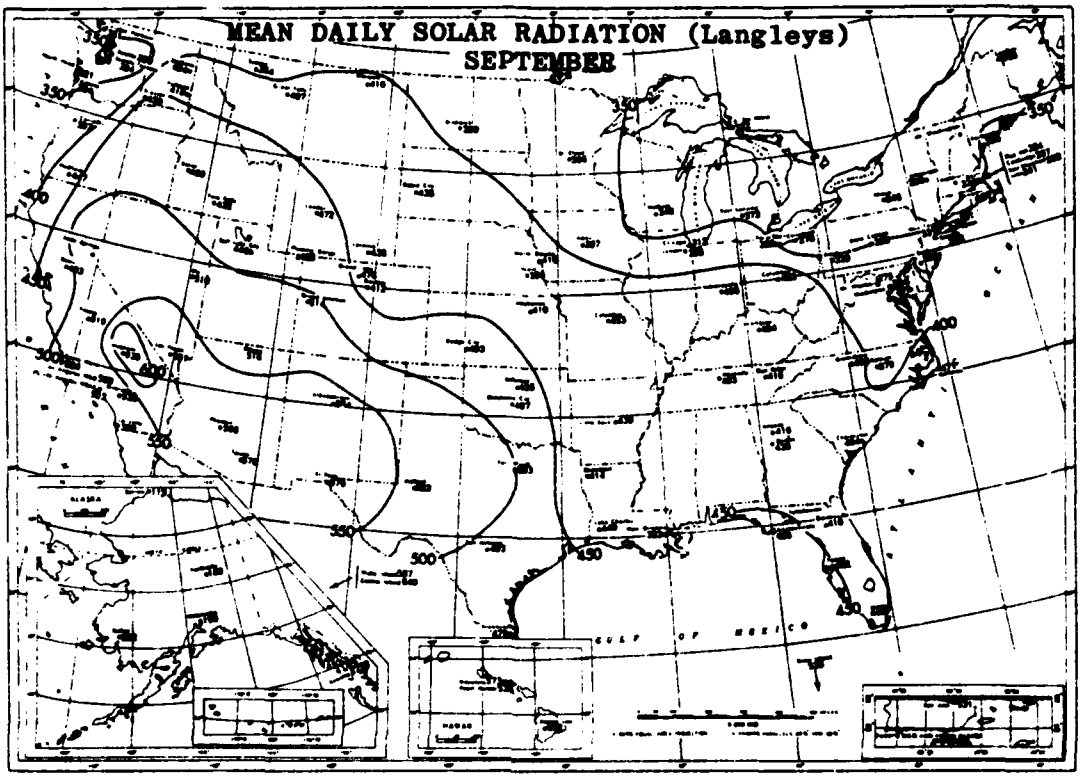

MEAN DAILY SOLAR RADIATION (Langleys)
APRIL
GULF OF MEXICO

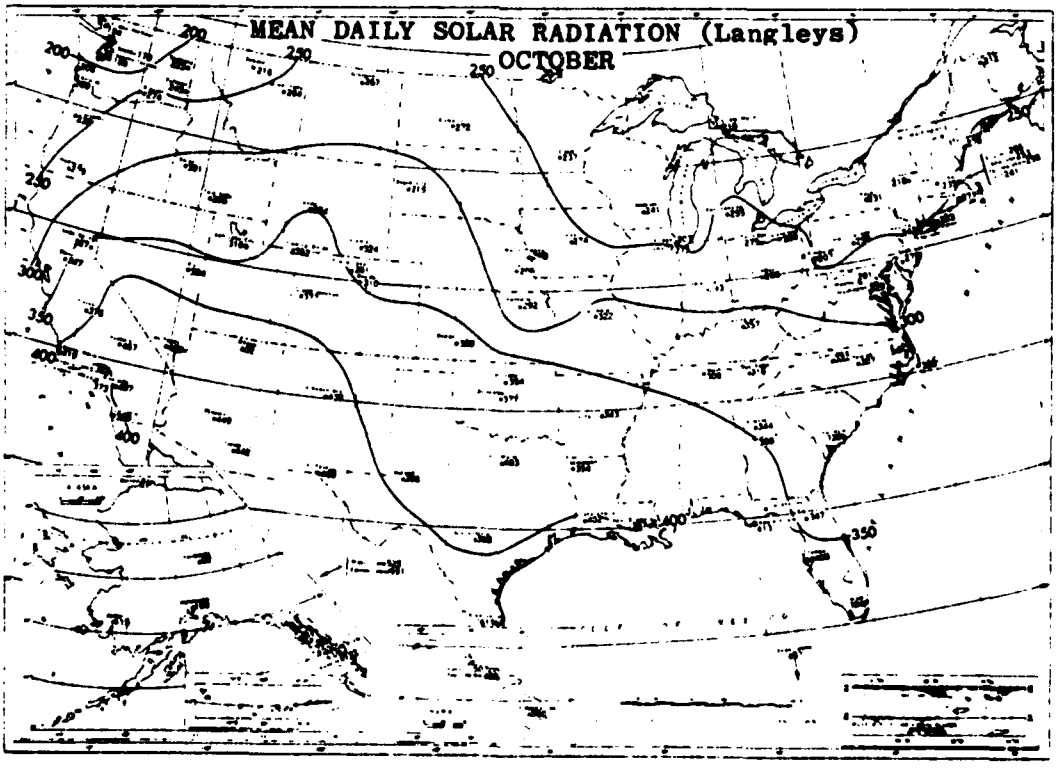

MEAN DAILY SOLAR RADIATION (Langleys)
MAY

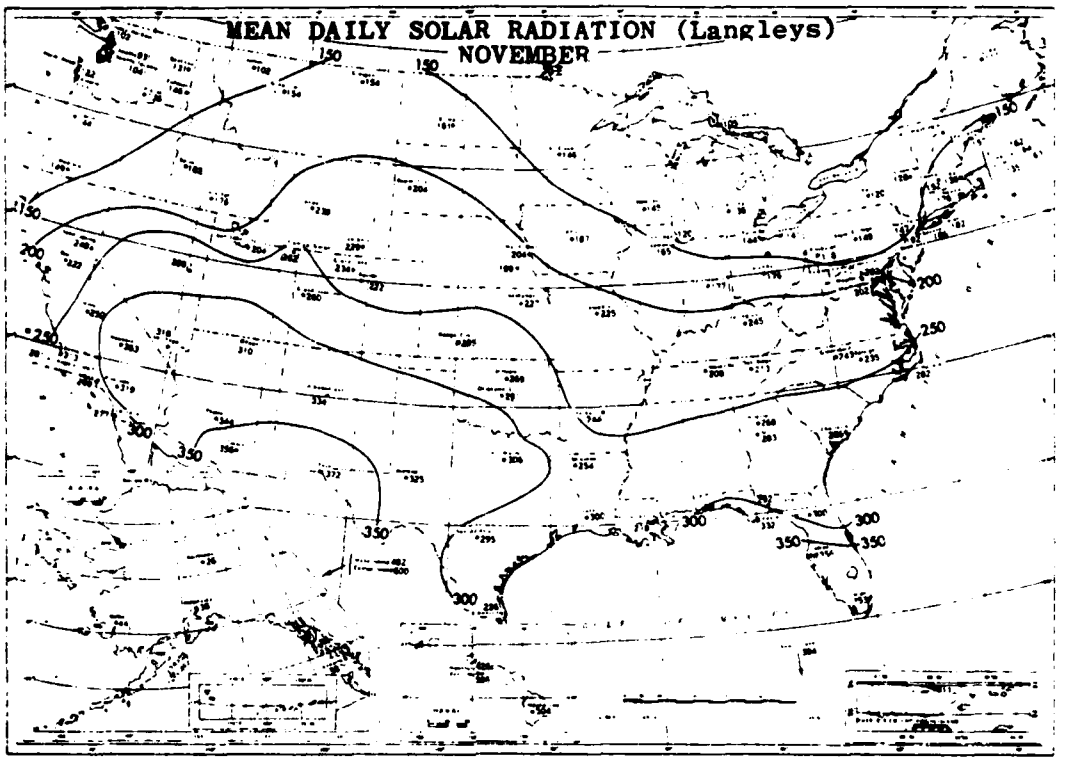

MEAN DAILY SOLAR RADIATION (Langleys)
JUNE


MEAN DAILY SOLAR RADIATION (Langleys)
JULY


MEAN DAILY SOLAR RADIATION (Langleys)
AUGUST


MEAN DAILY SOLAR RADIATION (Langleys)
SEPTEMBER


MEAN DAILY SOLAR RADIATION (Langleys)
OCTOBER



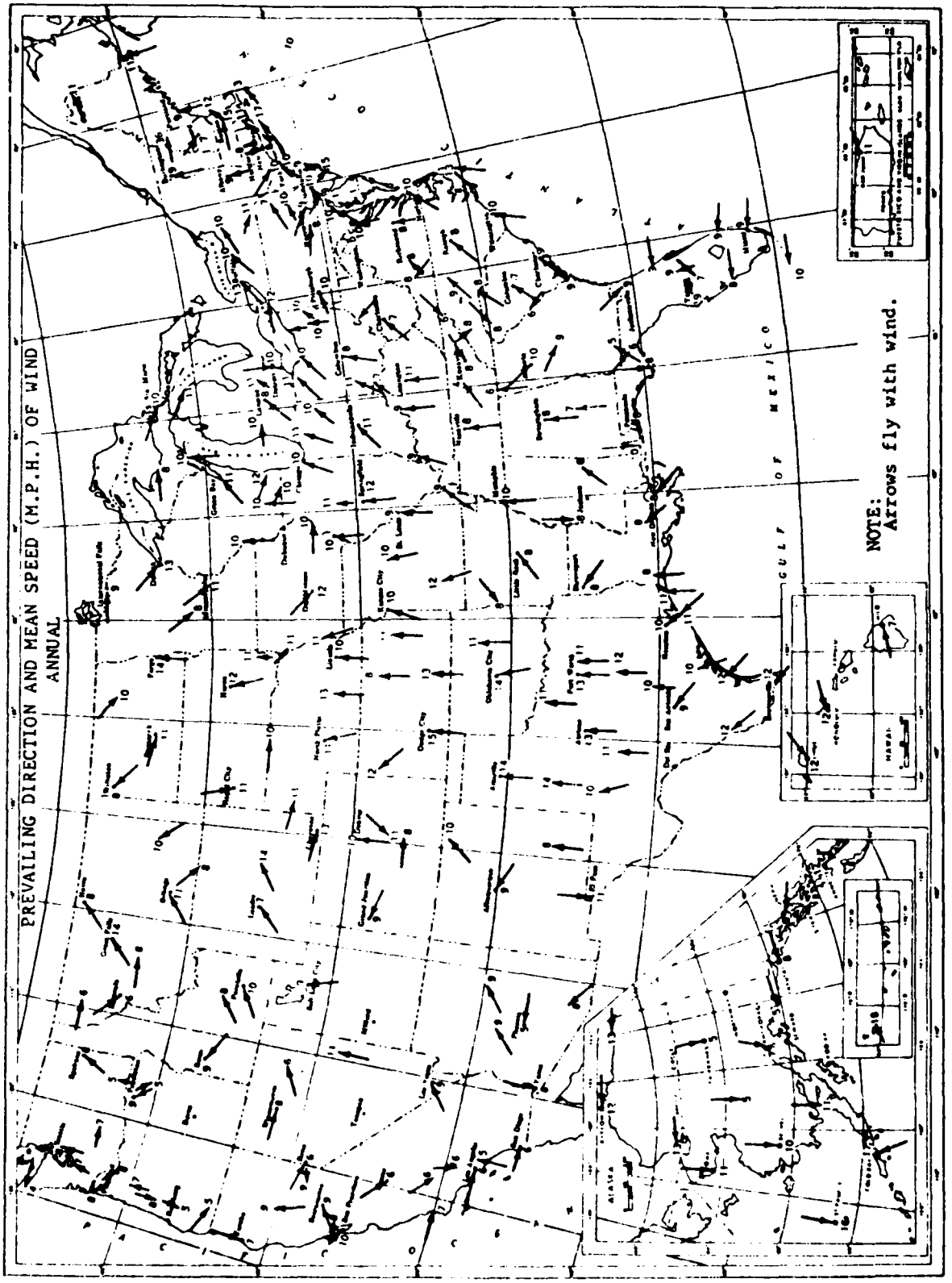
Scale for the 48 Contiguous States:

300 0 300 600 900 1200 1500 Miles

SCALE 1:30,000,000

ALBERS EQUAL AREA PROJECTION-STANDARD PARALLELS 29½° AND 45½°

APPENDIX D: WIND ENERGY AVAILABILITY MAP



Average Annual Wind Speed (mph) and Direction in the United States
(Source: U.S. Department of Commerce, 1979)

APPENDIX E: HEAT TRANSFER FLUIDS

Heat transfer fluids are used in solar systems to transfer heat from the solar collector to the storage medium and from the storage medium to the building. They are sometimes called "collector coolants" because they cool the collector as they absorb its heat. Mention of specific products in this appendix is for information only, and does not constitute an endorsement or recommendation.

COMPARISON OF HEAT TRANSFER FLUIDS BY GENERAL CHARACTERISTICS

Air

Air is one of the most commonly used transfer fluids in solar systems. It is free and will operate at any temperature the solar system will reach. Moreover, a leak in an air-based system will cause no damage, although it will degrade system performance. Since air has a low volumetric heat capacity, its flow rate through the system must be high. The power used to transfer a given amount of energy is higher for air than for most liquids. The major disadvantage of air is that it requires large duct size, which makes retrofitting difficult and provides more area for thermal losses. Air handling systems are also generally noisier than liquid-based systems.

Water

Water is a readily available fluid with good heat transfer properties (high heat capacity, high thermal conductivity, and low viscosity). Its major drawbacks are its high freezing temperature, its expansion upon freezing, and its corrosive effect on common engineering materials (except copper). Its low boiling point can cause high pressures within the collector system under zero flow conditions. Water has no adverse biological or environmental effects.

Ethylene Glycol

Other than water, the most commonly used heat transfer liquids in flat plate collectors are water/ethylene glycol solutions. These common, colorless, odorless antifreeze solutions are also used in many other applications. Ethylene glycol is relatively inexpensive and available from many manufacturers. (See listings in the *Thomas Register*.) With corrosion inhibitors, aqueous ethylene glycol solutions can reduce the corrosive action and freezing temperature of water. These solutions are usually available in a wide range of concentrations and inhibitor levels. The thermal properties of the solutions (heat capacity, thermal conductivity, and viscosity) are poorer than those of water.

The boiling and flash points of aqueous ethylene glycol mixtures are low and can be easily reached under zero flow conditions. Glycols can oxidize to organic acids (such as glycolic acids) when exposed to air near boiling temperatures. The inhibitors used are designed to neutralize these extremely corrosive acids. Periodic maintenance and addition of inhibitors must be done if these fluids are used. Another major drawback to the use of ethylene glycol is its high toxicity.[1] Most plumbing codes require that ethylene glycol solutions be separated from potable water by double-walled heat exchangers.

Propylene Glycol

Propylene glycol has properties similar to those of ethylene glycol, except that propylene glycol has higher viscosity and is less toxic. With inhibitors, propylene glycol can be used with most common engineering materials. Periodic maintenance must be performed and inhibitors must be added to limit corrosion. Propylene glycol will also form acids at high temperatures in oxygen-rich atmospheres. Because of its lower toxicity, propylene glycol has been widely used in the food industry. Most manufacturers who produce ethylene glycol also market propylene glycol. The higher viscosity of propylene glycol makes the heat transfer properties of aqueous propylene glycol mixtures poorer than those of ethylene glycol.

Other Glycols

Other glycol solutions have been used as heat transfer fluids in industrial applications. These include diethylene and triethylene glycol. With inhibitors, both of these fluids can be used with higher boiling points than ethylene glycol. The thermal properties of these aqueous solutions are similar to those of ethylene glycol at similar concentrations. The vapor pressure of each is slightly higher than that of ethylene glycol. Their toxicity is between that of ethylene and propylene glycol; their cost is slightly higher than that of ethylene and propylene glycol.

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[1] The U.S. Food, Drug and Cosmetic Act of 1938, a step in the formation of the U.S. Food and Drug Administraction (FDA), was prompted mainly by a poisoning episode in 1937 involving at least 73 deaths and perhaps as many as 107 deaths caused by diethylene glycol contained in a drug known as "Elixir Sulfanilamide" (according to W. G. Campbell). Diethylene glycol is somewhat less toxic than ethylene glycol.

Other glycol heat transfer compounds include polyalkylene glycols such as Ucon[2] brand fluids and Jeffox[3] brand fluids. With inhibitors, the corrosive action of these compounds upon common engineering materials can be reduced. They are low in toxicity and are available in a wide range of viscosities. Fluids of this type that are applicable to heat transfer purposes cost more than the other glycol compounds.

Petroleum (Mineral) Oils

Petroleum oils are also used as heat transfer fluids in industrial applications. They generally are designed to operate at high temperatures, although some are able to offer lower temperature operation. As a group, they have poorer heat transfer properties than water, with lower heat capacity and thermal conductivity and higher viscosity. The flash point and boiling point lie below possible zero flow temperatures of a collector. Upon exposure to air at high temperatures, these fluids are subject to oxidation and cracking, forming tars and other by-products that will reduce collector performance and increase corrosion. The toxicity of these fluids is generally low, and their prices are relatively low. Mobiltherm[4] Light brand fluid was chosen in this appendix as a good representative of this class of fluids for low temperature applications.

Silicone Fluids

Some flat plate collector installations have used silicone fluids for heat transfer. They are produced by Dow Corning and General Electric, among others. These fluids have low freezing and pour points, low vapor pressure, low general corrosiveness, good long term stability, and low toxicity. Their major drawbacks are high cost and high viscosity, causing poor heat transfer and requiring higher flow rates. Also, leakage through fittings can create problems because silicone fluids have lower surface tension than aqueous solutions. Joints and fittings must be tight to ensure that leakage is minimal.

Other Fluids

Another fluid for possible use in flat plate collectors is Dowtherm[5] J brand fluid. It is an alkylated aromatic compound with low viscosity, low heat capacity, and low thermal conductivity. It is relatively inexpensive but has low flash and fire points. Oxidation of Dowtherm J at high temperatures

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[2] Ucon is a trademark of Union Carbide Corporation.

[3] Jeffox is a trademark of Jefferson Chemical Company, Inc.

[4] Mobiltherm is a trademark of Mobil Oil Corporation.

[5] Dowtherm is a trademark of Dow Chemical Company.

upon exposure to air can lead to formation of insoluble materials and increased fluid viscosity. When the fluid is overheated, the flash point can be lowered and vapor pressure increased. If it is contaminated by other fluids (such as water), corrosion can be enhanced

(as in the case of water and steel). The toxicity of Dowtherm J is high. As with aqueous ethylene glycol solutions, double walls would be required to separate the potable water from the Dowtherm J.

Some other possible heat transfer fluids include Therminol[6] 44 brand ester-based fluid, Therminol 55 brand alkylated benzene fluid, and Therminol 60 brand hydrogenated aromatic fluid. They have low heat capacity, low thermal conductivity, high viscosity, and low freezing temperature. The flash points of these fluids are at the upper range of possible zero flow temperatures. The costs of Therminol 44 and 60 are relatively high, while Therminol 55 is much less costly.

Sun-Temp[7] brand fluid, a saturated hydrocarbon, is another possible heat transfer fluid available to flat plate collector users. It has low heat capacity, low thermal conductivity, high viscosity, a low freezing temperature, a high boiling temperature, low toxicity, low corrosiveness with aluminum, and low vapor pressure. It is relatively inexpensive. Because of its high viscosity, higher flow rates are required to produce turbulent flow and to increase heat transfer.

Inorganic aqueous salt solutions have recently been proposed for use as heat transfer fluids. According to K. W. Kauffman, 23-percent sodium acetate and 38-percent sodium nitrate aqueous solutions with suitable additives can be used as heat transfer fluids. These solutions have low toxicity. Their cost is comparable to that of ethylene glycol, and their heat transfer properties are similar to those of the glycols. Pumping costs for these fluids would be low. Like other aqueous solutions, they are subject to boiling at lower temperatures with high vapor pressures. These fluids are still being investigated for solar energy applications.

COMPARISON OF HEAT TRANSFER FLUIDS BY PHYSICAL PROPERTIES

The preceding discussion of heat transfer fluids pointed out general characteristics of each fluid studied. The following sections will discuss the following physical properties:

- Thermophysical properties.
- Cost.
- Toxicity.
- Flammability.
- Vapor pressure.
- Freeze protection.

[6]Therminol is a trademark of Monsanto Company.

[7]Sun-Temp is a trademark of Research Technology Corporation.

The heat transfer fluids discussed earlier will be compared to offer a quantitative description of probable performance in double-loop heat exchanger collector systems. In some sections, representative fluids are chosen for the comparison. For ethylene and propylene glycol a 50-percent aqueous solution with inhibitors is used, since this allows adequate freeze protection for most cases. For some applications, lower concentrations might be plausible; in such cases, the results found here will be slightly conservative for heat transfer and flow rate properties. Also, since the properties of diethylene and triethylene glycol are close to those of ethylene glycol, we did not consider it necessary to compare these fluids in every section.

Thermophysical Properties

The thermophysical properties of the fluids were found from the manufacturers' specifications over the operating temperature range of flat plate collectors. For heat transfer, water is the best fluid. It has a high heat capacity, high thermal conductivity, and low viscosity. Water and the other heat transfer fluids are compared in Figures E-1 through E-4 for the following thermophysical properties:

- Absolute viscosity.
- Heat capacity.
- Thermal conductivity.
- Density.

Generally, aqueous solutions (such as ethylene and propylene glycol) have better thermophysical properties than do the rest of the heat transfer fluids except Dowtherm J. Dowtherm J has a lower viscosity than glycol solutions but also has lower heat capacity and thermal conductivity. Other simple comparisons of the heat transfer fluids can be made from Figures E-1 through E-4.

Cost

In some applications, more expensive fluids can be competitive with less costly ones. In order to determine the relative cost of a heat transfer fluid, you must know the volume of fluid required for a particular application. For some applications (such as domestic hot water heating), the amount of heat transfer fluid required will be small, since the collector area needed is small. In traced tank systems more costly fluids can be used if their other properties are desirable.

Table E-1 shows the 1978 costs of many heat transfer fluids in single 55-gallon drum quantities. Note that the final costs will generally be lower for the glycol solutions, since it is not necessary to use them in 100-percent solutions. Thus Mobiltherm Light and the glycols are the least expensive heat transfer fluids for initial installation, while the silicone fluids are the most expensive.

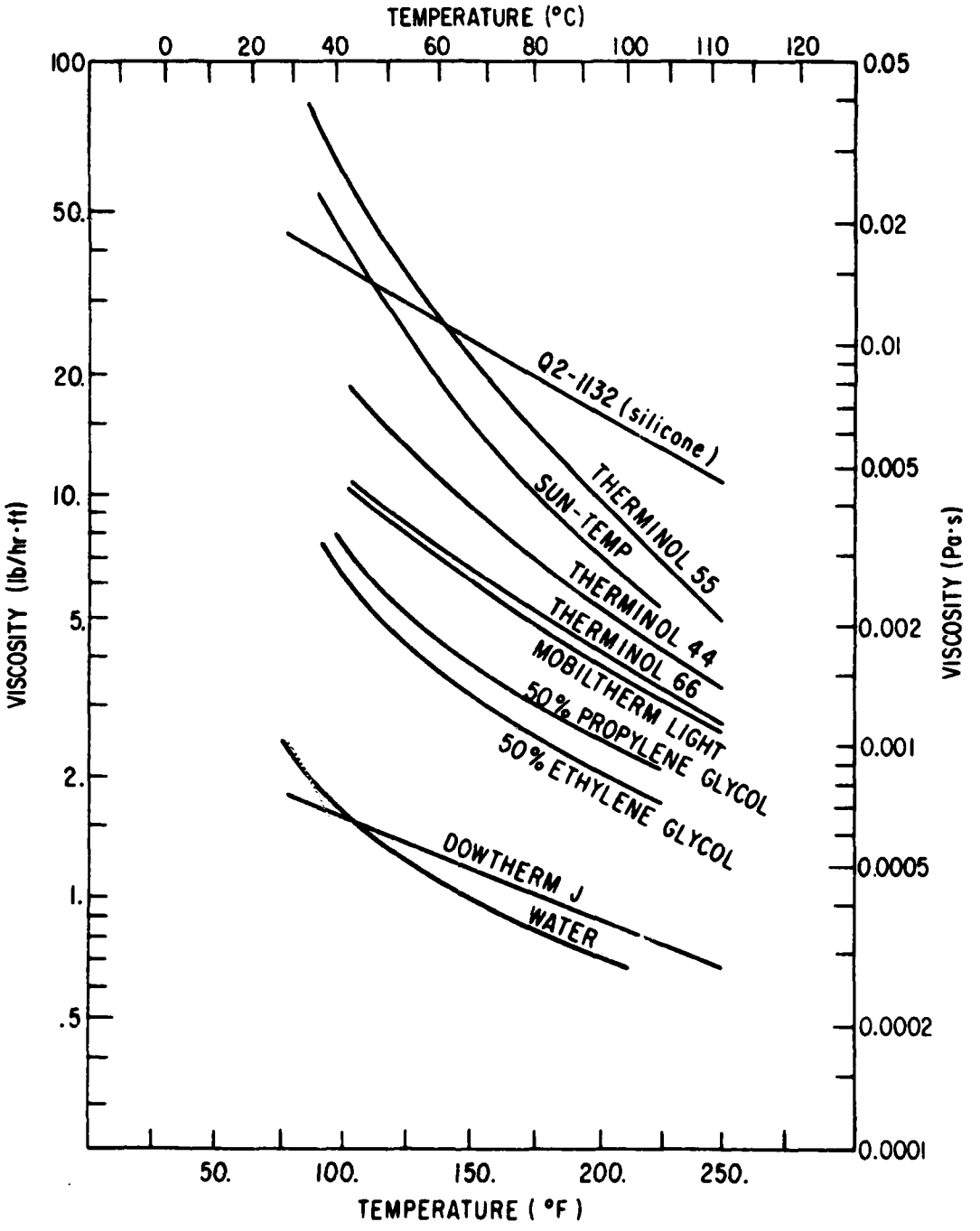


Figure E-1. Viscosity of Heat Transfer Fluids versus Temperature (Multiply viscosity in centipoise by 2.419×10^{-4} to get viscosity in lb/ft·hr.)

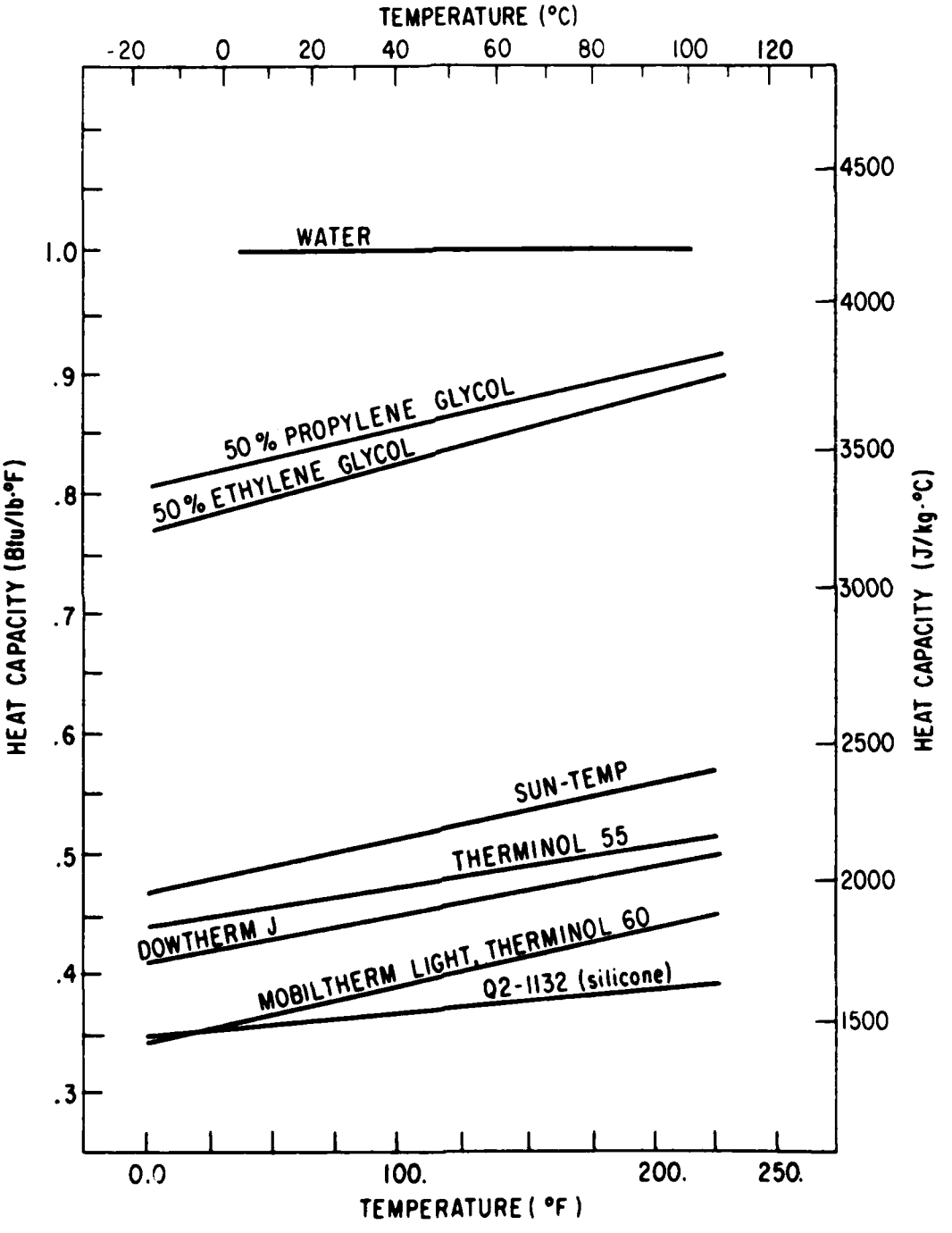


Figure E-2. Heat Capacity of Heat Transfer Fluids versus Temperature

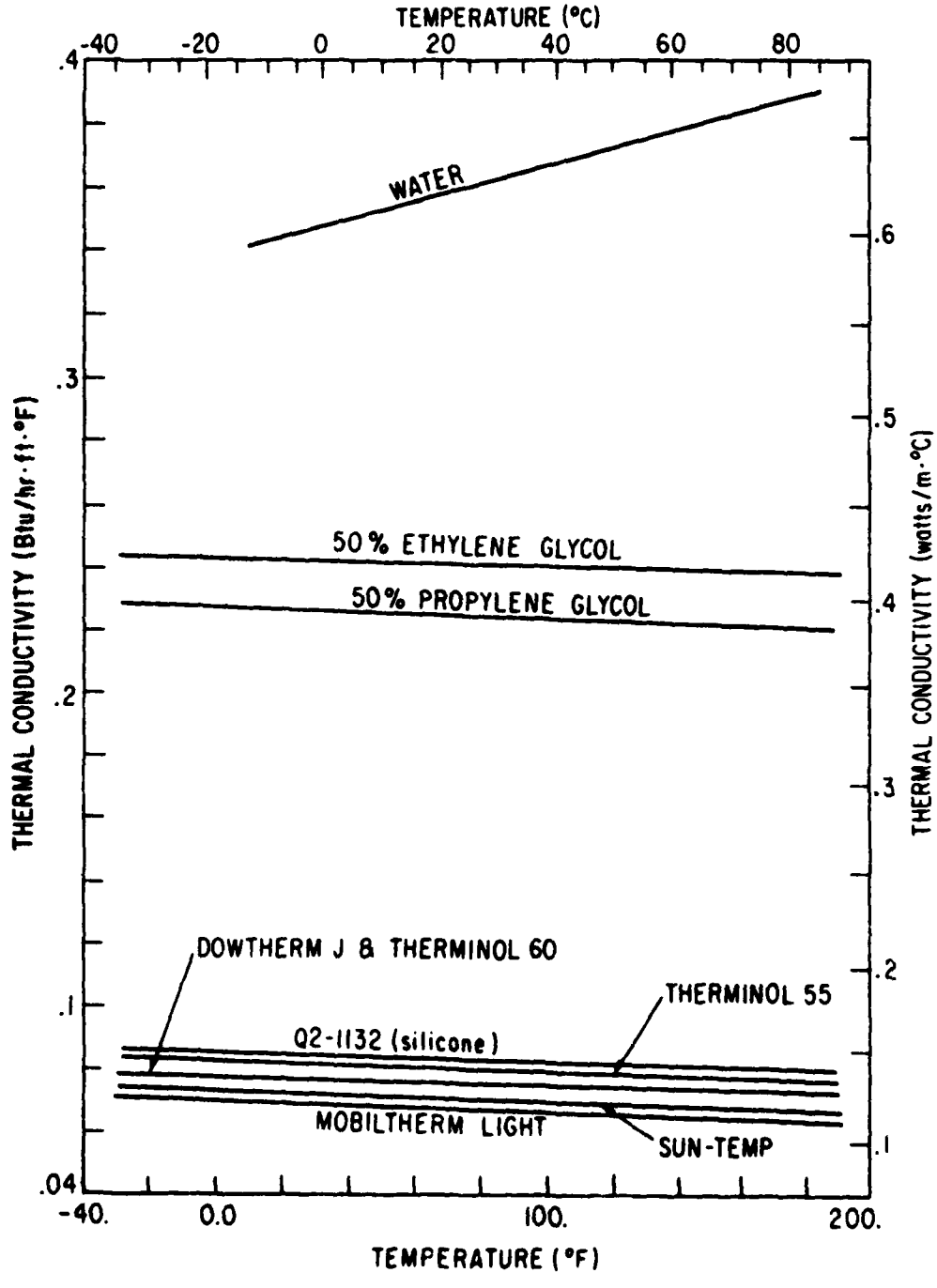


Figure F-3. Thermal Conductivity of Heat Transfer Fluids versus Temperature

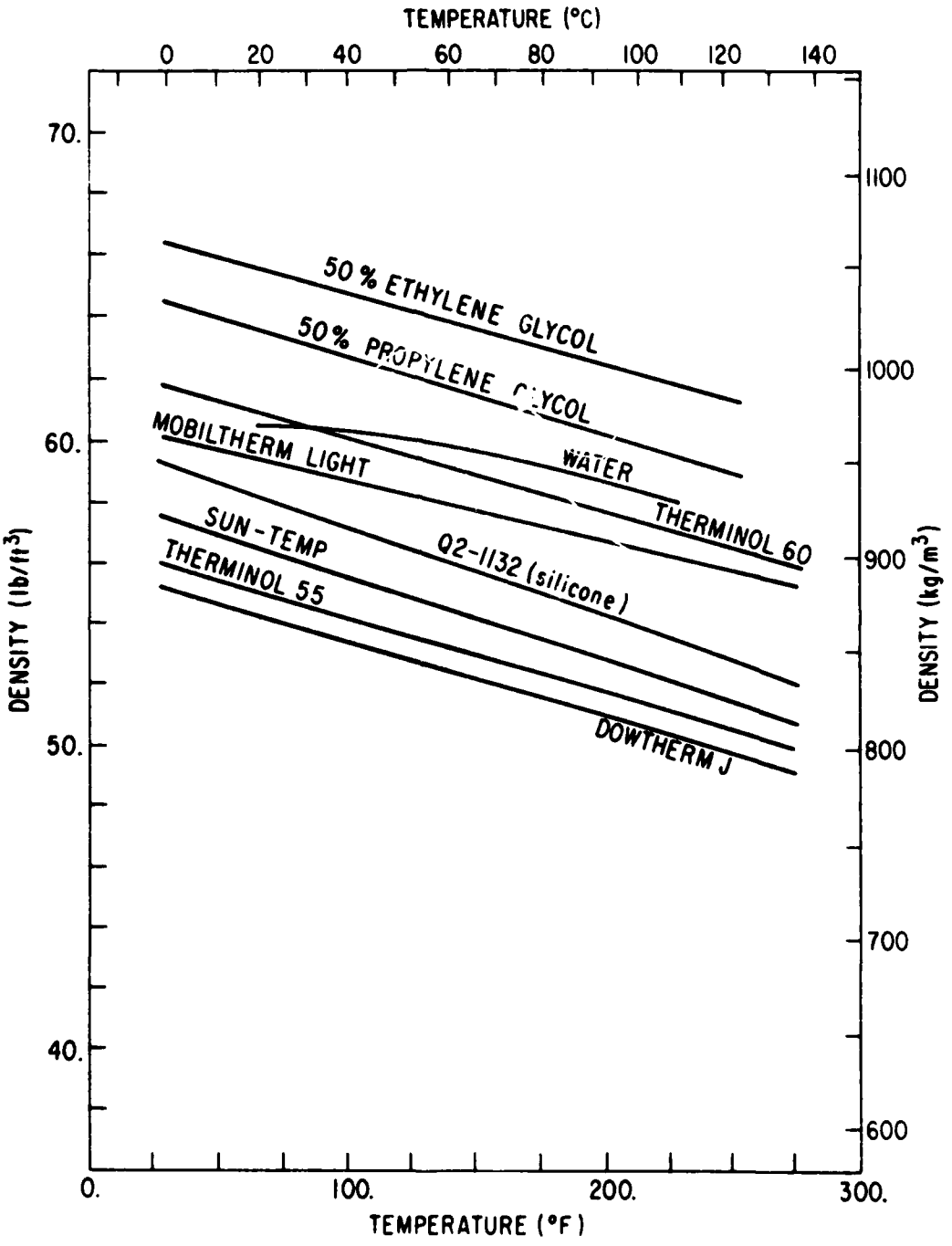


Figure E-4. Density of Heat Transfer Fluids versus Temperature

Table B-1. Initial Fillup Cost of Heat Transfer Fluids

Fluid	Cost per Gallon (single 55-gallon drum quantities)	Manufacturer
Water	--	--
100% Ethylene Glycol	$ 2.56	Union Carbide
100% Propylene Glycol	2.45	Union Carbide
100% Diethylene Glycol	2.82	Union Carbide
100% Triethylene Glycol	3.70	Union Carbide
100% Ucar[a] Thermofluid (ethylene glycol and inhibitors)	3.81	Union Carbide
100% Ucar Foodfreeze (propylene glycol and inhibitors)	3.63	Union Carbide
100% Dowtherm[b] SR-1 (ethylene glycol and inhibitors)	3.65	Dow Chemical
100% Dowfrost[b] (propylene glycol and inhibitors)	3.45	Dow Chemical
Mobiltherm[c] Light	1.29	Mobil Oil
SF-96(50) (silicone)	14.00	General Electric
Q2-1132 (silicone)	23.00	Dow Chemical
Dowtherm J	4.50	Dow Chemical
Therminol[d] 44	7.65	Monsanto
Therminol 55	2.80	Monsanto
Therminol 60	6.80	Monsanto
Sun-Temp[e]	3.50	Resource Technology Corporation

[a]Ucar is a trademark of Union Carbide Corporation.

[b]Dowtherm and Dowfrost are trademarks of Dow Chemical Corporation.

[c]Mobiltherm is a trademark of Mobil Oil Corporation.

[d]Therminol is a trademark of Monsanto Company.

[e]Sun-Temp is a trademark of Resource Technology Corporation.

There are other costs besides those of the initial fillup. Periodic maintenance and inhibitor addition, if needed, can add to the total cost of the fluid over a specific time period. Where inadequate corrosion and freeze protection might lead to collector failure, this additional cost must be considered. Also, more viscous fluids will require higher flow rates and increased pumping costs. The total investment in fluid over a given time period is equal to the sum of the initial cost of the fluid plus any additional costs of added fluid or inhibitor, increased pumping costs, maintenance, cost of replaced parts needed because of inadequate freeze or corrosion protection, or cost of reserve draindown or expansion tanks needed for some fluids.

Toxicity

The toxicity of a heat transfer fluid can greatly affect the design and operation of a double-loop flat plate collector system. Most plumbing codes require that double walls or vented surfaces separate a toxic fluid from potable water supplies. The possibility of poisonous fumes escaping from the heat transfer fluid must also be considered. These problems require the use of heat exchangers that transfer heat less optimally than those that can be used with nontoxic fluids. The following discussion describes the toxicity of the heat transfer fluids studied. The information was obtained from the manufacturers.

In a discussion of toxicity, the following definitions (from *United States Codes Annotated, 1974*) are useful.

A hazardous substance is any substance or mixture of substances that:

- Is toxic.
- Is corrosive (will cause destruction of living tissue by chemical action).
- Is an irritant.
- Is a strong sensitizer.
- Is flammable or combustible.
- Generates pressure through decomposition, heat, or other means.

A toxic substance is any substance that has the capacity to produce injury or illness to man through ingestion, inhalation, or absorption through any body surface.

A highly toxic substance is any substance that produces death within 14 days in half or more than half of a group of ten or more laboratory white rats, each weighing between 200 and 300 grams, at a single dose of 50 milligrams or less per kilogram of body weight when orally administered or when inhaled continuously for a period of 1 hour or less at an atmospheric concentration of 200 parts per million by volume or less of gas or vapor or 2 milligrams per liter by volume or less of dust or mist.

LD_{50} refers to the quantity of chemical substance that kills 50 percent of dosed animals within 14 days. Dosage is expressed in grams or milliliters per kilogram of body weight.

Single dose (acute) oral LD_{50} refers to the quantity of substance that kills 50 percent of dosed animals within 14 days when administered orally in a single dose.

Because the primary hazard in using heat transfer fluids is the possibility that they may leak into a potable water supply and be ingested, acute oral toxicity is the primary concern in this section. Table E-2 lists the LD_{50} values for selected fluids for acute oral toxicity. No substance listed is highly toxic according to the preceding definition, but several are quite toxic. Dowtherm J is the most toxic fluid listed in Table E-2, with the ethylene glycol mixture second. The least toxic fluids are silicone fluids, Sun-Temp, and propylene glycol.

Flammability

The possibility of the heat transfer fluid being a fire hazard must be considered. In a discussion of the flammability of a heat transfer fluid, the following definitions are useful.

Boiling point is the temperature at which the vapor pressure of a liquid equals the absolute external pressure at the liquid-vapor interface.

Flash point is the lowest temperature at which a combustible vapor above a liquid ignites and burns when ignited momentarily in air.

Fire point is the lowest temperature at which a combustible vapor flashes and burns continuously.

Self-ignition point is the temperature at which self-sustained ignition and combustion in ordinary air take place independent of a heating source.

Extremely flammable describes any substance that has a flash point at or below 20°F as determined by the Togliabue Open Cup Tester (TOCT).

Flammable describes any substance that has a flash point between 20°F and 80°F as determined by the TOCT.

Combustible describes any substance that has a flash point between 80°F and 150°F as determined by the TOCT.

Table E-3 lists the fluids and their boiling or flash points, whichever were supplied by the manufacturers. None of the fluids listed are extremely flammable or flammable. Only Dowtherm J is combustible, with a flash point of 145°F. With the exception of the silicone fluids, Sun-Temp, and Therminol

Table E-2. Acute Oral Toxicities of Heat Transfer Fluids

Fluid	LD_{50} (grams/kilogram of body weight)
Water	--
100% Ethylene Glycol (no inhibitors)	8.0
100% Propylene Glycol (no inhibitors)	34.6
100% Diethylene Glycol (no inhibitors)	30.0
100% Triethylene Glycol (no inhibitors)	30.0
100% Dowtherm SR-1	4.0
Mobiltherm Light	20.0
SF-96(50) (silicone)	50.0
Q2-1132 (silicone)	50.0
Dowtherm J	1.1
Therminol 44	13.5
Therminol 55	15.8
Therminol 60	13.0
Sun-Temp	No test information available

Table E-3. Flammability of Heat Transfer Fluids

Fluid	Boiling Point °F (°C)	Flash Point, °F (°C) (Cleveland Open Cup)
Water	212 (100)	--
100% Ethylene Glycol	388 (198)	240 (116)
50% Ethylene Glycol	225 (107)	--
100% Propylene Glycol	370 (188)	225 (107)
100% Diethylene Glycol	475 (246)	290 (143)
100% Triethylene Glycol	550 (288)	330 (166)
100% Dowtherm SR-1	325 (163)	240 (116)
50% Dowtherm SR-1	230 (110)	--
100% Dowfrost	--	214 (101)
Mobiltherm Light	250 (121)	--
SF-96(50)	--	600 (316)
Q2-1132	--	450 (232)
Dowtherm J	--	145 (63)
Therminol 44	425 (218)	405 (207)
Therminol 55	600 (316)	355 (179)
Therminol 60	650 (343)	310 (160)
Sun-Temp	500 (260)	310 (160)

44, most of the fluids have flash points below possible stagnation temperatures, a potential hazard.

The HUD Minimum Property Standards for FHA eligibility, according to K. W. Kauffman, preclude the use of fluids whose flash points are not at least 100°F higher than the highest temperature to which they might be exposed. Thus the use of fluids with low flash points is limited unless adequate safeguards limit the exposure of these fluids to high temperatures and exposure to the atmosphere.

Vapor Pressure

Under zero flow conditions within the collectors, temperatures may exceed 300°F. For aqueous solutions the vapor pressure under stagnation conditions can reach several atmospheres. Some collectors cannot withstand these pressures. Figure E-5 shows the absolute vapor pressure versus temperature for several of the fluids. The vapor pressures of the fluids are quite low, even under zero flow conditions, except for the aqueous solutions and Dowtherm J.

Freeze Protection

One of the major drawbacks of using water as a heat transfer fluid is its high freezing temperature. In the continental United States, few locations have had no recorded below-freezing temperatures.

Antifreeze solutions have been commonly added to water to lower its freezing temperature. In some cases these solutions can retard the expansivity of the water and create a slush that will not rupture the fluid vessel. Most nonaqueous fluids do not expand upon freezing and thus will reduce the risk of damaged piping.

Because some fluids become so viscous that their freezing temperatures are not easily measured, the pour point temperatures of the fluids are used as their lower operating limits. The pour point temperature is the temperature of the fluid at which it fails to flow when the container is tilted to horizontal and held for 5 seconds.

Freeze protection temperatures can best be obtained from the manufacturer for the particular fluid in question.

PRESSURE DROPS IN PIPES AND HEAT EXCHANGERS

One of the important parameters to be considered in selecting a heat transfer fluid is the operating pressure drop caused by friction in the fluid loop. This parameter will determine what size pump you buy and how much power will be needed to circulate the fluid.

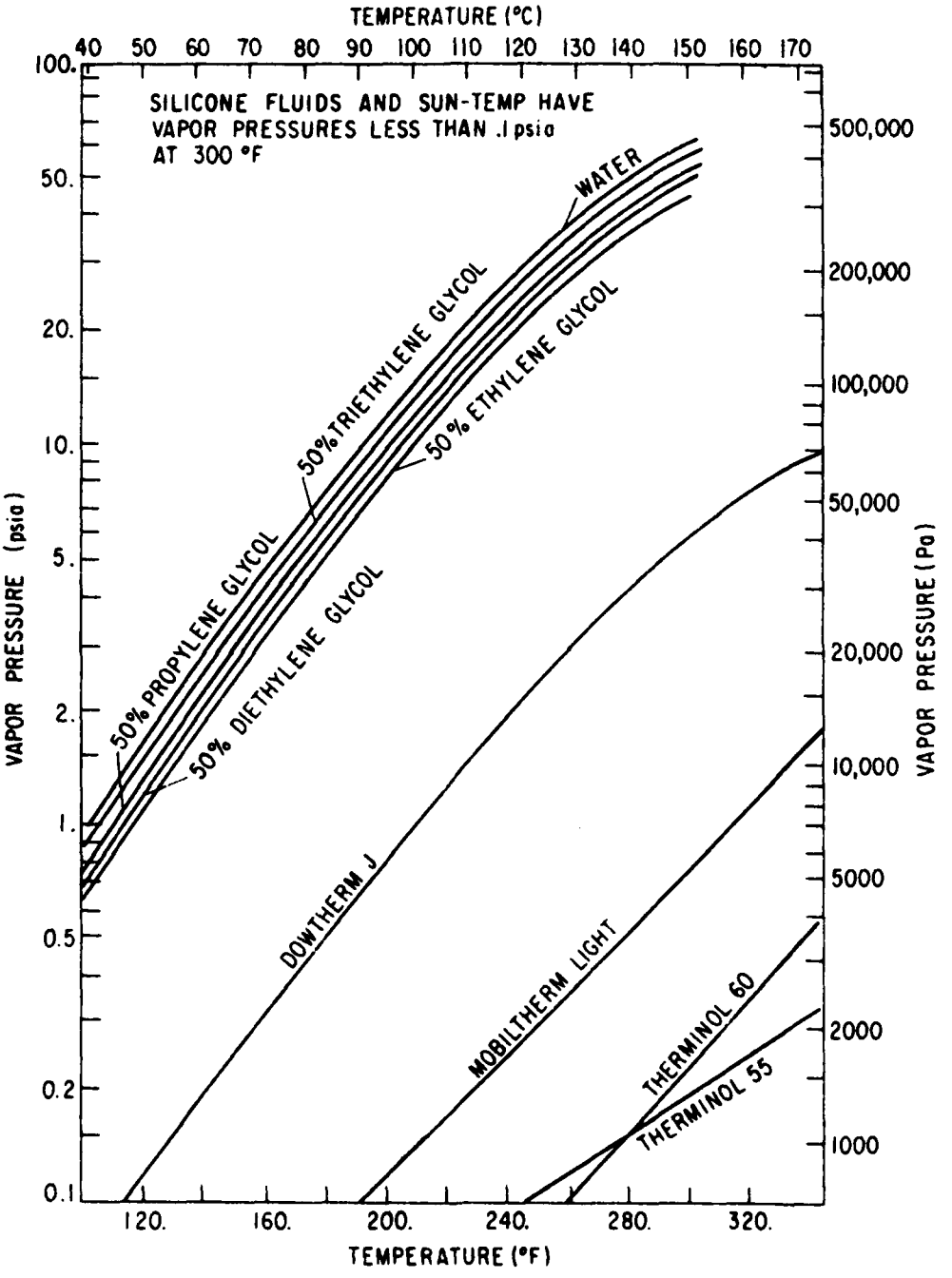


Figure E-5. Vapor Pressure of Heat Transfer Fluids versus Temperature

The Darcy-Weisbach equation gives the pressure drop per unit length in a tube:

$$\Delta p = f\frac{L}{D_i}\frac{\rho V^2}{2g} \qquad (E-1)$$

where:

Δp is the pressure drop in lbf/ft^2 (Pa).

L is the length of the tube in feet (meters).

D_i is its inside diameter in feet (meters).

ρ is the density of the fluid in lb/ft^3 (kg/m^3).

V is the fluid velocity in ft/hr (m/s).

g is the unit conversion factor,[8] 4.17×10^8 lb·ft/lbf·hr^2 (1.0 when metric units are used).

f is the friction factor, which is related to the Reynolds number, Re, in the following way:

$$f = \begin{cases} 16/Re < 2500 \text{ (laminar flow).} \\ 0.0014 + 0.125/Re^{0.32} \text{ for } Re > 2500 \text{ (turbulent flow).} \end{cases}$$

The Reynolds number is:

$$Re = \frac{\rho V D_i}{\mu} \qquad (E-2)$$

where μ is the fluid's absolute viscosity in lb/ft·hr (Pa·s).

The fluid velocity, V, is equal to the volume flow rate divided by the cross-sectional area of the tube:

$$V = \frac{4\,q}{\pi\,D_i^2} \qquad (E-3)$$

where q is the volume flow rate in ft^3/hr (m^3/s).

---

[8] See Appendix A for a discussion of units and conversions. Notice that the problem of converting pounds mass to pounds force is avoided in the metric system.

Combining Equations E-1 and E-3 gives the following equation:

$$\Delta p = \frac{8}{\pi^2} \frac{f \rho L q^2}{g\ D_i^5} \quad . \qquad \text{(E-4)}$$

This equation shows that the inside tube diameter, D_i, greatly affects the pressure drop per unit length within the tube. Replacing 1-inch pipe with 3/4-inch pipe will give more than four times the pressure drop in the pipe for a given flow rate.

When calculating the pressure drop in a system, add together the pressure drops of each component to get the total pressure drop. *Flow of Fluids* by the Crane Company and the *ASHRAE Handbook of Fundamentals* give the pressure drops in equivalent lengths of pipe for elbows and other types of pipe joints and for valves.

For shell and tube heat exchangers, the tube side pressure drop can be calculated by using the formulas given above. The shell side pressure drop can be calculated by using the following equation from D. Q. Kern:

$$\Delta p = \frac{8\ f \rho q_{max}^2\ (n_{baf} + 1)}{\pi^2 g\ D_s^3\ D_o} \qquad \text{(E-5)}$$

where:

q_{max} is the maximum volumetric flow rate in ft^3/hr (m^2/s) as defined in Appendix D.

n_{baf} is the number of baffles within the heat exchanger.

D_s is the inside diameter of the shell in feet (meters).

D_o is the outside diameter of the tubes in feet (meters).

The rest of the symbols are as defined previously.

SYMBOLS USED IN APPENDIX E

Main Symbols

D	tube diameter, ft (m)
f	friction factor
g	unit conversion factor, 4.17×10^8 lb·ft/lbf·hr^2 (1.0 if metric units are used)
L	length, ft (m)
n	number
q	volume flow rate, ft^3/hr (m^3/s)
Re	Reynolds number
V	fluid velocity, ft/hr (m/s)
ρ	fluid density, lb/ft^3 (kgm/m^3)
Δp	pressure drop, lb/ft^2 (Pa)
μ	absolute fluid viscosity, lb/ft·hr (Pa·s)

Subscripts

baf	baffles
i	inside of tube
max	maximum of several possible quantities
o	outside of tube
s	shell of heat exchanger

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APPENDIX F: PROPERTIES OF SOLID SENSIBLE HEAT STORAGE MATERIALS

storage medium	density lb/ft^3 ρ	heat capacity BTU/lb_mF C_p	volume heat capacity $Btu/ft.F$ PC_p	thermal conductivity $Btu/hr.ft.F$ K	thermal diffusivity ft^2/hr K/C_p
concrete	140	0.27	37.8	0.5-0.7	0.02
sand					
dry	95	---	---	0.2	---
10% wet	100	---	---	0.6	---
soil					
dry	90	0.20	---	0.20	---
wet	117	0.30	---	1.4	---
lime	105	0.22	23.10	0.4	0.02
stone					
granite	165	0.21	34.7	1-2.3	0.05
fireclay brick	144	0.20	28.80	0.6	0.02
chrome brick	246	---	---	0.8	---
magnesia brick	187	0.27	50.50	2.9	0.06
castiron brick	494	0.20	98.60	16.9	0.17
castiron	474	0.11	52.14	27.1	0.63
mildsteel	490	0.11	53.90	26.5	0.49
stainless steel	488	0.11	53.68	8.0	0.15
magnesium	109	0.23	25.59	91	3.60
Fe_2O_3	326	0.18	58.69	1.7	0.03
aluminum	169	0.21	35.15	117.6	3.33
Al_2O_3	250	0.20	49.94	1.5	0.03
brass	532	0.09	48.94	56	1.14
bronze	540	0.08	44.28	15	0.34
glass	170	0.20	34.00	0.5	0.01

APPENDIX G: ECONOMIC INSULATION THICKNESS

This appendix shows how you can calculate the economic thickness of insulation as an alternative to the SMACNA or HUD standards described in Chapter 1. To use the method, you must determine the collector and storage unit size by calculation as outlined in Chapter 1, and you should understand elementary calculus. In many cases, the thickness determined by this method will not differ greatly from the thickness required by the SMACNA standard limiting the loss of heat to 2 percent in 12 hours.

The basis of the method is to compare the cost of collecting an additional unit of energy with the cost of saving an equal unit of energy. If the cost of collecting energy is more than the cost of saving it, you should add insulation; but if the cost of collecting energy is less than the cost of saving it, you have too much insulation. The economic thickness of insulation is the one that makes the cost of saving energy equal to the cost of collecting it.

COST OF COLLECTING ENERGY

When you calculate the optimum collector area for your system using f-Chart, SOLCOST, or other methods, it is relatively easy to add an increment of collector area, ΔA_c, to the optimum collector area and determine the increment of useful energy ΔQ_c, the added collector area adds over a year's time. Since you supplied the cost data necessary to calclate the optimum collector area, you can easily determine the added cost to the customer, ΔC_c, of the added collector area. An economic factor, E_c, uses mortgage rates, inflation rates, length of service, tax rates, and the like to convert the customer's initial cost for the solar collector to an annual equivalent cost basis.

The ratio $E_c \Delta C_c / \Delta Q_c$ is the incremental cost of collecting energy in units of dollars per Btu (dollars per joule). It is the quantity that will be compared with the cost of saving energy. Later we will assume that the cost of saving energy will be calculated on the same economic basis (that is, the same mortgage rate, inflation rate, length of service, and so on) as the cost of collecting energy so that the economic factor can be ignored.

INSULATION ON FLAT SURFACES

Heat Loss

If the surface to be insulated is flat or its radius of curvature is large compared to the insulation thickness (a large tank, for example), the following analysis can be used. The energy lost from the surface, Q_L, is approximately:

$$Q_L = - \frac{At\ (\bar{T} - \bar{T}_a)}{R^* + sr^*} \quad . \qquad (G\text{-}1)$$

In Equation G-1, R^* is the thermal resistance of the fixed layers of insulation in °F·ft^2·hr/Btu (°C·m^2/W), s is the thickness in inches (centimeters) of the layer of insulation, and r^* is the thermal resistance per unit thickness of that layer in °F·ft^2·hr/Btu·in (°C·m^2/W·cm). The asterisks on R^* and r^* are a reminder that these quantities must be adjusted for parallel heat losses. A is the area of the surface to be insulated, and t is the total time in hours (seconds) the system will be heated during a year. If the storage unit is used for heating domestic hot water, it will probably be hot for the entire year, or 8760 hours (31.5 x 10^6 seconds); but if the system is used only for space heating, the storage unit will probably be hot only during the heating season. Ducts, pipes, and heat exchangers will be hot only while the collector is operating or while the building is being heated from storage.

$\bar{T}$ is the average storage, heat exchanger, pipe, or duct temperature during time t. The bar over the T indicates that it is averaged over the time the unit is operating. A reasonable estimate for $\bar{T}$ is the average of the maximum temperature, T_{max}, and the minimum temperature, T_{min}, while the unit is operating. $\bar{T}_a$ is the ambient temperature averaged during the time the unit is operating. The minus sign preceeding Equation G-1 indicates that the heat is lost from the system.

The derivative of the heat loss equation with respect to the thickness of the insulation layer will be used in a later step:

$$\frac{dQ_L}{ds} = \frac{Atr^*\ (\bar{T} - \bar{T}_a)}{(R^* + sr^*)^2} \quad . \qquad (G\text{-}2)$$

Cost of Insulation

The cost of the insulation, C_I, is assumed to include a fixed cost, C_1, and a cost that is proportional to the amount of insulation used:

$$C_I = C_1 + sAc \quad . \qquad (G\text{-}3)$$

The factor s is the thickness of the insulation layer, A is the area covered, and c is the installed cost to the customer per unit of area per unit of thickness (dollars per square foot per inch or dollars per square meter per centimeter). Differentiating Equation G-3 with respect to insulation layer thickness yields

$$\frac{dC_I}{ds} = Ac \quad . \qquad (G\text{-}4)$$

The incremental cost of saving energy which must be compared to the incremental cost of collecting energy is

$$E_I \frac{dC_I}{dQ_L} = \frac{E_I \, dC_I/ds}{dQ_L/ds} = \frac{E_I \, c(R^* + sr^*)^2}{(\bar{T} - \bar{T}_a)tr^*} \cdot \qquad \text{(G-5)}$$

E_I is an economic factor that converts the customer's initial cost for the insulation to an annual equivalent cost basis. Since the insulation and the collectors are part of the same solar system, they will have the same mortgage rate, inflation rate, tax rate, and length of service; therefore, we can assume that E_I is equal to E_c. When the incremental cost of collecting energy is equated to the incremental cost of saving energy, the two economic factors will cancel and can be neglected. If we had wanted to compare the cost of saving energy with the cost of conventional energy, it would not have been possible to neglect the economic factors in this manner.

The last step in deriving of the most economic thickness of an insulation layer is to equate the incremental cost of collecting energy with the incremental cost of saving energy and solve for the thickness:

$$s = \left[\frac{t(\bar{T} - \bar{T}_a)\Delta Cc}{c \; r^* \; \Delta Qc}\right]^{1/2} - \frac{R^*}{r^*} \cdot \qquad \text{(G-6)}$$

Example G-1: Determine the economic insulation thickness for a rock bed with walls similar to those described in Examples 4-4 and 4-5. Assume the collector sizing calculations show that by adding 53.8 square feet (5 square meters) to the collector at an added cost of $1076 you can collect an additional 5.05×10^6 Btu (5.32×10^9 joule) of usable energy per year. Since the heating season is 8 months long, the rock bed will be in use 5840 hours (21.0×10^6 seconds) per year. The average rock bed temperature is 115°F (46°C), and the rock bed is located in a basement where the ambient temperature is 68°F (20°C).

The thicknesses of gypsum board and plywood given in Example 4-4 contribute only to the fixed cost of the insulation, since they will be present regardless of the stud width. The method of changing the amount of insulation for this example is to change the width of the studs and the corresponding thickness of the fiberglass.

A 5-1/2-inch thickness of fiberglass has an R-value of

$$R_1 = 5.5 \times 3.15 = 17.33\ ^\circ F \cdot ft^2 \cdot hr/Btu\ (3.05\ ^\circ C \cdot m^2/W),$$

and a 2 x 6 stud has an R-value of

$$R_2 = 5.5 \times 1.25 = 6.88\ ^\circ F \cdot ft^2 \cdot hr/Btu\ (1.21\ ^\circ C \cdot m^2/W).$$

With the stud layout shown in Figure 4-2, the fiberglass-and-stud layer has a thermal transmittance (see Example 4-4) of

$$U = \frac{1}{16}\ \frac{13.12}{17.33} + \frac{2.88}{6.88}$$

$$= 0.0735\ Btu/hr \cdot ft^2 \cdot ^\circ F\ (0.417\ W/m^2 \cdot ^\circ C).$$

The adjusted thermal resistance per unit thickness, r^*, is

$$r^* = \frac{1}{Us} = \frac{1}{5.5 \times 0.0735}$$

$$= 2.47\ ^\circ F \cdot ft^2 \cdot hr/Btu \cdot in\ (0.172\ ^\circ C \cdot m^2/W \cdot cm).$$

The plywood, gypsum board, and surface resistance contribute $R^* = 2.52\ ^\circ F \cdot ft^2 \cdot hr/Btu\ (0.444\ ^\circ C \cdot m^2/W)$ to the insulation.

We assume that the difference in cost between using 2 x 6 studs and using 2 x 8 studs with thicker insulation is $0.36 per square foot, which gives a cost for the insulation layer of

$$c = \$0.18/ft^2 \cdot in\ (\$0.76/m^2 \cdot cm).$$

Substituting these figures into Equation G-6 gives the economic thickness of the stud-and-insulation layer:

$$s = \left[\frac{5840 \times (115-68) \times 2076}{0.18 \times 2.47 \times 5.05 \times 10^6}\right]^{1/2} - \frac{2.52}{2.47}$$

$$= 14.9\ in\ (37.9\ cm).$$

You should notice that two of the four prerequisites for relaxing the insulation requirement to the HUD standard have been satisfied in the statement of the example. That is, the rock bed is located inside a building, and collected heat will not be added to storage in the summer. If the rock bed will

not be used to heat domestic hot water, and occasional overheating near the rock bed in Spring and Fall can be tolerated, you can use the HUD standard allowing a 10 percent loss in 24 hours. This will allow you to use a 2 x 6 stud wall as shown in Example 4-4.

INSULATION ON A CYLINDRICAL SURFACE

Calculating the optimum insulation thickness for a cylindrical surface is more complicated than calculating for a flat surface, because both the heat loss equation and the insulation cost equation have more terms. Although we could not find a closed-form solution, we have included an equation that can be solved by trial-and-error or by numerical methods.

Heat Loss

The insulation is assumed to have three parts--an inside surface resistance, R_i, in °F·ft^2·hr/Btu (°C·m^2/W); a layer of insulation whose thickness, s inches (centimeters), we want to determine; and an outside surface resistance, R_o, in °F·ft^2·hr/Btu (°C·m^2/W). The heat loss is

$$Q_L = - \frac{\frac{\pi}{12} Lt \, (\bar{T} - \bar{T}_a)}{\frac{R_i}{D} + \frac{r^* \ln\left(1 + \frac{2s}{D}\right)}{2} + \frac{R_o}{D\left(1 + \frac{2s}{D}\right)}} \qquad \text{(G-7)}$$

where:

L is the length of the surface to be insulated in feet (meters).

t is the annual amount of time the cylinder will be hot in hours (seconds).

$\bar{T}$ is the average temperature of the cylinder in °F (°C).

$\bar{T}_a$ is the average ambient temperature surrounding the cylinder in °F (°C).

D is the diameter of the cylinder in inches (centimeters).

r^* is the thermal resistance of the insulation per unit thickness in °F·ft^2·hr/Btu·in (°C·m^2/W·cm) adjusted for parallel heat loss.

Taking the derivative of the heat loss with respect to insulation thickness yields an equation which will later be combined with the derivative of the cost:

$$\frac{dQ_L}{ds} = \frac{\frac{\pi}{12}Lt(\bar{T} - \bar{T}_a)\left[\frac{r^*}{D\left(1 + \frac{2s}{D}\right)} - \frac{2\,R_o}{D^2\left(1 + \frac{2s}{D}\right)^2}\right]}{\left[\frac{R_i}{D} + \frac{r^* \ln\left(1 + \frac{2s}{D}\right)}{2} + \frac{R_o}{D\left(1 + \frac{2s}{D}\right)}\right]^2}. \qquad \text{(G-8)}$$

Cost of Insulation

The cost of the insulation, C_I, is assumed to have three terms--a fixed cost, a cost that is proportional to the outer surface area of the insulation, and a cost that is proportional to the amount of insulation used:

$$C_I = C_1 + \frac{\pi}{12}(D + 2s)\,LC_2 + \frac{\pi}{48}\left[(D + 2s)^2 - D^2\right]Lc \qquad \text{(G-9)}$$

where:

C_1 is the fixed cost of the insulation in dollars.

C_2 is the cost of applying a cover to the insulation in dollars per square foot (dollars per square meter).

c is the cost of the insulation in dollars per square foot per inch (dollars per square meter per centimeter).

The derivative of insulation cost with respect to insulation thickness is

$$\frac{dC_I}{ds} = \frac{\pi L}{12}\left[2C_2 + D\left(1 + \frac{2s}{D}\right)\right]c \quad . \qquad \text{(G-10)}$$

Combining Equations G-8 and G-10 yields the incremental cost of saving energy:

$$E_I \frac{dC_I}{dQ_L} = \frac{E_I\, dC_I/ds}{dQ_L/ds}$$

$$= \frac{E_I\left[C_2 + D\left(1 + \frac{2s}{D}\right)\frac{c}{2}\right]\left[R_i\left(1 + \frac{2s}{D}\right) + \frac{r^*D}{2}\left(1 + \frac{2s}{D}\right)\ln\left(1 + \frac{2s}{D}\right) + R_o\right]^2}{\left[\frac{r^*D}{2}\left(1 + \frac{2s}{D}\right) - R_o\right]t(\bar{T} - \bar{T}_a)}. \qquad \text{(G-11)}$$

Since the incremental cost of saving energy, $E_I \frac{dC_I}{dQ_L}$, must equal the incremental cost of collecting energy, $E_I \frac{\Delta C_c}{\Delta Q_c}$, at the optimum insulation thickness, and since we assume that the economic factors are equal, Equation G-11 can be rewritten as follows:

$$\left[C_2 + \frac{cD}{2} Y\right]\left[R_i Y + \frac{r^*D}{2} Y \ln Y + R_o\right]^2$$

$$- \left[\frac{r^*D}{2} Y - R_o\right] t(\bar{T} - \bar{T}_a) \frac{\Delta C_c}{\Delta Q_c} = 0 \qquad \text{(G-12)}$$

where $Y = 1 + \frac{2s}{D}$.

Equation G-12 can be solved for Y in terms of C_2, $\frac{cD}{2}$, $\frac{r^*D}{2}$, R_i, R_o, and $t(\bar{T} - \bar{T}_a) \frac{\Delta C_c}{\Delta Q_c}$, either by trial and error or by more advanced methods such as the Newton-Raphson method. A program for a Hewlett-Packard HP-25 calculator is shown in Table G-1. The program solves Equation G-12 by the interval-halving method.

To use the program, load the six constants into registers 0 through 5, a high guess at Y in register 7, and a low guess in register 6. We recommend using 10 for the high guess and 1 for the low guess. Calculate (high guess-low guess)/2 and leave the result showing in the display. Press the following key sequence:

f, clear PRGM, R/S .

When the calculator stops running, after about a minute, the plus-or-minus accuracy of the Y calculation will be in the display. Press the rolldown key twice to obtain Y.

Example G-2: A one-inch pipe (with an actual outside diameter of 1.315 inches, or 3.34 centimeters) carries antifreeze solution at an average temperature of 130°F (54°C) from the collectors into a building. The average ambient temperature is 40°F (4°C), and the pipe will pass through an uninsulated attic. The pipe will be insulated with foam rubber costing $1.92 per square foot per inch of thickness ($8.14 per square meter per centimeter of thickness), and the cost of installing the insulation is $2.09 per square foot ($22.50 per square meter). Adding 53.8 per square foot (5 square meters) of collector will cost $1076 and will result in collection of 5.05×10^6 Btu (5.32×10^9J) of additional energy per year. The pipe will be hot an average of 6 hours per day during the 8-month heating season for a total of 1460 hours (5.26×10^6 seconds) per year.

Table C-1. HP-25 Calculator Program for Insulation Thickness

Register	Contents	Register	Contents	Register	Contents
0	C_2	3	R_o	6	low guess
1	$\frac{cD}{2}$	4	$\frac{r^*}{2}$	7	high guess
2	R_i	5	$t(\bar{T} - \bar{T}_a)\frac{\Delta C_c}{\Delta Q_c}$		

Step	Key	Step	Key	Step	Key
1	RCL 6	16	X	31	STO 6
2	+	17	RCL 0	32	GTO 35
3	ENTER ↑	18	+	33	R↓
4	ENTER ↑	19	X	34	STO 7
5	f ln	20	$x \overleftrightarrow{} y$	35	RCL 7
6	RCL 4	21	RCL 4	36	RCL 6
7	X	22	X	37	-
8	RCL 2	23	RCL 3	38	2
9	+	24	-	39	÷
10	X	25	RCL 5	40	EEX
11	RCL 3	26	X	41	CHS
12	+	27	-	42	5
13	g x^2	28	g x ≥ 0	43	$x \overleftrightarrow{} y$
14	$x \overleftrightarrow{} y$	29	GTO 33	44	f x ≥ y
15	RCL 1	30	R↓	45	GTO 01

The insulation's thermal resistance per unit of thickness is $r^* = 4.90$ °F·ft²·hr/Btu·in (0.340 °C·m²/W·cm). The thermal resistance on the inside of the pipe, R_i, is negligible, and the thermal resistance on the outside of the pipe, R_o, from Table 4-5, is 0.68 °F·ft²·hr/Btu (0.120 °C·m²/W). The six constants required for the calculation are

$$C_2 = \$2.09/ft^2 \ (\$22.50/m^2) .$$

$$\frac{cD}{2} = \frac{1.92 \times 1.315}{2} = \$1.26/ft^2 \ (\$13.59/m^2) .$$

$$\frac{r^*D}{2} = \frac{4.90 \times 1.315}{2} = 3.22 \ °F·ft^2·hr/Btu \ (0.568 \ °C·m^2/W).$$

$$R_i = 0 .$$

$$R_o = 0.68°F·ft^2·hr/Btu \ (0.120 \ °C·m^2/W) .$$

$$t(\bar{T} - \bar{T}_a) \frac{\Delta C_c}{\Delta Q_c} = \frac{1460 \times (130-40) \times 1076}{5.05 \times 10^6}$$

$$= \$28.0 \ °F·hr/Btu \ (\$53.1 \ °C/W) .$$

Using the calculator program in Table G-1 produces the solution Y = 2.160, and the outside diameter of the insulation is DY = 1.315 x 2.160 = 2.84 inches (7.21 centimeters). The thickness of the insulation is

$$s = \frac{D}{2}(Y - 1) = 0.76 \text{ in } (1.94 \text{ cm}).$$

Standard-sized 3/4-inch-thick insulation would be close to the economic thickness and would provide a thermal resistance of approximately 4.4 °F·ft²·hr/Btu (0.775 °C·m²/W) including the surface resistance. This is close to the Polytechnic Institute of New York recommendation of 4°F·ft²·hr/Btu (0.705 °C·m²/W) for 1-inch pipe.

Example G-3: A hot air duct passing through an uninsulated attic has the following parameters associated with it:

D = 12 in (30.5 cm).

$\bar{T}$ = 120°F (49°C).

$\bar{T}_a$ = 40°F (4°C).

R_i = 0.17°F·ft^2·hr/Btu (0.030°C·m^2/W).

R_o = 0.68°F·ft^2·hr/Btu (0.120°C·m^2/W).

r^* = 3.15°F·ft^2·hr/Btu·in (0.219°C·m^2/W·cm).

c = \$1.65/ft^2·in (\$6.99/m^2·cm).

C_2 = \$1.30/ft^2 (\$13.99/m^2).

ΔC_c = \$1076.

ΔQ_c = 5.05 x 10^6 Btu (5.32 x 10^9 J).

t = 1460 hr (5.26 x 10^6 sec).

The solution is Y = 1.263, and the outside diamater of the insulation is

$$DY = 12 \times 1.263 = 15.16 \text{ in } (38.5 \text{ cm}) .$$

The economic thickness of insulation is

$$s = \frac{D}{2} (Y-1) = 1.58 \text{ in } (4.0 \text{ cm}) .$$

A 1-1/2-inch-thick wrapping of fiberglass would have a thermal resistance of approximately 5.8 °F·ft^2·hr/Btu (1.03 °C·m^2/W) including the inside and outside surface resistances.

SYMBOLS USED IN APPENDIX G

Main Symbols

A	area; $ft^2(m^2)$
C	cost, $
c	unit cost insulation, $\$/ft^2 \cdot in$ ($\$/m^2 \cdot cm$)
D	storage unit, pipe, or duct diameter, in (cm)
E	economic factor to convert initial cost to annual equivalent cost
L	length of storage unit, pipe, or duct, ft (m)
Q	thermal energy, Btu (J)
R	thermal resistance of insulation, $°F \cdot ft^2 \cdot hr/Btu$ ($°C \cdot m^2/W$), not corrected for parallel heat loss
R^*	thermal resistance of insulation, $°F \cdot ft^2 \cdot hr/Btu$ ($°C \cdot m^2/W$), corrected for parallel heat loss
r^*	thermal resistance of insulation per unit thickness, $°F \cdot ft^2 \cdot hr/Btu \cdot in$ ($°C \cdot m^2/W \cdot cm$), corrected for parallel heat loss
s	insulation thickness, in (cm)
$\bar{T}$	temperature, °F (°C), averaged over time t
t	time, hr (sec)
U	thermal conductance of insulation, $Btu/hr \cdot ft^2 \cdot °F$ ($W/m^2 \cdot °C$)
Y	ratio of outside diameter of insulation to diameter of storage unit, pipe, or duct
ΔA	increment of area, ft^2 (m^2)
ΔC	increment of cost, $
ΔQ	increment of energy, Btu (J)

Subscripts

a	ambient condition
c	collector
I	insulation
i	inside surface
L	loss
max	maximum condition
min	minimum condition
o	outside surface
1, 2, etc.	first, second, etc.

GLOSSARY

absolute pressure	Also called absolute head. Pressure measured relative to vacuum. Absolute pressure is the gauge plus the atmospheric pressure.
absorption chiller	device that uses the heat input to drive a 2-component chemical cycle that results in a cold output.
absorption cooling	cooling provided by evaporating a volatile fluid which is then (1) absorbed in a strong solution, (2) desorbed under pressure by outside heat, and (3) recondensed at a temperature high enough that the heat of condensation can be rejected to the outside.
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	A solar collector that uses circulating air as a heat transfer fluid.
ambient temperature	The temperature of the surroundings as measured by a dry-bulb thermometer...temperature of air which surrounds object on all sides.
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.
aquifer	a permeable body of rock capable of yielding quantities of groundwater to wells and springs -- also providing a zone useful for energy storage.
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 heat when solar energy alone is insufficient, a backup system.
base load	The minimum load of a utility (electric or gas) over a given period of time.
battery	An electrical energy storage system frequently using reversible chemical reactions.
Btu	British thermal units. The amount of heat required to raise 1 pound of water 1 degree Fahrenheit; the basic unit of heat in the English system of units.
capital cost	Cost of equipment, construction, land, and other items required to construct a facility; different from recurrent operating and maintenance costs.
CAS	Compressed air storage in which air is forced into reservoir under pressure.
catalyst	Substance that alters the velocity of a chemical reaction and may be recovered essentially unaltered in form and amount at the end of the reaction.
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 fluid is flung outward by the rotation of an impeller. See positive displacement pump.
champagne effect	Aeration problem associated with compressed air storage in which air dissolves into the water causing storage loss.
charging	(1) in general refers to a process in which energy is going to the storage. (2) with respect to battery - process of supplying electrical energy for conversion to stored chemical energy.

chemical compatibility	Ability of materials and components in contact with each other to resist chemical degradation, such as that caused by electrolytic action or plasticizer migration.
chemical heat pump	Functionally similar to mechanical heat pump-turns thermal energy into heat using two different reversible reactions with a common gaseous product.
chiller	A device which cools fluids to temperatures below those obtainable with ordinary cooling water by using the vaporization of a refrigerant.
coefficient of performance (COP)	Ratio calculated for both heating and cooling equipment by dividing capacity in watts by power input in watts.
coil-in-tank heat exchanger	A coil of tubing submerged 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	A heat transfer fluid used in solar collectors.
component	Smallest identifiable element of a solar heating or cooling subsystem, such as a valve, a control, or a container.
congruent melting	Behavior of phase change materials that do not change their chemical nature when they melt.
controls	Devices such as thermostats and temperature-sensing elements used to manipulate fans, pumps, and dampers in solar systems.
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.

convector	A heat exchanger that uses natural convection of air to transfer heat from water or steam to a building.
cooling season	The time of year (usually June to September, but varying with climate) when air conditioning is desirable to maintain comfortable room temperatures.
cooling tower	A towerlike device in which atmospheric air circulates and cools warm water, generally by direct contact.
daily temperature range	The difference between the warmest storage temperature attained in a day and the coolest storage temperature reached on the same day.
desiccant cooling	Cooling provided by evaporatively cooling air that has been dried with a desiccant (a liquid or solid with a strong affinity for water). The desiccant is then "regenerated" for reuse by heating it to a high temperature to drive off the water.
DHW	Potable domestic hot water.
dielectric bushing	An electrically insulating pipe connector used to connect dissimilar metals in plumbing.
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).
dimensionless variable	A quantity which does not have dimensional units and is therefore has the same value in any system of units.
discharging	(1) in general, refers to a process in which energy is leaving storage. (2) with respect to battery - withdrawl of electrical energy so that connected equipment can be operated.
diurnal storage	Referring to energy storage strategy adequate for 24 hours.
domestic water heating system	A solar heating system which supplies a portion of the energy needed to heat water used for domestic purposes, such as bathing and washing.

drainback system	A method of protecting the solar collectors against freezing by draining the collector water into the storage tank on cold nights.
draindown system	A method of protecting the solar collectors against freezing by draining the collector water into the sewer on cold nights.
effectiveness	The ratio of actual heat transferred in a heat exchanger to the maximum possible heat that could be transferred in a perfect heat exchanger.
elastomeric energy storage	New mechanical energy storage option based on elastic expansion.
electrolytic solution	A liquid that conducts electricity via ions and electrons. When dissimilar metals are in contact with an electrolytic solution, galvanic corrosion can occur. See dielectric bushing.
eutectic	Having the lowest fusing (freezing) points of a mixture of several salts or metals; these substances can be utilized in heat-of-fusion energy storage.
evaporator	Heat exchanger that interfaces chiller with either cold storage or the space to be cooled.
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 at the University of Wisconsin for calculating the performance of solar energy systems.
flat-plate collector	A collector of diffuse and beam solar radiation, with five basic component parts: 1. Glazing to reduce heat loss from the illuminated surface of the absorber. 2. Tubes for directing the heat-transfer fluid from the inlet to the outlet. 3. Absorber plate to which tubes are attached in in a manner producing a good thermal bond. 4. Insulation to minimize heat loss from the nonilluminated side of the absorber plate. 5. Housing surrounding the foregoing components and keeping them free of dust and moisture.

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.
flue gas	Gas from the combustion of fuel.
fluid	A substance that cannot retain its shape without an external container; a gas or a liquid.
fluidized bed	A container of solid particles in which a fluid flows strongly enough that the particles lose permanent contact with each other and are suspended in the fluid.
flywheel	Massive wheel that stores rotational kinetic energy.
forced convection	See convection, forced.
fouling factor	A factor ($°F \cdot ft^2 \cdot hr/Btu$) that expresses the degradation of heat exchanger performance caused by scaling or biological fouling. The fouling factor is the inverse of the scaling coefficient.
FRP	Fiberglass-reinforced plastic.
fuel cell	A device in which hydrogen is fed to an electrode where it is catalytically converted to hydrogen ions, releasing electrons to flow through an external circuit. The electrons flowing through the external circuit constitute an electric current. Upon return to the fuel cell the electrons react to form oxygen ions which, in turn, unite with hydrogen ions to form water. The process is, essentially, the reverse of electrolyis.
galvanic corrosion	Caused by an electrochemical reaction between two or more different metals in a system.
galvanic series	metals ranked from electrically positive to electrically negative. Measures relative "tendency to corrode" of each metal when used in a multimetal system.
gauge pressure	Pressure measured relative to atmospheric pressure.

head	The maximum distance a liquid can rise in a pipe. Head is used as a measure of pressure.
heat capacity	The amount of heat (Btu) required to raise the temperature of 1 pound of a substance 1 degree Fahrenheit. Heat capacity is measured in units of Btu per pound per degree Fahrenheit. Compare the definition of heat capacity with the definition of specific heat.
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 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 loss	Heat lost from a building by conduction through the windows, walls, roof, and floor or foundation, and by the infiltration of air.
heat pump	Reversed (or reversible) air-conditioning device employing refrigeration equipment in a manner that enables usable heat to be supplied to a space during winter and extracted from the space during summer.
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 ($Btu/hr \cdot °F \cdot ft^2$).
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.

hybrid system	A solar energy system that combines features of both active and passive systems.
hydraulic stone	Concrete type compound impervious to water.
hydroelectric plant	A plant in which the kinetic energy of falling water is used to turn a generator producing electricity.
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, fancoil units, floor panels, or ceiling panels.
inhibitors	Additives to storage water to prevent algal growth and corrosion.
insolation	The amount of solar energy incident on a unit of surface area per unit of time ($Btu/hr \cdot ft^2$). 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.
iteration	A method of solving mathematical equations that do not have closed-form solutions. Iteration involves making an initial guess at the solution and refining the guess by repeated application of the mathematical formulas until the change in the refined guess becomes negligible.
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 amount of heat per unit of 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. Items usually included are first cost, mortgage interest, fuel, electricity, repairs, and other taxes.
liquid-to-liquid heat exchanger	Device for heat exchange between the liquid streams.
liquid-type collector	A solar collector that uses a circulating liquid as a heat transfer fluid.
load	Space or domestic water heating energy need, or the cooling energy need, which is to be supplied by solar or conventional energy.
mature technology	System that is commercially available, reliable, well-proven in efficiency.
maximum operating 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.
minimum operating temperature	The lowest temperature at which useful heat can be extracted from storge.
mixing valve	Automatic thermostatic valve that can be installed downstream of a solar heated hot-water storage tank to prevent accidental scalding of user.
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.

open-loop system	Some part of the system is vented to the atmosphere, or the system contains fresh or changeable water.
operating temperature range	The difference between maximum operating temperature and minimum operating temperature for a specified length of time. See daily temperature range.
outgassing	Generation of vapors by materials, usually during exposure to elevated temperatures and/or reduced pressures.
Pacific Regional Handbook method	A method devised at Los Alamos Scientific Laboratory for calculating the performance of solar systems.
parasitic losses	The power required to circulate heat transfer fluids and operate controls.
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.
PCM	Phase change material.
peak power	Output of a solar-energy system on a clear day with the peak solar radiation at normal incidence. Peak solar radiation is usually assumed to be 1000 watts per square meter, or about 320 Btu per hour per sq. ft.
pH	Measure of acidity or alkalinity: simply speaking, pH<7 is acid, pH=7 is neutral, pH>7 is alkaline.
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.

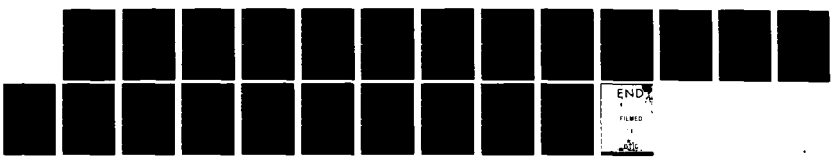
photovoltaic cell	Semiconductor device in which the absorption of light creates a separation of electrical charges. This results in an electrical potential that can be tapped by allowing electrons to flow through an external circuit. The net effect is direct conversion of light into electricity.
pipe friction	Wall roughness of pipes or ducts which causes a loss of pressure in the direction of flow and a resultant evolution of heat.
pitting	Localized loss of material by erosion or chemical decomposition.
plenum (plural plena)	A space at the inlet or outlet of a rock bed used to distribute the air uniformly to the rocks.
positive displacement pump	A pump in which the fluid is squeezed between solid parts such as a piston and cylinder. Compare with centrifugal pump.
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.
primary	Battery which cannot be recharged efficiently or safely after any amount of discharge.
psi	Pounds per square inch; a unit of pressure. Unless otherwise specified, pressure is measured relative to atmospheric pressure.
pumped hydroelectric storage	A means for the large-scale storage of electrical energy. Excess electricity produced during periods of low demand is used to pump water up into an elevated reservoir. When demand is high the water is released to operate a hydroelectric generator.
Rankine chiller	Device cooling fluids that is based on the Rankine cycle.
Rankine cycle	A closed heat engine cycle using various components, including a working fluid pumped under pressure to a boiler where heat is added; and expander (turbine) where work is generated; and a condenser used to reject low-grade heat to the environment.

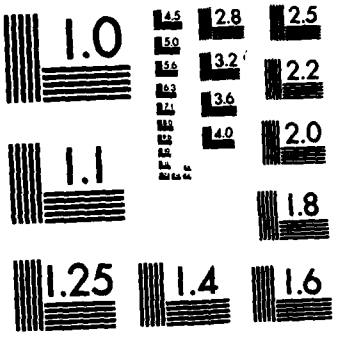
resistance heating	A method of heating with electricity in which electricity passing through a resistor is converted directly to heat.
retrofit	As used here, retrofit means to install a solar energy system in an existing building or in a building not originally designed for solar energy.
Reynolds number	The dimensionless number $\rho vd/\mu$ where ρ = fluid density, v = fluid velocity, and d = a characteristic distance or diameter, and μ = fluid viscosity. The Reynolds number is closely related to the ratio of inertial force to viscous force in the fluid. Laminar flow occurs at low Reynolds numbers, and turbulent flow occurs at high Reynolds numbers.
rock bed face velocity	The volumetric air flow rate divided by the gross cross-sectional area of the rock bed. The face velocity is an abstract quantity that does not equal the actual air velocity in the small passageways between the rocks.
R-value	Resistance of insulation to heat conduction given in units of $°F \cdot ft^2 \cdot hr/Btu$.
scaling coefficient	A factor ($Btu/hr \cdot °F \cdot ft^2$) that expresses the degradation of heat exchanger performance due to formation of scale on the heat exchange surfaces. The scaling coefficient is the inverse of the fouling factor.
sealed system	A solar system that excludes oxygen by closing all vents and inlets and outlets for liquids. Exclusion of oxygen in this manner limits one type of corrosion, but requires an expansion tank to limit pressure.
secondary	Battery which can be recharged after being discharged under specified conditions of use.
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 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.
solar ponds	Ponds of stratified water that collect and retain heat. Convection, normally present in ponds, is suppressed by imposing a stable density gradient of dissolved salts.
SOLCOST	Method for estimating the size and cost of solar systems for heating domestic water, and making comparisons with conventional systems.
space heating	Heating a building to maintain a comfortable indoor temperature.
specific heat	The ratio of the heat capacity of a substance to the heat capacity of water (1 Btu per pound per degree Fahrenheit). Unlike heat capacity, specific heat is a dimensionless (unitless) quantity. Compare the definition of specific heat with the definition of heat capacity.
SPHS	Surface pumped hydro storage. Pumped hydro strategy with both reservoirs above ground.
storage capacity	The amount of energy which can be stored by a solar heating system to be used at a later time for space or water heating.
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, pumps, valves, and other components necessary to transfer heat into and out of the storage medium.
supercooling	Melting behavior problem characterized by PCM temperature dropping below the melting point without a change in phase.

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MICROCOPY RESOLUTION TEST CHART
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tank-in-tank heat exchanger	A tank containing a heat transfer liquid submerged in a tank containing another heat transfer liquid. Natural convection occurs on both sides of the inner-tank wall.
temperature, dry-bulb	Temperature of the air as indicated by an ordinary thermometer.
temperature stratification	Thermal stratification.
temperature, wet-bulb	Temperature of the air as indicated by a thermometer whose bulb is surrounded by a wet gauze and past which air is blown rapidly. The difference between the wet-bulb and dry-bulb temperatures is used to find the atmospheric humidity. When the atmosphere is completely saturated with water vapor (humidity = 100%), the wet-bulb temperature equals the dry-bulb temperature.
tempering valve	A valve that limits the temperature of water flowing from a domestic hot water tank by mixing it with cold water.
TES	Thermal energy storage.
thermal conductivity	Amount of heat transferred by conduction through a cube with unit sides per unit time per degree of temperature difference; e.g. calories transferred per second through a one-centimeter cube having a one degree C temperature difference between its front and back faces.
thermal stratification	Separation of hot and cool parts of the storage medium within the storage unit.
thermistor	A type of temperature sensor.
thermochemical energy storage	Energy storage that uses reversible chemical reactions to store and release thermal energy.
thermodynamics, laws of	first law states that heat can be converted into work and work into heat; there is a constant relationship between the two quantities which is known as the mechanical equivalent of heat (one calorie equals about 4.2 joules). The second law states that when a free exchange of heat takes place between two objects, the heat is always transferred from the warmer to the cooler object.

thermosiphoning	Motion of a fluid caused by buoyancy of its warmer parts; natural convection.
thermosiphon system	A pumpless solar system in which buoyancy, acting on water heated by the collector, causes the water to rise into the storage tank. Thermosiphon 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.
tidal power	Electricity produced by using the rise and fall of ocean tides.
ton of refrigeration	The removal of heat at the rate of 12,000 Btu/hr; derived from melting a ton (2000 lb) of ice, requiring 2000 x 144 Btu over a 24-hr period.
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.
traced tank	A type of heat exchanger in which the heat transfer fluid is carried in a tube wrapped around the storage tank; a wraparound heat exchanger.
traction battery	One that can suffer deep discharging and be used for sustained work.
transitional flow	Fluid flow that is on the borderline between laminar flow and turbulent flow.
transmittance	The ratio of the radiant energy transmitted by a given material to the radiant energy incident on a surface of that material; depends on the angle of incidence.
turbulent flow	Fluid flow in which mixing between adjacent layers is prevalent. Turbulent flow occurs at high Reynolds numbers. See laminar flow and transitional flow.
U-factor	The number of Btu which pass through one square foot of solid in one hour if there is a one degree (F) difference between the two sides. Used to express heat transmission. The reciprocal of "R" (U = 1/R).

UPHS	Underground pumped hydro storage. Pumped hydro scheme in which second reservoir is underground.
vapor	Gaseous phase of substances that are either liquids or solids at the same temperature but higher pressure.
void fraction	The ratio of air space volume in a rock bed to the total volume of the rock bed.
volumetric heat capacity	The amount of heat a unit volume of a storage medium contains per unit change of temperature, expressed as $Btu/ft^3 \cdot °F$.
water-air heat exchanger	Device in which air is either heated or cooled by flowing water.
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.

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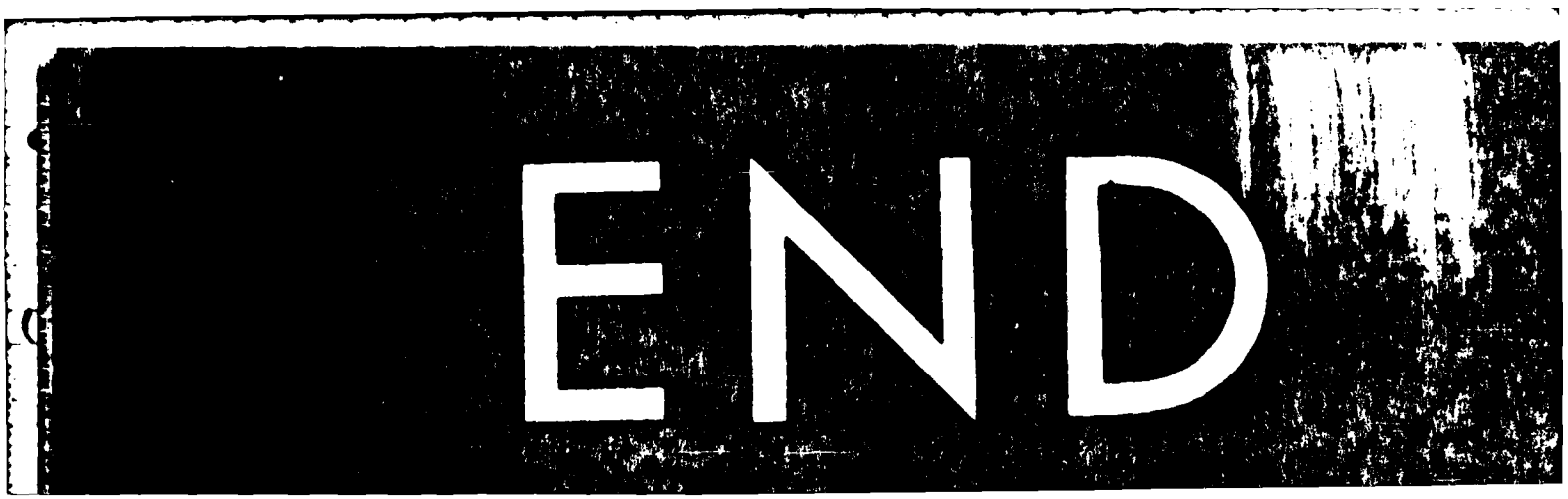
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