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TECHNICAL GUIDELINES FOR ENERGY CONSERVATION

CENTER FOR BUILDING TECHNOLOGY
INSTITUTE FOR APPLIED TECHNOLOGY
NATIONAL BUREAU OF STANDARDS
WASHINGTON D.C. 20234

JUNE 1977

Final Report For Period December 1975 to March 1977

Approved for Public Release; Distribution Unlimited.

AIR FORCE CIVIL ENGINEERING CENTER

TYNDALL AIR FORCE BASE
FLORIDA 32403

AFCEC-TR-77-12

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This report provides detailed technical material on various energy conservation actions for existing Air Force facilities and utility systems. It is specifically tailored to serve as a working document for Base engineers and technical personnel. The report covers energy conservation for Air Force facilities, including the equipment for providing hot water, space heating and cooling, lighting, and humidification. It also covers central plant systems and underground distribution systems (hot water, steam, and chilled water). It does not cover energy conservation measures for tactical or mission-related equipment such as ground vehicles or fighter aircraft.

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PREFACE

This report is the technical portion of a two-report Energy Conservation effort prepared by the Staff of the Center for Building Technology of the National Bureau of Standards (NBS) for the Air Force Civil Engineering Center (AFCEC), Tyndall AFB, Florida.

AFCEC-TR-77-11 is designed for use by managers to enhance their understanding of energy conservation measures and to assist them in developing a comprehensive energy program. This report provides the necessary engineering information for Staff engineers to evaluate and implement appropriate energy conservation measures.

This report, the results of research performed during the period December 1975 to March 1977, was submitted by NBS to AFCEC in March 1977. The NBS Program Manager was Clinton W. Phillips, Office of Housing and Building Technology, and the NBS project leader and coordinator was Douglas M. Burch, Thermal Engineering Section, Building Environment Division.

VOLUME II. TECHNICAL GUIDELINES FOR ENERGY CONSERVATION. Separate chapters were prepared and reviewed by staff with particular expertise in each area, as indicated:

CHAPTER 1. INTRODUCTION. Robert Jones and D. Burch, reviewed by C. W. Phillips.

CHAPTER 2. ENERGY CONSERVATION MEASURES FOR EXTERIOR BUILDING ENVELOPES. D. M. Burch and R. Baumgardner*, reviewed by F. J. Powell.

CHAPTER 3. MODIFYING MECHANICAL SYSTEMS AND OPERATING PRACTICES FOR ENERGY CONSERVATION. D. M. Burch and ENVIRO-MANAGEMENT & RESEARCH, INC.**, in consultation with R. Beausoliel, A. Camacho, J. B. Coble, D. Garbern, J. Kao, M. McNeil, and S. Treado, reviewed by R. Jones.

CHAPTER 4. CONDUCTING THE BUILDING SURVEY. R. Baumgardner*, reviewed by F. J. Powell.

CHAPTER 5. MEASUREMENTS FOR IDENTIFYING ENERGY CONSERVATION POTENTIAL. D. M. Burch, reviewed by T. Kusuda.

CHAPTER 6. THE ECONOMIC ANALYSIS. Stephen Petersen, reviewed by H. E. Marshall.

APPENDICES

A. HEAT TRANSFER FUNDAMENTALS. B.A. Peavy, reviewed by J. Hill.

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B. SOLAR ENERGY SYSTEMS FOR AIR FORCE APPLICATIONS. James Hill, reviewed by T. Kusuda.

C. HEAT AND CHILLED WATER DISTRIBUTION SYSTEMS. T. Kusuda, reviewed by D. M. Burch.

D. SURVEY OF COMPUTER PROGRAMS FOR EVALUATING BUILDING AND SYSTEM PERFORMANCE. William Carroll, reviewed by F. J. Powell.

E. BIBLIOGRAPHY OF ENERGY CONSERVATION DOCUMENTS. Mary Reppert.

Valuable assistance was given for field measurements by Walter Ellis and James Allen, and for editing by M. Reppert, and for typing and copying, by S. Baile, R. Schmeit, S. Wyand, G. Linton, D. Dawson, and Sara Gill.

The AFCEC Project Officer was Captain Larry W. Strother.

This report has been reviewed by the Information Office (OI) and is releasable to the National Technical Information Service (NTIS). At NTIS it will be available to the general public, including foreign nations.

This report has been reviewed and is approved for publication.

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TECHNICAL GUIDELINES FOR ENERGY CONSERVATION

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TECHNICAL GUIDELINES FOR ENERGY CONSERVATION

CHAPTER 1. INTRODUCTION

1.1 Purpose

This report provides detailed technical material on various energy conservation actions for existing Air Force facilities and utility systems. It is specifically tailored to serve as a working document for base engineers and technical personnel.

1.2 Scope

This report covers energy conservation for Air Force facilities and family housing, including the equipment for providing hot water, space heating and cooling, lighting, and humidification. It also covers central plant systems and underground distribution systems for hot water, steam, and chilled water. It does not cover energy conservation measures for tactical or mission-related equipment such as ground vehicles or fighter planes. Whereas this report addresses the retrofit of existing buildings and systems, its use is also encouraged in the design of new buildings, and extensions or remodeling of existing buildings and systems.

1.3 Background on the Energy Problem

It has been estimated that between 1971 and 2001, the United States will consume more energy than it has in its entire history. By 2001 the annual U. S. demand for energy in all forms is expected to double, and the annual worldwide demand will probably triple.*

Because the great increase in energy consumption in the past century has taken place chiefly in the advanced countries, it is instructive to examine the trends in the U.S. The annual consumption of all forms of energy in the United States has increased seventeen-fold in the past century, with a corresponding population increase of a little more than five-fold. During this period, fuel sources have shifted steadily (see figure 1-1). Wood was the dominant fuel energy source in 1850;

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* Energy and Power, Scientific American, Inc., W. H. Freeman and Company, 600 Market Street, San Francisco, California, 1974.

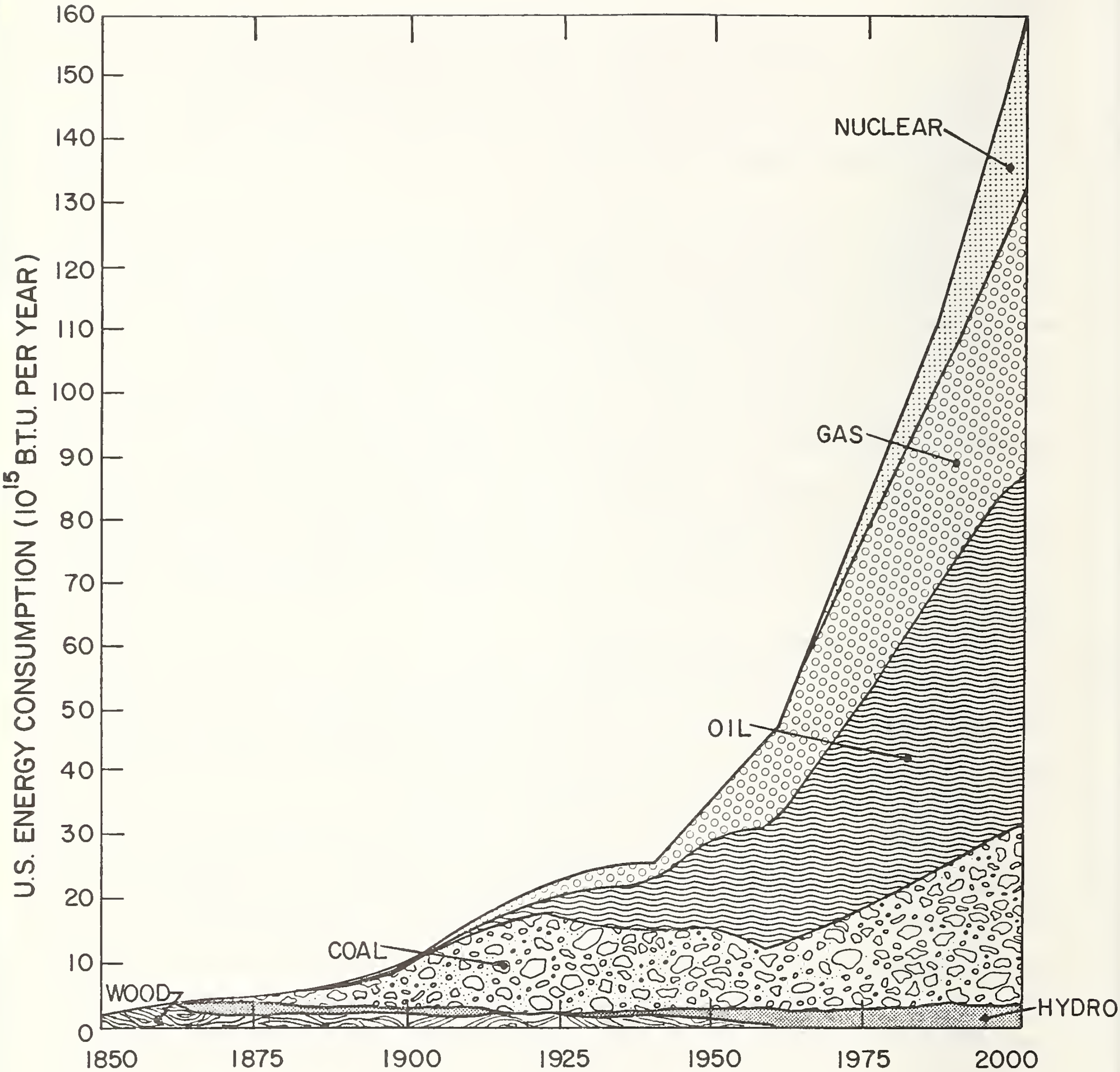

Figure 1-1. U. S. energy consumption. (From Energy and Power, Scientific American, Inc., 1974.)

consumption and wood had declined to some 10 percent. In the 50 years between 1910 and 1960, coal lost its leading position to natural gas and oil. Today nuclear power is emerging as a national energy source.

Escalating world energy consumption has resulted in spiraling Air Force energy costs. A graphic example of this is given in table 1-1.

TABLE 1-1. AIR FORCE ENERGY CONSUMPTION

Fiscal Year	Total A.F. Energy Consumption 10^{12} Btu	Energy Costs 10^{6} Dollars
1973	242	163
1974	209	185
1975	204	311
1976	197	355

Note that Air Force energy conservation programs have resulted in approximately a 19-percent reduction in energy consumption. However, due to increased energy costs, Air Force energy costs have risen by a factor of over 2.0.

1.4 Motivation

The effective use of this report depends upon BCE personnel, their knowledge, experience, and motivation. The operating and maintenance personnel must be called upon to examine mechanical and electrical systems to increase their energy efficiency. Operating procedures long considered standard are being questioned. Even the requirements of codes and regulations are being subjected to reevaluation. In the process, operating engineers and designers are finding out that their systems can be improved considerably, while still providing adequate performance.

In order to assure that all potential measures are identified and that the most cost-effective and energy-efficient measures are selected, responsible BCE personnel should follow the procedures described in this report for audit, identification, and analysis. The experience of successful programs to implement energy conservation measures indicates that the key can be a thorough and systematic investigation of all installations, all systems, and all equipment. Nothing can be excused or overlooked, and each must be evaluated against the respective performance criteria contained in this report. In this way, the wealth of information assembled in the report can be used to complement the experience of BCE personnel.

1.5 Use of this Technical Report

It will be helpful for the user of this report to familiarize himself with the Table of Contents and the general content of each of the chapters. However, each of the chapters is designed to be self-contained, so the user need not read the entire report for treatment of each topic. AFCEC-TR-77-12 consists of six chapters, including this introduction, and five appendices.

Chapter 2 provides technical information on energy conservation measures for exterior building envelopes.

Chapter 3 provides technical information in modifying mechanical equipment and utility systems for energy conservation.

Chapter 4 explains how to conduct a physical reconnaissance of a building to identify potential target areas for energy conservation.

Chapter 5 describes key measurements which can be performed for identifying energy conservation potentials.

Chapter 6 presents a methodology for determining how to allocate a given energy conservation budget to optimize the dollar return (energy savings) for a particular dollar investment.

In the appendices, some special information is developed, which will be of particular interest to those who would like to go into greater detail on the following items:

A. Heat Transfer Fundamentals
B. Solar Energy Systems
C. Heat and Chilled Water Distribution Systems
D. Computer Programs for Evaluating Buildings and System Performance.

Finally, appendix E provides a bibliography of energy conservation documents.

The expected result of using AFCEC-TR-77-11 and AFCEC-TR-77-12 is the planning and development of comprehensive energy conservation programs, which will identify in a consistent manner throughout Air Force installations energy-efficient and economically feasible retrofit measures for implementation. In this way, not only will the national need to conserve resources be addressed, but also the burden of rising utility costs at United States Air Force Bases will be reduced.

CHAPTER 2. ENERGY CONSERVATION MEASURES FOR EXTERIOR BUILDING ENVELOPES

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CHAPTER 2. ENERGY CONSERVATION MEASURES FOR EXTERIOR BUILDING ENVELOPES

2.1 Assessing the Problem

Before deriving a detailed energy conservation plan for modifying the exterior envelope of a building, it is a good practice to assess whether the building has predominantly a heating or cooling problem or possibly both. The treatment of these two types of problems is usually quite different. The following general guidelines are provided.

Buildings located on Air Force Bases having more than 6000 heating degree days* will usually consume much more energy for space heating than for space cooling (see figure 2-1). In such instances, more significant reductions in annual energy consumption can usually be obtained by treating the heating problem than the cooling problem. In treating the heating problem, emphasis should be placed on reducing heat loss due to air infiltration and reducing conduction losses through the exterior building envelope. The latter is conventionally accomplished through the installation of storm windows and thermal insulation to the exterior walls, ceilings, and floors of a building.

On the other hand, buildings located on Air Force Bases having less than 3000 heating degree days will often consume much more energy for space cooling than for space heating. In treating such a building, emphasis should be placed on reducing internal heat gains from equipment and lighting, reducing solar radiation through windows (through the use of shading devices), and reducing heat gains due to air infiltration.

Buildings located on Air Force Bases having heating degree days between 3000 to 6000 will often have both heating and cooling problems which should be treated.

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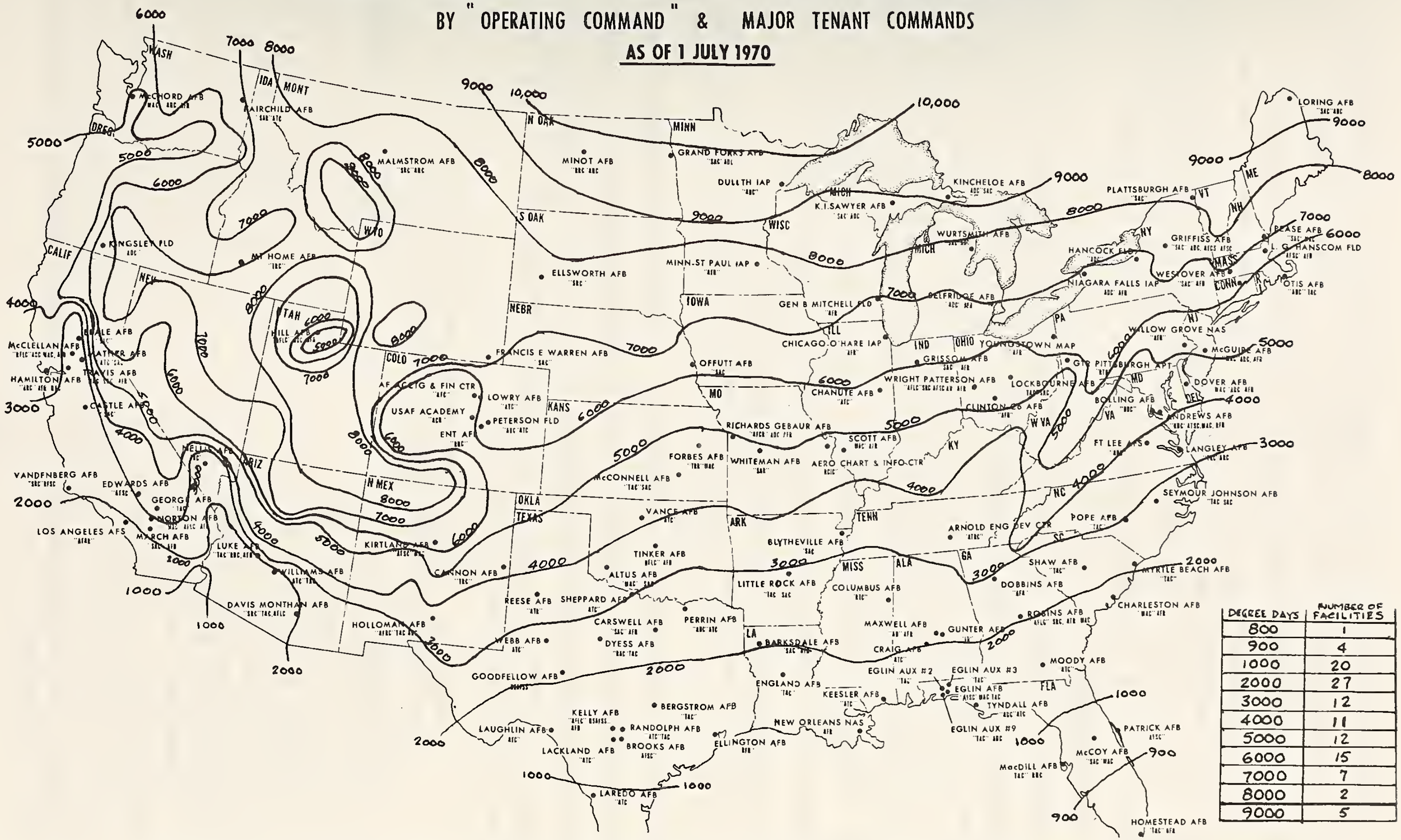
* A degree day is a unit based upon temperature difference and time used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65° F, there exist as many degree days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65° F.

USAF ACTIVE MAJOR INSTALLATIONS

BY "OPERATING COMMAND" & MAJOR TENANT COMMANDS

AS OF 1 JULY 1970



DEGREE DAYS	NUMBER OF FACILITIES
800	1
900	4
1000	20
2000	27
3000	12
4000	11
5000	12
6000	15
7000	7
8000	2
9000	5

Figure 2-1. Heating degree-day map of United States showing the location of major Air Force installations.

A map of the United States showing the location of active major Air Force installations is given in figure 2-1. Lines of constant heating degree days have been superimposed on this map. By locating the particular Air Force Base on this map, one can determine whether the buildings on the Base have predominantly a heating or cooling problem, or possibly both. Once the type of problem has been identified, a more appropriate treatment program to reduce energy consumption may be devised.

2.2 Fenestration

A heating problem may be alleviated by reducing the thermal transmittance of the exterior building envelope of a building by using storm windows or insulating glass units, by installing storm doors, and by reducing air infiltration through cracks around windows and doors. On the other hand, cooling problems are treated most effectively by reducing solar radiation transmitted through glass by using shading devices and by reducing air infiltration through cracks around windows and doors.

2.2.1 Reducing Heat Conduction through Windows

The heat conduction through an existing single-pane window may be reduced by simply adding another pane, forming an air space between the two panes. Another retrofit option is to replace an existing single-pane window system with a double-pane insulating-glass system which consists of a hermetically sealed two-pane system with an air space between the panes. The coefficient of transmission (U) for various window systems is given in table 2-1.

TABLE 2-1. COEFFICIENT OF TRANSMISSION (U) FOR VARIOUS WINDOW SYSTEMS $Btu/h \cdot ft^2 \cdot °F$ [1]*

Description	Winter	Summer
Flat glass, single pane	1.13	1.06
Insulating glass-double pane		
3/16-inch air space	.69	.64
1/4-inch air space	.65	.61
1/2-inch air space	.58	.56
Insulating glass-triple pane		
1/4-inch air space	.47	.45
1/2-inch air space	.36	.35
Storm windows		
1 to 4-inch air space	.56	.54

* Numbers placed in brackets denote literature references cited at the end of the chapter.

The heat-loss rate (q) through a window is governed by the relation:

$$q = U \cdot A \cdot (T_i - T_o) \tag{1}$$

where U = coefficient of transmission

A = area of the window

$T_i - T_o$ = inside-to-outside temperature difference.

Reductions in thermal transmittance (U) are seen to be directly proportional to the heat-loss rate. Note that the coefficient of transmission for double-pane window systems is only slightly improved by increasing the thickness of the air space. As a general rule, the addition of a second pane reduces the heat conduction by approximately one-half.

The installation of storm windows provides reductions in heating energy requirement not only by reducing heat loss from a building, but also by reducing the number of heating hours. When the thermal transmittance of the building envelope is reduced, the heat from the lighting equipment and people can provide a larger proportion of the needed heat supply.

When reductions in heating energy requirements obtained from having storm windows installed are considered for buildings having only moderate internal heat release rates (family-housing units and dormitories), payback period as a function of energy price is plotted for various heating degree days in figure 2-2. Here the term "energy price" is defined as the dollar cost per 100,000 Btu (therm) of heating energy delivered to the conditioned space. Note that the payback period for storm windows decreases as the energy price increases and as the number of heating degree days increases.

It should be pointed out that the addition of storm windows may not always save energy. One such case is a building for which doors remain open during periods when the building is heated (such as a guard-gate building or some warehouses). For such buildings, heat is readily transferred due to massive air leakage, and often the indoor temperature may actually approach the outdoor temperature. Since the temperature difference across the windows in this instance is small, the heat

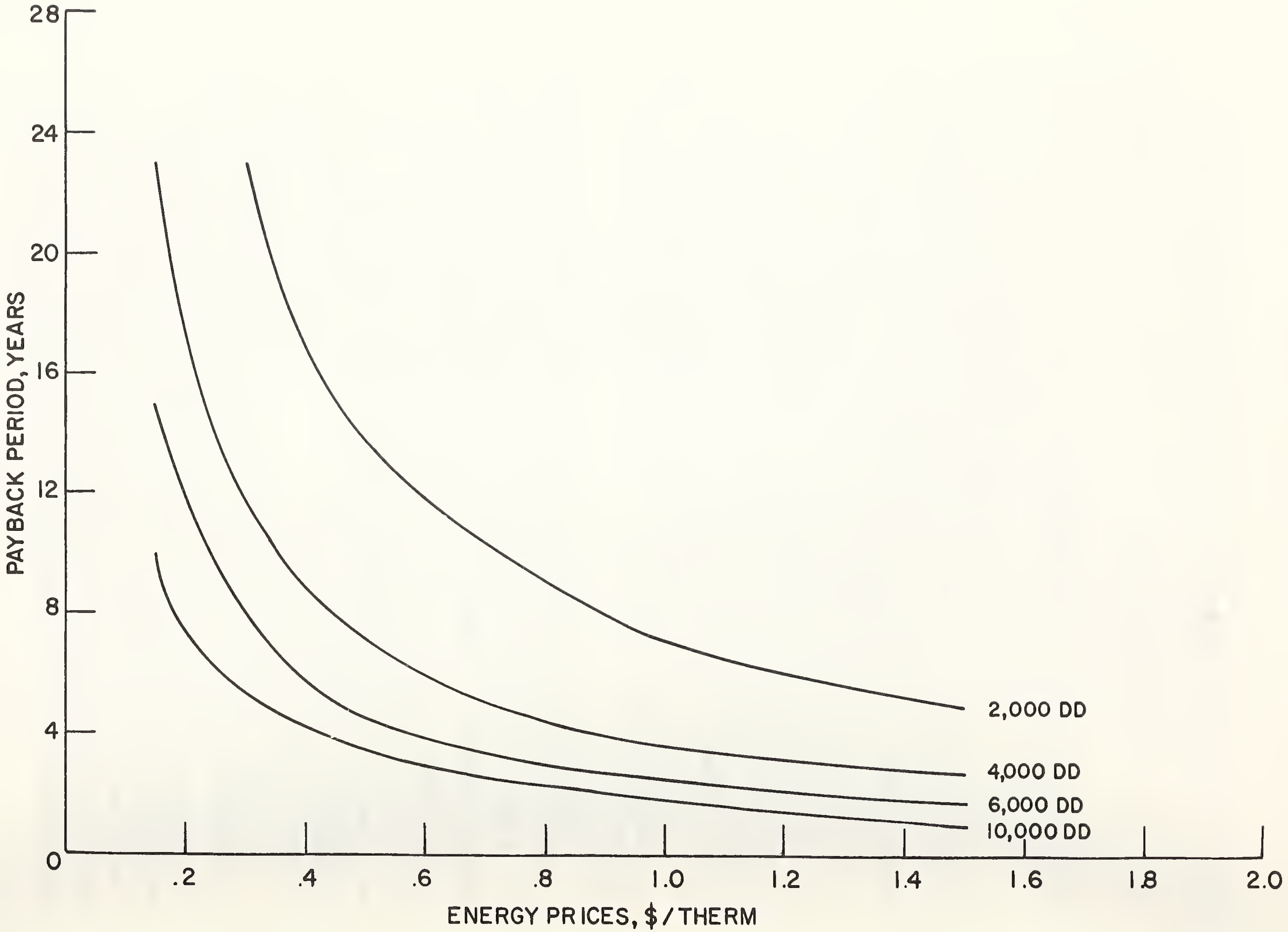


Figure 2-2. Payback periods for storm windows [2].

transfer through the window is small and the benefit derived from the installation of storm windows is negligible. The addition of a storm window to a building may actually increase the annual energy requirements in the case of a building having cooling requirements during periods when the outdoor temperature is lower than the indoor temperature. This condition occurs in a building having a large internal equipment load (such as a computer facility, a laundry facility, or a plant shop). The condition also occurs for buildings at geographic locations having hot summer days and cool nights during which natural cooling of the structure may occur. In these instances, the effect of installing a storm window may be to trap internal heat inside the structure and reduce the effect of natural cooling when there is no provision for introducing outside make-up air. A complete analysis of the annual benefit to be derived from storm windows is best accomplished with a computer simulation (see Appendix D).

2.2.2 Solar Shading

Solar shading devices can reduce the solar radiation transmitted through windows and thereby reduce the cooling loads for a building. The following energy conservation measures will provide reductions in transmitted solar radiation:

Interior shading devices
- draperies
- venetian blinds
- pull shades

Exterior shading devices
- awnings
- trees

Others
- heat-absorbing glass
- solar films and reflective coatings
- solar screens.

In evaluating a solar-shading device, the shading coefficient, defined as the fraction of solar radiation that is transmitted through a window system with a particular solar-shading device installed, may be used as a criterion. Shading coefficients for various shading devices are presented in reference [1]. Another factor which should be considered is the reduction of natural light that occurs when solar-shading devices are installed.

If designed properly, exterior shading devices are generally more effective than interior shading devices. The use of deciduous trees and shrubbery as a solar shading device for south-facing windows has the additional advantage that in the winter solar radiation is transmitted and will provide a free source of heat.

Internal shading devices such as blinds, curtains, and solar films present a somewhat less effective way of reducing solar radiation.

The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) [1], gives the reduction in solar heat gain that can be expected for a single plate-glass window when using internal shading devices as follows:

VENETIAN BLINDS		ROLLER SHADE		
		Opaque		Translucent
Medium	Light	Dark	White	Light
36%	45%	41%	75%	61%

Similar results are given by ASHRAE for closed draperies, showing that they can reduce solar gain by as much as 65 percent. These internal shading devices are relatively inexpensive and yet can be very effective. Of course they must be used properly; that is, shades and drapes should be closed during the cooling season on sunlit surfaces and opened when needed during the heating season.

Solar shading devices should be installed only for those window systems that are exposed to solar radiation. Since windows on the north side of a building are seldom exposed to solar radiation, shading devices need not be installed on north windows. However, interior shading devices such as insulated draperies will reduce the heat transfer due to air-to-air temperature difference.

Solar transmission through a window system may be substantially reduced by applying a reflective coating or film to one side. For instance, installing a storm window with a reflective coating in front of an existing single-pane window can reduce the overall daytime heat transmission through the window by a factor of 4 [5].

Other types of glass available for energy conservation applications are tinted glass and heat-absorbing glass. Tinted glass reduces light levels and glare. However, it absorbs solar energy, making it a hot radiant-heating panel when it is exposed to the sun. This may be desirable under certain climatic conditions. Reflective and heat-absorbing glass rejects solar energy and does not become hot. It reflects up to 80 percent of the radiant energy, which is very helpful for cooling in summer, but results in a loss of useful heat in winter. Natural light is lost when tinted or reflective glass is used. The effects on yearly consumption should be considered before a choice is made.

Summary of Energy-Conservation Measures for Fenestration

Minimal Expenses

° Inspect condition of indoor shading devices such as draperies and blinds which can reduce heat gain by as much as 50 percent. Keep indoor shading devices clean and in good repair.

° During the heating season, close all interior shading devices to reduce nighttime heat losses.

° Use opaque or translucent insulating materials to block off and thermally seal all unused windows.

Significant Expense

° Consider adding reflective and/or heat-absorbing film to glazing to reduce solar heat gains by as much as 80 percent. Be aware that such films will substantially reduce the benefits of natural lighting. Investigate their predicted years of durability.

° Consider adding reflective materials to the window side of draperies to reflect solar heat when draperies are drawn.

° Install indoor shading devices to reduce the transmission of solar radiation in the summer. They should be light-colored and opaque. In order to retard winter heat loss, draperies should be insulated or lined.

° Consider installation of outdoor shading devices, such as sunshades, awnings, balconies, and trees, which intercept solar heat before it has a chance to enter the building, and which dissipate heat outdoors rather than indoors. Adjustable sunshades permit solar radiation to enter during the heating season.

° Consider installation of storm windows if practical.

° If storm windows are not practical, consider reglazing with double- or triple-glazing, or with heat absorbing and/or reflective glazing materials. In selecting a replacement, choose a window system that has a low heat transmission coefficient.

° Consider the installation of solar screens.

° Use operable thermal shutters which decrease the composite "U" value to 0.1.

° Allow direct sun on windows from November through March. Shade windows from direct sun from April through October.

° Plant deciduous trees for summer and winter sunshading and windbreak effects.

2.3 Reducing Air Infiltration

The purpose of this section is to present background material and specific energy conservation measures for reducing air infiltration in buildings.

Air infiltration is the exchange of indoor air of a building with the outdoor air caused by indoor-to-outdoor temperature difference and wind forces. When the indoor air of a building is warmer than the outdoor air, it will be lighter than the outdoor air and will tend to rise and leak out of the top levels of a building while colder outdoor air enters the building at the lower levels. This driving force is called the stack effect. When an exterior surface of a building decelerates moving outdoor air, an inside-to-outside pressure difference is developed across the building surface which in turn causes air infiltration on one side and air exfiltration on the lee side.

In treating a building for air infiltration, one needs to establish priorities for energy conservation measures. Defects in the construction should be given first priority. This category includes items such as broken windows, holes in walls, large louvered ventilation exhaust openings frozen in an open position, window systems which have become warped to such an extent that they can no longer be closed, etc.

Of second priority are the less obvious cracks and small openings such as around windows, doors, etc. Such items are usually treated by applying caulking and weatherstripping.

An important point that should be made is that the use of conventional measures to reduce air infiltration on a building that is of tight original construction may not be effective in reducing air infiltration. National Bureau of Standards studies [3,4] have shown that conventional measures to reduce the obvious leaks and cracks in wood-frame residences were only able to reduce the air infiltration rate to one half air change per hour.

Energy conservation measures for reducing air infiltration in buildings are summarized in the following:

Windows and Skylights

Minimal Expense

° Replace broken or cracked window panes.

° Replace worn or broken weatherstripping around operable windows. If possible, install weatherstripping where none was installed previously.

° Weatherstrip operable sash if crack is evident.

° Caulk around window frames (exterior and interior) if cracks are evident.

° Rehang misaligned windows.

° Be certain that all operable windows have sealing gaskets and cam latches that are in proper working order.

° Consider posting a small sign next to each operable window instructing occupants not to open window while the building is being heated or cooled.

Exterior Surfaces

Minimal Expense

° Caulk, gasket or otherwise weatherstrip all exterior joints, such as those between wall and foundation or wall and roof, and between wall panels.

° Caulk, gasket or otherwise weatherstrip all openings, such as those provided for entrance of electrical conduits, piping, through-the-wall cooling and other units, outside air louvers, etc.

° When window and through-the-wall air conditioning units permit outdoor air to enter the building cover them when not in use. Specifically designed covers can be obtained at relatively low cost.

Elevator Shafts

Minimal Expense

° Seal elevator shafts around cabs at floor stations.

Windbreaks

Minimal Expense

° If open space is available, consider planting trees or large shrubs to act as windbreaks on the windward side of the building. The windbreak can have positive value on reducing wind impact, at least on lower floors. Trees and shrubs also can be used to reduce solar penetration.

Fireplace Damper

Minimal Expense

° Close the fireplace damper when not in use.

° Repair fireplace damper if it does not seal properly.

Doors

Minimal Expense

° Replace any worn or broken weatherstripping. Install weatherstripping where appropriate.

° Rehang misaligned doors.

° Caulk around door frames.

° Inspect all automatic door closers to ensure they are functioning properly. Consider adjustment for faster closing.

° Inspect gasketing on garage and other overhead doors. Repair, replace, or install as necessary.

° Consider placing a small sign next to each door leading to the exterior or unconditioned spaces advising occupants to keep door closed at all times when not in use.

° Consider installing signs on exterior walls near delivery doors providing instructions to delivery personnel on operation of doors.

° Establish rules for all building personnel regarding opening and closing of doors, directing them to keep them closed whenever possible.

° Consider installing automatic door closers on all doors from conditioned spaces leading to outdoor environment.

Significant Expense

° If the building has a garage, but does not have a garage door, consider installing one, preferably motorized to enable easier opening and closing.

° Consider use of a card-, key-, or radio frequency-operated garage door which stays closed at all times except when in use.

° Consider making delivery entrances smaller. The larger the opening, the more air that infiltrates when doors are open.

° Consider using an expandable enclosure for delivery ports. It reduces infiltration when in use because it can be adjusted to meet the back of a truck, reducing substantially the amount of air which otherwise would infiltrate.

° Consider installation of an air curtain, especially in delivery areas. The device prevents penetration of unconditioned air by forcing a layer of air of predetermined thickness and velocity over the entire entrance opening. (An expert in the field should be consulted before obtaining such a device, especially when highrise structures are involved. The degree of stack effect, among other things, determines its usability.)

° Consider installation of a vestibule for the entrances of a building, where practical. It should be fitted with self-closing weatherstripped doors.

° Consider utilizing revolving doors for the entrances. Studies have shown that such devices allow far less air to infiltrate with each entrance or exit. Use of revolving doors in both elements of a vestibule is even more effective. If high-peak traffic is involved, swinging doors can be used to supplement revolving doors.

Ventilation Exhaust Openings

Minimal Expense

° Permanently close off ventilation exhaust openings that are not essential or never used.

° Close off ventilation exhaust openings when not in use.

° Check spring-activated ventilation exhaust devices to make sure that they are in proper operating condition.

Significant Expense

° Consider replacing existing ventilation exhaust devices with spring-activated ventilation exhaust devices.

2.4 Retrofitting Building Surfaces for Energy Conservation

Often the most significant reductions in the energy requirement for space heating can be obtained by installing insulation into existing walls, ceilings, and floors of buildings. However, such energy conservation measures are not always cost effective due to the high initial cost required for the modification.

Energy conservation measures which increase the thermal resistance of the exterior envelope of a building envelope reduce the heating energy requirement by the following two mechanisms:

° reduction of heat-loss rate;

° reduction of heating hours.

When thermal insulation and storm windows are added to a building, there will exist a larger number of hours during periods of moderately cold weather for which space heating can be entirely provided by the lights, people, and equipment instead of the heating plant. The number of heating hours is thereby reduced. This latter phenomenon provides substantial reduction in the heating energy requirement which is often not considered in evaluating the cost effectiveness of energy conservation measures.

The addition of thermal insulation may not always provide reduction in the energy requirement for space cooling. In geographic locations having a large number of cooling days for which the nighttime outdoor air temperature falls below the indoor temperature of the building, wall insulation may not be effective in reducing the energy requirement for space cooling. An explanation is given in the following:

Wall insulation unquestionably reduces heat gain through the wall during the day. However, the insulation may eliminate a previous benefit of natural cooling during nighttime periods. When this occurs, the internal heat from lighting, equipment, people, and residual stored solar heat from the previous day must be removed by the air conditioning equipment. Before the insulation had been installed, a major portion of these internal gains could have been removed by heat loss through the walls to the cool outdoor air. In many instances the reduction of natural cooling during the night can actually offset the reduction in heat gains during the day.

Another effect of adding thermal insulation in the walls is to increase the number of cooling hours. For every building there is an outdoor temperature at which no energy is required for space cooling. At this condition, the heat loss through the building envelope is sufficient to remove the internal heat gains from lighting, equipment, people, and transmitted solar radiation. This outdoor air temperature is called the "outdoor balance temperature". When insulation is added to the walls of a building, the heat-loss rate from the building is reduced, and the outdoor temperature must reach a lower level before natural cooling is sufficient to remove the internal heat gains. Under such conditions, the number of hours for space cooling (by the air-conditioning equipment) is essentially increased.

The foregoing situation generally occurs at geographic locations having moderate summer energy requirements. For locations having extremely hot summers, for which the nighttime outdoor temperatures are usually hotter than the indoor temperatures of the buildings, wall insulation may produce reductions in the energy requirement for space cooling.

Additional topics on retrofitting of building envelopes are discussed in the following sections.

2.4.1 Minimum Requirements for an Insulating Material to be Eligible for Use by the Air Force

Insulating materials eligible for use by the Air Force must meet the appropriate Federal Specification and American Society for Testing Materials (ASTM) Tests for the particular material. Federal Specifications and ASTM designations for various insulation materials are summarized in table 2-2. If an insulation material does not have a Federal Specification and/or an ASTM test, a special waiver for its use should be obtained from the appropriate Major Command Headquarters for the Base.

In addition to the foregoing, every thermal insulation used in an Air Force building must have a flame-spread rating less than 25 when tested according to ASTM E 84-70 flame-spread test method, and it must have a smoke-development rating not over 30, except when the insulation is isolated from the interior of the building by masonry walls, in which case it may have a flame-spread rating up to 100, and no smoke-development limitation is required.

2.4.2 Thermal Resistance of Insulation Materials

The heat transfer rate (q/A) in $Btu/h \cdot ft^2$ through a wall, ceiling, or floor is customarily expressed by the relation:

$$q/A = \Delta T/R \qquad (2)$$

where ΔT = air-to-air temperature difference, °F

R = thermal resistance, $h \cdot ft^2 \cdot {}^\circ F/Btu$

Note that increasing the thermal resistance decreases the heat-transfer rate.

In selecting an insulating material, it is desirable to minimize the heat-transfer rate by using an insulation material with the largest R-value. Thermal-resistance values (R-values) per inch thickness for various insulation materials are given in table 3 of chapter 20 of the ASHRAE Handbook on Fundamentals. To find the total R-value for a particular thickness of insulation, multiply the R-value for the material by the thickness of the specimen.

TABLE 2-2. FEDERAL SPECIFICATIONS AND ASTM TEST DESIGNATIONS FOR VARIOUS INSULATIONS

Type of Insulation	Number	Federal Specification Title	ASTM Designation
Mineral Fiber	HH-I-521E	Insulation Blankets, Thermal (Mineral Fiber, for Ambient Temperatures)	C665-70
	HH-I-526C	Insulation Board, Thermal (Mineral Fiber)	C612-70 C726-72
	HH-I-545B	Insulation, Thermal and Acoustical (Mineral Fiber, Duct Material)	--
	HH-I-1030B	Insulation, Thermal (Mineral Fiber for Pneumatic or Poured Application)	C764-73
Mineral Cellular	HH -I-529B	Insulation Board, Thermal (Mineral Aggregate)	--
	HH-I-551E	Insulation, Block and Board, Thermal (Cellular Glass)	C552-73
	HH-I-574A	Insulation, Thermal (Perlite)	C549-73
	HH-I-585B	Insulation, Thermal (Vermiculite)	C516-67
Organic Fiber	HH-I-515C	Insulation Blanket, Thermal; and Insulation, Thermal (Loose-fill for Pneumatic or Poured Application) Cellulose, Vegetable and Wood Fiber	C739-73
	HH-I-528B	Insulation Batts and Blankets, Thermal (Vegetable Fiber)	--
	LLL-I-535A	Insulation Board, Thermal and Insulation Block, Thermal	C208-72
Organic Cellular	HH-I-524B	Insulation Board, Thermal (Polystyrene)	C578-69
	HH-I-530A	Insulation Board, Thermal	C591-69
	HH-I-573B	Insulation, Thermal (Flexible Unicellular Sheet and Pipe Covering)	C534-70
Air Spaces	HH-I-1252A	Insulation, Thermal, Reflective (Aluminum Foil)	--

For example, find the R-value for 6 inches of cellulose. From reference [1], the R-value per inch is found to be 3.7. For 6 inches the total thermal resistance is (3.7) (6) = 22.2.

2.4.3 Energy Conservation Measures for Exterior Building Envelopes

Some measures for reducing heat transmission through walls, roofs, and floors are given below:

Minimal Expense

° When exterior surfaces of a building are repainted, select a light color paint to provide small absorptance for solar radiation.

° When a new exterior roof surface is installed, select a light-colored exterior roof surface to provide small absorptance for solar radiation.

° Plant trees around structure to serve as a windbreak and provide natural shading of building surfaces from solar radiation.

° Repair major cracks and openings through exterior surfaces.

Significant Expense

° Where roof insulation is not practical, consider insulating the top floor ceiling. This can be done easily with blown insulation. In most cases, ceiling insulation also will require a vapor barrier placed on the warm side of the ceiling--if not integral with the insulation--to prevent structural damage caused by rot, corrosion or expansion of freezing water.*

° Consider blowing more insulation into cavity walls.

° Consider installing thermal insulation in the floor over unconditioned spaces (such as crawl spaces, unheated basements, and unheated garages).

° Consider installing thermal insulation around the perimeter edges of slab floors placed on grade.

° Consider insulating non-cavity walls.

---

* Caution: The installation of ceiling insulation may cause a reduction in attic temperatures and thereby increase the likelihood for pipes located above the insulation to freeze.

2.5 Figuring an Economically Optimum Allocation of Energy Conservation Resources for Building Envelopes

In determining the economically optimum level for a particular energy conservation investment (e.g., how much ceiling insulation to be installed), the methodology should be based on an incremental analysis. The energy savings and cost for each additional increment of investment should be calculated. Increments of investment should be added until the incremental cost of the investment exceeds the incremental saving in energy (expressed in dollars). This procedure is outlined in detail in chapter 6.

Based on the foregoing technique, a procedure was developed for estimating the economically optimum level of investment for various energy conservation actions for the exterior envelopes of family-housing units and dormitories. This procedure was specifically developed for these two types of buildings and in many cases will not be applicable to other types of buildings. The procedure is given in the following:

(a) Locate your Air Force Base on the heating zone map given in figure 2-3.

(b) Locate your Air Force Base on the cooling zone map given in figure 2-4. If your Air Force Base is on the borderline of two zones, select the zone in which the climate is more typical of your area.

(c) Locate the heating index from table 2-3 by finding the number at the intersection of your heating zone row and the heating fuel cost column (to the nearest cost shown).

(d) If the building being considered is air conditioned, locate the cooling index for attics from table 2-4 by finding the cooling zone and cooling costs to the nearest cost shown.

(e) If the building being considered is air conditioned, locate the cooling index for walls from table 2-5 by finding the cooling zone and cooling costs to the nearest cost shown in the table.

(f) Find the sum of the heating and cooling index for attics.

(g) Find the sum of the heating index and cooling index for walls.

(h) Find the resistance value of insulation recommended for the attic and around attic ducts from table 2-6.

(i) Find the recommended level of insulation for floors over unheated areas from table 2-7. Using table 2-7, check to see whether storm doors are economical for the building. Storm doors listed as optional may be economical if the doorway is heavily used during the heating season.

(j) Find the recommended level of insulation for the walls and ducts in unheated areas from table 2-8. Table 2-8 also shows the minimum economical storm window size in square feet for triple-track storm windows.

EXAMPLE

Consider an example house located on the maps of figures 2-3 and 2-4. It is located in heating zone III (figure 2-3) and in cooling zone B (figure 2-4). This house currently uses fuel oil at a cost of 34¢ a gallon to heat. It uses electricity at 4¢ a kilowatt hour to cool.

(a) From figure 2-3, the example house is located in heating zone III.

(b) From figure 2-4, the example house is located in cooling zone B.

(c) From table 2-3, heating index is found to be 20.

(d) From table 2-4, cooling index for attics is 5.

(e) From table 2-5, cooling index for walls is found to be 2.

(f) The sum of the heating index and cooling index for attics is 25.

(g) The sum of the heating index and cooling index for walls is 22.

(h) From table 2-6, the recommended resistance value is R-30 for the ceiling and R-16 for ducts.

(i) From table 2-7, the recommended resistance for a floor over an unheated space should be R-19. A storm door is seen to be optional.

(j) From table 2-8, the example house should have full wall insulation if none existed previously, and R-16 around ducts located in unheated regions other than the attic.

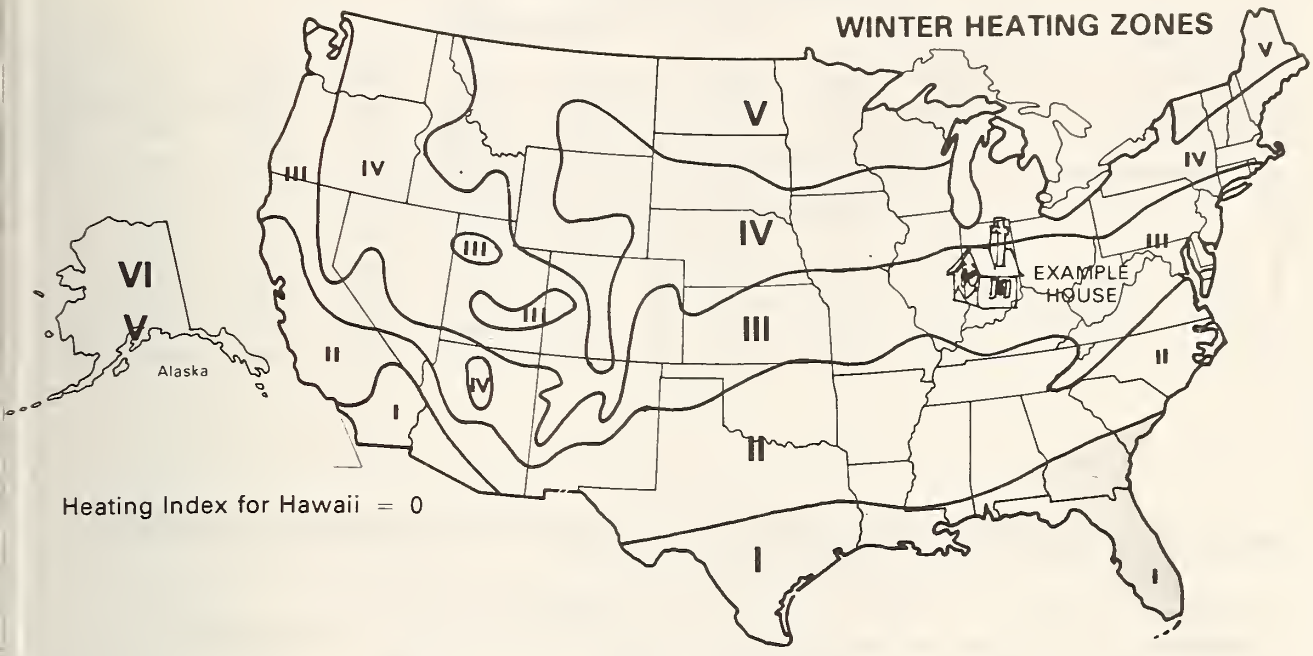


Figure 2-3. Winter heating zones.

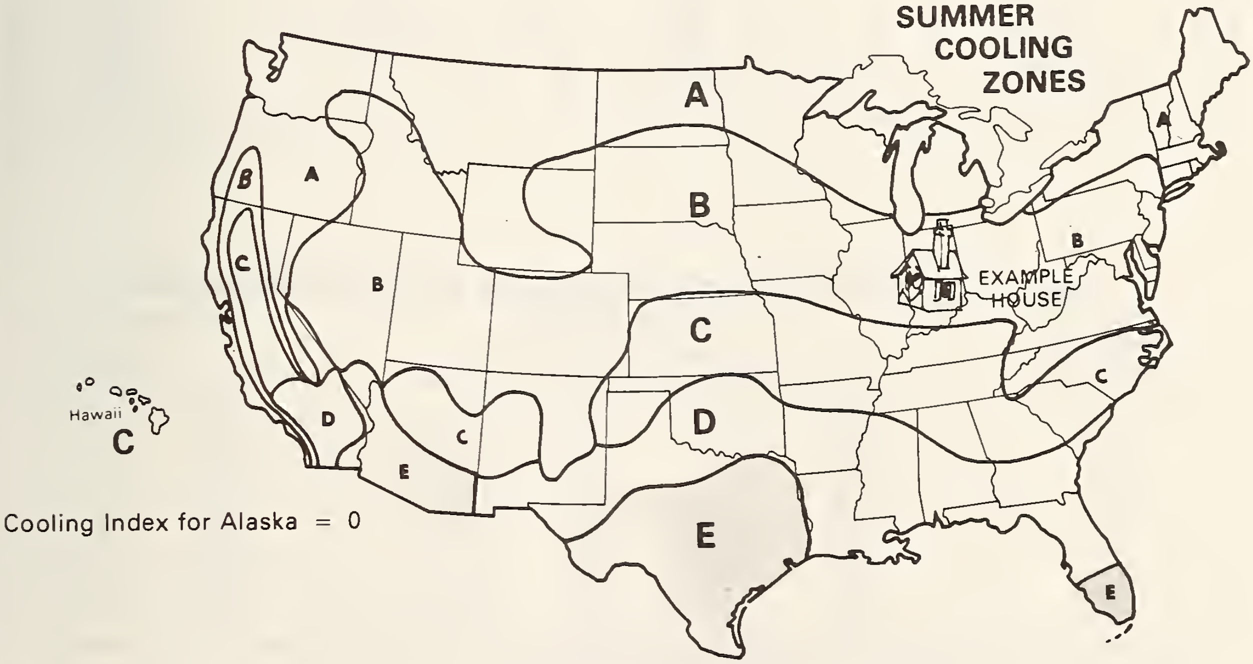


Figure 2-4. Summer cooling zones.

TABLE 2-3. HEATING INDEX

Type of fuel:		Cost per unit*									
Gas (therm)		9¢	12¢	15¢	18¢	24¢	30¢	36¢	54¢	72¢	90¢
Oil (gallon)		13¢	17¢	21¢	25¢	34¢	42¢	50¢	75¢	$1.00	$1.25
Electric (kWh)					1¢	1.3¢	1.6¢	2¢	3¢	4¢	5¢
Heat pump (kWh)		1¢	1.3¢	1.7¢	2¢	2.6¢	3.3¢	4¢	6¢	8¢	10¢
HEATING ZONE	I	2	2	3	3	4	5	6	9	12	15
	II	5	6	8	9	12	15	18	27	36	45
	III	8	10	13	15	20	25	30	45	60	75
	IV	11	14	18	21	28	35	42	63	84	105
	V	14	18	23	27	36	45	54	81	108	135
	VI	22	28	36	42	56	70	84	126	168	210

TABLE 2-4. COOLING INDEX FOR ATTICS

Type of air conditioner:		Cost per unit*						
Gas (therm)		9¢	12¢	15¢	18¢	24¢	30¢	36¢
Electric (kWh)		1.5¢	2¢	2.5¢	3¢	4¢	5¢	6¢
COOLING ZONE	A	0	0	0	0	0	0	0
	B	2	2	3	4	5	6	7
	C	3	5	6	7	9	11	13
	D	5	6	8	9	12	15	18
	E	7	9	11	14	18	23	27

TABLE 2-5. COOLING INDEX FOR WALLS

Type of air conditioner:		Cost per unit*						
Gas (therm)		9¢	12¢	15¢	18¢	24¢	30¢	36¢
Electric (kWh)		1.5¢	2¢	2.5¢	3¢	4¢	5¢	6¢
COOLING ZONE	A	0	0	0	0	0	0	0
	B	1	1	2	2	2	3	4
	C	2	2	3	4	5	6	7
	D	3	3	4	5	7	8	10
	E	4	5	6	8	10	13	15

TABLE 2-6. ATTIC FLOOR INSULATION AND ATTIC DUCT INSULATION

INDEX	ATTIC INSULATION	Approximate Thickness			DUCT INSULATION*	
Heating Index Plus Cooling Index for Attics	R-Value	Mineral Fiber Batt/Blanket	Mineral Fiber Loose-Fill**	Cellulose Loose-Fill**	R-Value	Approximate Thickness
1-3	R-0	0"	0"	0"	R-8	2"
4-9	R-11	4"	4-6"	2-4"	R-8	2"
10-15	R-19	6"	8-10"	4-6"	R-8	2"
16-27	R-30***	10"	13-15"	7-9"	R-16	4"
28-35	R-33	11"	14-16"	8-10"	R-16	4"
36-45	R-38	12"	17-19"	9-11"	R-24	6"
46-60	R-44	14"	19-21"	11-13"	R-24	6"
61-85	R-49	16"	22-24"	12-14"	R-32	8"
86-105	R-57	18"	25-27"	14-16"	R-32	8"
106-130	R-60	19"	27-29"	15-17"	R-32	8"
131-	R-66	21"	29-31"	17-19"	R-40	10"

* Use Heating Index only if ducts are not used for air conditioning ** High levels of loose-fill insulation may not be feasible in many attics. ***Assumes that joists are covered; otherwise use R-22.

TABLE 2-7. INSULATION UNDER FLOORS AND STORM DOORS

INDEX Heating Index Only	INSULATION UNDER FLOORS* R-Value	Mineral Fiber Batt Thickness	STORM DOORS
0-7	0**	0"**	None
8-15	11**	4"**	None
16-30	19	6"	Optional
31-65	22	7"	Optional
66—	22	7"	On all doors

* If your furnace and hot water heater are located in an otherwise unheated basement, cut your Heating Index in half to find the level of floor insulation.

** In Zone I and II R-11 insulation is usually economical under floors over open crawlspaces and over garages; in Zone I insulation is not usually economical if crawlspace is closed off

TABLE 2-8. WALL INSULATION, DUCT INSULATION, AND STORM WINDOWS

INDEX Heating Index Plus Cooling Index for Walls	WALL INSULATION (blown-in)	INSULATION AROUND DUCTS IN CRAWLSPACES AND IN OTHER UNHEATED AREAS (EXCEPT ATTICS)* Resistance and Approximate Thickness	STORM WINDOWS (Triple-Track) Minimum Economical Window Size
0-10	None	R-8 (2")	none
11-12	Full-Wall Insulation Approximately R-14	R-8 (2")	20 sq. ft.
13-15		R-16(4")	15 sq. ft.
16-19		R-16(4")	12 sq. ft
20-28		R-16(4")	9 sq. ft.
29-35		R-16(4")	6 sq. ft.
36-45		R-24(6")	4 sq. ft.
46-65		R-24(6")	All windows**
66—		R-32(8")	All windows**

* Use Heating Index only if ducts are not used for air conditioning. **Windows too small for triple-track windows can be fitted with one-piece windows.

REFERENCES

1. ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigeration, and Air Conditioning Engineers, 1972.

2. Retrofitting Existing Housing for Energy Conservation: An Economic Analysis, S. R. Petersen, Building Science Series 64, National Bureau of Standards, Dec. 1974.

3. Retrofitting an Existing Wood-Frame Residence for Energy Conservation--An Experimental Study, D. M. Burch and C. M. Hunt, Building Science Series (in review Jan. 1977), National Bureau of Standards.

4. Comparison of Measured and Computer-Predicted Thermal Performance of a Four-Bedroom Wood-Frame Townhouse, B. A. Peavy, D. M. Burch, and F. J. Powell, Building Science Series 57, National Bureau of Standards, April 1975.

5. Technical Options for Energy Conservation in Buildings, NBS Technical Note 789, National Bureau of Standards, July 1973.

CHAPTER 3. MODIFYING MECHANICAL SYSTEMS AND OPERATING PRACTICES FOR ENERGY CONSERVATION

CONTENTS

CHAPTER 3. MODIFYING MECHANICAL SYSTEMS AND OPERATING PRACTICES FOR ENERGY CONSERVATION

3.1 Modifying Building Operation

3.1.1 Night Temperature Setback

Energy requirements for space heating of family-housing units and dormitories may be reduced 5 to 15% by setting back the indoor temperature 10° F during an 8-hour night-time period. Night temperature setback will also produce substantial saving in other types of Air Force buildings. Clock thermostats are commercially available and range in price from $50 to $100*. The fuel savings for the first couple of years will usually pay for the initial cost of the clock thermostat; fuel savings after this short payback period will represent money in the bank. The heat loss from buildings is proportional to the difference in temperature between the indoor and outdoor air. Night-time reduction of indoor air temperature will provide significant reductions in the night-time fuel consumption. However, when the indoor temperature is set up in the morning, the heating plant must run a long time to reheat the building. Thus, some of the night-time fuel savings are offset by increased fuel consumption in the morning to reheat the building.

Recently Nelson of Honeywell conducted a research study [1]** to investigate the heating energy savings for residential buildings achieved by night temperature setback of the indoor air temperature. The Nelson data is plotted to show fuel savings versus annual heating degree days in figure 3-1. The top line gives the fuel savings for a 10° F, 8-hour setback, whereas the bottom line gives the fuel savings for a 5° F, 8-hour setback.

Notice that increasing the amount of setback significantly increases the fuel savings. Also, the percent fuel savings becomes greater as the annual heating degree days decrease. A surprising finding of the Nelson study [1] was that the percent fuel savings resulting from night

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* 1976 prices.
** See references cited at end of chapter.

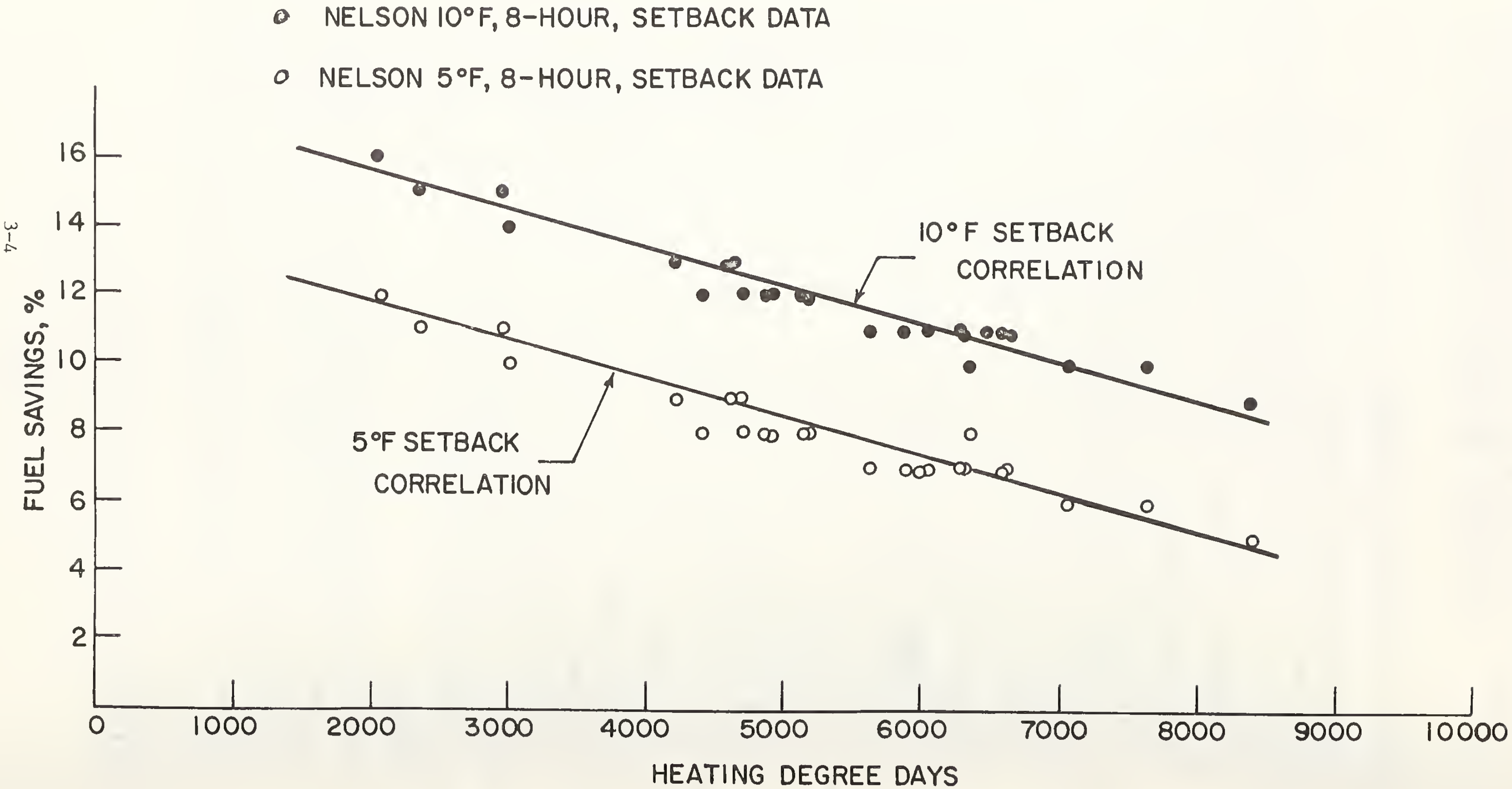

Figure 3-1. Fuel savings achieved by night temperature setback.

temperature setback was practically independent of the degree of insulation of the building. Thus, the percent fuel savings would be the same for a poorly insulated building as for a well insulated building.

The data from Ref. [1], in the strict sense, are applicable to residential buildings and, in most cases, may be applied to dormitories. Night temperature setback for other types of Air Force buildings will usually provide substantial energy savings. However, the amount of savings for these other-type buildings will depend on the amount of internal loads and the characteristics of the heating plant.

3.1.2 Reducing Indoor Temperature

For space heating applications, the energy savings that may be achieved by reducing the indoor temperature from one level to another may be estimated from the relation:

$$\% \text{ Savings} = \frac{\Delta T_i}{T_B - T_o} \tag{1}$$

where ΔT_i = reduction in indoor temperature, °F

T_B = outdoor balance temperature, °F

T_o = winter average outdoor temperature, °F.

The outdoor balance temperature is the outdoor temperature for which no space heating energy from the heating plant is required (usually 60° F to 65° F). At this condition the internal heat from lights, equipment, occupants, and transmitted solar radiation is sufficient to heat the building. Winter average outdoor temperatures for a wide range of geographic locations are given in table 1 of chapter 43 of the 1973 ASHRAE Systems Volume [2].

Example:

The indoor temperature of a dormitory located in Washington, D.C. is reduced by 4° F. Calculate the reduction in the heating energy requirement.

The outdoor balance temperature for the dormitory is taken as 65° F, and the winter average temperature for Washington, D.C. is 45.7° F. Thus,

$$\% \text{ savings} = \frac{4}{65-45.7} = 20.7\%$$

Caution: For interior spaces and for spaces having large internal loads (due to lights, environment, people, etc.), a lowering of the thermostat can have a negative effect on energy consumption; in this situation, know your system before making adjustments.

For space cooling applications, raising thermostat settings to 80° F and humidistat settings to 60% RH could reduce the space-cooling energy requirement by an estimated 15% compared with operation at 75° F and 50% RH. (Note: in some reheat systems internal operating temperatures must also be raised along with thermostat settings, to effect the desired energy savings.)

3.1.3 Economizer Cycles

The economizer cycle provides for the introduction of outdoor air into a conditioned space of a building in order to reduce the number of operating hours of the refrigeration equipment. When the temperature and relative humidity of the outdoor air are within prescribed limits, and the thermostat is calling for cooling, the economizer cycle can introduce outdoor air into the building instead of providing space cooling with air conditioning equipment. Use of an economizer cycle in residential buildings can make a considerable reduction in compressor operating time, according to an analog computer study recently completed by Honeywell [3]. Figure 3-2 gives the calculated savings when the economizer cycle is used on a residential building.

3.1.4 Modifying Humidity Control of Mechanical Equipment

Perhaps the most definitive experimental study on thermal comfort is one conducted on over 1600 college students at Kansas State University over the past several years [4]. The students were exposed

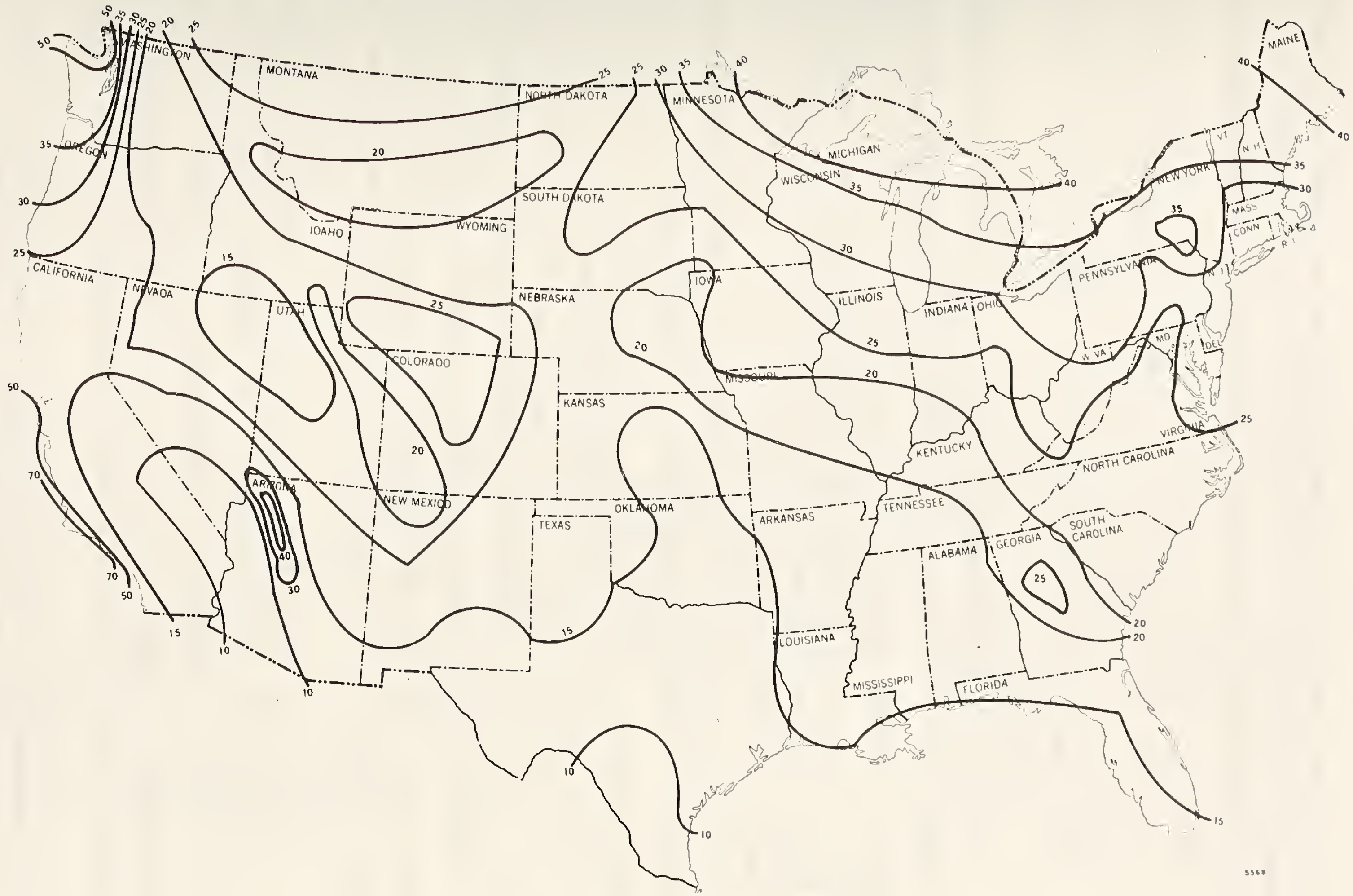

Figure 3-2. Percent savings with air economizer used in a residential building [3].

(over a 3-hour period) to uniformly heated or cooled environment (wall temperatures identical to the air temperature) with moderate air velocities, and asked to vote on a scale from 1 to 7 which ranged from cold to hot:

1. cold
2. cool
3. slightly cool
4. comfortable
5. slightly warm
6. warm
7. hot.

All subjects, male and female alike, were clothed in cotton twill shirts and trousers (shirts worn outside of the trousers), and cotton sweat socks without shoes. The net insulating value of the clothing ensemble, as computed by means of an electrically heated copper manikin, was 0.6 clo*, which has been generally accepted as the standard clothing for thermal clothing studies.

Even though the results are available for men and women separately and also for the votes taken at each half-hour interval for the entire 3 hours of exposure, the data shown in figure 3-3 is an average for the votes of men and women taken together at the end of the 3 hours. One should note the very slight dependence of comfort on relative humidity. Studies such as these have led to the conclusion that temperature is far more important than relative humidity at comfort conditions. It is interesting to note that at the comfort condition (KSU=4), a relative humidity change of 13% is equivalent to a dry-bulb temperature change of 1° F.

In specifying the comfortable operating conditions for a building, one approach is to select bounds for the relative humidity (say between 20 and 60%) and KSU upper and lower bounds (say $3 \leq KSU \leq 5$). The cross-hatched region of figure 3-3 includes all the combinations of dry-bulb temperatures and relative humidities that produce the above-defined

---

* clo represents a measured resistance to heat transfer of the clothing ensemble of 0.88 $ft^2 \cdot h \cdot {}^\circ F/Btu$ or equivalent of an overall heat-transfer coefficient = 1.13 $Btu/h \cdot ft^2 \cdot {}^\circ F$.

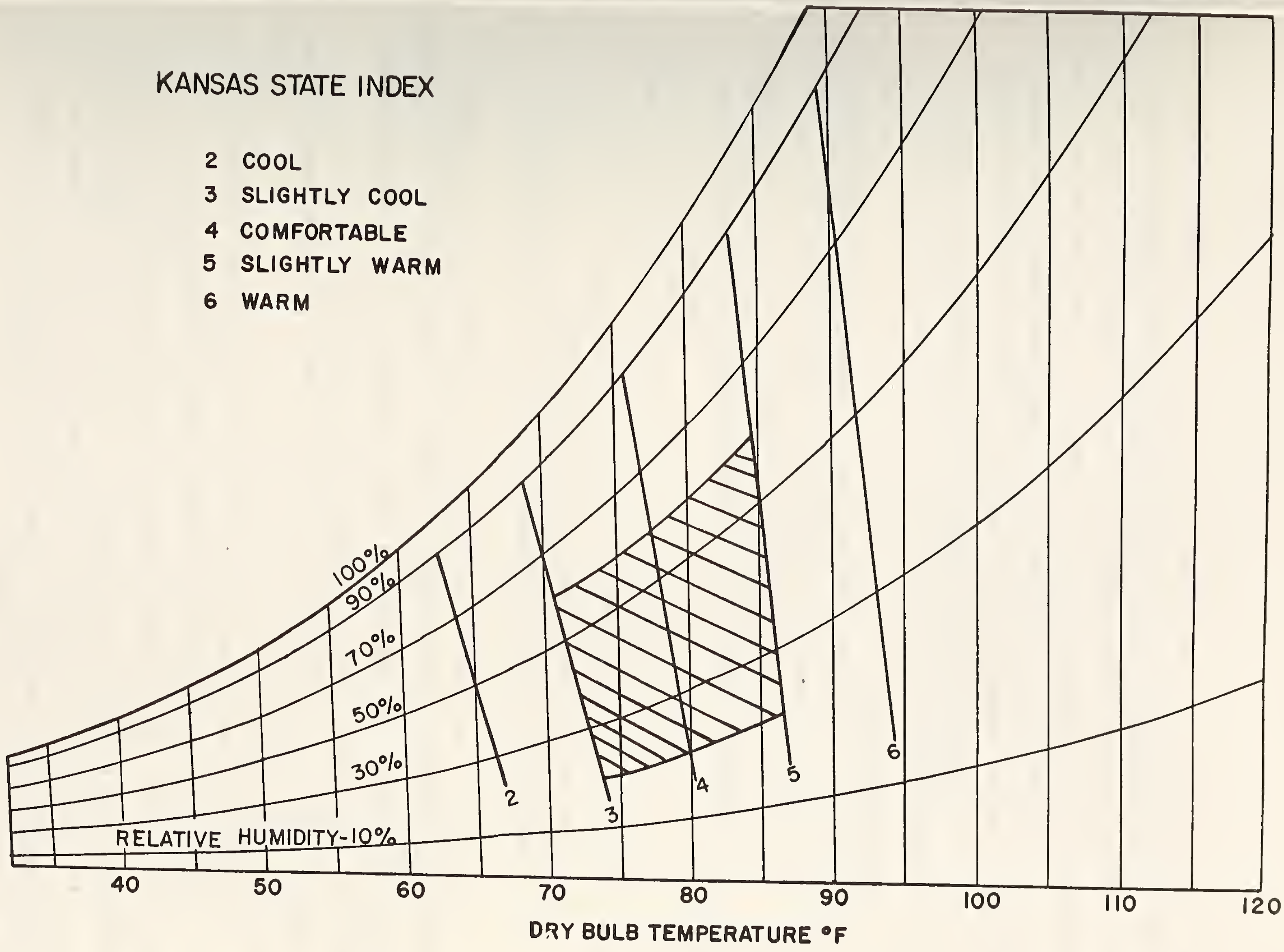

Figure 3-3. Various comfort levels shown on psychrometric chart.

comfort conditions. The mechanical systems of some buildings presently in use are designed to provide indoor humidity control within a rather narrow range. Such systems expend considerable amounts of energy for humidification and dehumidification processes. In many instances, these mechanical systems can be modified so as to maintain indoor conditions within the comfort range and at the same time reduce unnecessary humidification and dehumidification, thereby reducing heating and cooling energy requirements.

3.1.5 Ventilation Air

Air circulation systems are provided in buildings to supply or remove heat from a space, to provide outside air to remove odors and contaminants, and to provide oxygen for occupants and combustion. It has been common practice to determine required air quantities for heating and cooling spaces from the equation:

$$Q = H/(1.08 \cdot \Delta T) \qquad (2)$$

where Q = air quantity, ft^3/min

H = heat removed (or added) to the space, Btu/h

ΔT = temperature difference between supply air and space air, °F.

The quantity of outside air introduced frequently has been based on regulations that state requirements in cfm/ft^2, cfm/person, or air changes per hour. Many of the regulations call for generous air quantities that result in extravagant use of energy; some systems employ even more excessive outside air quantities due to poor design, faulty installation, or inattentive operation.

Significant energy savings can often be achieved by reducing outside air quantities to recommended Air Force criteria levels given in table 3-1. Also, large savings can be achieved by shutting down systems completely when building is unoccupied. It must be noted that the interaction of related (and frequently interconnected) supply, return, make-up, and exhaust systems must be considered when systems are shut down or dampers are closed or adjusted.

TABLE 3-1. OUTSIDE AIR VENTILATION STANDARDS [5].

	AIR COND		MECH VENT		REMARKS
	cfm/sf	cfm person	Summer cfm/sf	Winter cfm person	
Auditorium, Theater	....	5	6–12	5	....
Automotive Shop, Garages	....	....	....	....	See para 6–63 and 6–69
Bowling Alleys (Personnel occupied space only)	....	15	5–10	15	....
Chapel	NONE	NONE	6	....	....
Chapel Annex (Classroom)	0.25	....	5–10	5	....
Classrooms	....	5	5–10	5	....
Conference and Briefing Rms Cocktail Lounge	....	25	5–10	25	....
Data Processing and Electronic Equipment Room	....	25	....	....	See chapter 20
Dining Hall, Messes, Clubs, Restaurant, and Snack Bar	0.25	....	5–10	....	....
Dormitory, Officers Quarters	....	5	3–6	....	See atch 12
Flight Control Center	....	25	....	....	....
Gymnasium	....	....	6	5	....
ACND Areas	....	....	15	....	....
Kitchens (Dining Hall and Hospital)	....	....	....	....	See paras 6–65 and 6–82h
Medical Facilities	....	....	....	....	See section J
Mechanical Equip Rm	....	....	3	....	Total air supply to limit rm temp rise to 10° max depending on satis equip operation
Offices, Drafting Rm, Library	....	5	3	10[1]	Interior rooms only[1]
Ozalid Room	....	....	3[2]	....	Exhaust duct connected[2] to equip, summer and winter
Paint Finishing Rm Paint Hangar, Paint Spray Booth	....	....	....	....	See section H
Parachute Wrapping and Storing	....	5	3	....	....
Security and Switchboard Rms	....	5	4	10[3]	Interior rooms only[3]
Shops, Aircraft Maintenance	....	5	6–12	5	....
Shops, Civil Engineering	....	....	6–12	....	....
Toilets and Locker Rooms	....	....	2.0	....	Required year-round
Shower Rooms	....	....	2.5	....	Required year-round

3.2 HVAC Equipment in Large Buildings

Energy consumption by HVAC equipment in large buildings can be lowered by the following measures:

- reducing energy consumption for fans;
- reducing energy consumption for pumps;
- reducing distribution system heat losses;
- eliminating steam leaks;
- eliminating unnecessary reheat;
- flashing condensate into low-pressure steam;
- increasing condensate return to boilers;

These measures are discussed in the following sections.

3.2.1 Reduce Energy Consumption for Fans

Implementing energy conservation methods which reduce heating and/or cooling loads permits reduction of air volume handled by the central HVAC system. Whenever the volume of air is reduced, the resistance to air flow in that section of the system is also reduced. This occurs because the resistance of any system depends upon the quantity of air circulated in the system as well as the characteristics of the various portions of the system including:

- intake and discharge louvers;
- heating and cooling coils;
- filters;
- apparatus casings;
- ductwork system; and
- registers, grilles, and diffusers.

The total resistance to air flow is the sum of the resistance of the individual parts. Although straight lengths of duct generally offer little or no opportunity for resistance reductions, the potential of other items is substantial.

The resistance to air flow through filters is a function of filter construction, type of media, area of media per unit volume (proportional to actual velocity through the media, not face velocity) and the dirt load at any given time. In general, resistance to air flow increases with filter efficiency, although there are some high efficiency filters which also have low resistance. To determine the "dirty" resistance limit, contact the manufacturer of the particular filter. Consider installing a manometer across each filter bank to indicate when the filters should be changed. If the building contains many air systems, consider the use of an alarm system to report filter conditions at a central point.

Many filters impose high resistance due to their inherent characteristics and/or because they exceed required filtration standards. Determine filtration requirements in terms of the ambient air conditions and space air quality requirements. One of the current accepted methods for determining air filtration efficiency is the ASHRAE Atmospheric Dust Spot Efficiency Test Method [6]. Check present filter installation to determine if an alternative type of filter will meet building needs at operating costs sufficiently reduced to provide an economic payback.

Resistance to air flow through coils depends largely upon the number of rows or depth required for adequate heat transfer. Cooling coils are usually many rows deep because of the small temperature differences between coil and air, and consequently offer a high resistance to air flow. Heating coils which have a higher temperature difference are usually fewer rows deep than cooling coils and impose less resistance to air flow.

In some air handling systems the coils are placed in series and impose combined resistance year-round, even though only one coil may be used at any given time. Where the cooling medium is chilled water and the heating medium is hot water, it may be possible to remove the heating coil and repipe the cooling coil to provide both heating and cooling (though not simultaneously). Eliminating the heating coil will reduce the system resistance and thereby achieve reductions in fan horsepower. A side benefit of using the cooling coil for both heating and cooling is that the extra heat-transfer surface of the cooling coil provides opportunities to lower the hot-water temperatures and use low-grade waste heat.

In other installations, it may be possible to arrange the coils for parallel flow. However, in all instances, consideration must be given to the function of the coils, their operating limits and the danger of freeze-up.

Resistance to air flow caused by duct fittings on the inlet and discharge side of fans can be reduced and fan performance significantly increased by modifying the shape of the fittings (see Ref. [7] for recommendations). Where space permits, modify abrupt changes in sections of duct work where velocities exceed 2000 ft/min to long taper-fittings, and install turning vanes in square bends.

Supply and exhaust fans must operate at a pressure sufficient to overcome the resistance to air flow through the ducts serving the outlet farthest from the fan (index outlet). When the index outlet is remote from other parts of the system and is served by a long run of duct, it imposes a power consumption penalty on the whole system. It is often worthwhile to replace this section of duct with one of larger cross section and lower resistance.

To achieve correct volumes at each grille or register, dampers in low resistance branches are adjusted until all branches are of resistance equal to the index run. Many systems when initially started have dampers that are closed more than they need be, adding unnecessary resistance. Also, it is common practice to reduce fan volume by closing down dampers on the fan inlet or outlet rather than by reducing fan speed.

Check whether these situations occur in the building and, if necessary, rebalance the system by first opening fully any dampers which are not used to proportion air flow between branches or outlets (main dampers close to fan). Identify the index outlet and fully open any dampers between this outlet and the fan. Measure the volume at the index outlet and adjust branch dampers successively (starting with the next longest run) until proportional volumes are achieved at each outlet. Each damper adjustment will affect flow rates in branches already adjusted, but two or three successive adjustments of the whole system will give good balance.

Reducing the resistance to air flow in an existing system results in an increase of air delivery which, in turn, permits a reduction in fan speed to bring air volume down to the original level. In the case of centrifugal fans (the type most commonly used), the power input varies directly with the cube of the speed. As such, any reduction in speed, which is proportional to air volume, results in a substantial decrease in power input.

The basic fan laws for centrifugal fans are:

- volume varies directly with speed;
- pressure varies directly with square of speed;
- power input varies directly with cube of speed; and
- volume varies directly with the square of pressure.

Reductions in the volume of air flow will provide three-fold reductions in fan brake horsepower (BHP). Accordingly, if the volume of air in a system is reduced by 10% the reduction in fan BHP will be about 30%:

In summary, to reduce energy consumption of fans:

- reduce heating and cooling loads;
- reduce resistance to air flow, as is feasible, in each of the areas listed above;
- measure the resulting new air volume and determine the new air volume required to meet new loads;
- reduce fan speed as appropriate. Where motor sheave is adjustable, open the "v" to reduce its effective diameter and adjust motor position or change belts to maintain proper tension. If no further adjustment is possible, change motor sheave. If the full load on the motor is less than 60% of the nameplate rating, consider changing the motor to the next smaller size.

When installing new motors:

- match the phase and frequency with electrical distribution system characteristics;
- when motors cannot exactly match system voltage, use the following criteria:
  - For motor loaded to under 75% of its capacity, select one having a voltage rating slightly more than system voltage.

-- For motor loaded to more than 75% of its capacity, select one having a voltage rating slightly under the system voltage. Problems may arise if nameplate rating of motor selected were to be in excess of system voltage by more than 5%--a likely situation in these days of brownout and voltage cutbacks.

° Ascertain that motor has sufficient starting torque for the fan. This caution applies to fans with large wheels having considerable inertia.

Other measures to consider for reducing energy consumption of fans include:

° Determine the tradeoff between raising supply air temperatures during the cooling season to reduce refrigeration power and lowering the temperature to decrease the volume of air to reduce fan energy.

° Reduce air volume and horsepower of existing variable air volume systems; convert other systems to variable volume.

° Seal off unused portions of exhaust hoods in kitchens and cafeterias to reduce air volume and fan energy where it is difficult or uneconomic to change fan speeds.

° Eliminate preheat coils whenever possible without increasing freezeup possibilities.

° When the same system is used for both heating and cooling, consider installing a control to reduce total air volume during the heating season. More air is required during the cooling cycle because of lower temperature differentials.

° In large buildings where steam is available, consider turbine-driven fans with variable speed control.

° In high-bay buildings, consider distribution systems that can reduce stratification. Stratification is undesirable, since it increases heat loss from top portions of the zones of the building during the winter.

3.2.2 Reduce Energy Consumption for Pumps

Pumps used to circulate hot water, chilled water and condenser water are governed by basic laws similar to those for fans. Therefore, a reduction in flow rate or in system resistance to flow results in reduction in pumping power.

However, since water in hydronic systems is moved through closed circuits, and the flow rate through the various elements may be varied by means of throttling, mixing or by-pass valves, these systems are not subject to the same balancing techniques as are air systems. Furthermore, most pumps for these systems are direct-driven by constant speed motors. Therefore, one's ability to make adjustments for pumps, as can be done for fans, is sharply reduced.

But in view of the high potential for energy savings due to the number of operating hours and to the frequency of overdesign to provide safety margins, it is well worth while to consider a complete engineering analysis to determine if modifications to the system will pay off in energy savings. One measure that can be tried in installations having parallel pump systems is to reduce the number of pumps in operation at one time; even this operation demands careful analysis of its effect on the performance of the system and its components.

3.2.3 Reduce Distribution System Heat Losses

Heat losses from boilers, steam and hot water piping can comprise a sizable portion of the total energy consumed in a building.

In the case of exposed piping, examine insulation carefully. Repair or replace damaged insulation and consider adding insulation to that which already exists as well as installing it on pipes which currently are bare. Table 3-2 indicates heat losses in pipes without insulation and with varying amounts of insulation. Table 3-3 indicates minimum recommended insulation for various different types of piping systems.

In the case of buried pipe, addition of insulation is a costly procedure, even if possible. Usually it will be necessary to replace the underground piping system. Heat loss can best be determined by measuring flow and temperature at inlet and outlet of buried runs. Grass that is different in color along underground runs, or runs where snow melts more quickly than in adjacent areas, often indicate underground distribution systems with failing pipe insulation. Where there is a considerable amount of buried pipe, the use of aerial surveys using an infrared camera is an effective technique (see Chapter 5).

TABLE 3-2. HEAT LOSS FROM BARE AND INSULATED PIPE*

conditions: 250° F pipe temperature, 80° F ambient temperature, calcium-silicate insulation

Pipe Size in	Bare pipe, Btu/ft·hr	Insulated pipe, Btu/ft·hr — Thickness of insulation, In: 1	2	3	4	5	6
1	262	35	26	27	19	17	17
2	456	53	36	29	25	23	21
3	657	72	46	36	32	28	26
4	833	87	55	43	36	32	29
6	1,202	125	75	56	46	40	35
8	1,543	158	92	69	55	48	43
10	1,902	192	108	80	66	56	50
12	2,246	215	125	93	75	64	57

*"The 1975 Energy Management Guidebook" published by Editors of Power Magazine, McGraw Hill Inc., N.Y., N.Y., 1975.

TABLE 3-3. MINIMUM RECOMMENDED INSULATION FOR VARIOUS PIPING SYSTEMS*

Types of piping systems	Temperature range, °F	Insulation thickness, In		
		Pipe sizes, in.		
		Up to 2	2 1/2 to 4	Over 4
Heating systems				
H-P steam[1]	306-400	3	4	4 1/2
M-P steam[2]	251-305	2	3	3 1/2
L-P steam[3]	Up to 250	1 1/2	2	2 1/2
Condensate[4]	190-220	1 1/2	2	2 1/2
Hot water	Up to 200	1	1 1/2	2
Hot water	Over 200	1 1/2	2	2 1/2
Cooling systems				
Chilled water	40-60	1	1	1
Refrg. brine	Below 32	1 1/2	1 1/2	1 1/2

1. High pressure
2. Medium pressure
3. Low pressure
4. For feed water

*"The 1975 Energy Management Guidebook" published by Editors of Power Magazine, McGraw Hill Inc., N.Y., N.Y., 1975.

3.2.4 Eliminate Steam Leaks

As shown in table 3-4, steam leaks can be very costly. The entire system should be inspected to determine if there are leaks. Malfunctioning steam traps are also a common source of losses.

TABLE 3-4. THE COST OF STEAM LEAKS IN A 125 psi SYSTEM AT $4/1,000 lb [8]

Size of Orifice	Steam Wasted Per Month, lb	Total Cost (Per Month)	Total Cost (7 Months)
1/2 inch	708,750	$2,040	$14,280
3/4 inch	400,000	1,600	11,200
1/4 inch	178,000	712	5,000
1/8 inch	44,200	176	1,232
1/16 inch	11,200	45	315
1/32 inch	3,000	12	84

Note: For steam at 50 lb pressure, the waste is about 75% of the figures given; at 20 lb it is about 50%, and at 5 lb, the loss is about 25%.

Identify the location of a leaking trap by testing the temperature of return lines with a surface pyrometer and measuring temperature drop across the suspected trap. Lack of drop indicates steam blow-through. Excessive drop indicates the trap is holding back condensate. Repair or replace all faulty traps.

3.2.5 Assess Energy Use for Humidification Processes

The mechanical systems of some buildings are designed to provide indoor humidity control within a rather narrow range through the use of reheat. Such systems expend considerable amounts of energy for humdification and dehumidification processes. In many instances, adjustment of controls can be made to reduce reheat load without degrading the indoor environment of the building. (Sec. 3.1.4)

3.2.6 Flash Condensate to Low-Pressure Steam

Where medium- and high-pressure condensate is vented to the atmosphere or discharged, a significant portion of its heat content can be reclaimed by flashing that condensate so as to provide a source of low-pressure steam to be used for space heating or heating of domestic hot water. The quantity of 15 psig steam that can be produced by a condensate flashing system is shown in figure 3-4. In addition to saving heat, the flashing of condensate may reduce makeup boiler feed water.

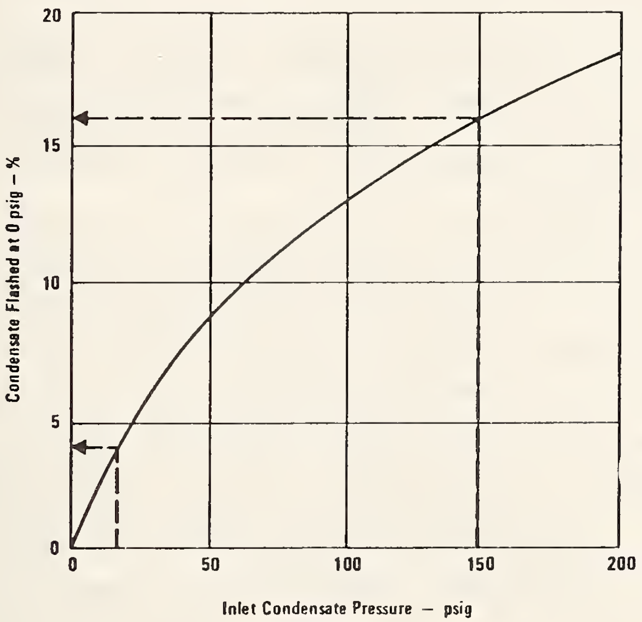

Figure 3-4. Flashing steam condensate (calculated from steam tables)

3.2.7 Increase Condensate Return to Boilers

Loss of condensate represents a waste of heat and of valuable high purity water. Identify all sources of condensate and evaluate economic feasibility of installing pump and insulated piping to return condensate to boiler feedwater tank. If condensate is contaminated, evaluate possible clean-up. Repair leaking condensate return systems.

3.3 HVAC Equipment in Family Housing

3.3.1 Heating Plants

The portion of the input energy of a combustion heating plant that is not delivered to the occupied space of a family housing unit is lost through one of the following mechanisms:

- jacket losses from the heating plant
- duct losses (or pipe losses)
- flue losses.

In the case of a forced-air electric furnace, losses occur by way of jacket and duct losses. Energy conservation measures for improving the performance of heating plants should be aimed at reducing the foregoing loss mechanisms.

Duct losses occur by way of air leakage at the seams where duct sections are joined and by heat conduction through the ducts themselves. Leaks in warm-air supply ducts may permit as much as 10% of the heating energy to escape. Duct leaks may also cause the occupied space of residential buildings to be under either negative or positive pressure, thereby substantially increasing the rate of air infiltration when the furnace operates. The conductive heat-loss rate through uninsulated ducts located in unheated crawl spaces may be as much as 120 $Btu/h \cdot ft^2$. At this rate every 2.8 ft^2 of duct surface would lose heat at a rate equivalent to that for a 100-watt light bulb.

Air leaks from ducts are easily corrected by the application of a suitable duct sealant and/or duct tape at the seams where the duct sections are joined. Conductive heat-losses from ducts located in unheated spaces may be substantially reduced by installing thermal insulation around the ducts. Similarly, the heat loss from pipes located in unheated spaces may be reduced by installing pipe insulation.

Losses up the flue or chimney may be reduced by lowering the mean temperature of the stack gases and by improving the combustion efficiency that occurs in the combustion chamber. A burner tune-up will frequently improve combustion efficiency. When the heat output rate of the furnace is too large for the heating load of the living space, the burner and fan will cycle on-off quite frequently. During the off portion of the fan cycle, heat stored in the heat exchanger is lost up the flue. This loss may be reduced by reducing the firing rate of the burner and adjusting the fan controls to cut in and cut out at lower temperatures. In addition, an automatic damper may be installed in the chimney which closes when the burner is not operating, thereby reducing the stack losses substantially.

3.3.2 Central Air Conditioning Equipment

Energy waste occurs when a central air-conditioning system is installed on an existing furnace system which has insufficient blower capacity. An evaporator coil introduces a considerable pressure drop and may substantially reduce the air-delivery rate of an existing furnace blower. Furthermore, the evaporator requires a certain air flow rate passing through it in order to obtain proper air-side convective heat transfer. Measure the delivered air quantity and compare with that required for the evaporator that is installed (usually 350-400 cfm/ton for unitary residential equipment). For belt-driven blowers, it may be possible to increase the air-delivery rate by installing a larger pulley on the motor shaft. For such modification, it is very important to monitor the current supplied to the blower motor to make sure that it remains below the maximum rated value for the motor. If increasing the pulley size is unable to produce a sufficient air-delivery rate, then a larger size blower/motor system should be installed.

Low air conditioning performance can also be caused by improper installation of the evaporator. It should be installed so that all the air is forced to pass through the evaporator. If it has small gaps around its perimeter, significant quantities of air may leak around the evaporator instead of passing through it. This situation would result in low suction temperature of the refrigerant line leaving the evaporator.

With regard to the outdoor condenser units, make sure that the airflow passages of the unit are not obstructed (by vegetation or inadvertently placed objects) or improperly enclosed. Also, note whether solar radiation strikes the outdoor unit during a significant portion of the day. Some improvement in air conditioning performance can often be achieved by simply shading the outdoor unit.

Also, note the location of the indoor thermostat. A thermostat placed at an unusually hot location of a building may cause overcooling of other parts of the building, which will cause energy waste.

Another way in which energy waste may occur is for an air-conditioning system to be installed so that it uses a part of the building as a return plenum. In such instances, significant energy waste will occur if the return plenum communicates through leaks to hot spaces such as attics, crawl spaces, and the outdoors. Warm air drawn from such spaces not only introduces sensible heat, but also introduces moisture-laden air which represents an increased latent load for the air-conditioning system.

After the installation of an air-conditioning equipment has been examined, inspect the ducts for air leaks. Duct leaks may represent a significant energy waste. Serious duct leaks can usually be detected by feeling with the fingers around the seams where duct sections are joined together. These duct leaks can readily be eliminated by the application of a suitable duct sealant and/or duct tape.

Consideration should also be given to insulating ducts, particularly if the supply ducts pass through hot attic spaces. Procedures for estimating the optimum amounts of insulation for ducts located in attics and other unheated spaces (such as crawl spaces) is presented in Chapter 2.

Routine preventive maintenance should be established, with the following items included as part of the maintenance program, performed prior to the air-conditioning season:

- clean air filters
- check the refrigerant charge of the system
- oil all moving parts
- check the overall operating condition of the system.

3.3.3 Summary of Energy Conservation Measures for HVAC Equipment in Family Housing

Heating Plants

(1) Seal duct leaks with a suitable duct sealant and/or duct tape and insulate supply air ducts that pass through unheated spaces.

(2) If the heating plant is located within the living unit, consider providing outdoor makeup air for combustion-air and draft-diverter-air requirements.

(3) Consider installing a clock thermostat for reducing indoor temperature at night.

(4) If an oversized furnace was originally installed or if the implementation of energy conservation measures has caused the maximum heating load to be considerably reduced, consider reducing the firing rate of the fuel burner.

(5) Implement a pre-heating season preventive maintenance schedule which includes: a burner tune-up and cleaning and replacement of air filters.

(6) Make sure the thermostat and controls are operating properly to preclude overheating.

(7) If a part of the building serves as a return plenum for the heating plant, make sure that cold air from unheated spaces is not inadvertently drawn into the air return.

Cooling Plants

(1) Correct improper installations such as

- Evaporator which permits air to flow around the evaporator instead of through it.
- Under-capacity blower that cannot provide sufficient air volume through the evaporator.

(2) Take corrective action to free inadvertently obstructed air passages of the condenser unit.

(3) Consider shading the condenser unit to reduce solar heat gain.

(4) Turn off air-conditioning system when the building is unoccupied.

(5) Seal duct leaks with a suitable duct sealant and/or duct tape.

(6) Insulate supply air ducts located in the attic.

3.4 Domestic Hot Water*

3.4.1 Background

Average Usage

Hot water utilization factors (expressed in gallons of hot water consumed per person per day) for various type buildings are given in table 3-5.

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* Much of this material of this section was taken from "Guidelines for Saving Energy in Existing Buildings," ECM 1 [8].

TABLE 3-5. HOT WATER UTILIZATION FACTORS (F) [8]

Office Buildings	
(Without kitchen or cafeteria services)	2 to 3 gallons per capita per day for hand washing and minor cleaning (based on an average permanent occupancy which includes daily visitors)
Base Exchanges	
(Without kitchen or cafeteria services)	1 gallon per customer per day
Food Serving Facilities	
Dishwashing, rinsing and hand washing	3 gallons per meal plus 3 gallons/ employee/day
Schools	
Boarding	25 gallons per capita per day
Day	3 gallons per capita per day (Does not include cafeteria or athletic facilities)
Dormitories and VOQ's	20 to 30 gallons per capita per day
Hospitals	30 to 50 gallons per capita per day

Average Temperature

The usual temperature at which hot water is supplied varies from 120° F to 150° F. It usually is too hot to be used directly and must be mixed with cold water at the tap. For dishwashing and sterilization the delivery temperature is generally 160° F or higher. Often hot water supplied to all faucets is at temperature required for the kitchen. Hot water, generated and stored in tanks at 150° to 160° F, loses heat by conduction and radiation from the tank and piping, even before the delivery.

When hot water is supplied by a tankless heater, it may be within 5° or 6° F of the boiler water (or steam) temperature maintained to heat the building. A mixing valve is often used to control the delivery temperature, but frequently the temperature at which it is set is excessive.

Energy Conservation Measures

The opportunities to conserve energy for heating domestic hot water can be summarized as follows:

(1) Reduce the load

- decrease the quantity of domestic hot water used
- lower the temperature of the domestic hot water

(2) Reduce the system losses

- repair leaks and insulate piping and tanks
- reduce recirculating pump operating time

(3) Decrease the losses from domestic hot water generators

(4) Utilize waste heat to temper the cold water supply to heaters and, where feasible, to generate hot water.

3.4.2 Reduce the Temperature of Domestic Hot Water Supplied to Taps

Lowering the temperature of hot water can reduce both the domestic hot water load, and the distribution losses. The annual domestic hot water load (Q), not including distribution losses, is given by the relation:

$$Q = 8.3 \cdot F \cdot N_o \cdot N_d \cdot C_p \cdot (T_s - T_g) \tag{3}$$

where 8.3 = pounds of water per gallon

F = water utilization factor, gal per person per day (see table 3-5)

N_o = number of building occupants

N_d = number of days per year that the building is occupied

C_p = specific heat of water, Btu/lb·°F; use 1.00

T_s = temperature at which water is supplied to the tap, °F

T_g = temperature at which water is supplied to water heater (approximately equal to annual average ground water temperature), °F.

A chart for estimating annual hot water heating load is given in figure 3-5. In this chart annual hot water heating load per capita (Q/N_o) is plotted as a function of water utilization factor (F) for

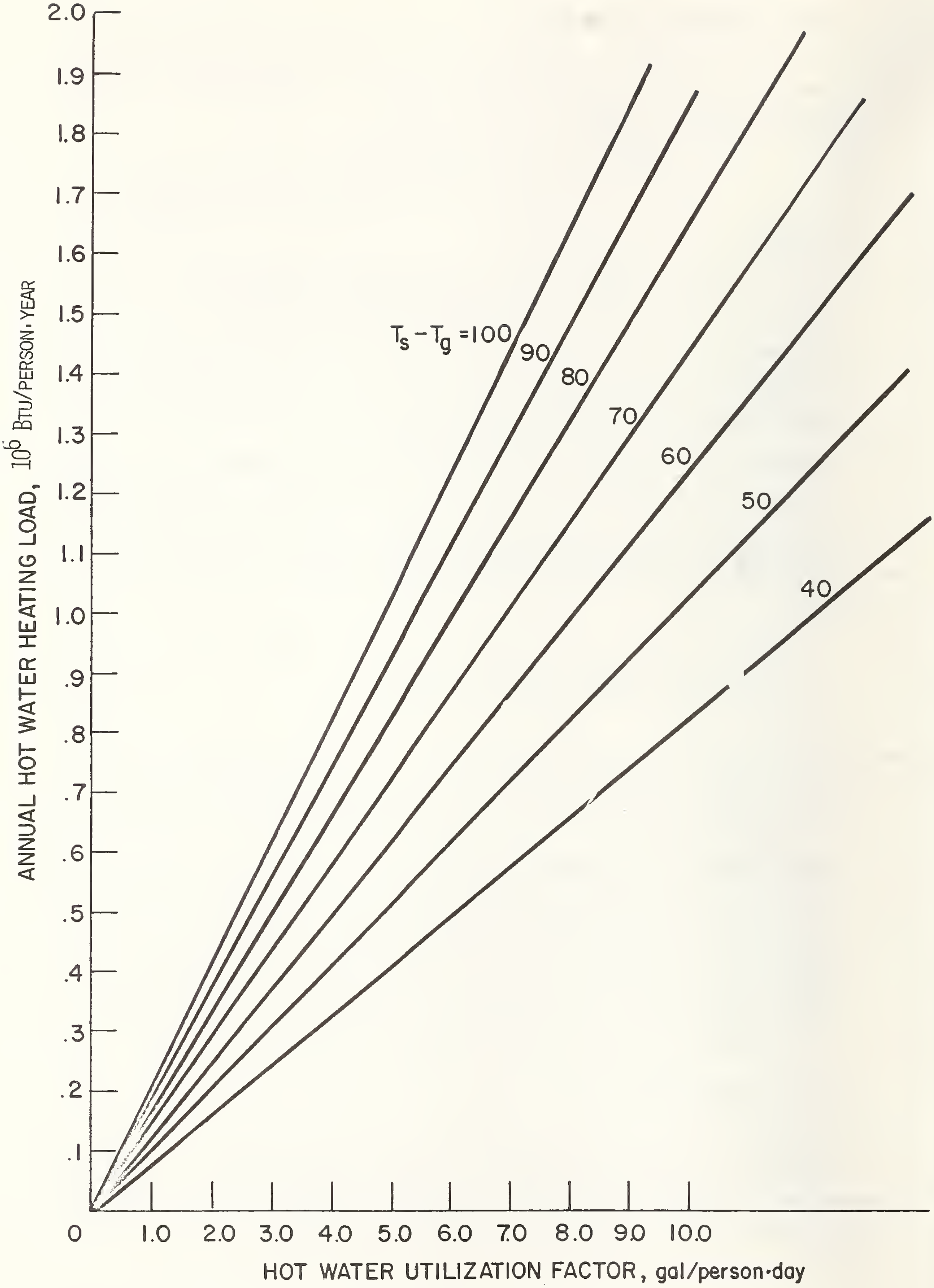

Figure 3-5. Chart for estimating annual hot water heating load.

various temperature heating increments ($T_s - T_g$). To find the annual hot water heating load, find the appropriate utilization factor (F) (from table 3-5) on the horizontal axis. Draw a line vertically upward until it intercepts the appropriate temperature heating increment ($T_s - T_g$) curve. The annual hot water heating load per capita (Q/N_o) is found by reading horizontally across to the vertical axis.

From equation (3) the percentage reduction in the hot water heating load ($\Delta Q/Q$) achieved by lowering the hot water supply temperature is seen to be

$$\Delta Q/Q = \Delta T_s/(T_s - T_g) \qquad (4)$$

Here ΔT_s is the reduction in the hot-water supply temperature. The foregoing technique for calculating reductions in hot water heating load is conservative, since it does not include the reduction in distribution losses that also occur when the hot-water supply temperature is reduced. However, it is also subject to some overestimation since for some usages (baths, showers, etc.) the user will tend to draw more hot water in order to obtain desired temperature.

The annual energy consumed by a fuel-fired hot-water heating device is equal to the annual hot-water heating load divided by the efficiency of the hot-water heater. Efficiency factors for several hot-water heating plants are given in table 3-6.

TABLE 3-6. EFFICIENCIES FOR SEVERAL HOT-WATER HEATING PLANTS [8]

Oil-fired heating boilers used year round, but with domestic hot water as the only summer load.	.45
Oil-fired heating boilers used year round, but with a summer load such as absorption cooling in addition to domestic hot water.	.70
Gas-fired heating boilers used year round, but with domestic hot water as the only summer load.	.50
Gas-fired heating boilers used year round, but with a summer load such as absorption cooling in addition to domestic hot water.	.75
Separate oil-fired domestic water heaters.	.70
Separate gas-fired homestic water heaters.	.75
Separate electric water heaters.	.95
Separate coal-fired water heaters.	.45

Consider the following examples:

An office building has 500 occupants for 250 days per year. The hot-water supply temperature is 130° F and enters the heater at 60° F (an average for the year). Domestic hot water is used year round with domestic hot water as the only summer load. The fuel is #2 oil, which has a heating value of 138,000 Btu's per gallon. Find the gallons of fuel oil saved by reducing the hot-water supply temperature to 100° F.

(1) From table 3-5, the hot-water utilization factor (F) ranges between 2 and 3 gallons per person per day. For this example, take it to be 3 gallons per person per day.

(2) From figure 3-5, the annual hot-water heating load for 3 gallons of hot-water supplied at 130° F (or $T_s - T_g = 70°$ F) is 430,000 Btu per occupant per year. For 500 occupants, the annual hot-water heating load is 215,000,000 Btu per year. The required quantity of fuel oil is

$$\frac{215{,}000{,}000 \text{ Btu}}{(0.45)\ (138{,}000 \text{ Btu/gal})} = 3{,}500 \text{ gallons}$$

where 0.45 is water heater efficiency.

(3) Repeating the procedure of (2) for a hot-water supply temperature of 100° F, the required quantity of fuel oil is found to be 2,000 gallons.

(4) The savings in fuel oil is seen to be 1,500 gallons. At \$.36 per gallon, this fuel saving represents \$540 per year.

3.4.3 Reduce the Quantity of Domestic Hot Water Used

From equation (3), it is seen that the percentage reduction in hot-water heating load ($\Delta Q/Q$) due to reduced consumption is given by the relation:

$$\Delta Q/Q = \Delta F/F \tag{4}$$

Here ΔF is the reduction in the hot-water utilization factor. A secondary benefit is that less water need be handled in the water-supply treatment and sewage-treatment plants, whether on-Base or off-Base.

For the preceding example, calculate the reduction in hot-water heating energy achieved by installing flow-reducing devices only in the lavatory faucets to reduce the water-utilization factor (F) from 3 to 2 gallons per person per day. The percent reduction in hot-water heating energy is

$$\Delta Q/Q = \Delta F/F = 1/3 = 0.33$$

The reduction in hot-water-heating energy is found by multiplying the original hot-water-heating energy requirement by the foregoing reduction factor ($\Delta Q/Q$), or

$$\text{Fuel oil saved} = (3{,}500)\ (0.33) = 1{,}170 \text{ gallons}$$

3.4.4 Improve the Efficiency of the Storage and Distribution Systems

Repair or replace damaged insulation to reduce heat loss. Heat losses from uninsulated hot water distribution systems can be substantially dependent upon the temperature differential between pipe and ambient air and the amount of exposed piping.

Exposed piping in basements and equipment rooms is relatively simple to insulate. Piping in ceiling spaces may also be accessible by removing ceiling panels. Preferably, the entire piping system should be insulated, but inaccessible portions may be left bare providing they are a small percentage of the total.

The loss of heat from storage type heaters, storage tanks, and external heaters must be offset by the addition of heat to maintain a ready supply of hot water. This heat loss occurs 24 hours a day whether the building is occupied or not. Storage tanks and external heaters should be covered with a minimum of 3 inches of insulation (thermal resistance, R = 10). Insulate bare tanks and apply additional insulation to tanks having less than 3 inches. Storage type heaters with factory insulated jackets can be covered with an additional layer of insulation. This insulation may consist of ordinary insulating blanket material, or insulation jackets ready-cut and available from material suppliers; in any event, it is likely to pay out in a few years.

3.4.5 Generate Hot Water More Efficiently

All of the measures for improving combustion units for space heating apply equally well to hot-water heaters. Keep in mind that when more than one heater is installed on a project, it is more efficient to operate a single unit for the total load (if it can carry it), rather than to operate more than one unit at partial loads. When existing water heating equipment is replaced, consider the use of modular units.

3.4.6 Replace Central Systems by Local Heating Units

Commercial hot-water systems frequently require hot water for short periods of heavy use at various locations within the building. It is often more efficient to provide water heaters close to the usage points instead of using central units having long runs of hot-water piping.

Analyze the hot water use within the building to determine the patterns of demand to determine whether installation of local units is advantageous. Estimate the energy losses of the existing system and calculate the savings derived by installing local units.

3.4.7 Install Temperature Boosters

When multiple temperature requirements are met by a central domestic hot-water system, the generation temperature is determined by the maximum usage temperature. Lower temperatures for other usages are attained by mixing hot water with cold water at the tap. Where most of the hot water usage is at the lower temperatures and only a few specific locations require high temperature hot water, consider installing booster heaters in the hot-water lines serving these locations.

3.4.8 Install Separate Boilers for Summer Generation of Domestic Hot Water

In many buildings, the space heating system boilers also provide heat for the domestic hot water system. While this is satisfactory during the heating season when the boiler is firing at high-efficiency, demand for boiler heat in summer may be limited to hot-water generation only. Operating large boilers at low part-loads results in low boiler efficiency.

To reduce energy losses due to low boiler efficiency in summer, install a separate hot-water heater or boiler, sized for the hot-water demand. Shut down the space heating boiler in the summer and generate hot water at improved efficiency.

3.4.9 Waste-Heat Recovery for Hot-Water Heating

(1) Hot-Gas Heat Exchanger

A typical refrigeration machine with a water-cooled condenser rejects approximately 15,000 Btu/h for each 12,000 Btu/h of refrigeration. An air-cooled condenser rejects up to 17,000 Btu/h. Up to one third of the heat rejected from either system can be recaptured. To recover heat of compression, install a heat exchanger in the hot-gas line between the compressor and condenser of the chiller. Such a waste-heat recovery system would be applicable to most Base commissary facilities. A typical arrangement in conjunction with a domestic hot-water system is shown in figure 3-6. Hot-gas temperature depends on head pressure but is usually on the order of 120° to 130° F.

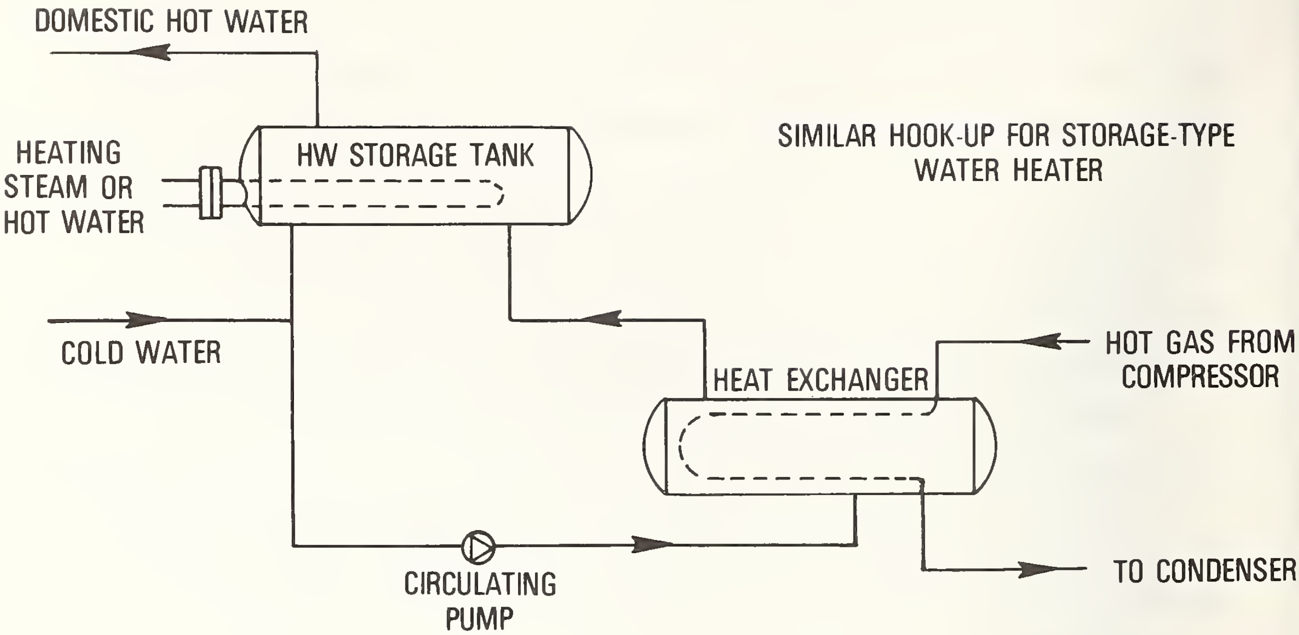

Figure 3-6. Schematic of hot-gas heat exchanger heat-recovery system.

(2) Hot-Drain Heat Exchanger

Buildings with kitchens, laundries, and other service facilities which utilize large quantities of hot water, in many cases discharge hot waste water to drains. By installing a heat exchanger, heat can be recaptured for use to preheat domestic hot water. One type of heat reclamation system is shown schematically in figure 3-7.

In general, special equipment to preheat water from 50° F to 105° F will often be cost effective. The hot water at 105° F can then be fed into the domestic hot-water tank for further heating to utilization temperature, if required. The heat-reclamation system saves the heat required to heat water from 50° F to 105° F which otherwise would have been provided by other heat sources.

For example, in a laundromat, the average daily wash load is 2,000 lbs/day. Water usage is 2.5 gallons of 170° F water and 1.0 gallon of 50° F water per lb of wash load. The waste water discharged to drain is 3.5 gallons of 136° F water per lb of wash load.

Assuming that the laundromat operates 365 days/yr and that domestic hot water is generated by an electric hot-water heater, the yearly saving is 833×10^6 Btu. At 90% efficiency and \$.03 kWh, this represents \$8,130 cost saving/year.

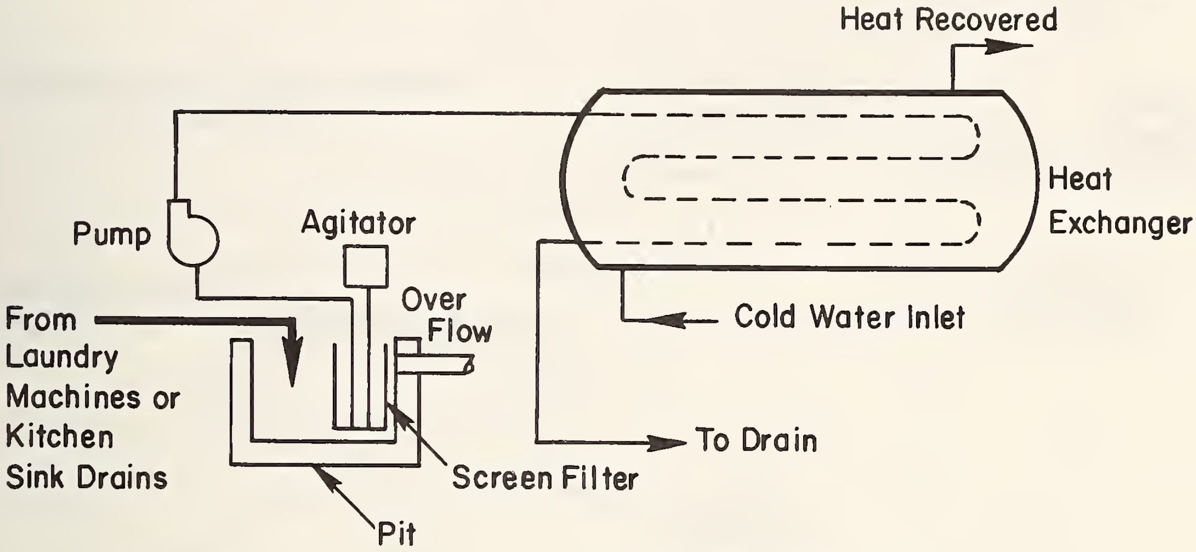

Figure 3-7. Schematic of laundry and kitchen hot-water heat-recovery system.

(3) <u>Hot-Condensate Heat Exchanger</u>

The condensate return portion of many steam systems exhausts large quantities of heat in the form of flash steam when the hot condensate is reduced to atmospheric pressure in the condensate receiver. Recover waste heat by installing a heat exchanger in the condensate-return main ahead of the receiver to reduce condensate temperature to approximately 180° F. Use the heat recovered to preheat water as required. Figure 3-8 shows the schematic installation of a condensate heat exchanger.

The quantity of heat recovered depends on the quantity and temperature of the condensate. Consider the following example:

<u>Example</u>

Condensate return volume is equal to 6 gpm at 260° F. A heat exchanger is installed to reduce the condensate temperature from 260° F to 180° F and the quantity of heat recovered is:

$$Q = 6 \cdot (260 - 180) \cdot 500 = 240{,}000 \text{ Btu/h}$$

where 500 converts gpm to pounds per hour.

In each case, the feasibility of a waste heat recovery system depends upon being able to use the recovered heat; i.e., there must be a need for the domestic hot water that is generated.

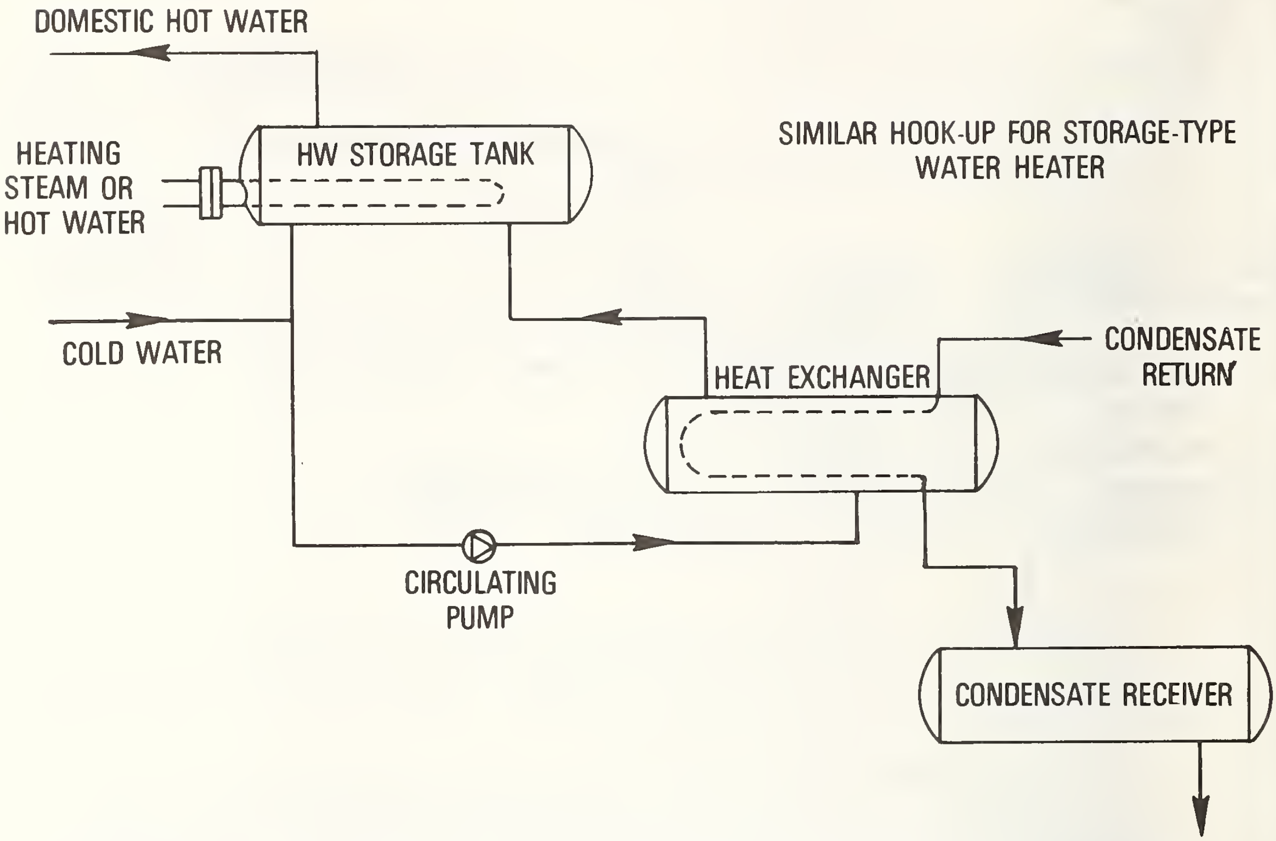

Figure 3-8. Waste-heat recovery using a hot-condensate heat exchanger.

3.5 Lighting*

3.5.1 Background

Electrical energy used for lighting accounts for a significant portion of the total energy usage in Air Force buildings. When lights are left on in areas which are unoccupied or unused for lengthy periods of the week (i.e., religious buildings and outdoor parking lots), the inadvertent waste of energy often approaches or exceeds the amount of energy used by other building systems during the week; the cost of this waste for one year may equal the initial cost of installing automatic controls to eliminate it.

* Much of this material was taken from "Guidelines for Saving Energy in Existing Buildings," Building Owners and Operators Manual, ECM 1 [8].

Electric lighting is also the major contributor to internal heat gain in Base exchanges and commissaries, and core areas of office buildings. Since it increases the cooling load, electric lighting causes additional energy expenditure to operate the refrigeration system; it would not be unusual for an air conditioning system to expend up to 2 kWh of electrical power to remove the heat resulting from 3 kWh of electric lighting.

Significant reductions in energy consumption for lighting can be attained from the following measures:

- Conserve energy by turning off lights;
- Improve the effectiveness of existing lighting installations;
- Use daylight for illumination;
- Add switches and timers to shut off lights;
- Remove unnecessary lamps;
- Reduce lighting loads by relamping;
- Reduce illumination levels.

These items are discussed in the following sections.

3.5.2 Conserve Energy by Turning Off Lights

Letting a light burn rather than turning it off does not save electrical energy. When electric lighting is not required, it should be switched off. Figure 3-9 illustrates the energy savings achieved by turning off lights during unoccupied periods for an office lighting system of 1,000 fluorescent luminaires.

The full system uses 92 kW. The upper graph shows typical usage when lights are permitted to burn continuously from about 8:30 a.m. to 7:30 p.m. The lower graph shows efficient use of the same system leaving lights on only when used. The energy saving amounts to 267.5 kWh daily or 5,875 kWh per month. At $.03 per kWh, this figure corresponds to a savings of $176 per month.

3.5.3 Improve the Effectiveness of Existing Lighting Installations

Increase the effectiveness of the present lighting system, then reduce the number or wattage of lamps required to produce the required illumination. In order to reduce the number and/or wattage of lamps, consider the following measures to increase the maintained foot-candles provided by existing lamps and fixtures:

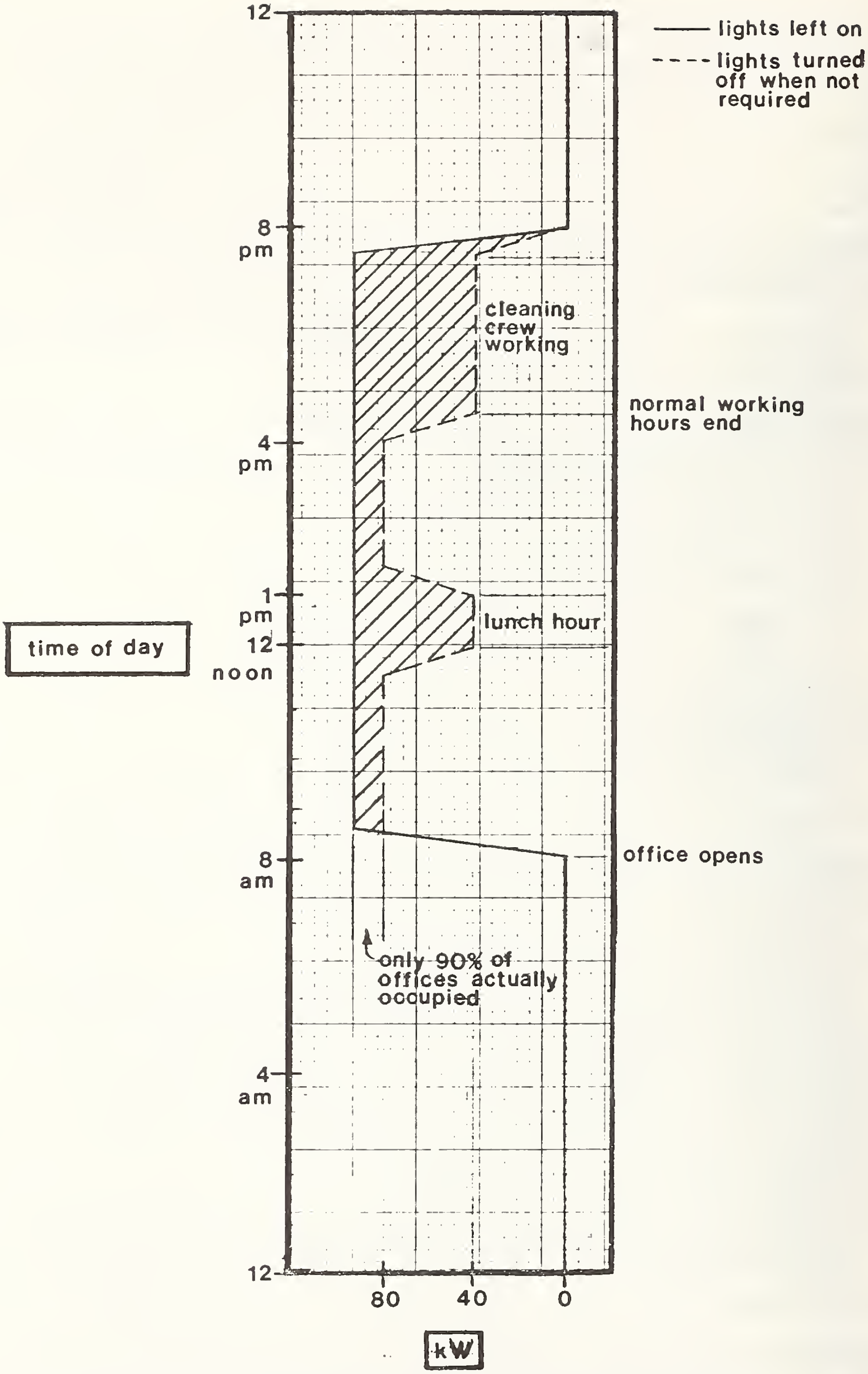

Figure 3-9. Demonstration of energy saving achieved by turning off light during unoccupied periods [8].

Improve the lumens of output by regularly cleaning fixtures and lamps and instituting a relamping program. Clean fixtures at least every time they are relamped. (When relamping and cleaning are combined, the additional labor is very small.) In extremely dirty atmospheres, clean fixtures between lamp replacements, as well. Clean the lenses and interior housings. Use an anti-static compound to reduce electrostatic dirt collection.

A group relamping program will produce a higher level of maintained foot-candles, because lamp lumens will not depreciate to low levels at the end of their lives. The effect of dirt on the output lumens as a function of time is shown in figure 3-10.

When there are not critical reading or writing tasks, and where some glare from exposed lamps is acceptable, illumination levels may be increased by removing louvers or lenses from fixtures. Consider this suggestion, for instance, in corridors, storage areas, spaces with high ceilings, equipment rooms, and snack bars. In many instances, this may permit a reduction in the number of lamps used.

Increase the reflectance of ceilings, walls, and floors by cleaning or by painting with colors of higher reflectances: more illumination on tasks, with the same lumen output, will result. Increasing the reflectivity of the interior surfaces may open the possibility of reducing the number of lamps and luminaires while maintaining recommended lighting levels. Greater reflectances enhance the performance of natural daylight as well as artificial lighting. For example, a 25 x 25-foot room with a 9-foot ceiling and a ceiling reflectance of 80%, wall reflectance of 70%, and floor reflectance of 30% will provide 35% more foot-candles (with no increase in power consumption) than the same room with a ceiling reflectance of 50%, wall reflectances of 30%, and floor reflectance of 10%.

Caution: High gloss finishes may reduce visual comfort by increasing glare. Discretion should be used when redecorating.

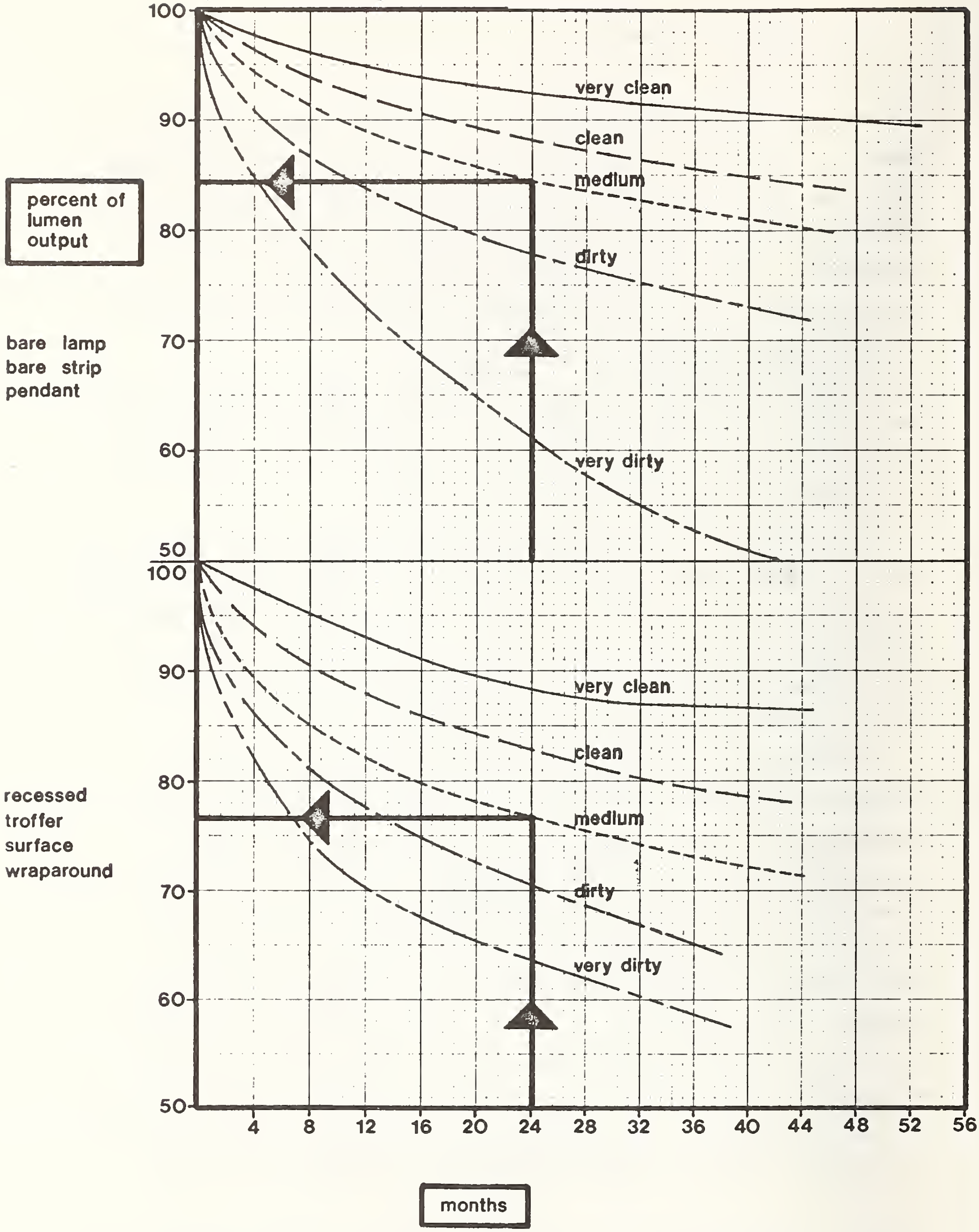

Figure 3-10. The effect of dirt on output lumens as a function of time [8].

3.5.4 Use Daylight for Illumination

Use windows effectively as a primary source of illumination in perimeter spaces. The amount of available daylight in a building is a function of operating hours, latitude, weather, time of year, air quality, window size and location, shading and glazing details, reflectivity of interior surfaces and furnishings. Control of natural light, for effective use and integration with artificial light, is important, because the amount of natural light available varies. Control may be manual or automatic. Often blinds or drapes, which should be closed only for short periods of excessive sunshine or glare, remain closed all day with the electric lights turned on, when daylight could be used. Emphasis should be placed on proper control of daylight; otherwise the heat gain imposed on the cooling system may outweigh savings in energy for lighting. Indirect daylighting, however, generates less heat per lumen of light than electrical lighting.

The luminous efficiency (lumens per watt) of various light sources as compared to daylight is shown below (no allowance for ballast of luminaires):

Low Pressure Sodium	183 lumens/watt
Natural Light	120 lumens/watt (varies)
High Pressure Sodium	105-120 lumens/watt
Metal Halide	85-100 lumens/watt
Fluorescent	67-91 lumens/watt
Mercury vapor	56-63 lumens/watt
Incandescent	17-22 lumens/watt

3.5.5 Add Switching and Timers to Shut Off Lights

If there are insufficient switches for unwanted lights to be turned off, consider installing additional switches.

There are many types of surface-mounted flat ribbon conductors available for installation in existing spaces; they can be installed with the minimum dislocation of existing wiring or damage to interior decorations. Locate new switches near doors or where they are most convenient for use: switches in inconvenient locations will not be used. If switches are group-mounted, label each one to indicate the area it controls.

Provide time switches for areas which are commonly used for short times, and in which lighting is inadvertently but frequently left on (i.e., reference rooms and stock rooms). At a pre-determined time after the switch has been turned on, it will automatically shut off; if the area is to be used for a long period of time, the switch can be manually overridden.

To ensure the success of any switching program, instruct all occupants in the use of available switching.

3.5.6 Remove Unnecessary Lamps

Remove unnecessary lamps when those remaining can provide the desired level of illumination. When removing fluorescent or high-intensity-discharge lamps, also remove the ballast (or disconnect it in place). If it is left connected to the power source, it will continue to consume energy even though it serves no useful purpose. When two-lamp fluorescent fixtures are mounted in a row, remove lamps in alternate fixtures of the rows (rather than removing one entire row) to derive higher quality lighting. In order to maintain the recommended lighting level it may be necessary, after removing some lamps, to use increased output lamps in place of some of the remaining lamps. Even so, the measure should result in net savings of energy. In many cases it will be possible to relamp the remaining fixtures with more efficient lamps to provide more lumens per watt. Refer to figure 3-11 to determine resulting foot-candle output with some common lamp replacements.

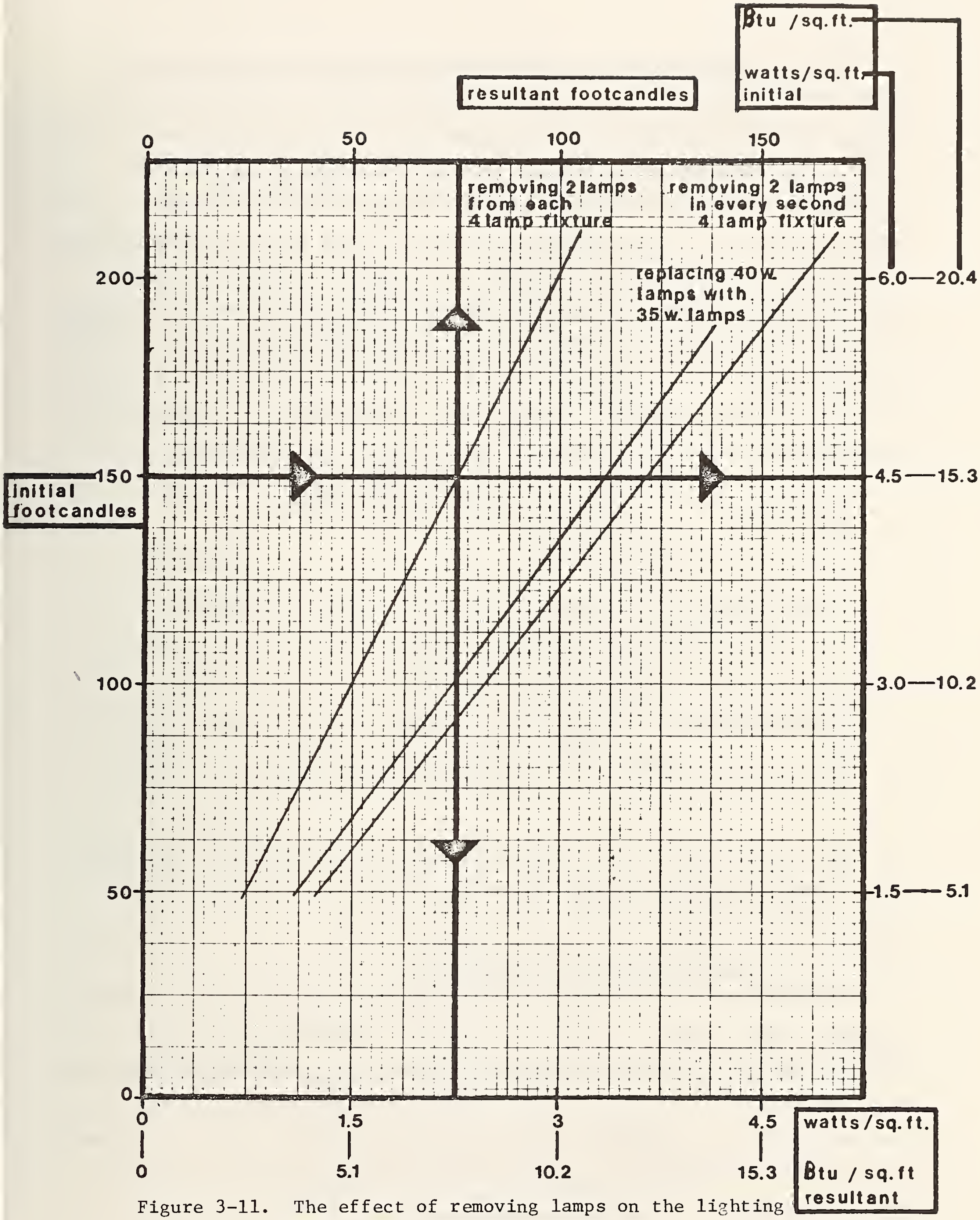

Figure 3-11. The effect of removing lamps on the lighting density and the illumination [8].

Lamp efficiencies vary between lamps consuming the same number of watts and lamps of the same type. Different colors, shapes, gaseous fills, cathode construction and internal coatings cause those variations. For example, a fluorescent 40-watt, T-12 lamp, rated 430 milliamperes, can have a rating from around 53 lumens to around 92 lumens per watt. Or, for example, a natural white fluorescent lamp at 2100 lumens (53 lumens/watt) provides 1/3 less foot-candles than cool white fluorescent lamps at 3200 lumens (80 lumens/watt). In some cases, it may be desirable to relamp with lower wattage lamps instead of removing lamps.

An example of energy and money saved by removing two lamps from every other fixture (and disconnecting ballasts) is given below:

Example: Building Type - Base Exchange	
Number and type of fixtures (4-40W) 2' x 4'	375
Floor area	24,000 ft^2
Illumination level before change	100 FC
Hours of operation per year	3720 hours
Watts saved by the change = $80W \times \frac{375}{2}$ =	15,000 W
Energy saved/year = $\frac{15,000W}{1,000} \times 3720$ hours =	55,800 kWh

At an electricity cost of 4c/kWh, dollar savings =55,800 x .04
=$2,232

Additional savings in energy for cooling, due to reduced heat gain from lighting, will compensate for loss of heat from lighting which occurs in the winter.

3.5.7 Reduce Lighting Loads by Relamping

Efficiency of lamps is measured in lumens or usable light produced per watt input. Selecting lamps with higher lumens per watt could permit the removal of some lamps, providing the lumens produce the required foot-candles. More efficient lamps also impose smaller heat loads on the air-conditioning system. In winter, any heat loss by the reduction in wattage can generally be supplied more efficiently by the heating system itself.

For certain type installations where the type of fixture and the usage is identical, it is desirable to adopt a group relamping program, so that when a majority of the fluorescent tubes reach the point of only 70% of their rated output they will be replaced. Labor costs are generally less for group relamping than for individual relamping throughout the year.

When relamping, it is important to select replacement lamps which produce illumination suitable for the space. For instance, using high intensity discharge lamps in an office space with a low ceiling may produce glare and shadow which will cause eyestrain. Recommended applications for various types of lamps are given in table 3-7.

TABLE 3-7. ENERGY CONSERVING LAMP APPLICATIONS [8].

Lamp Type	Applications
Incandescent	1. Short time uses such as decorative display lighting or out-of-doors (Christmas trees). 2. Religious worship halls. 3. In projection lamps for illuminating work closets or other very confined spaces. 4. Stage spotlighting. 5. Where small light source are required to light the task.
Fluorescent	1. Offices and other relatively low ceiling applications. 2. Flashing advertising signs. 3. Service islands at service stations. 4. Display cases in stores. 5. Desk lamps. 6. Classrooms or training centers. 7. Cafeterias (with color correction).
High Intensity Discharge	1. Stores and other relatively high ceiling applications. 2. Auditoriums. 3. Outdoor area lighting. 4. Outdoor floor lighting. 5. Outdoor building security lighting. 6. Marking of obstructions. 7. High bay work areas. 8. Hangars.

High-intensity discharge (HID) lamps are more efficient (i.e., they produce more illumination per watt input) than many other types of lamps. Therefore, substantial reductions in lighting energy can often be achieved by converting existing lighting systems to HID systems.

There are three types of HID lamps available: mercury vapor, metal halide and high-pressure sodium lamps. For each type there are several sizes, wattages and bulb shapes available.

The HID lamps provide satisfactory and efficient illumination systems for various applications such as industrial facilities (shops), auditoriums, street lighting, gymnasiums, warehouses and some commercial interiors. Recent information indicates lighting fixture manufacturers have developed high-pressure sodium lamp luminaires for administrative and classroom areas. For this type installation, detailed information on luminaire efficiency, light distribution and visual comfort index is still under study.

Selection of a satisfactory HID lighting system for a particular application depends on the characteristics of each type of lamp, such as wattage, size, color quality and the light utilization of the luminaire specified. The design of the lighting systems should conform to the recommended standards for quality of illumination, light distribution, shadows, source brightness and visual comfort of the Illuminating Engineering Society.

Caution: HID lighting systems which are not properly designed may produce shadow and glare, particularly for low-ceiling applications.

3.5.8 Reduce Illumination Levels

Energy consumption for lighting may sometimes be lowered by reducing illumination levels to the following Air Force criteria values:

Description of Space	Illumination level, Foot-candles
work stations	50
work areas	30
non-working areas	10

If several tasks requiring different levels of illumination occur within the same space, first consider their visual requirements and then modify maintenance procedures, redecorate the area, and implement changes to the lighting system while reducing illumination levels to the appropriate level for each task. A uniform modular lighting pattern of general illumination, throwing light equally on all areas regardless of task, may waste up to 50% of the energy used for lighting in the building. Orient lighting to suit the tasks to be performed or rearrange work areas (furniture, etc.) to best utilize the lighting system in place.

Lighting in existing administrative open-space offices usually consists of long rows of fluorescent fixtures which provide a uniform lighting pattern in the building. This system of lighting is often inefficient since it provides over-illumination of some areas and insufficient light in others. One solution to this situation is "task lighting." The "Federal Design Matters" Pamphlet, Issue No. 5, October 1975, reported that "task lighting" reduces the total energy requirement for lighting and air-conditioning by 30 percent. Additionally, the pamphlet reports "task lighting" speeds and lowers the cost of construction, alterations and maintenance, as well as improving acoustics and fire protection, and helps staff members to focus on their work.

Light levels in standard foot-candles can be determined with a illumination meter such as a photovoltaic cell connected to a meter calibrated in foot-candles (i.e., foot-candle meter). These instruments may be of an inexpensive type, portable, easy to use and suitable to provide information for preliminary assessment of illumination levels.

Further information on energy conservation techniques relating to lighting may be found in references[8,9].

3.6 Central Plants

The purpose of this section is to provide technical information on ways of improving the efficiency of central steam, high-temperature water (HTW), and chilled-water generating plants.

In reviewing the material of this section, recognize that not all modifications are applicable to all systems. In addition, many of these modifications involve substantial expense and the possibility of service interruptions. Accordingly, feasibility studies should be conducted as necessary to consider the following factors:

- effect of modification on other components of the system and on other systems;
- impact on environmental conditions and related impact on comfort, productivity, safety, and health;
- existing restrictions in terms of environment, zoning, building/fire requirements;
- cost of removing existing system or components, modifying space, design, installation (including equipment, materials and labor).
- salvage value of components removed;
- energy consumption and cost without modification;
- problems to be expected during installation and start-up.

Although the costs of these modifications generally are high, the amount of energy and energy dollars to be conserved also will be high, thereby making modifications justified in many cases. Recognize too that some substantial savings can be obtained with very little investment.

It also should be noted that no substantial modification should be made before the operating and maintenance practices associated with existing equipment are optimized to their fullest extent. To evaluate a system based on poor operating and maintenance practices does not provide an accurate base for determining its true efficiency. In fact, the importance of proper operating and maintenance practices cannot be overemphasized. Not only do they help ensure maximum energy efficiency, but they also serve to ensure optimum system performance and response to environmental and related needs, reduced breakdowns and associated problems, and longer life for all equipment.

3.6.1 Steam and HTW Generating Plants

Boiler plants at U.S. Air Force installations include, among others, high-pressure steam plants; high-pressure and low-pressure steam heating plants, and high-temperature hot-water (HTW) plants.

In many steam and HTW generating plants, inefficient equipment design and non-optimal operating practice cause substantial energy waste. Fortunately, some types of energy waste can be easily detected and corrected by improved operation and minimum investment. Various energy conservation measures for improving the efficiency of central heat plants are discussed in the sections below.

3.6.1.1 Limit Flue-Gas Losses

High flue-gas temperature in central heat plants is one of the major indicators of fuel waste. When natural gas is the fuel, the loss with a flue gas temperature of 1000° F is equivalent to 179 ft^3 of gas for each 1000 ft^3 of gas burned. If the flue-gas temperature is reduced to 500° F, 81 ft^3 of gas are saved for each 1000 ft^3 of gas burned. Substantial energy saving can be realized by reclaiming a portion of the heat content of the flue gases for pre-heating combustion air and boiler feed water.

A boiler stack economizer, as shown in figure 3-12, consists of a heat exchanger placed inside the exhaust stack for the purpose of preheating boiler feed water. Heat captured from the exhaust gases increases the temperature of the feed water which is returned to the boiler. In past years this technique was limited to systems using low or nonsulphur content fuels to avoid build-up caused when the economizer cooled the exhaust gases to the sulphur dew point. This problem can be overcome by controlling the volume of feed water passing through the transfer coils. As a general rule, when an economizer is installed to preheat boiler feed water, every 10 to 11° F increase in feed-water temperature produces approximately a 1% increase in boiler efficiency.

Preheating primary and secondary air also increases boiler efficiency by reducing the cooling effect when the air enters the combustion chamber. As shown in figure 3-13, a 100° F increase in the temperature

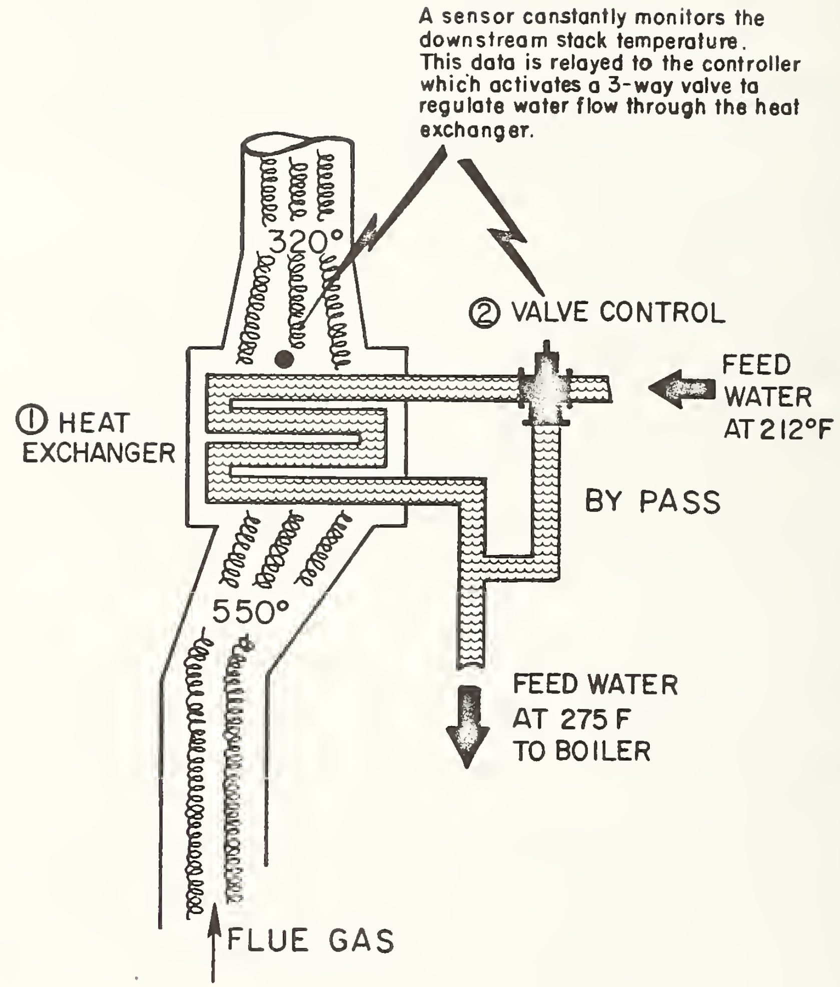

Figure 3-12. Schematic of a boiler stack economizer with stack temperature controls.

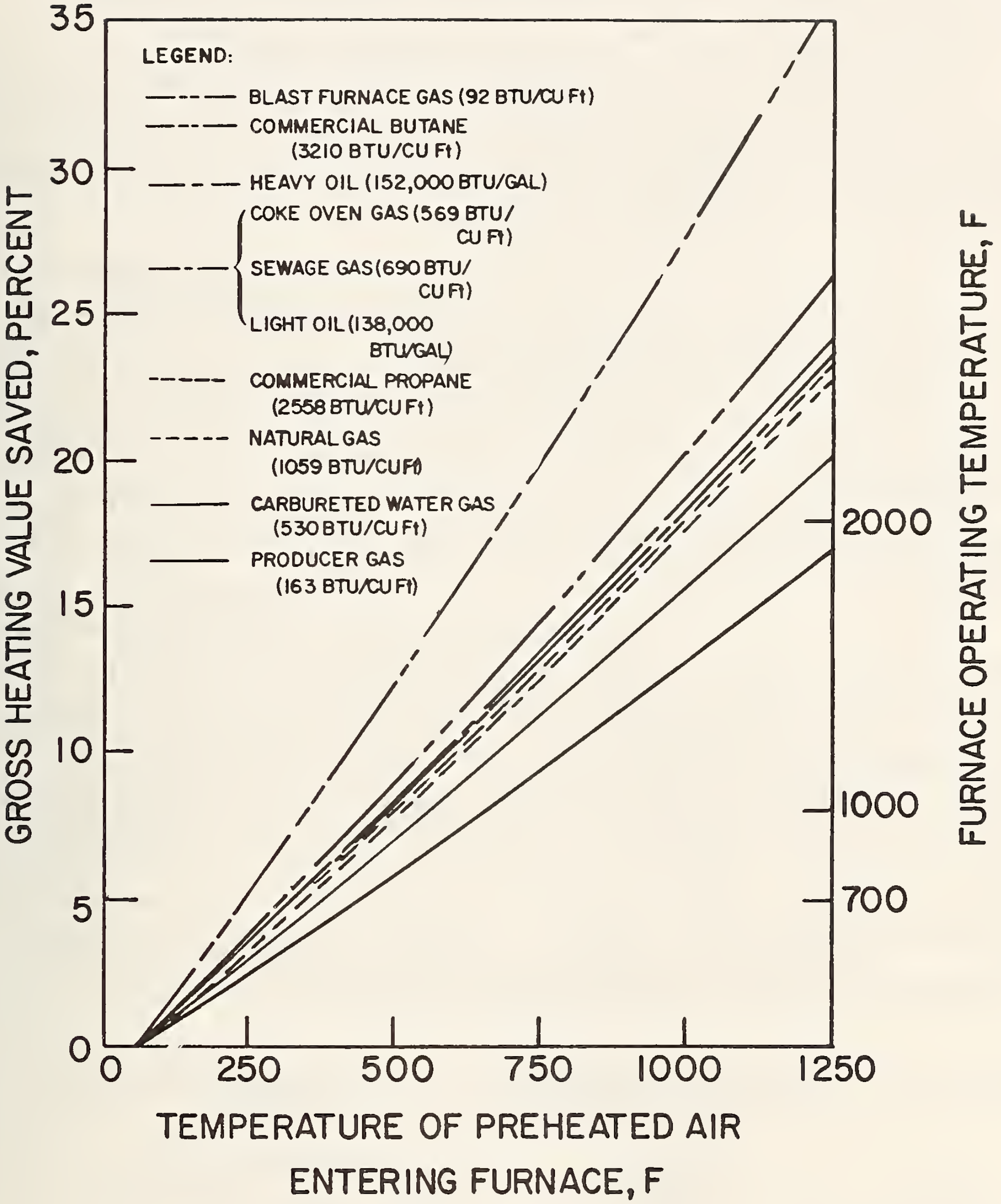

Figure 3-13. Fuel savings resulting from use of preheated combustion air [10].

of the combustion air entering the boiler produces approximately a 2% reduction in boiler loss for natural gas and light oil.

Combustion air can be preheated up to 350° F for stoker-fired coal, oil, and gas. The maximum temperature permissible is determined by the type of construction involved and the materials of the firing equipment. Manufacturer's recommendations should be followed in all cases.

There are several sources of heat to perform the preheating function. As an example, ambient air in most boiler rooms is heated incidently by boiler and pipe surfaces and tends to collect below the ceiling. This air can be utilized directly as preheated combustion air by ducting it down to the firing level and directing it into the primary and secondary air intakes. Other methods available, discussed in section 3-11, involve recovering waste heat from boiler stacks, from condensate, and from blowdown hot wells, among other sources.

One word of caution concerning waste-heat recovery from combustion gases should be made. If the temperature of the flue gases is reduced too much, sulfur trioxide (SO_3) will combine with moisture in the flue gas to produce sulfuric-acid mists. The temperature at which this happens is called the acid dew point temperature. When sulfuric acid mist occurs, it will adhere to cooler surfaces (such as recuperator tubes), entrap fly carbon and other particles in the flue gas, and initiate low-temperature corrosion. Minimum stack temperatures for trouble-free operation when burning oil are given in table 3-8.

TABLE 3.8. MINIMUM STACK TEMPERATURES FOR TROUBLE-FREE OPERATION

Fuel	Sulfur %	Minimum Stack Temperature °C	°F
Diesel type	1.0	138	280
Heavy oil	2.5	155	310
Heavy oil	3.5	160	320

In most cases, however, the corrosion problem will not be serious if the temperature of the metal is kept above 240° F. Exposed metal ductwork can be protected through insulation. A metal stack can be protected by an aluminum outer shroud which provides an air space, or it can be left bare and replaced as necessary. Masonry stacks also are exposed to such corrosive action and may be protected by a glazed tile liner.

3.6.1.2 Limit Combustion Losses

The loss of total heat input as a function of the percent (by weight) of excess air for a natural-gas-fired boiler is given in figure 3-14. Note that good burner performance occurs when the percent excess air is between 0 and 6% (by weight). Excess air in the amount of 30% increases the loss of total heat input by approximately 2%. This shows the need for measurement and adjustment of primary and secondary air to burners.

3.6.1.3 Preheat Oil to Increase Efficiency

In the case of oil-fired boilers, preheating oil can increase the efficiency of the boiler by as much as 3%, depending on the particular constituents of the oil involved. Table 3-9 gives the minimum preheating temperatures for various grades of oil to attain complete atomization.

TABLE 3-9. MINIMUM OIL PREHEATING TEMPERATURES REQUIRED FOR COMPLETE ATOMIZATION

Grade Oil	Temperature, °F
No. 4	135° F
No. 5	185° F
No. 6	210° F

Fuel oils must be preheated to give optimum viscosity at the burner. Heating oil beyond the temperatures indicated will increase efficiency even more, but is not generally recommended as it can result in vapor locking and flame-out.

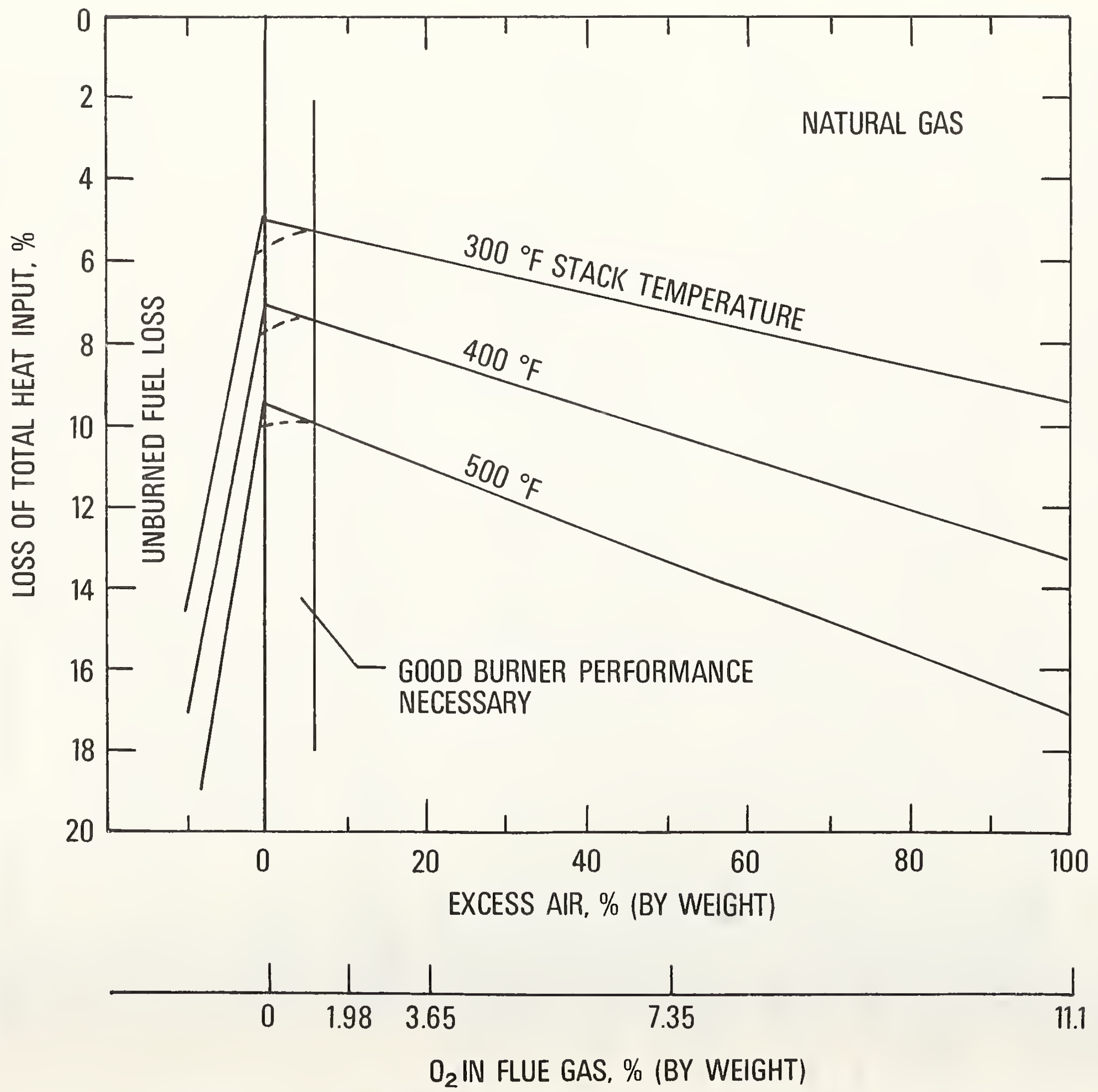

Figure 3-14. Combustion loss variations as a function of excess air for a natural-gas-fired boiler.

3.6.1.4 Reduce Blowdown Losses

The purpose of blowing down a boiler is:

- to maintain a low concentration of dissolved and suspended solids in the boiler water, and
- to remove sludge in the boiler.

The two principal types of blowdown are intermittent manual blowdown and continuous blowdown. Manual blowdown (or sludge blowdown) is necessary whether or not continuous blowdown equipment is installed. The frequency of manual blowdown depends on the amount of solids in the boiler makeup water and the type of water treatment used. Continuous blowdown results in a steady energy drain because makeup water must be heated.

Blowdown energy losses can be minimized by the use of automatic blowdown controls that monitor the conductivity or PH of the boiler water periodically and blowdown the boiler only when required to maintain acceptable water quality. Further savings can be realized if the blowdown water is piped through a heat exchanger or a flash tank with a heat exchanger. In this way, heat from the boiler-blowdown flash tank can be used for preheating feed water.

3.6.1.5 Isolate Off-Line Boilers

Small heating loads on a multiple boiler installation are often met by one boiler on line, with the remaining boilers idling on stand-by. Idling boilers consume energy to meet stand-by losses. In many cases these losses are increased by a continuous induced flow of air through the idling boilers and up the stack. Unless a boiler is about to be used to meet an expected increase in load, it should be secured and isolated from the heating system (by closing valves) and from the stack (by closing dampers).

A large boiler can be fitted with a by-pass valve or a regulating orifice (see figure 3-15) to allow the minimum flow required to keep it warm and avoid thermal stress when it is brought on-line again. When boiler waterside is isolated, it is important to prevent air flow through the stack. It is possible for a back flow of cold air to freeze the boiler.

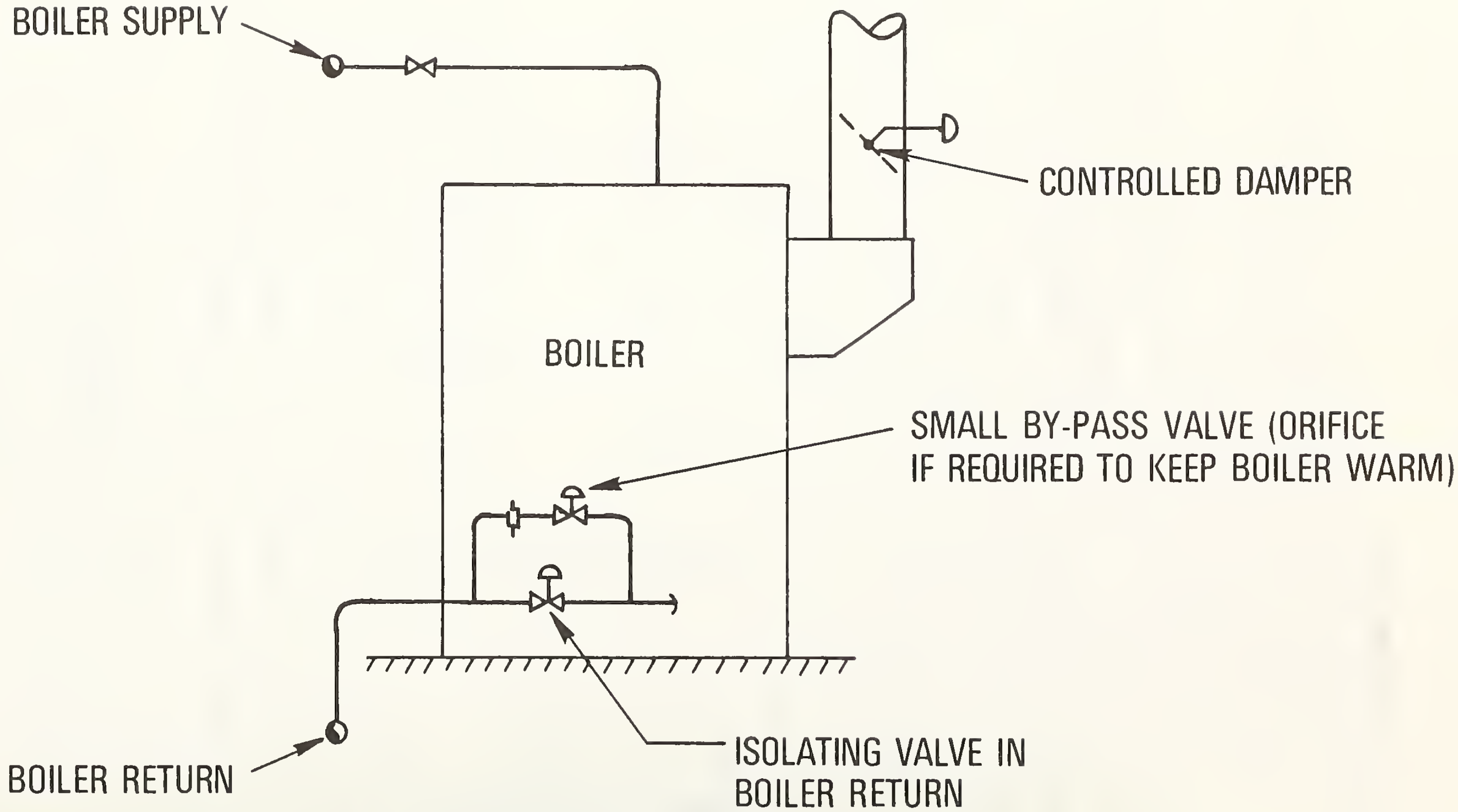

Figure 3-15. Techniques to isolate an off-line boiler.

3.6.1.6 Replace Boilers at or Near End of Useful Life

Boilers which are at or near the end of their useful life should be replaced by modern boilers which are matched to the current and projected needs of the installation involved. In most cases 80% efficiency can be obtained with new boilers on the market. Increased efficiency can be gained by specifying multiple boilers with high efficiency burners. Replacement burners should be selected on the basis of life-cycle cost rather than first cost. Increased cost of fuel, labor and materials should be considered in developing life-cycle-cost projections. Also consider installation of a dual-fuel system to reduce problems related to shortages or curtailments.

3.6.1.7 Replace Existing Boilers with Modular (Multiple) Boilers

Most boilers achieve maximum efficiency only when running at their rated output. In most cases, however, full boiler capacity is seldom required because heat load is 60% or less than full load 90% of the time (see figure 3-16). Given this situation, large capacity boilers in single units operate intermittently for the major part of the heating season. Although high-low firing capabilities may reduce cycling, the boilers can reach their design efficiency only for short periods, resulting in low seasonal efficiencies (see figure 3-17).

A modular (multiple) boiler system comprised of two or more boiler units will increase seasonal efficiency. Each boiler is fired near its capacity as much as possible. As the heating load increases, the separate boilers are brought on line in sequence until the load is satisfied. In a typical installation, replacement of a single-unit, large-capacity, boiler with modular (multiple) boilers may improve the seasonal efficiency by 5 to 10%. A good time to consider making this replacement is when the existing boiler system is at or near the end of its useful life.

3.6.1.8 Convert from Steam Atomizing Burners to Air-Atomizing Burners

Steam-atomizing burners consume at least 1.5% of the total boiler capacity to atomize the fuel oil. By contrast, air-atomizing burners rely on an air compressor, which costs far less to operate. Accordingly, when an existing steam-atomizing burner is at or near the end of

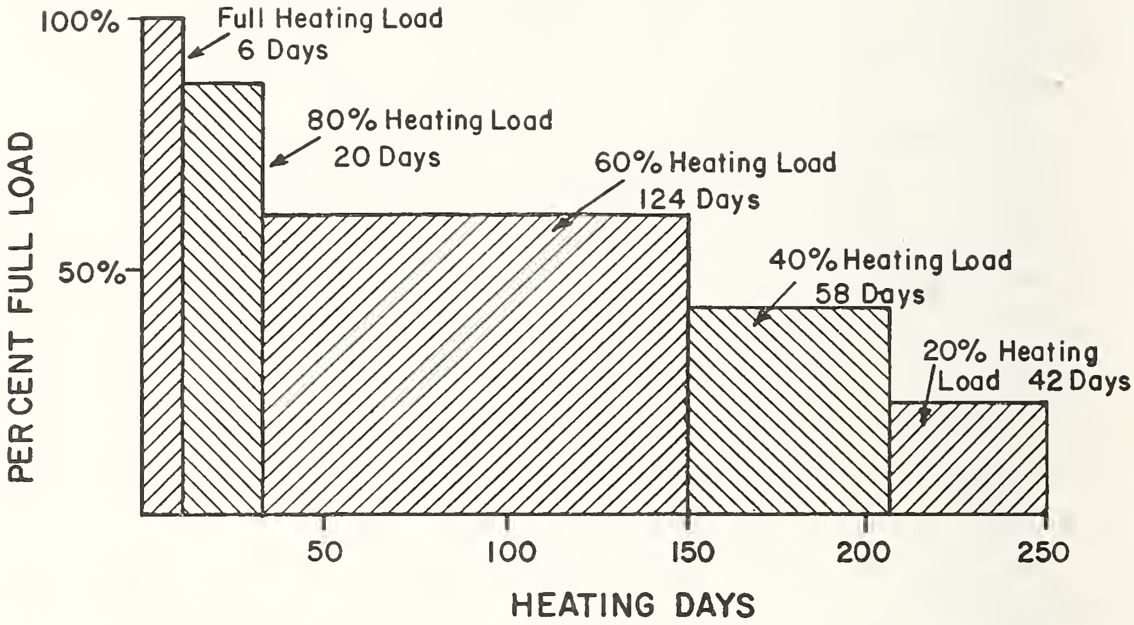

Figure 3-16. Typical heating load distribution (250-day season in 6000 degree-day zone).

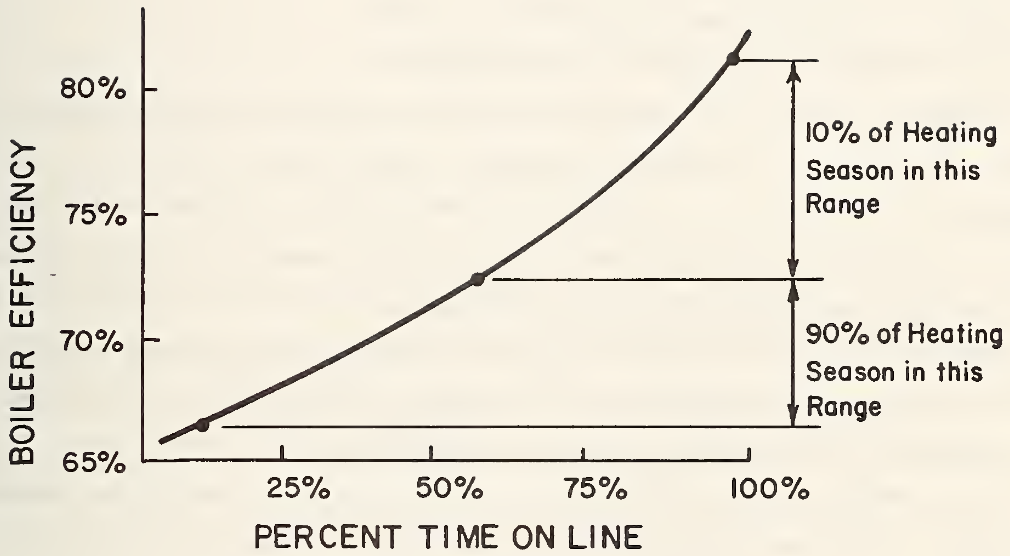

Figure 3-17. Effect of cycling to meet part loads.

its useful life, an air-atomizing replacement should be considered. In certain cases the savings to be obtained from an air-atomizing burner will warrant its installation prior to the scheduled replacement of the existing system.

3.6.2 Chilled-Water Plants

For both absorption and vapor-compression systems, the parameter characterizing the operating efficiency of the refrigeration system is the coefficient of performance (COP), which is defined by the relation:

$$COP = \frac{T_L}{T_H - T_L} \tag{5}$$

where T_L = absolute temperature of evaporator
and T_H = absolute temperature of condenser.

A chilled water plant can be made to operate more efficiently by increasing its COP. This may be accomplished by decreasing the condenser temperature (T_H) which in turn may be achieved by reducing the outlet water temperature of the cooling tower. The cooling tower is usually sized to give the required capacity at the maximum outside design wet-bulb temperature for the location involved. This maximum wet-bulb temperature is seldom reached, and a majority of the time it is possible to operate the condenser water system at lower temperatures.

The COP may also be increased by increasing the evaporator temperature (T_L). This may be accomplished by simply elevating the temperature of chilled water supplied to the chilled water distribution system. Most building equipment for providing space cooling is designed for the hottest day of the year, so that the equipment is able to operate satisfactorily using chilled water at elevated temperature on many days without any loss of human comfort.

Energy conservation measures for chilled-water plants are summarized below:

- ° Consider the use of lower condenser water temperatures to reduce the electric energy consumption for refrigeration equipment;
- ° Select and operate chillers so as to maximize the operating time at full-load conditions. Many types of refrigeration systems

operate at maximum efficiency at full load. When the load decreases, a reduction in efficiency occurs;

- Utilize elevated chilled-water temperatures during times when humidity conditions permit;
- Consider using heat exchangers to recover heat from condenser water.

3.7 Electrical Load Management

3.7.1 General

There are two key areas where the control of electrical utilization can provide energy savings. They are:

- Load curtailment (end-use conservation);
- Load management (shifting of usage patterns).

Load curtailment involves actual reduction in consumption by users of electricity through several means including reducing lighting, reducing temperature, controlling equipment usage and using more efficient electric appliances and equipment. These methods have the joint benefit of reducing generating requirements, and also reducing the consumption of fossil fuels needed to generate the electricity.

Load management is directed mainly at reducing peak generating capacity requirements. Users of electricity are encouraged to shift the times when they use electricity from peak-usage periods to off-peak periods, resulting in a flattened electrical daily-load-profile curve. In this way, the generating capacity requirement (which are determined by generating requirements at peak time) can be reduced and demand charges, employed by the utility company, can be reduced.

3.7.2 Selection of Interruptable Loads

Loads essential to the Base buildings operations, as for example, ventilation in hazardous areas, should not be deferred to other times.

Most loads that may be considered for control fall into one of the following categories:

(1) Inhibited loads - loads which are not used constantly and may be scheduled to be supplied at certain times of the day, such as battery charging banks;

(2) Switchable loads - loads which are locally controlled such as furnace in response to a thermostat control;

(3) Cyclic loads - loads that turn on-off at intervals throughout the day, such as refrigeration compressors.

Loads most commonly selected for interruption include:

Water heaters	Non-essential lighting
Air conditioners	Large pumps
Electric heaters	Ventilating fans
Furnaces	Clothes dryers
Boilers	Washing machines

3.7.3 Major Loads that can be Deferred without Consumer Dissatisfaction

In the United States, at present, the only consumer load segment having a substantial energy storage capacity is electric water heaters. The volume of a typical water heater tank is about 50 gallons, which is sufficient to avoid any great inconvenience to the customer under normal usage, if the heater is disconnected from the electrical supply for only a few hours. Since the average diversified demand of a water heater during peak hours is about 1 kilowatt, an Air Force Base with 300 water heaters under control could thus reduce its load by about 300 kilowatts during the peak hours.

3.7.4 Load Management Control Techniques

Three different load-management techniques are currently employed:

(1) Time Control

The simplest load-management device is a clock-activated switch at each individual load unit to be shed. It is the least expensive.

To make a change in the on-off times of the controlled load is a time-consuming and costly process, depending on the number of such units on the Base. On the other hand, the main advantage of systems using clocks is their simplicity, which is reflected in the relatively low cost of the hardware.

A more complex time-control system is a demand controller. It is the most common device used to defer loads. A demand controller basically is a comparator. It is an electronic load-programming unit that monitors the rate of energy consumption and compares this rate against a preset desired rate of energy use. The unit can control many electrical circuits, depending on its complexity. Input to the controller is derived by an electrical connection to both the kilowatt-hour meter and the demand meter. The demand meter provides the controller with an end-of-interval signal derived from either a built-in timer or an external clock. Some of the limitations on the use of this equipment are physical requirements, accessibility of utility-metering equipment for making connections, and number and strength of pulses.

(2) Radio Control

The principle of radio control is simple. Radio transmitters are placed in suitable locations to cover the Base with a sufficiently strong signal. The transmitted waves are modulated by a code containing an address for selecting the desired receiving group, as well as the control message itself. At the load under control, a radio receiver detects and decodes the message and switches on and off the load.

(3) Ripple Control

In this method of load control, a coded narrow-band audio frequency signal is introduced into the power grid. This signal conveys switching commands to receivers placed at the controlled load.

The ripple control system is more expensive than the radio control system. At present, there are no United States manufacturers producing this hardware system commercially.

3.8 Energy Monitoring Control Systems*

3.8.1 Background

Energy Monitoring Control Systems (EMCS) currently represent a sizable investment in Air Force dollars. If the capabilities of these systems are properly understood and applied at Base level, the EMCS concept can be a tremendous benefit in saving money and manhours. If they are misunderstood or improperly used, these savings are lost. An EMCS can be an automated watchdog to conserve energy for the Base Civil Engineer (BCE) and permits more work to be accomplished with less resources.

EMCS have proven their worth in both civilian and military applications by reducing the operating costs of heating, ventilating and air

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* Much of this material was extracted from reference [11].

conditioning (HVAC) equipment; by improving reliability; and by reducing manhours in accomplishing work. Documented utility savings at Luke AFB for only one month (July 1975) amounted to over $12,000 (which does not include savings from peak demand charges) [11].

The Air Force has begun to develop the potential reductions in energy consumption on its Bases. Automatic load shedding, night setback, CN/OFF control programs, and reduced ventilation rates in buildings are some of the major EMCS areas that can yield up to a 30 percent total annual energy savings per Base.

3.8.2 Description of Typical System

A typical EMCS is shown in figure 3-18. The main components of the system are:

(1) transducers: These devices monitor the environmental conditions within a building and the operating conditions of its HVAC, hot water heating, and other type equipment.

(2) remote building terminals: These units receive signals from the various transducers and process these signals for transmission to the central-processing terminal.

(3) transmission cable: This part of the system carries processed signals from the remote building terminals to the central-processing terminal.

(4) central-processing terminal: This part of the system contains either a computer for converting processed-signal data into engineering units (or electronic hardware for performing this function) and input/output devices such as teletype printers, cathode-ray-tube displays, and a line printer.

3.8.3 Considerations for Procurement of EMCS

An Air Force guide specification is now available for the procurement of an EMCS. Before purchasing an EMCS, it is necessary that a detailed input/output summary be carefully developed at the Base level by the EMCS project engineers who are familiar with both the electrical and mechanical systems of the Base (see figure 3-19). The I/O summary should identify (by priority of economic payback) which pieces of equipment and what environmental conditions will be monitored/controlled by the EMCS. This is an important step in the planning process and must be carefully

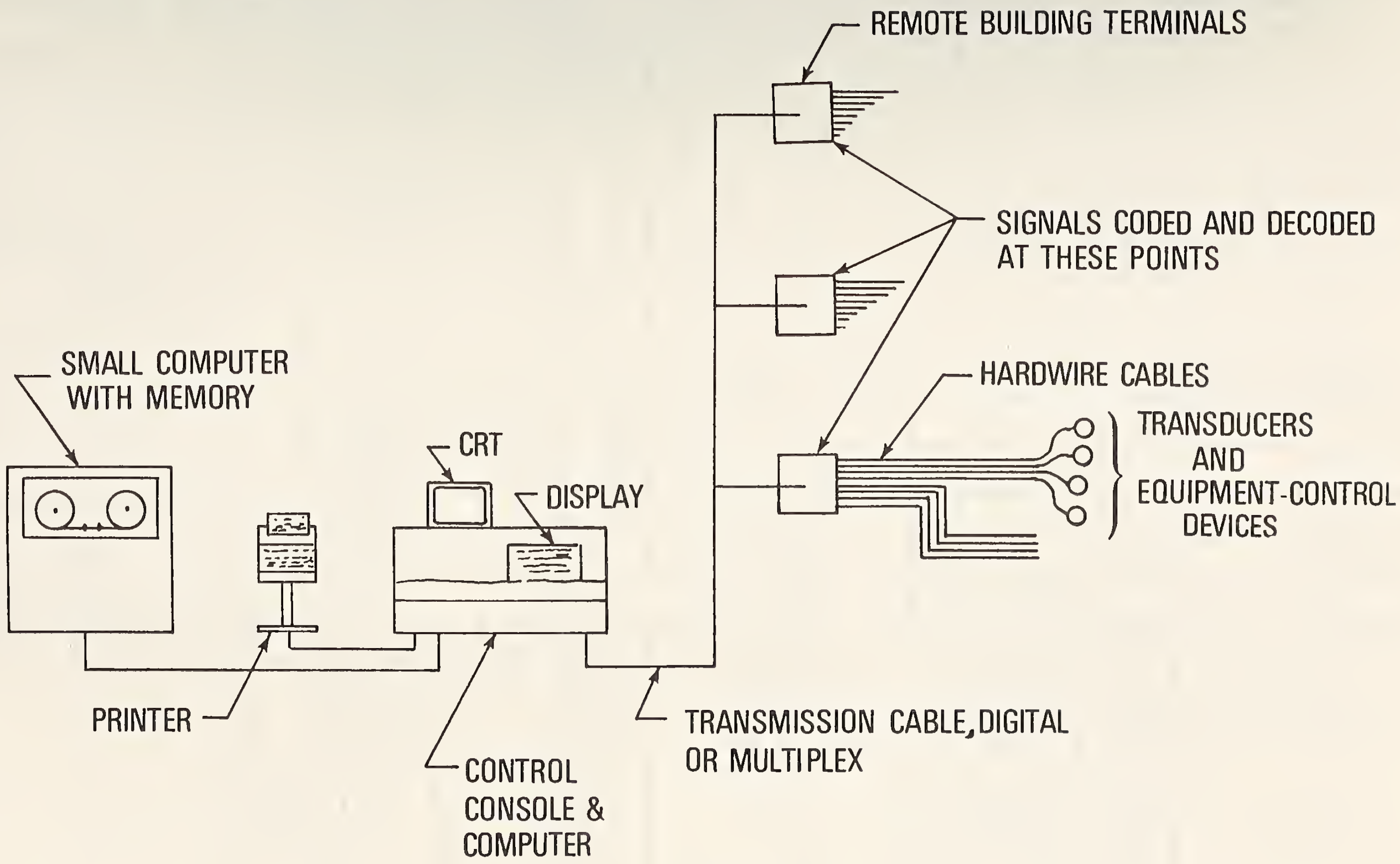

Figure 3-18. Typical energy monitoring control system.

	Diagrammatic Slide or Graphic	Intercom Station	Analog Indication					Alarm					Control							REMARKS
			Temperature	Pressure	Relative Humidity			Digital (control)	Analog			Maintenance	Start/Stop & Status	Status		Program Control	Reset Control PT ADJ (OPA)	Damper POS ADJ (off)	Position Feedback for OPA on OPA	
SUPPLY SYSTEM	X	X																		
SUPPLY FAN								A					A			A				
RETURN FAN								A					A			A				
RETURN AIR TEMP.			B						B											
MIXED AIR TEMP.			B						B											
DISCHARGE AIR			B						B								C		C	
FILTER												2B								Diff. Pressure Across Filters
LOW TEMP. ALARM (Freeze)								B												
ZONE TEMP. INDI.			C														C		C	Specify zones
MINIMUM O.A.			D															C	C	
REHEAT													C							
RETURN AIR R.M.				D					D											
RETURN AIR FIRE ALARM								D												

Figure 3-19. Typical input/output summary. A - high return on investment. B - highly desirable points, able to monitor system operations. C - provides data and control capability to perform manual optimizing functions. D - points that allow more complete picture of total system operation [11].

done to insure maximum benefits and minimum payback period for the initial investment. Lack of proper planning in this area can mean an inefficient system.

Often small EMCS systems are initially purchased due to funding limitations. As more funds become available, systems are often expanded to include more channels and increased data processing capabilities. It is strongly recommended that in adding on to such complex systems, an attempt be made to purchase the add-on equipment from the manufacturer of the original system, in order to avoid the following problems associated with hybrid systems:

- interfacing problems between components manufactured by different companies;
- legal disputes concerning who owns maintenance problems;
- loss of warranty due to system modification.

Another important consideration in procuring an EMCS is to obtain a system with the capability for future expansion. In many instances, a system having expansion capability will cost only a small percentage more than a system without this capability. On the other hand, if this capability is not provided for, then the expansion of the system at some future time may require the procurement of new major components at the cost of essentially a new system.

3.8.4 Select and Train an Operator

Console operator training must not be a last minute "rush" job. During start-up of a new EMCS, there will undoubtedly be some "teething pains." Consequently, the operator must have enough technical knowledge of the system to do some trouble shooting. He cannot expect to learn the system solely through trial and error methods or on-the-job training (OJT). It would be best if the operator were already trained by the time the contractor starts to install the EMCS; thus, contractor personnel and the operator could work together as the system is being built up.

The system operator must be familiar with both the electrical and mechanical systems on Base. Preferably, he should be civilian, rather than military, to avoid the interruptions resulting from PCS moves. To the extent possible, initial training requirements should be included in the original EMCS project funds with training provided by the equipment supplier. All console operators should attend at least one commercial training course prior to system start-up or before assuming EMCS duties.

In addition to the manufacturer's training school, an EMCS Mobile Training Team is now available at Sheppard AFB, Texas, to assist Base personnel and provide on-site OJT during the EMCS start-up period.

Another aspect of the EMCS training program involves general education/orientation sessions given to BCE shop personnel and management. Shop personnel play an important part in the success of EMCS. They must be pre-briefed and have a good working knowledge of the system's capabilities prior to start-up. Users of the system should also be pre-briefed on what the equipment can and cannot do. All levels of management should be educated to insure that a positive attitude exists toward the new EMCS.

3.8.5 Communications Lines

An EMCS system requires the use of telephone lines (pairs) of coaxial cable to communicate data between the field panels (located in various facilities throughout the Base), and the EMCS console which is usually located in the BCE Control Center/Service Call Area. It is therefore essential that the EMCS project engineer(s) closely coordinate with the local Base communications people early in the planning stage. Past experience has shown that serious project delays can result if the Base communications people are left in the dark or told at the last minute about the EMCS requirements. Communications and electronics plans to install new telephone equipment for the EMCS can take up to 12-18 months for approval. The telephone pairs must be high quality; the coaxial cable must also be high quality and properly installed. Failure to closely monitor this phase of the project will quickly jeopardize system effectiveness.

An important consideration for an EMCS is to provide lightning protection for above ground signal-transmission and power lines. Lightning that strikes these lines may induce transient electrical surges which can do substantial damage to the EMCS equipment. The following measures are recommended to protect these lines:

- install signal transmission lines underground instead of above ground.
- install arrestor (or voltage-clipping circuits) on incoming power lines.
- when signal transmission lines are installed above ground, utilize shielded cable with the exterior-shield grounded to the earth.
- install lightning rods at high places where signal-transmission lines are located. The purpose of these rods is to bleed off electrical charge, instead of accumulating and creating a transient voltage surge on signal-transmission lines.

3.8.6 Back-up Equipment

One manufacturer has a test kit that provides a back-up capability for the EMCS in the event of equipment failure. This suitcase-size test kit ($3,000) may be taken anywhere on base and connected into the EMCS. It has a control capability but does not monitor input data. If, for some reason, a field panel is one of the Base facilities should fail, the test kit could be taken out to the facility and connected into the system, bypassing the faulty field panel. Likewise, if the EMCS console should shut down, the test kit could be connected into the main system, bypassing the console; thus, the kit provides the operator with a control capability until the console is brought back on line. Meanwhile, the mechanical and electrical equipment tied into the EMCS will continue to operate at the same settings that were being used prior to the failure. Therefore, a loss of the EMCS console would not result in the shutdown of the entire Base-wide HVAC systems.

3.8.7 Service/Maintenance Contracts

Any piece of electronic equipment will require periodic servicing and maintenance; an EMCS is no exception. The console operator can perform a limited amount of servicing and minor maintenance on the equipment if he has been properly trained at the manufacturer's school. This is now being done at Luke AFB, for example, with the approval of the EMCS manufacturer. Only qualified personnel should be allowed to work on EMCS equipment. An in-house capability for total EMCS servicing does not generally exist at Base level; therefore the manufacturer's personnel will be needed for most servicing requirements. Contract costs vary considerably. As a rule of thumb, an annual service/maintenance contract will cost anywhere from 5 to 10 percent of the initial EMCS purchase price. To help reduce or avoid these annual service contract costs, the Air Force is considering as a long-range goal, having a total in-house service repair capability (24 hours a day) for EMCS equipment at each Base.

New EMCS's normally have a one-year warranty with the respective manufacturer which may or may not include maintenance on the EMCS equipment. It is best to get a service contract option at the time of bidding the EMCS contract. After the initial warranty period, some type of annual service/maintenance contract will be required. Again, this involves advance planning and budgeting; it must not be put off until the last minute. It is important that the warranty not be allowed to expire without a follow-up service contract.

3.8.8 EMCS Manning

The EMCS must have adequate manning depth by qualified people to insure stable, continuous use of the system. Current Base level EMCS manning is frequently only one-man deep. There must be back-up manning for the console operator position to allow for TDY, leave, sickness, or injury.

The EMCS concept is a new and efficient element of utility management. It is necessary to understand its capabilities and limitations if we are to properly use these systems as an integral part of Base energy conservation programs. Technical expertise is available within the various Air Force commands for anyone wanting assistance in developing a viable EMCS program.

Motivation is a key factor in producing a successful EMCS program. Both management and labor must collectively agree to make the system work. Disinterest on the part of middle and top level management at Base level will practically ensure a slow death; disinterest, mistrust, or misunderstanding of EMCS principles by the operators and shop technicians on Base will also guarantee that the program will fail.

It is necessary to carefully and thoughtfully consider better ways to manage our scarce resources. An EMCS provides us with innovative ways to save energy and avoid spending utility dollars. A great flexibility in operating and maintaining our utility systems is also gained.

3.9 Energy Recovery through Waste POL Utilization

One proven and practical procedure which can result in substantial energy savings is the utilization of waste POL's (petroleum, oils, lubricants) as supplementary heating fuel in oil- and natural-gas-fired boilers. Under this system, waste POL's are blended with the existing fuel type and the mixture is combusted in the existing boiler. Millions of gallons of contaminated fuels and lubricants are generated each year on Air Force Bases. Examples of waste POL's include:

- waste aviation-piston-engine oil;
- waste aviation-turbine lube;
- simple mixtures of waste aviation-piston-engine oil, synthetic turbine oil, hydraulic fluid;
- contaminated jet fuel (i.e., JP-4, JP-5); engine oil, synthetic turbine oil, automotive crankcase oil, and non-halogenated solvent.

These waste POL's have a Btu content approximately equivalent to that of No. 2 or No. 5 fuel oil (18,500 Btu/lb). Therefore, their inclusion in the fuel supply can result in a gallon-for-gallon reduction in fuel oil purchases or similar reductions in the demand for natural gas.

This technique has been well tested and documented, and a complete description of the test facilities, design details, and testing results can be found in technical report AFCEC TR-76-2 [12]. Recent studies conducted at three Air Force Bases have shown that combustion of waste POL in standard fuel mixtures does not impair boiler performance. In fact, it may actually increase the boiler efficiency. In addition, utilization of controlled percentages of waste POL's does not increase stack emissions over federal air-quality standards.

The economic feasibility of a waste POL system at any individual Base depends on several factors, including length of the heating season, current price of the fuel oil or natural gas used, and the resale value of the waste POL's. A cost analysis, comparing cost of installation versus annual savings in fuel purchases, will determine if the waste POL system is financially attractive. Large installations will likely find rapid amortization of capital expenditures for installation.

The steps required for waste POL utilization are:

(1) Collection, segregation and storage of waste POL's;

(2) System modifications to provide comingling of the POL's with the standard fuel immediately before injection into the boiler.

Required alterations to existing boilers are usually minimal (i.e., a simple tee connection for oil burners, a bit more for gas-fired boilers).

It is important that JP-4 be segregated and stored in a separate steel tank. Therefore, in order to be assured of proper segregation during all phases of transportation, collection, and unloading, a segregation plan should be devised in accordance with TO 42B-1-23 to analyze and segregate waste POL's. It is also important that halogenated-hydrocarbon components not be included with other waste POL for utilization, since the combustion of these components results in the production of hydrogen-chloride or hydrogen-fluoride gases which are highly corrosive and will severely damage boiler components.

Appropriate consideration should be given to the location of piping where it enters the heating plant. Due to the volatility of JP-4, special care is required in order to maintain a safe system free of leaks or malfunctions. Since the waste POL pipelines in the vicinity of the boilers have a tendency to elongate, poorly installed connections could deflect or distort, resulting in extremely dangerous vapor leaks. Techniques for minimizing the possibility of such vapor leaks are given in reference [12].

3.10 Power-Factor Adjustments

3.10.1 Background

Electric equipment having inductive loads such as induction motors (especially when operated at less than full loads), transformers, arc welders, rectifiers, arc furnaces, fluorescent lamps, and various types of electronic equipment is magnetized from the electrical distribution system causing a lagging component in their current. Energy spent in building up the magnetic fields of these equipments flows back and forth between the generator and the load. This out-of-phase magnetizing current causes the total current in the line to lag behind the voltage giving rise to the power factor (PF) in the system. Power factor is defined as the ratio of working active current to total current (which includes both the working active current and the magnetizing current).

Power factors for various equipment are given in table 3-10. From the definition of the power factor, a low power factor occurs when a piece of electric equipment requires a large magnetizing current. From table 3-10, it is seen that equipment having low power factor or high magnetizing current includes induction motors, arc welders, arc furnaces, and induction furnaces.

The magnetizing current drawn by inductive equipment does not contribute to the work or heat output of electric equipment, but it nonetheless does constitute a KVA load on transformers of the electrical distribution system. Thus, magnetizing current does cause the voltage of an electrical distribution system to be reduced. This voltage drop will decrease the performance of other electric equipment and reduce illumination from incandescent-light sources. This magnetizing current must also be transported through the distribution system, and therefore increases copper losses (I^2R losses). Since the magnetizing current does represent a KVA load on generating equipment, it has the effect of using up valuable generating capacity. Power companies, therefore, often impose a demand charge for lagging power factors that are a part of the customer's load.

TABLE 3-10. TYPICALLY HIGH LEADING AND LAGGING POWER-FACTOR LOADS [13].

High loads	*Approximate PF*
Incandescent lamps	1.0
Fluorescent lamps (with built-in capacitors)	0.95-0.97
Resistor heaters	1.0
Synchronous motors	1.0 or leading
Rotary converters	1.0
Leading loads	
Synchronous motors	0.9; 0.8; etc., leading, depending on rating
Synchronous condensers	Nearly 0, leading
Capacitors	0
Lagging loads	
Induction motors (full-load)	
Split-phase fractional hp	0.55 to 0.75
Split-phase (1-10 hp)	0.75 to 0.85
Split-phase condenser type	0.75 to 1.0
Polyphase squirrel-cage motors	
High speed (1 to 10 hp)	0.75 to 0.90
High speed (10 hp or more)	0.85 to 0.92
Low speed	0.70 to 0.85
Wound rotor	0.80 to 0.90
Groups of induction motors	0.50 to 0.85
Welders	0.50 to 0.70
Arc furnaces	0.80 to 0.90
Induction furnaces	0.60 to 0.70

The simplest and most economical method of improving power factor at the Base level is to use capacitors. When properly applied to an electrical distribution system, a power-factor-correcting capacitor supplies a leading reactive magnetizing current that counteracts the lagging reactive current produced by induction equipment. This improves the overall power factor. Power-factor-correcting capacitors, when properly located, will release electrical system capacity (KVA), raise the voltage level, and reduce system losses. This will in many cases permit additional loads to be added to an existing system without the need for system capacity to be increased.

Consider the induction motor shown in figure 3-20. When the motor is operated without power-factor correction, a magnetizing current of 60 amps must be supplied from the electrical distribution system. The

60-amp magnetizing current represents a KVA load on the electrical distribution system. It causes the voltage output of transformers in the electrical distribution system to be reduced, thereby wasting valuable KVA capacity of the electrical distribution system. Also, reduced voltage may cause other equipment to operate less efficiently. Note that the installation of a power-factor-correcting capacitor causes the magnetizing current not to appear as a load on the electrical distribution system.

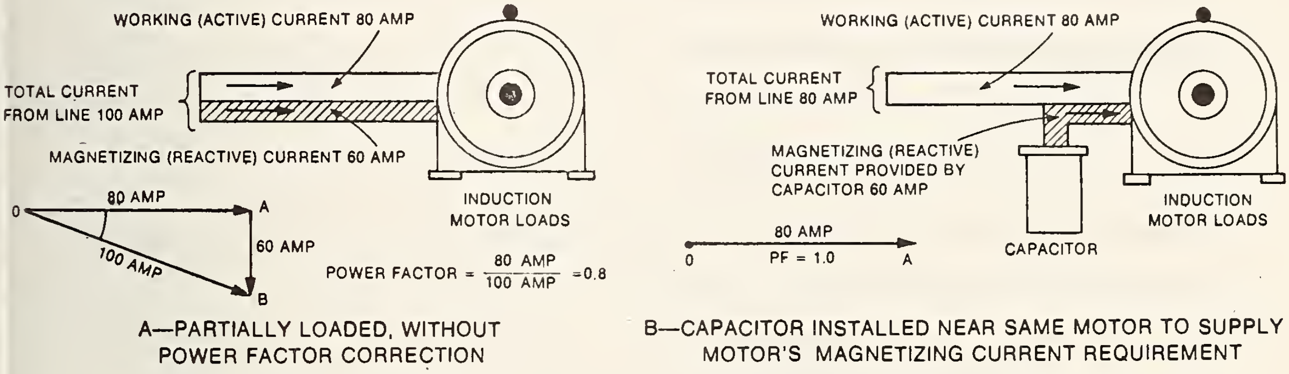

Figure 3-20. Power factor correction for an induction motor [13].

3.10.2 Power-Factor Fundamentals

Inductive loads require two current components: the magnetizing current (reactive current) and the power producing current (active current). The total current (I_T) is related to these two components by the relation:

$$I_T^2 = I_a^2 + I_R^2 \tag{6}$$

where I_a = active or power-producing current, amps

I_R = reactive or magnetizing current, amps .

At a common voltage, the total power component (P_T) is given by the relation:

$$P_T^2 = P_a^2 + P_R^2 \tag{7}$$

where P_a = active or actual power, KW

P_R = reactive power, KVAR

The power factor (PF) is defined as the ratio of active power (P_a) to the total power (P_T) or

$$PF = P_a/P_T \tag{8}$$

From trigonometric considerations,

$$PF = \cos\theta = P_a/P_T \tag{9}$$

Combining equations (7) and (9), we may relate reactive power to the real power by the relation:

$$P_R = P_a \cdot \sqrt{\left(\frac{1}{PF}\right)^2 - 1} \tag{10}$$

If capacitors are used to increase the power factor from one level to another, then the reactive power (ΔP_R) introduced by the capacitor must be equal to the difference between the reactive power levels calculated from (10), or

$$\Delta P_R = P_a \cdot \left\{ \sqrt{\left(\frac{1}{PF_i}\right)^2 - 1} - \sqrt{\left(\frac{1}{PF_f}\right)^2 - 1} \right\} \tag{11}$$

where the subscripts "i" and "f" denote the initial and final power factors, respectively.

The additional KVA capacity (ΔP_T) that is released by a power factor correction is equal to:

$$\Delta P_T = \frac{P_a}{PF_i} - \frac{P_a}{PF_f} = P_a \cdot \frac{PF_f - PF_i}{PF_f \cdot PF_i} \tag{12}$$

3.10.3 Analysis of Power-Factor Problems

To assist Air Force personnel in evaluating power-factor problems several charts are presented. Table 3-11 presents KW multipliers to determine kilovars required for power factor correction. To use this chart, enter the left vertical column at the original power factor that existed prior to the adjustment. Then move to the right along the row containing the original power-factor value until you reach the column containing the desired corrected power factor. The figure in the table at this location is the KW multiplier for determining the capacitor kilovars required for power-factor adjustment.

Figure 3-21 may be used to find the additional system capacity that is released as a result of power-factor correction. To find the percent additional capacity, enter the figure at the point on the horizontal axis coinciding with the corrected power-factor value. Proceed vertically upwards until you arrive at the curve for the original power factor. Interpolate when necessary. Read the percent additional capacity from the vertical axis. And finally, table 3-12 gives the 1975 prices of power-factor-correction capacitors.

TABLE 3-12. COST OF POWER-FACTOR-CORRECTING CAPACITORS (1975)

Voltage of Capacitor	Cost of Capacitor ($/KVAR)
240	20 - 30
480 - 600	10 - 15
2,400 - 13,800	5 - 10

Note that the dollars per kilovars is much cheaper for higher voltage capacitors. The following example is presented to demonstrate the use of the foregoing tables and figures:

TABLE 3-11. KW MULTIPLIERS TO DETERMINE CAPACITOR KILOVARS REQUIRED FOR POWER FACTOR CORRECTION [13].

Orig-inel power factor	Corrected power factor																				
	0.80	0.81	0.82	0.83	0.84	0.85	0.86	0.87	0.88	0.89	0.90	0.91	0.92	0.93	0.94	0.95	0.96	0.97	0.98	0.99	1.0
0.50	0.982	1.008	1.034	1.060	1.086	1.112	1.139	1.165	1.192	1.220	1.248	1.276	1.306	1.337	1.369	1.403	1.440	1.481	1.529	1.589	1.732
0.51	0.937	0.962	0.989	1.015	1.041	1.067	1.094	1.120	1.147	1.175	1.203	1.231	1.261	1.292	1.324	1.358	1.395	1.436	1.484	1.544	1.687
0.52	0.893	0.919	0.945	0.971	0.997	1.023	1.050	1.076	1.103	1.131	1.159	1.187	1.217	1.248	1.280	1.314	1.351	1.392	1.440	1.500	1.643
0.53	0.850	0.876	0.902	0.928	0.954	0.980	1.007	1.033	1.060	1.088	1.116	1.144	1.174	1.205	1.237	1.271	1.308	1.349	1.397	1.457	1.600
0.54	0.809	0.835	0.861	0.887	0.913	0.939	0.966	0.992	1.019	1.047	1.075	1.103	1.133	1.164	1.196	1.230	1.267	1.308	1.356	1.416	1.559
0.55	0.769	0.795	0.821	0.847	0.873	0.899	0.926	0.952	0.979	1.007	1.035	1.063	1.093	1.124	1.156	1.190	1.227	1.268	1.316	1.376	1.519
0.56	0.730	0.756	0.782	0.808	0.834	0.860	0.887	0.913	0.940	0.963	0.996	1.024	1.054	1.085	1.117	1.151	1.188	1.229	1.277	1.337	1.480
0.57	0.692	0.718	0.744	0.770	0.796	0.822	0.849	0.875	0.902	0.930	0.958	0.986	1.016	1.047	1.079	1.113	1.150	1.191	1.239	1.299	1.442
0.58	0.655	0.681	0.707	0.733	0.759	0.785	0.812	0.838	0.865	0.893	0.921	0.949	0.979	1.010	1.042	1.076	1.113	1.154	1.202	1.262	1.405
0.59	0.619	0.645	0.671	0.697	0.723	0.749	0.776	0.802	0.829	0.857	0.885	0.913	0.943	0.974	1.006	1.040	1.077	1.118	1.166	1.226	1.369
0.60	0.583	0.609	0.635	0.661	0.687	0.713	0.740	0.766	0.793	0.821	0.849	0.877	0.907	0.938	0.970	1.004	1.041	1.082	1.130	1.190	1.333
0.61	0.549	0.575	0.601	0.627	0.653	0.679	0.706	0.732	0.759	0.787	0.815	0.843	0.873	0.904	0.936	0.970	1.007	1.048	1.096	1.156	1.299
0.62	0.516	0.542	0.568	0.594	0.620	0.646	0.673	0.699	0.726	0.754	0.782	0.810	0.840	0.871	0.903	0.937	0.974	1.015	1.063	1.123	1.266
0.63	0.483	0.509	0.535	0.561	0.587	0.613	0.640	0.666	0.693	0.721	0.749	0.777	0.807	0.838	0.870	0.904	0.941	0.982	1.030	1.090	1.233
0.64	0.451	0.474	0.503	0.529	0.555	0.581	0.608	0.634	0.661	0.689	0.717	0.745	0.775	0.806	0.838	0.872	0.909	0.950	0.998	1.068	1.201
0.65	0.419	0.445	0.471	0.497	0.523	0.549	0.576	0.602	0.629	0.657	0.685	0.713	0.743	0.774	0.806	0.840	0.877	0.918	0.966	1.026	1.169
0.66	0.388	0.414	0.440	0.466	0.492	0.518	0.545	0.571	0.598	0.626	0.654	0.682	0.712	0.743	0.775	0.809	0.846	0.887	0.935	0.995	1.138
0.67	0.358	0.384	0.410	0.436	0.462	0.488	0.515	0.541	0.568	0.596	0.624	0.652	0.682	0.713	0.745	0.779	0.816	0.857	0.905	0.965	1.108
0.68	0.328	0.354	0.380	0.406	0.432	0.458	0.485	0.511	0.538	0.566	0.594	0.622	0.652	0.683	0.715	0.749	0.786	0.827	0.875	0.935	1.078
0.69	0.299	0.325	0.351	0.377	0.403	0.429	0.456	0.482	0.509	0.537	0.565	0.593	0.623	0.654	0.686	0.720	0.757	0.798	0.846	0.906	1.049
0.70	0.270	0.296	0.322	0.348	0.374	0.400	0.427	0.453	0.480	0.508	0.536	0.564	0.594	0.625	0.657	0.691	0.728	0.769	0.817	0.877	1.020
0.71	0.242	0.268	0.294	0.320	0.346	0.372	0.399	0.425	0.452	0.480	0.508	0.536	0.566	0.597	0.629	0.663	0.700	0.741	0.789	0.849	0.992
0.72	0.214	0.240	0.266	0.292	0.318	0.344	0.371	0.397	0.424	0.452	0.480	0.508	0.538	0.569	0.601	0.635	0.672	0.713	0.761	0.821	0.964
0.73	0.186	0.212	0.238	0.264	0.290	0.316	0.343	0.369	0.396	0.424	0.452	0.480	0.510	0.541	0.573	0.607	0.644	0.685	0.733	0.793	0.936
0.74	0.159	0.185	0.211	0.237	0.263	0.289	0.316	0.342	0.369	0.397	0.425	0.453	0.483	0.514	0.546	0.580	0.617	0.658	0.706	0.766	0.909
0.75	0.132	0.158	0.184	0.210	0.236	0.262	0.289	0.315	0.342	0.370	0.398	0.426	0.456	0.487	0.519	0.553	0.590	0.631	0.679	0.739	0.882
0.76	0.105	0.131	0.157	0.183	0.209	0.235	0.262	0.288	0.315	0.343	0.371	0.399	0.429	0.460	0.492	0.526	0.563	0.604	0.652	0.712	0.855
0.77	0.079	0.105	0.131	0.157	0.183	0.209	0.236	0.262	0.289	0.317	0.345	0.373	0.403	0.434	0.466	0.500	0.537	0.578	0.626	0.685	0.829
0.78	0.052	0.078	0.104	0.130	0.156	0.182	0.209	0.235	0.262	0.290	0.318	0.346	0.376	0.407	0.439	0.473	0.510	0.551	0.599	0.659	0.802
0.79	0.026	0.052	0.078	0.104	0.130	0.156	0.183	0.209	0.236	0.264	0.292	0.320	0.350	0.381	0.413	0.447	0.484	0.525	0.573	0.633	0.776
0.80	0.000	0.026	0.052	0.078	0.104	0.130	0.157	0.183	0.210	0.238	0.266	0.294	0.324	0.355	0.387	0.421	0.458	0.499	0.547	0.609	0.750
0.81		0.000	0.026	0.052	0.078	0.104	0.131	0.157	0.184	0.212	0.240	0.268	0.298	0.329	0.361	0.395	0.432	0.473	0.521	0.581	0.724
0.82			0.000	0.026	0.052	0.078	0.105	0.131	0.158	0.186	0.214	0.242	0.272	0.303	0.335	0.369	0.406	0.447	0.495	0.555	0.698
0.83				0.000	0.026	0.052	0.079	0.105	0.132	0.160	0.188	0.216	0.246	0.277	0.309	0.343	0.380	0.421	0.469	0.529	0.672
0.84					0.000	0.026	0.053	0.079	0.106	0.134	0.162	0.190	0.220	0.251	0.283	0.317	0.354	0.395	0.443	0.503	0.646
0.85						0.000	0.027	0.053	0.080	0.108	0.136	0.164	0.194	0.225	0.257	0.291	0.328	0.369	0.417	0.477	0.620
0.86							0.000	0.026	0.053	0.081	0.109	0.137	0.167	0.198	0.230	0.264	0.301	0.342	0.390	0.450	0.593
0.87								0.000	0.027	0.055	0.083	0.111	0.141	0.172	0.204	0.238	0.275	0.316	0.364	0.424	0.567
0.88									0.000	0.028	0.056	0.084	0.114	0.145	0.177	0.211	0.248	0.289	0.337	0.397	0.540
0.89										0.000	0.028	0.056	0.086	0.117	0.149	0.183	0.220	0.261	0.309	0.369	0.512
0.90											0.000	0.028	0.058	0.089	0.121	0.155	0.192	0.233	0.281	0.341	0.484
0.91												0.000	0.030	0.061	0.093	0.127	0.164	0.205	0.253	0.313	0.456
0.92													0.000	0.031	0.063	0.097	0.134	0.175	0.223	0.283	0.426
0.93														0.000	0.032	0.066	0.103	0.144	0.192	0.252	0.395
0.94															0.000	0.034	0.071	0.112	0.160	0.220	0.363
0.95																0.000	0.037	0.079	0.126	0.186	0.329
0.96																	0.000	0.041	0.089	0.149	0.292
0.97																		0.000	0.048	0.108	0.251
0.98																			0.000	0.060	0.203
0.99																				0.000	0.143
																					0.000

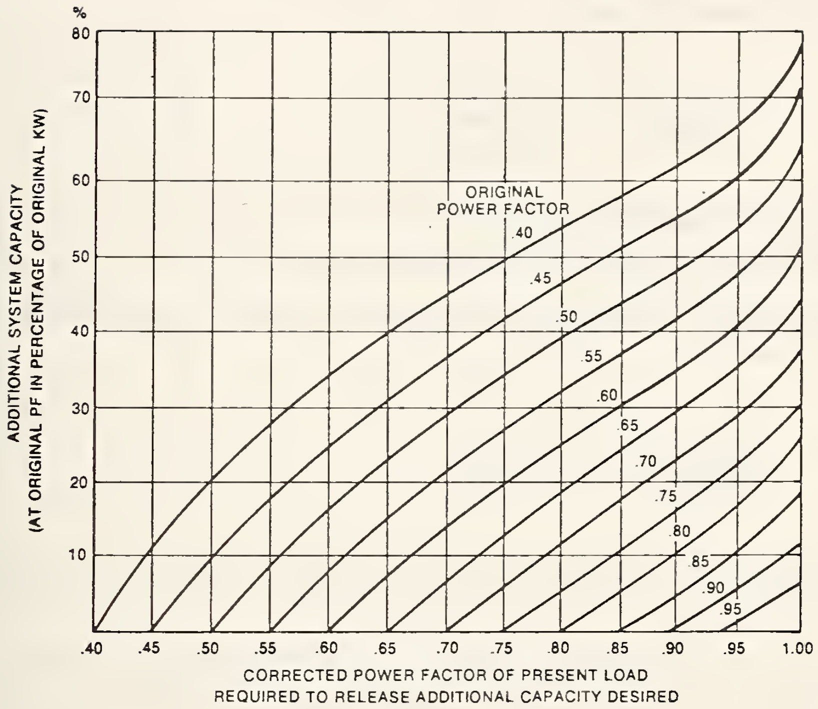

Figure 3-21. Additional system capacity after PF correction [13].

Example:

Here's how improving the power factor can save money. Assume that a plant's load is 1,500 KW at 0.75 power factor and the utility has a cost-rate schedule based on \$2/KVA, with a maximum billed power factor of 0.9. The billed KVA is:

$$\frac{1{,}500\ \text{KW}}{0.75\ \text{PF}} = 2{,}000\ \text{KVA}$$

The utility demand charge is:

2,000 KVA x \$2/KVA = \$4,000 a month.

The minimum KVA on which the plant's 1,500 KW demand cost can be based is:

$$\frac{1{,}500\ \text{KW}}{0.9\ \text{PF}} = 1{,}667\ \text{KVA}$$

The demand charge at 1,667 KVA would be \$3,334 a month (1,667 KVA x \$2/KVA).

This means that by improving the power factor from 0.75 to 0.9, the demand charge will be reduced \$666 a month (\$4,000 - \$3,334 = \$666).

From table 3-11, the KW multiplier to determine capacitor kilovars required for correcting the power factor from 0.75 to 0.9 is found to be 0.398. The number of kilovars to achieve the required power-factor correction is (0.398) (1500) = 597 KVARS.

From table 3-12, the cost of a 480/600 - volt static capacitor is \$10/KVAR or \$5970. Therefore, a \$5970 investment will produce \$666 monthly savings. The payoff time is less than one year.

Using figure 3-21, the additional system capacity after the power-factor correction is found to be (1500) (.18) = 270 KVA.

3.10.4 Selection of the Most Suitable Location for Power-Factor-Correction Capacitors

Consider the electrical distribution system diagram shown in figure 3-22. The induction motors and the small motors shown in the diagram introduce a lagging power factor. Power-factor-correction capacitors may be located at the following locations:

- at the motor (localized correction);
- at the 480-V Bus (group correction);
- at the 2.4 KV Bus (group correction);
- at 13.8 KV primary side of main transformer (group correction).

One of the main advantages of locating the power-factor-correction capacitor at either the primary side of the main transformer or at the 2.4 KV bus is that the cost of the capacitor is 5 to 10 dollars per kilovar, which is much less than at other locations. The cost of the capacitor at either the induction motor location or the 480-V bus approximately doubles.

Localized correction at the motor also has several advantages. Additional system capacity is released for all parts of the electrical distribution system located above the point of correction. This means that power factor correction will permit more electrical consuming equipment to be added to the distribution system without the need of expanding the system capacity. However, if the power-factor correction is achieved at the primary side of the main transformer, additional load capacity is achieved for the generating plant of the power company, but none is made available for the Air Force power-distribution system. An important point is that additional system capacity is only made available for those parts of the distribution system located above or upstream from the power-factor-correction capacitors. However, a useful analysis of all technical and economic aspects (including load utility-rate schedules, released load KVA, reduced losses, voltage improvements, and prices of power-factor corrective devices) needs to be performed to select the best location for power-factor correction.

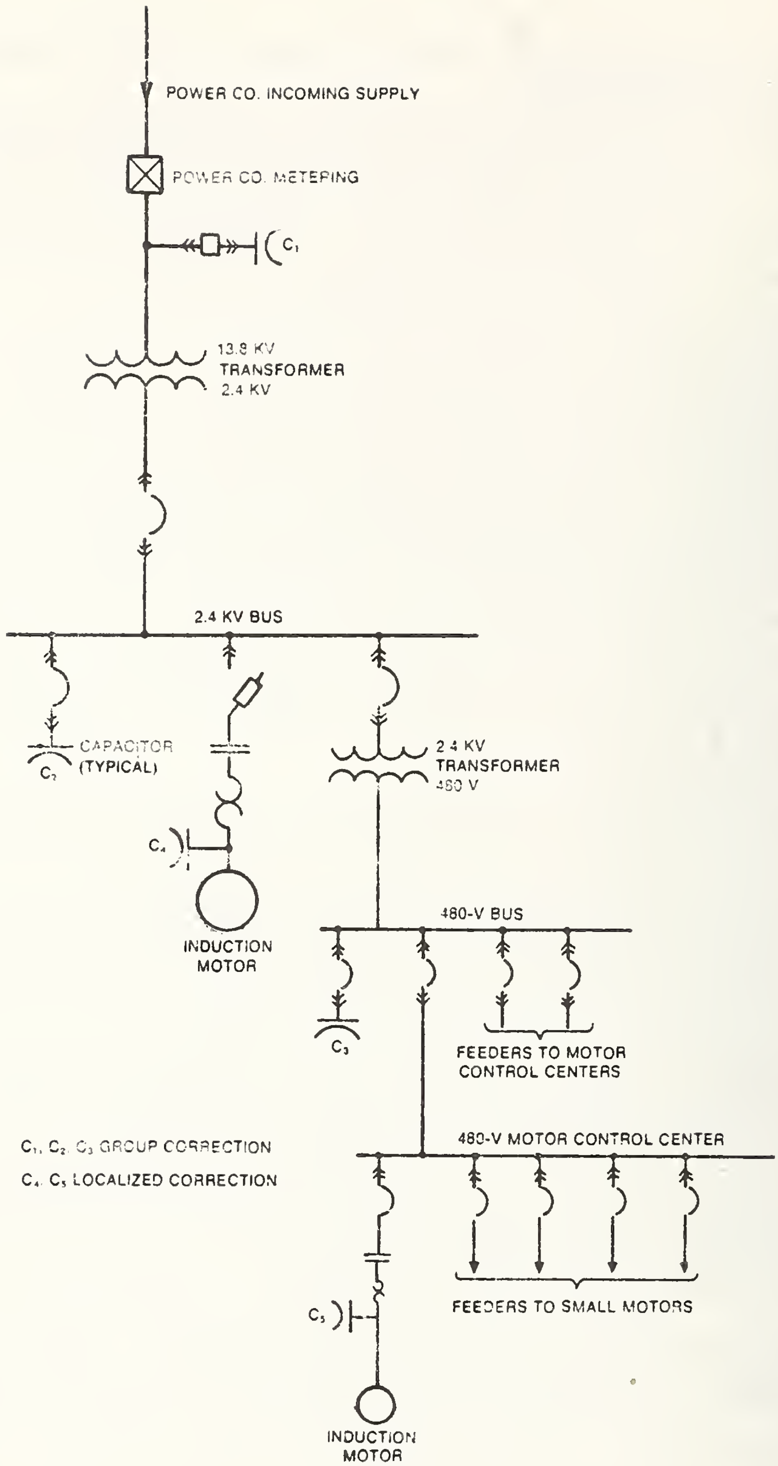

Figure 3-22. Possible locations of power-factor- correction capacitors [13].

3.11 Waste-Heat-Reclamation Systems

Waste-heat reclamation is the recovery or utilization of heat energy that would otherwise be rejected. As such, heat recovery conserves energy, reduces operating costs, and reduces peak loads. This section describes prime targets for waste-heat recovery systems and various waste-heat-recovery devices which are available. Due to the high cost of waste-heat-recovery devices, it is recommended that a feasibility study be first performed to investigate the cost effectiveness of such a modification.

3.11.1 Prime Targets for Waste-Heat Recovery

In examining a characteristic facility for places to use waste heat, there are several prime targets. First consider using hot gases to preheat boiler-makeup water. This modification is almost always cost effective, especially when there is a large continuous flow of makeup water. One reason why this is so attractive is that the engineering needed to retrofit an economizer (feed-water preheater) onto an existing boiler is fairly straightforward. Preheating combustion air is another alternative that can greatly reduce fuel costs, but in this case there is generally a need not only for a great deal of ductwork but also for re-engineering the burners. The cost of such a project is likely to be high.

An often overlooked source of profitable waste-heat utilization in cold or hot areas is the use of heat exchangers to heat or cool the make-up air that is introduced into a building by the mechanical system. This can be accomplished with any of a number of heat exchangers. Waste heat utilization for make-up air has the advantage that neither fluid stream contains chemically reactive components or is at a temperature which is likely to cause material damage.

3.11.2 Waste-Heat Recovery Devices

(1) Rotary Heat Exchanger

The rotary heat exchanger shown in figure 3-23 uses a heat-transfer medium in the form of a cylindrical drum or wheel. When rotated slowly between the supply-air and exhausted-air streams, it absorbs heat from the warmer air stream and rejects heat to the cooler air

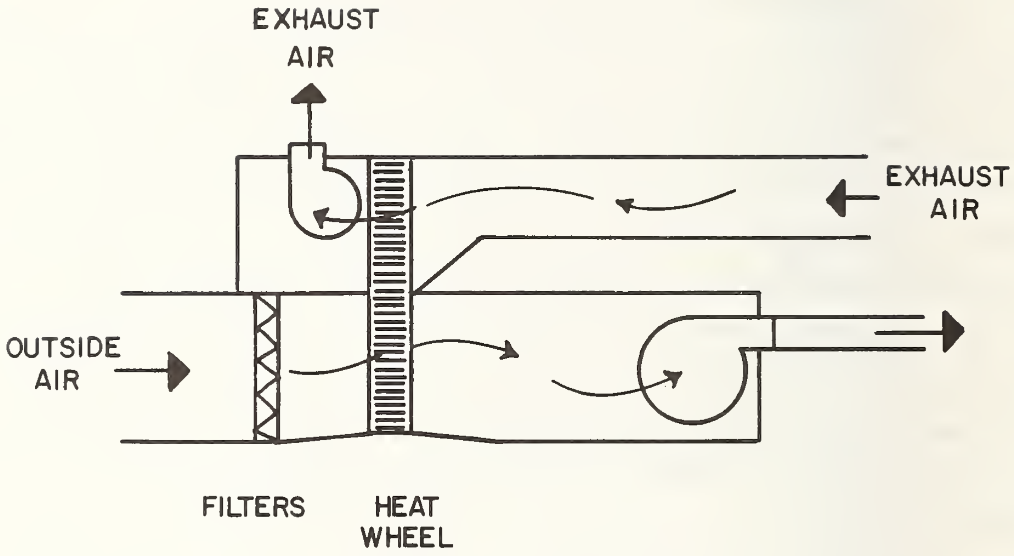

Figure 3-23. Rotary heat exchanger.

stream. In addition to sensible heat transfer, the wheel can be hygroscopically treated with a desiccant for transferring water vapor from the humid air stream to the drier stream. Nonhygroscopic wheels transfer water vapor when the temperature of one air stream is below the dew-point temperature of the other and there is direct condensation of water vapor. During operation at temperatures below 32° F, freezing usually is prevented by preheating the supply air.

Rotary heat exchangers are usually economical for applications having ventilation-air rates of 4000 cfm or more. When operated at approximately 8 to 10 rpm with a face velocity of 550 fpm, a non-hydroscopic rotary exchanger recovers 70 to 80% of the sensible heat and 40 to 60% of the latent heat. Hygroscopic or enthalpy exchangers recover 70 to 80% total heat.

The location of the supply and exhaust ductwork in close proximity to each other can be an obstacle in many applications, especially when exhaust streams are contaminated. In these cases, purge sections must be added to the wheel and special filtration may be necessary in the supply air.

(2) Air-to-Air Heat Exchangers

Air-to-air exchangers transfer heat directly from one air-stream to another through direct contact on either side of a metal heat-transfer surface. This surface may be either a plate with fins (more common for low temperature use in HVAC system) or tube as shown in figure 3-24 (more common for boiler flue-gas heat transfer). Air-to-air heat exchangers may be purchased as packaged units or can be custom made. They transfer sensible heat only and are not designed for cooling applications. Size is limited only by the physical dimensions of the space available.

The efficiency of each installation and cost should be calculated based on the particular circumstances that apply. Although efficiency of air-to-air heat exchangers generally is below 50%, it must be recognized that they are relatively inexpensive, have low resistance to air flow, require no motive power input, and are trouble-free and durable.

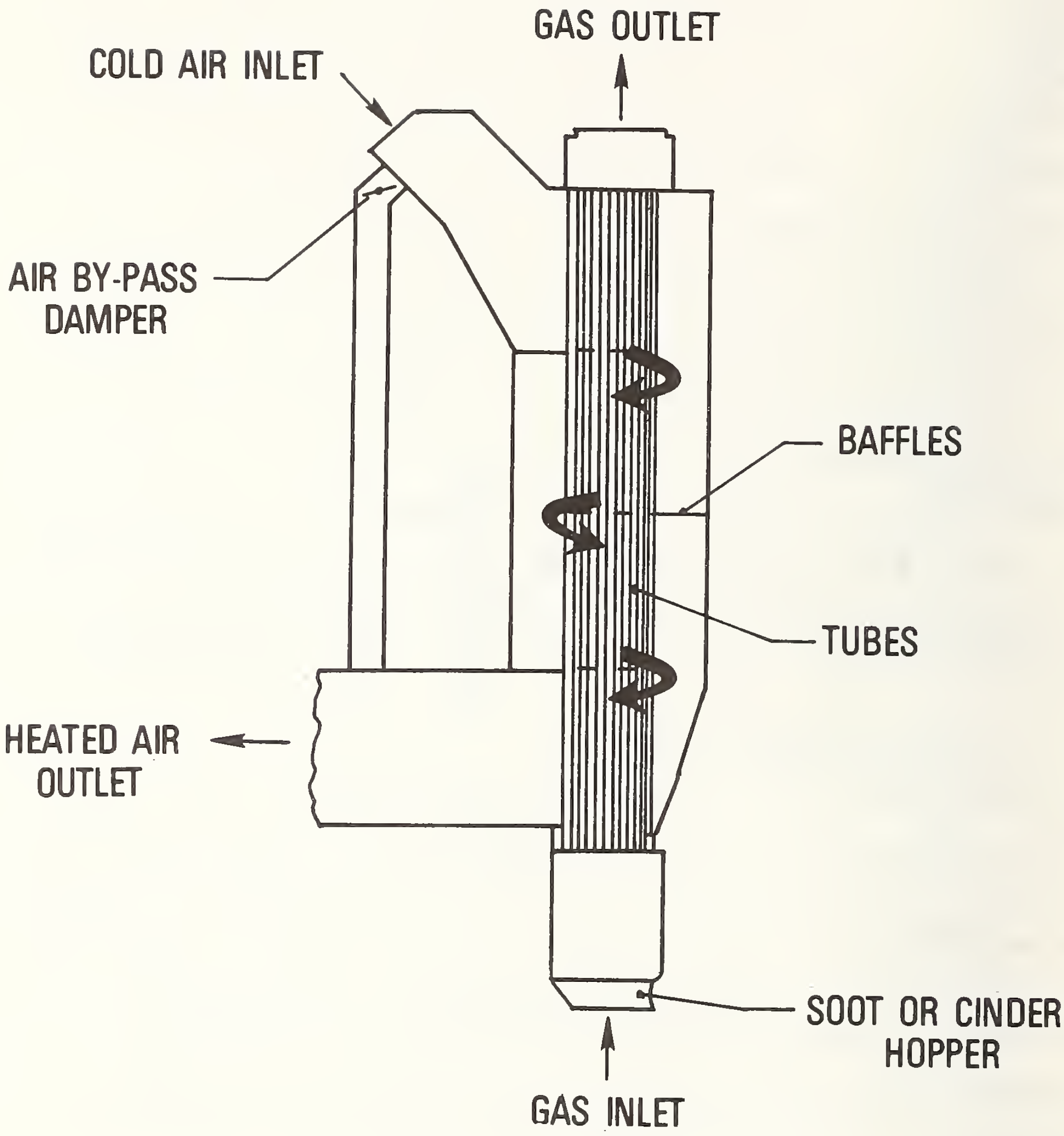

Figure 3-24. Flue-gas heat exchanger.

(3) Run-Around System

A limitation associated with rotary and air-to-air heat exchangers is that the supply and exhaust ductwork must be close together. In some applications this may be physically impossible or very costly. In addition, cross contamination caused by leakage from exhaust to supply streams may be unacceptable when the exhaust contains noxious or poisonous fumes, or if there is a need for cleanliness.

Both of these disadvantages can be overcome by using a run-around system (see figure 3-25) which consists of finned-tube water coils located in the exhaust and supply air streams, and a pump circulating water or a water/antifreeze solution between the coils.

In this form the system is for sensible heat recovery only. It is seasonably reversible, preheating in the winter and precooling in the summer. As with other heat-recovery devices, the coils are subject to corrosion, condensation and the possibility of freeze-up.

Coils and pump are normally selected to achieve sensible recovery efficiencies of 40% to 60%. Greater efficiencies can be achieved by adding more coils in the heat exchangers to increase their capacity. When finned tubing is added, however, the pressure drop across the coil increases and more fan power is required for the supply and exhaust systems. Additional tubing also increases the pumping energy required, so gains in efficiency will be partially offset by increased energy requirements.

If latent recovery is necessary, the system can be modified by replacing the water coils with a cooling-tower surface; an additional solution pump is needed to complete the system. This provides total heat or enthalpy transfer as the solution absorbs heat and water vapor from the air streams, and also acts as an air washer or scrubber.

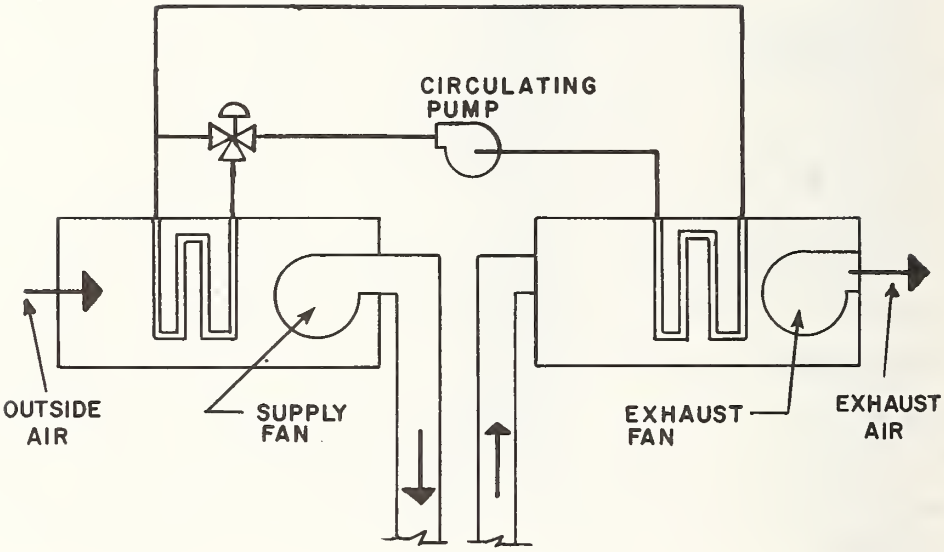

Figure 3-25. Run-around system.

(4) Heat Pipe

A heat pipe is a passive heat exchanger which involves a closed fluid within a sealed tube. A schematic of a heat pipe apparatus is shown in figure 3-26.

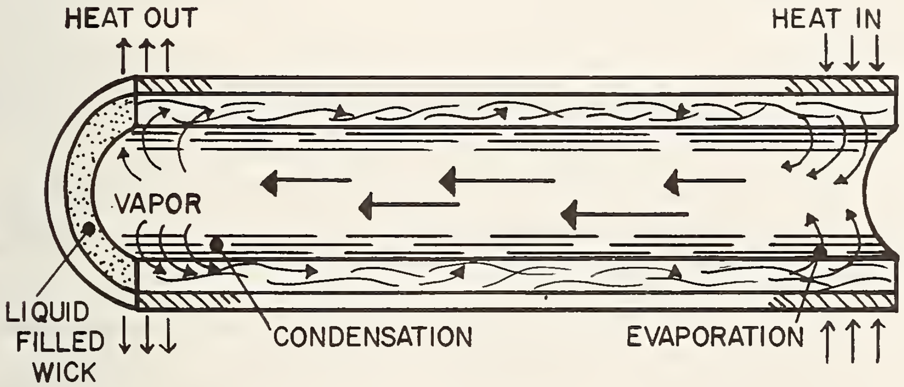

Figure 3-26. Schematic of heat pipe apparatus.

Cool air passing over one end of the tube is heated when it condenses the fluid enclosed in the tube. The warmer air stream passing over the other end of the tube is cooled when it evaporates the fluid contained in the tube. The action is reversible and operates whenever there is a temperature difference between either end of the tube.

To use a heat pipe apparatus for a heat recovery application, air intake and air exhaust ducts for a HVAC system of a building must be located close to each other. Heat recovery is achieved by installing a group of heat pipes between the air intake and the air exhaust ducts.

Any range of supply air cfm can be handled by the addition of more tubes to the heat exchanger rack. In addition, they contain no moving parts so there is minimal leakage between air streams. Mechanical energy is not used except in the form of increased fan energy to overcome static pressure losses.

(5) Shell and Tube Heat Exchangers

Shell and tube heat exchangers can be used to exchange heat in the following configurations:

- liquid-to-liquid
- steam-to-liquid, and
- gas-to-liquid

All three configurations are commercially available in a wide range of sizes and outputs and with reliable heat exchange data.

Particularly favorable applications are to capture energy from hot condensate, hot refrigerant gas, condenser water, and hot drain lines from kitchens and laundries.

The heat exchanger should be insulated to prevent unnecessary heat loss and should be constructed of materials to suit the application.

(6) Waste-Heat Boilers

Waste-heat boilers are used in conjunction with high-temperature heat recovery as from engine exhaust and exhaust from gas turbines used to drive refrigeration equipment or electric generators. In the case of the turbine, a duct, similar to a breeching, is connected directly from the exhaust of the turbine to the boiler. After passing through the boiler the gases are discharged to the atmosphere. Gas turbines produce about twice as much recoverable heat as reciprocating engines of the same size. Generally, a turbine yields 7 to 13 pounds of steam per hour (15 psi steam) per kilowatt generated. An overall system efficiency of slightly over 65% is possible in total energy plants,* if all the recoverable heat is used.

* A total energy plant is a plant which utilizes recovered heat from electrical power generators to provide a portion of the energy requirement for space heating, domestic hot water, and space cooling.

(7) Double Bundle Condenser System

A double-bundle condenser is constructed with two entirely separate water circuits enclosed in the same shell (see figure 3-27). Hot-refrigerant gas from the compressor is discharged into the condenser shell where its heat is absorbed by either one of the water circuits or by both simultaneously depending on the requirements of the system at a given time.

One of the circuits is the building-water circuit and the other, the cooling-tower circuit. The reason for splitting the condenser into two independent hydronic circuits is to prevent contamination of the building-water system with cooling-tower water, which may contain dirt and undesirable chemicals. When a double-bundle condenser is used in conjunction with a standard refrigeration machine, the heat rejected by the compressor is made available to the building-water circuit.

In certain heat recovery application the amount of heat that can be reclaimed during occupied hour may exceed the daytime heating requirements of perimeter zones. This excess heat can be stored for release during times when the building is unoccupied.

The cost of installing a double-bundle condenser system is such that, as a replacement, it probably will be most feasible for a system which already is at or neat the end of its useful life.

For more extensive discussions of these topics, and for a review of the economic basis for decision making in waste-heat recovery, the reader is referred to the National Bureau of Standards Handbook 122, "The Waste Heat Management Guidebook" [14].

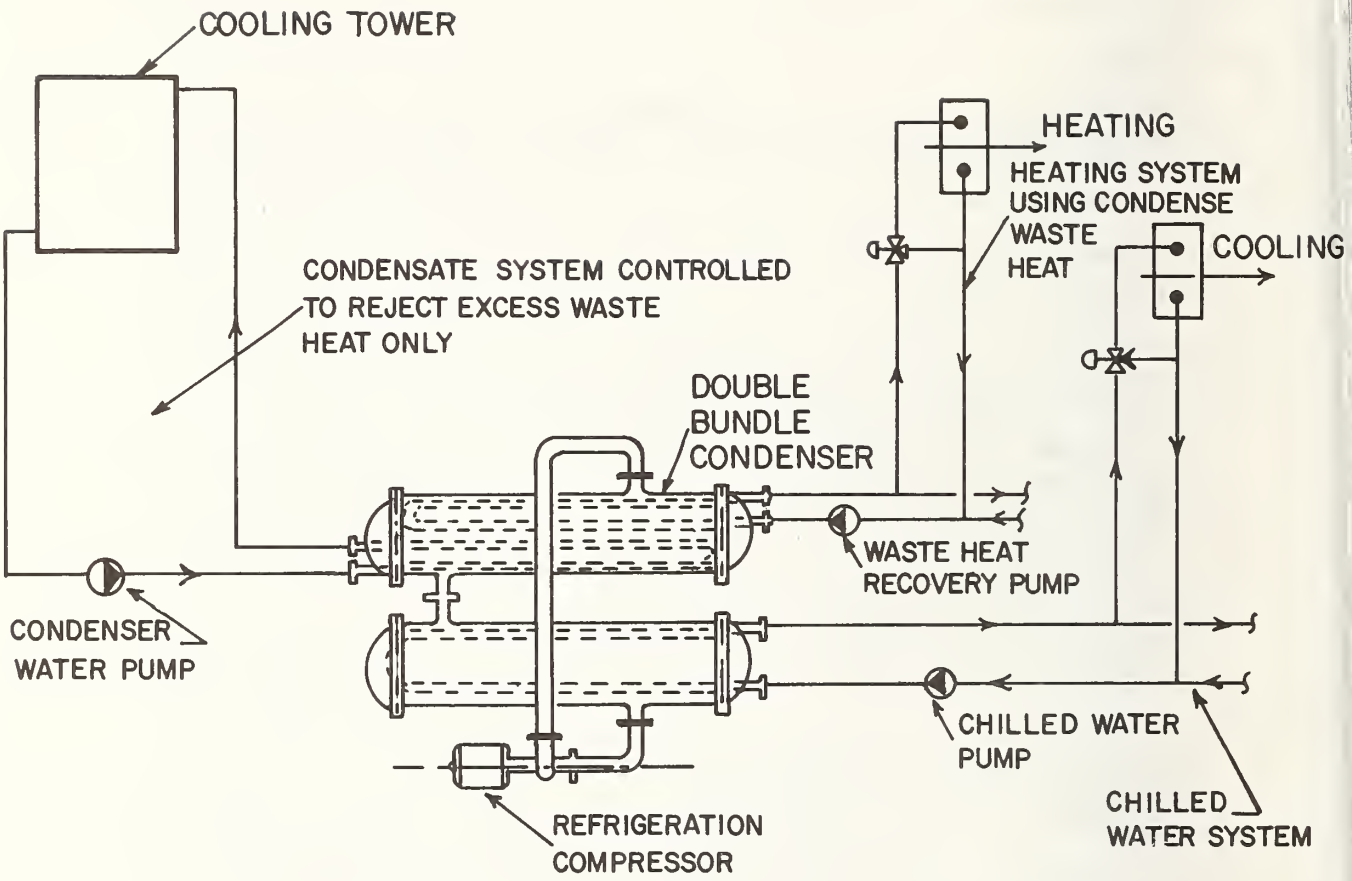

Figure 3-27. Waste-heat recovery from central chiller using double-bundle condenser.

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CHAPTER 4. CONDUCTING THE BUILDING SURVEY

CONTENTS

CHAPTER 4. CONDUCTING THE BUILDING SURVEY

4.1 Introduction

The building survey is a physical reconnaissance of a building for the purpose of identifying defects in the building and non-energy-conserving practices which when corrected or modified will bring about cost-effective reductions in energy requirements. The building survey must be thorough enough to accurately assess the major factors which contribute to the consumption of energy in the building. To assist the Base Civil Engineer in carrying-out the building survey, a data sheet is given in the appendix at the end of this chapter.

After a Base Civil Engineer has surveyed the buildings on his Base, he should be in a position to identify a vast number of energy conservation measures for the various buildings. An energy and economic analysis of all the reported technical options for energy conservation would be very tedious and time consuming. To assist the Base Civil Engineer in quickly identifying those areas having the greatest impact on or potential for energy conservation, some major target areas* are outlined in section 4-2.

Another important aspect of the building survey is the identification of condensation problems in roofs, walls and floors. For example, accumulation of moisture within the sidewalls of buildings often leads to paint failure; excessive moisture accumulation in some instances may contribute to rotting of wood siding and in masonry construction cause mortar to loosen or degrade caulking and weatherseals. Accumulated moisture may also wet insulation and reduce its insulating properties. In addition, the accumulation of moisture in wood siding may cause buckling and warping of wood siding which in turn may create cracks in the construction through which air infiltration may occur. Before additional insulation is installed, it is imperative that potentially serious

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*Note: The Air Force has several high energy-consuming testing complexes which consume excessive amounts of energy. A discussion of energy conservation measures for such facilities is beyond the scope of this report.

condensation conditions such as improper attic ventilation be identified and corrected, since more insulation may aggravate the problem. The identification of condensation is covered in section 4-3.

4.2 Prime Targets

Viewed from either within or without, prime targets are:

1. Buildings wasting energy by heat and cooling simultaneously.
2. Buildings with structural defects.
3. Facilities used for purposes other than originally designed.

The three targets are discussed in the following sections.

4.2.1 Buildings Wasting Energy by Heating and Cooling Simultaneously

Energy wasted under this condition is caused by excessive heating or cooling due to improperly regulated heating, ventilating, and air conditioning (HVAC) equipment. Unusual occupant reactions are a clue to this problem. The normal reaction of a human being exposed to uncomfortable temperature extremes is to "adjust" the physical conditions or move to a more hospitable environment. Windows and doors deliberately opened to the outside when they should be normally closed are an obvious clue to an improperly regulated heating or cooling system. There are other defenses erected by building occupants such as blocking ventilator openings, by-passing controls, etc. These energy-wasting conditions may exist in any type of Air Force building and may be confined to one or two rooms, or they may affect the complete building. Occupant reaction varies, but if windows are opened in an attempt to regulate the heating or cooling system, the result is usually wasted energy and an unbalanced HVAC system. Ideally, the best time to identify the foregoing is while temperatures are high or low. Identifying such conditions is easy; finding the cause may not be so easy. A brief check list of probable causes includes:

A. Malfunctioning temperature or regulator controls.

B. Adjustment of temperature or regulator controls by unauthorized personnel.

C. Controls improperly influenced by the environment.

The cause may be direct radiation from heat sources including the sun, excessive air movement over the thermostat or behind the wall in or on which the thermostat is mounted.

D. Malfunctioning heating system.

The design of a heating system may be the cause of overheating or underheating. Such a condition may be found in buildings altered from their original function. It may occur in a system using chilled water coils over which preheated air is passed. Control malfunction or modification of the original design may result in a preheat cycle not matched to the load requirements.

E. Fresh air intake dampers.

Motorized controls used to regulate fresh-air intake dampers should be periodically checked for proper operation. Dampers should normally return to a closed position when fresh air is not required. In certain buildings no fresh air is required for unoccupied periods. Prior to the energy shortage, many systems were designed with first costs in mind. As a result, energy savings controls were frequently deleted from the design, resulting in the loss of conditioned air due to improperly regulated dampers. There are control systems available which will remedy this problem. They will return their initial costs in energy savings within a few years. Control specialists are available who can provide guidance in the selection of control system packages.

F. Building exhausting conditioned indoor air to the atmosphere.

Obvious vapor plumes caused by indoor air exhausted into cold atmospheres may be visible evidence of energy waste. Normally, however, air exhausted from a building often is not readily detectable. The volume of indoor air exhausted to the atmosphere depends on the building function. Dining halls require almost constant ventilation, whereas electrical maintenance shops do not. Therefore, each type building must be considered separately. The effect of exhausting large amounts of air from a building is to create a negative pressure and to substantially increase the infiltration of outside air, thereby substantially increasing the energy requirement for space heating and space cooling. The air pressure difference in extreme cases may make opening and closing of entry doors difficult and dangerous. A pronounced inside-to-outside pressure difference often creates a moaning or whistling sound around doors and windows as the air is pulled inward. This sound is a clue to investigate further. The effect on the heating and cooling system is to create an unbalanced condition causing erratic temperature control. Under extreme conditions, the infiltrating air (especially if it is cold) will cause a variety of occupant reactions. These vary from erecting artificial wind barriers to blocking distribution outlets and returns. Controls affected by infiltrating air are often the cause of overheating or overcooling.

G. <u>Freight doors remaining open during loading and unloading operations</u>.

It is obvious that a great deal of energy is wasted by attempting to provide space heating for a warehouse when one side of it may be literally open to the outdoors during loading or unloading operations. There are, however, certain corrective alternatives available such as the use of expanded rubber dock-doors that form an air seal to the back entrance of large trailer trucks.

Several side effects show up as a result of the freight doors remaining open. Not only is indoor air flushed from the bay which is served by the open door, but it may also be flushed from other areas under certain conditions as shown in figure 4-1. Due to the extraordinary volume of air exchanged under such circumstances, dirt-laden air is circulated and deposited on heating system components. This tends to further aggravate energy waste by reducing the operating efficiency of the heating and cooling equipment.

H. <u>Hangars</u>

The heating or cooling of a whole hangar to comfort temperature levels may be inconsistent with the Base energy conservation programs. There may be circumstances, however, when it is required. Normally it is not economical to maintain comfort levels in high-bay structures with loose fitting doors typical of aircraft hangars. Hangars are obvious energy consumers. Recommended procedures are to maintain temperature levels in the main hangar space at 55° F or lower and office space at 68° F during occupied periods in the winter.

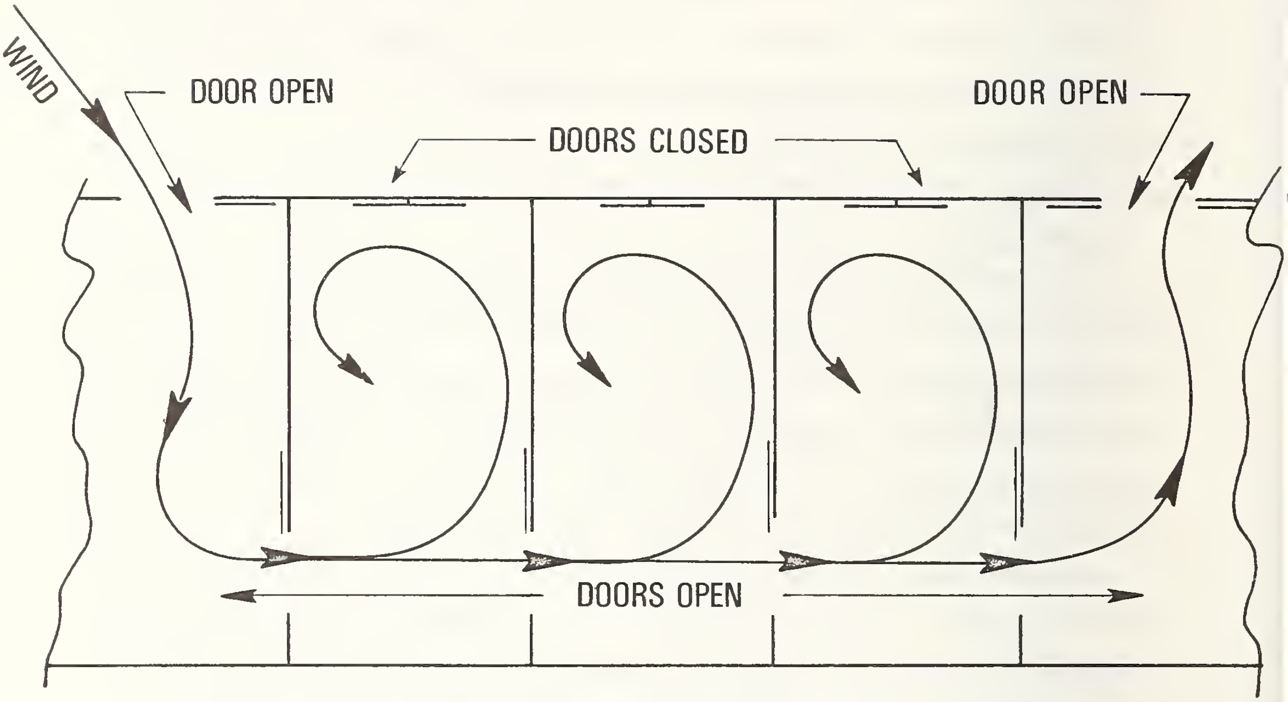

Figure 4-1. Effect of multiple open doors on heated air in adjoining bays of a warehouse.

4.2.2 Building with Structural Defects

Structural defects and the accumulation of many smaller cracks may cause a high rate of air infiltration which in turn contributes significantly to the energy requirement for space heating and space cooling. There is the obvious hole in the wall and the not-so-obvious hole. A gaping 6-sq ft hole is more readily identified and repaired than are a series of cracks through which air leaks, even though the sum of these cracks may be equal to a 12-sq ft hole. These "little openings" tend to be overlooked because they are cracks; however, each one is a defect and will very likely get worse, not better, leading to further deterioration. The greater the difference between inside and outside temperatures, the more likely that small fissures will become big problems, especially in a zone in which many freeze/thaw cycles occur during the winter. Keep in mind that air infiltration into heated spaces will contribute much more to the energy requirement for space heating and space cooling than air infiltration into unheated spaces such as crawl spaces. Structural defects resulting in significant heat loss due to air infiltration are described below:

4.2.2.1 Foundations and Basements

There is often a crack between the sill plate and the top of the foundation wall of wood-frame buildings. It may not be readily seen, but the moving air can often be detected by holding your hand in the suspected area. Smoke from a cigar or cigarette may also be used to identify air infiltration. Trace all water, gas, electrical conduit, and communication equipment wiring to the respective entry points and check for gasketing or properly installed caulking around the penetration. Other sources of infiltration are windows, coal access doors, outside entries to the basement area, and underground trenches carrying steam lines or communication wiring.

A check also should be made for old furnace flues now unused because of replacement of the heating plant with a non-combustion type. These flues should be capped at both ends. The effect of air passing over an open chimney creates a negative pressure at the base of the flue and induces air infiltration.

If the basement is under only part of the building, look for access holes to other unheated regions such as crawl spaces. These regions may or may not be vented. Some typical sources of heat loss in basements are depicted in figure 4-2.

4.2.2.2 Slabs on Grade

Air leaks occur where the wall has parted from the slab or where there is a penetration through the slab. These openings may not be visually detected from either the inside or the outside unless smoke is used, as outlined in the section under basement air leaks. Telltale signs of air leakage are inside traces of moisture at the wall/floor interface. Also look for water stain marks on basement trim or on the walls. There may be evidence of water damage to flooring materials at the perimeter edge which is also a likely indicator of an air leak. Caution: The lack of or defective perimeter slab insulation may also be the cause for condensation or frost, and result in water damage along slab perimeters. Also, under certain adverse conditions, hydraulic water pressure may force water up through small hair-line cracks in the slab, with obvious results.

4.2.2.3 Unheated Crawl Spaces

Crawl spaces may be found under bachelor quarters, dormitories, or family housing units. Access is usually difficult and detailed inspection is a less-than-pleasant chore. The crawl space is frequently one of the major contributors to, or direct causes of, other structural or operational problems. In freezing climates, crawl spaces should be checked for the following:

- All plumbing, including traps on sanitary sewer lines should be insulated to Air Force specifications for the area.
- Air infiltration may be occurring around plumbing pipes, especially to bath tubs. Check the plumbing access panel behind the tub to verify this condition.
- Air may also be infiltrating around chimney flues which have their foundation beginnings in the crawl space.

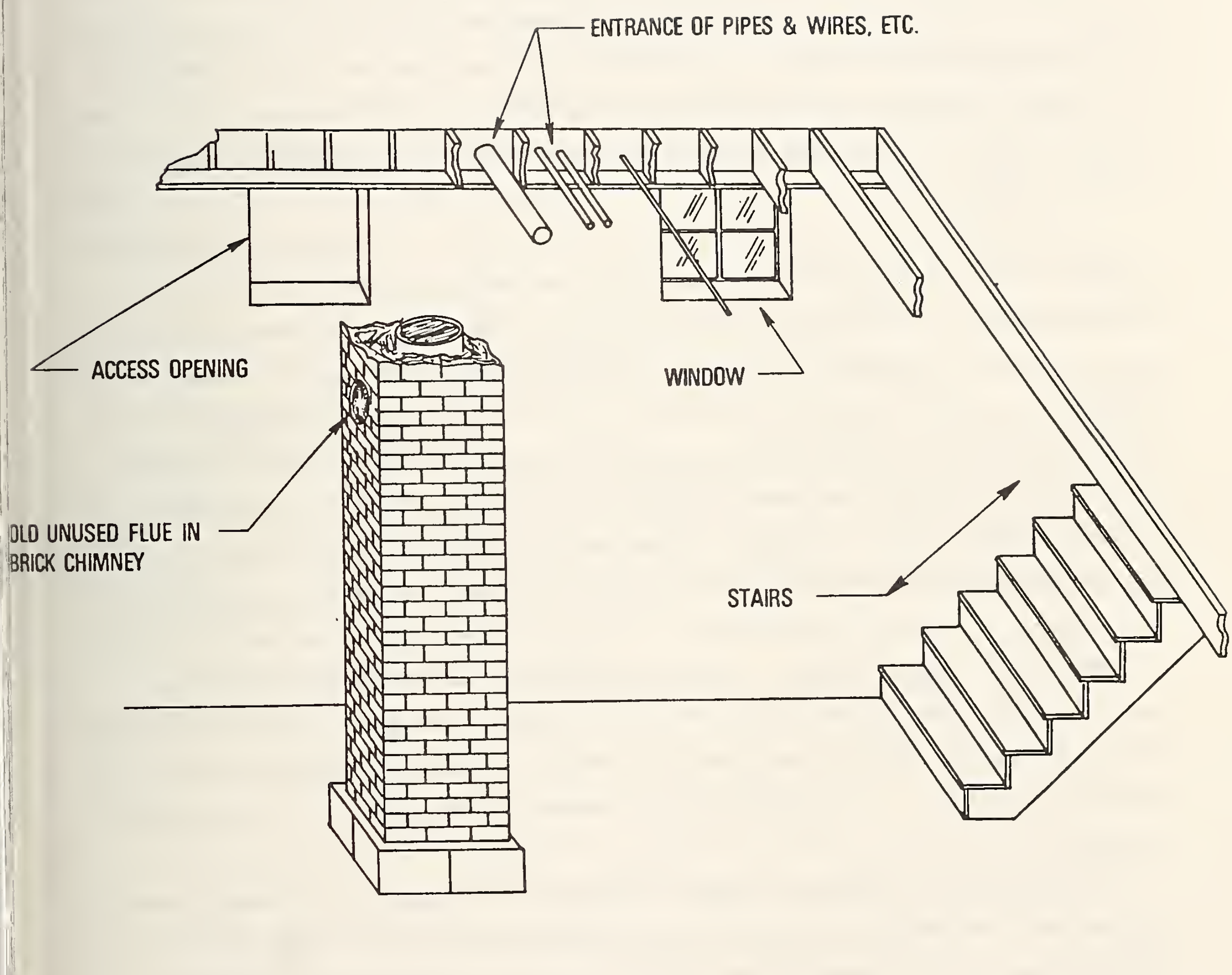

Figure 4-2. Typical sources of air infiltration in a basement.

° Holes for electric wires or conduit resulting in drafts in interior walls may be detected by feeling around interior wall outlets or switch plates. The worst draft conditions (potential freeze-up) will usually be found where oversized openings have been made to accommodate plumbing and heating pipe runs.

4.2.2.4 Structures Supported on Blocks/Posts

Buildings supported on blocks or posts are either temporary or located in environments where this is a practical building technique due to the environment. Since the climates permitting this type of structure are mild to tropical, the primary need is for air conditioning. The comments under crawl space are applicable to this type construction.

4.2.2.5 Walls

The search for telltale signs on walls should be conducted on the exterior first and interior last. Walk completely around the building noting its orientation to the points of the compass. Weathering of various exposed sides of buildings will be repeated on wall surfaces exposed to similar weather conditions, i.e. south and west exposures have more surface weathering, since they face prevailing winds, rain and sun. The north side may show indications of mold, mildew or moss due to less intense solar exposure. Some geographic areas may produce unique weather characteristics on building surfaces due to exposure extremes such as desert sand blasts, salt air, or artic temperature.

Note structural faults, such as broken or cracked foundations and walls. These faults may be the result of neglect or poor maintenance. They may also indicate a more serious problem behind the facade of the building envelope. It is important to ascertain the cause of structural faults.

A crack in a large plate glass window, dark stains on siding, calcification of masonry surfaces, and paint failure may be signs of serious problems. For instance, did the plate glass crack due to physical impact, thermal shock, or excessive wind load; or is the foundation settling and distorting the window frame? Are the dark stains dirt or mildew, or are they caused by water leaks which left a residue of foreign substances? Are white stains on masonry surfaces (calcification) caused by a water leak on the back side of the facade or because moisture migration forced salts to the surface? Is paint failure due to poor painting practices, or is it caused by moisture migration? Is rust streaking the result of fastener heads oxidizing or is there a leak at the joint where the fastener has worked loose?

The observer will certainly note the Air Force requirements for double glazing, window and door weatherstripping, and the need to caulk minor cracks, wherever they appear.

Accumulated ice dams on roof edges, valleys, gutters, and downspouts will force water into building sidewalls, roofs and ceilings. External damage from these conditions is usually evidenced by water marks, severely weathered surfaces, distorted flashing and gutters, and curled shingles.

Air conditioning equipment should be checked for unusual noise and abnormal accumulation of debris inside equipment cabinets. If winterizing covers are needed, they should fit snugly to the unit.

4.2.2.6 Ceiling/Roofs

An important aspect of the inspection of the building interior is the space between the finished ceiling and the roof. It may be six inches or six feet in height, but without exception it must incorporate two components. They are insulation and adequate ventilation. Check AFM 88-15 for specific requirements.

A visual inspection may be difficult due to space restrictions, but look for evidence of unvented moisture collecting on the underside of the roof sheathing or decking. Water stains, rusty fasteners, mildew, and wood rot are all signs of high moisture levels.

Wherever there is a penetration through the roof, look for evidence of water leaks. Rain water that leaks through a defective roof and wets ceiling insulation may substantially increase heat transmission through the ceiling. In geographical areas of 7000 degree days or greater, snow loads on roofs can exceed the rating of structural support. Broken or distorted support members are readily identified.

The roof is often the "forgotten" member of the building. "Out of sight, out of mind" is a trite but apt expression epitomizing the attention normally given to this important component. The "if it doesn't leak, it must be OK" theory may be reassuring, but it is no substitute for knowing. . . especially if a leak-forming condition is in the early stage. For shingle roofs, check the ridge lines for sagging and look for curled shingles (evidence of moisture in roof sheathing), torn or missing shingles, and loose flashing.

For built-up roofing systems, moisture accumulation in the insulation may substantially increase heat transmission through the roof. Signs to look for which indicate a possible moisture problem include standing water, plugged storm drains, imperfectly sealed expansion joints and imperfectly sealed flashing at roof edges, parapet walls and roof penetrations. Not easily detected are small bubbles, wrinkles, or heaving which may be caused by conditions of moisture, temperature, stress, or other structural faults already mentioned.

4.2.3 Facilities Used for Purposes Other Than Originally Designed

When buildings are used for purposes other than originally designed, often modifications may have been performed without proper consideration for energy conservation. For instance, office spaces in hangars may be installed without wall and ceiling insulation. In such instances, when the interior temperature within the hangar is maintained at a low level and the office spaces are heated, excessive heat loss may occur. Other typical situations in which buildings are used for purposes other than originally designed include:

A. Dormitories converted to offices

B. Warehouses converted to offices

C. Sheds converted to service shops

D. Basement rooms converted to day rooms.

There are other conversions not mentioned, but the important question is, does the conversion operate satisfactorily? Was the reason for the conversion to meet a temporary need or was there a long range plan?

If the installation has become "permanent," examine it for structural defects as previously outlined.

4.3 Identifying Condensation Problems

The identification of moisture problems is an important aspect of the building survey. Moisture accumulation in wood siding may lead to paint failure, as indicated in figure 4-3. Excessive moisture accumulation may actually lead to wood rot which may substantially shorten the life of a building. Moisture may also accumulate in the sidewall insulation material, thereby reducing its insulating properties. In addition, the accumulation of moisture in wood siding may lead to warped and bowed boards which in turn may create cracks in the construction through which air infiltration can occur.

Paint failure may also be caused by poor surface preparation and poor paint systems. It is necessary to look for other symptoms before diagnosing a moisture problem. In some frame construction, a close examination of a newly painted surface may reveal water blisters, as shown in figure 4-4. Bowed and warped boards may also be observed, as shown in figure 4-5. If paint peeling is observed without the presence of any of the foregoing symptoms, there is a good chance the paint peeling is not moisture related. Moisture condensation within building surfaces is usually not seen. It becomes obvious when sufficient moisture has accumulated to cause discoloration, rusting, or other water damage to building components. Excessive wetting of components such as insulation, structural framing, or masonry cause these components to function in a changed design mode. They become a means of conducting heat. They change structurally, assume new forms, emit odors, and may sustain permanent damage.

Winter condensation problems in attics and sidewalls tend to become more severe as one proceeds north. Generally, winter condensation problems are considered to be significant at those locations for which the January average temperature is 35° F or lower. A map showing the condensation zone of the United States is given in figure 4-6.

Whenever condensate is present, it is most important to identify the cause and take appropriate corrective action.

Figure 4-3. An example of paint failure for which moisture may be a contributing factor.

Figure 4-4. Blistering of paint due to the outward movement of water vapor.

Figure 4-5. Bowed and warped boards which may create cracks in the construction through which air infiltration can occur.

Figure 4-6. Winter condensation problems occur where the average temperature for January is 35° F or lower (taken from "Condensation Problems: Their Prevention and Solution," by L. A. Anderson, U.S.D.A. Forest Service Research Paper FPL 132, Forest Products Laboratory, Madison, Wicsonsin, 1972).

The two principal causes of excessive sidewall condensation in most buildings are the lack of any or an improperly installed vapor barrier, or excessive indoor humidification. During the building survey, the presence of indoor humidifiers should be noted. A good indication of excessive humidification is water marks on the inside wall surface under the windows. These marks are due to water that has condensed on the window panes during the winter season. If signs of moisture problems are observed during the building survey, an inspection of the affected area should be made to determine the cause.

Installing a vapor barrier in an existing building is not feasible unless major renovation is under way. If such is the case, the best barrier or membrane is one of 2.0 to 4.0 mil polyethylene film. Installation should be done with great care to avoid any cracks or openings through or around the film. The air inside a building in the winter contains a large amount of water vapor compared to the outdoor air and tends to seek minute openings and pass through them.

If the above technique is not practical, consider the use of vapor-impermeable paints or finishes such as waterproof paints, spar varnishes, and aluminum paint. These products are applied only to interior surfaces of walls exhibiting moisture problems and then covered with a cosmetic finish.

If none of the above is practical then consider the use of an exhaust fan regulated by a humidistat. Controlling the indoor relative humidity of a building will effectively reduce the movement of water vapor through the building envelope.

It is essential to the energy conservation program that buildings do not accumulate excessive amounts of moisture. The accumulation of moisture in wall insulation may increase heat transmission through the wall. In addition, moisture accumulation in wood siding may cause warping and bowing of the siding which in turn may create cracks in the construction through which air infiltration can occur.

During the building survey, it is important to note the status of attic ventilation openings. In the past the attic may have been closed off to prevent freezing of pipes or for other reasons. The addition of insulation to a completely closed-off attic will often lead to serious condensation problems. The increased amount of ceiling insulation substantially reduces attic temperatures and thereby significantly increases the likelihood for water vapor to condense on cold surfaces. It is extremely important that closed ventilation openings be reopened before more insulation is installed in an attic.

In an obvious effort to improve thermal performance of roof decks, it is customary to add additional insulation. However, before any insulation is installed underneath or on top of an existing roof deck, expert advice should be sought. Adding insulation without full knowledge of the present roof condition, moisture membrane, potential temperature difference and shifting of dew point, as well as roof loading factors, may negate any energy savings promised by the addition of more insulation.

APPENDIX

DATA SHEET FOR THE BUILDING SURVEY

EXTERIOR Form #

Bldg. Type & Location	Survey Date	Survey By	Weather Condition Temp.____R.H.____	Action Required	Reviewed By

Structural Problems	Foundation					Wall					Windows & Doors					Roofs	
	N	S	E	W		N	S	E	W		N	S	E	W		#1 ELEV.	#2 ELEV.
Cracks																	
Spalling																	
Water Leaks																	
Efflorescence																	
Caulking																	
Drainage - Defective Operation																	
Equip. Mounted to Bldg.- Loose																	
Mildew																	
Mold																	
Paint Peeling																	
Flashing																	

EXTERIOR CONTINUED

Form #

Structural Problems	Foundation					Wall					Windows & Doors					Roofs	
	N	S	E	W		N	S	E	W		N	S	E	W		#1 ELEV	#2 ELEV
Gutters																	
Downspouts																	
Parapet Walls																	
Ventilators																	
Double Glazing																	

Remarks:

MECHANICAL

Form #

Bldg. Type & Location	Survey Date	Survey By	Weather Condition Temp.____ R.H.____	Action Required	Reviewed By

Check Mechanical Components	Location			
Filters				
Unusual Operating Noise				
Unsafe or Hazardous Condition				
Missing Insulation on Pipes, Ducts, Tanks, etc.				
Obvious Alterations to Mechanical Components				
Motors				
Condensing Coils				
Heating Coils				
Electrical Controls				
Pneumatic Controls				

MECHANICAL CONTINUED

Form #

Check Mechanical Components	Location			
Drive Mechanisms				
Ducts				
Dampers				
Combustion Air Intake				
Exhaust Air				
Remarks:				

INTERIOR

Basement & Slab

Form #

Bldg. Type & Location	Survey Date	Survey By	Comfort Temperature Maintained	Action Required	Reviewed By
			Yes______ No______ Note:		

Structural Problems	Wall N	Wall S	Wall E	Wall W	Floor
Efflourescense					
Cracks					
Water Leaks					
Severe Drafts					
Mildew					

Remarks:

INTERIOR CONTINUED — Crawl Space & Under Floor Space — Form #

Standing Water	
Vapor Barrier	
Ventilation	
Exposed Plumbing	

Remarks:

INTERIOR CONTINUED — Rooms: Walls - Floor - Ceilings - Windows — Form #

Room # or Name	Excessive Drafts	Structural Cracks	Water Stains	Mildew	Dirt Streaks	Condensation	Frost	Fireplaces	Operating Controls	Delaminated Trim, Plaster & Paneling	Other		

Remarks:

INTERIOR CONTINUED

Attics & Under Roof Cavities

Form #

Frost	
Mildew	
Water Stains & Leaks	
Ventilation	
Missing Insulation	
Access Openings - Not Protected	

Remarks:

CHAPTER 5. MEASUREMENTS FOR IDENTIFYING ENERGY CONSERVATION POTENTIALS

CONTENTS

CHAPTER 5. MEASUREMENTS FOR IDENTIFYING ENERGY CONSERVATION POTENTIALS

5.1 Introduction

This chapter presents several simple measurement techniques for assessing the thermal transmission characteristics of exterior building envelopes; the thermal performance of HVAC and hot water heating equipment; and the operating efficiency of central hot water, steam, and chilled water generating plants--for the purpose of identifying and evaluating their respective energy conservation potentials.

5.2 Assessing the Thermal Transmission Characteristics

5.2.1 Infrared Thermography

Infrared (IR) thermography is a powerful technique for identifying potential areas for energy conservation. In this technique, an IR television camera is used to produce a thermal picture (or temperature profile map) of a surface or region under study. The thermal picture is displayed on a television screen which may be photographed for record and documentation. Regions of high heat loss and high energy waste stand out in marked contrast with respect to other parts of the surface being viewed and therefore may be readily identified. Once suspected regions of energy waste have been identified, a physical reconnaissance of these regions is necessary to verify the extent of energy waste and to develop recommendations for corrective actions to reduce energy consumption at those points.

Infrared thermography is based on the principle that all objects radiate thermal energy (E) according to the relation:

$$E = e \cdot \sigma \cdot T^4 \qquad (1)$$

where e = emissivity of the surface

σ = Stefan-Boltzmann constant

T = absolute temperature

An IR television camera senses the thermal radiation emitted from a surface. When the emissivity of the surface is known, the thermal picture produced by an IR television camera may be converted into a temperature-profile map of the surface being viewed. Regions of high heat loss on buildings are comparatively warmer than other parts of the buildings when viewed from the exterior and will therefore stand out in contrast with respect to other regions.

An important point is that regions may appear different in the thermal picture due to differences in the emissivities of the surfaces being viewed. It is fortunate that most building materials and paints (with the exception of metallic paints) have emissivities which range between 0.85 and 1.0. Differences due to emissivities over this range will have a small effect on the thermal picture produced by the IR television camera. However, one must become concerned with emissivity effects when unpainted metallic objects are viewed by an IR television camera.

5.2.1.1 The Infrared-Television System

Generally, an IR television system consists of an IR television camera, a black-and-white display console, and possibly a color-display console. These components can be housed as separate pieces of equipment or combined into a single module.

The IR television camera works very much like a studio television camera. A surface is scanned by the IR camera by a series of horizontal trace lines. A video signal is produced by the camera. At any instant, the intensity of this video signal is a function of the thermal radiation entering the IR camera. The video signal from the IR camera is processed in the black-and-white display console where it is converted into a thermal picture in which the gray tones in the picture approximately correspond to surface temperature.

If the system has a color monitor, the color monitor will assign separate colors to various levels of the video signal. The resulting picture displayed on the color television screen is a thermal picture in which the temperature range has been subdivided into various regions, each coded with a different color. The advantage of a system having a color monitor is that an abrupt change in contrasts occurs upon passing from one region to another of slightly higher or lower temperature.

In terms of 1976 prices, IR television systems range from $30,000 to $60,000. Many companies also rent equipment. There are a large number of consulting firms which could perform infrared surveys of Air Force facilities.

5.2.1.2 Applications

An aerial IR survey of the facilities of an Air Force Base will be useful in :

- identifying defective pipe insulation and steam leaks in underground heat-distribution systems
- locating regions of built-up roofing systems having wet insulation.

These applications are discussed below.

A National Bureau of Standards aerial survey of the underground heat-distribution system at Arnold Air Force Base showed that the earth above pipes with good insulation buried 6 to 10 feet could not be distinguished in the thermal picture. On the other hand, the earth above pipes with defective insulation stood out in marked contrast with respect to the ground surface 15 to 20 feet away from the pipe.

In the case of built-up roofing systems, a major problem is leaks which develop in the top membrane that permit rain water to enter the roofing system. Since the bulk of the top membrane remains impervious to water vapor, the escape of moisture is a very slow process. The build-up of moisture in many instances wets the roof insulation and significantly reduces its insulating capability, thereby permitting high heat loss through those regions having wet insulation. Infrared thermography has been shown to be effective in identifying regions of a built-up roofing system that have wet insulation [1]*. Once such regions are identified, repairs can be performed selectively on those regions having defective membranes and wet insulation underneath. This

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* Numbers in brackets indicate references at the end of chapter. This report also discusses the use of nuclear moisture meters.

is significantly less costly than installing a whole new roof system on a large building.

Another application of IR thermography is the location of major heat leaks in buildings. Figure 5-1 shows a conventional photograph in the visible spectrum of a test house at the National Bureau of Standards. The same view of the test house is shown in the IR spectrum in figure 5-2. Note that the major heat leaks (the windows, chimney, and louvered ventilation opening) stand out in marked contrast with respect to other regions displayed in the thermal picture. Further information on the use of IR thermography to identify heat leaks in buildings may be found in references [2-4].

Still another area of application for IR thermography is the survey of utility services devices such as:

- defective electrical connectors
- defective steam traps
- aboveground pipe insulation.

As an example, figure 5-3 shows a photograph in the visible spectrum of electrical connectors on a transformer. The same group of electrical connectors is shown in the IR spectrum in figure 5-4. Note that the center electrical connector, which is defective, stands out in marked contrast with respect to the others.

When interpreting IR photographs, other factors in addition to variation in the surface heat-loss rate may produce differences in the thermal picture. As previously discussed, variations in surface emissivity may also produce changes in the thermal picture. In addition, when a thermographic survey is conducted during the day, surfaces which have different solar absorptivities will be heated to different degrees by solar radiation. This latter effect may cause dark-colored surfaces to be warmer than light-colored surfaces. For this reason, when information on relative heat-loss rates through various building surfaces is sought, it is best to conduct IR surveys at night in order to eliminate solar radiation effects.

Further information on the various measurement techniques using IR thermography is given in reference [2].

Figure 5-1. Conventional photograph of a test house at NBS [4].

Figure 5-2. Infrared picture of the NBS test house shown in figure 5-1 [4].

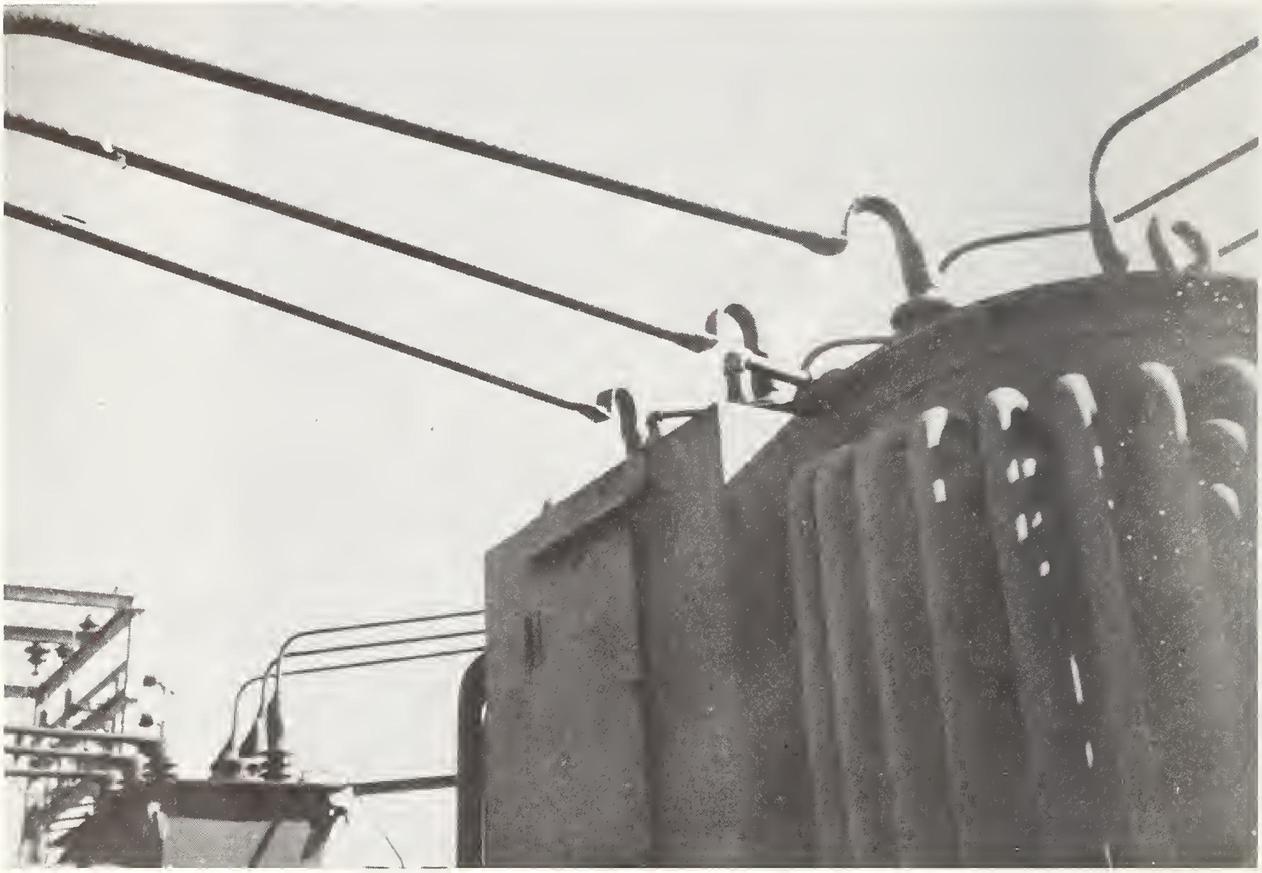

Figure 5-3. Conventional photograph of electrical connectors on a transformer.

Figure 5-4. Infrared picture of electrical connectors on the transformer shown in figure 5-3.

5.2.2 Measuring Air Leakage in Buildings

Air exchange between the indoor and outdoor air usually accounts for a quarter to a half of the heating and cooling loads of buildings. An important part of a measurement program for identifying energy conservation potentials for buildings is to assess the air-leakage characteristics of various buildings located on the Base. Those buildings having leaky construction need to be identified, so that retrofit actions can be implemented to reduce their air-leakage rates. In addition, once buildings of leaky construction have been identified, further measurements can be performed to locate major air-leakage paths.

In general, air leakage occurs through cracks around doors and windows and through openings (such as ventilation exhaust openings). Another path, which is often not considered, is air permeation through the seemingly solid construction. This latter air-leakage process becomes significant when construction materials having high permeability are used.

The driving forces which cause air leakage are the inside-to-outside temperature difference and wind velocity. In the winter, the air inside a building is considerably warmer and therefore lighter than the outdoor air. Since it is lighter, the indoor air tends to rise and leak out through the top parts of a building. Cold air is drawn into the building through cracks and openings at the lower level. When wind strikes a building, it produces an elevation in pressure on the exposed side and a depression in pressure on the leeward side. These induced inside-to-outside pressure differences also cause air infiltration to occur.

This section describes a few measurement techniques for assessing the air tightness of buildings and for locating specific air-leakage paths.

5.2.2.1 Measuring Natural Air-Infiltration Rates

The air-infiltration rate of a building is customarily expressed in air changes per hour. A technique for measuring the rate of air infiltration is the tracer-gas method.

The procedure for performing such a measurement is to release a small quantity of tracer gas (such as sulfur hexafluoride or helium) inside a building and to allow this tracer gas to become uniformly mixed with the indoor air. This is often conveniently accomplished by injecting the tracer gas into the return duct of the air-distribution system. Allowing the air distribution system to run continuously will usually provide uniform mixing of the tracer gas with the indoor air. The concentration of the tracer gas decays as the building exchanges air with the outdoor environment. By measuring this decay as a function of time, the air-infiltration rate for the building may be determined. If the air-delivery system is operated continuously, the concentration of tracer gas in the indoor air may be conveniently measured by sampling the air in a major supply duct.

The following factors may produce errors in tracer-gas techniques:

- non-uniform mixing of the tracer gas throughout the occupied space of the building
- errors in the measurement of tracer-gas concentration.

A complete discussion of the foregoing factors and other error-producing mechanisms may be found in reference [6].

An advantage of the tracer-gas technique is that it measures the air changes per hour for a building under natural driving forces of wind velocity and inside-to-outside temperature difference. Additional information on the tracer-gas technique may be found in references [5,6].

5.2.2.2 <u>Measuring the Air-Tightness Characteristics of Buildings</u>

An important part of an air-infiltration measurement program at a Base is to rank buildings according to their air-tightness characteristics. It is very difficult to obtain meaningful comparative data on the air-leakage characteristics of buildings using the tracer-gas technique, since the measured air-changes per hour for every building depend significantly on the wind velocity and indoor-to-outdoor temperature difference. It is not practical to attempt to measure air-infiltration rates for a group of buildings at identical wind velocities and temperature differences. A more suitable technique for ranking the air-tightness of a group of buildings is the pressurization technique.

The procedure for the pressurization technique is to introduce outdoor air into the interior of a building, thereby imposing an induced air-infiltration rate. This may be accomplished by connecting a centrifugal air blower to a building through a flexible hose, as shown in figure 5-5. The air-delivery rate of the blower is adjusted until the inside-to-outside pressure difference is 0.3* inches of water column as measured with a water manometer. For residential buildings, a recent study [8] found the required ft^3/min to be between 0.08 to 0.17 ft^3/min per cubic foot of indoor volume, depending on the air tightness of the house. Once this condition is attained, the air-delivery rate is measured with a suitable flow-measuring device (see diagram of figure 5-5) and the number of induced air changed per hour (I) is calculated from the relation:

$$I = W \cdot 60/V \qquad (2)$$

where W = air-delivery rate of blower, ft^3/min

V = inside volume of building, ft^3.

A single rating figure which characterizes the air-tightness of a building is called the "air-tightness figure ", which is defined as the number of induced air changes per hour per unit surface area of the exterior building envelope. The air-tightness figure is obtained by dividing the induced air changes per hour (I) by the surface area of the exterior building envelope. Retrofit actions to reduce air leakage should be applied first to those buildings having the highest air-tightness figures.

When buildings are pressurized for this technique, the principle driving force for air infiltration is the inside-to-outside pressure difference. Under natural conditions, the driving forces for air infiltration are the stack effect and wind-induced pressure gradients across various components of the building. Under natural conditions, the stack effect causes air to exfiltrate at the top levels of a building and to infiltrate at lower levels. On the other hand, when a building is pressurized, air exfiltrates at all levels. Therefore, the air-infiltration patterns are different under pressurized conditions in contrast to natural conditions. Further information on the pressurization technique may be found in references [7,9].

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*For very large and leaky buildings such as hangars, it may be more practical to perform measurements at a lower ΔP.

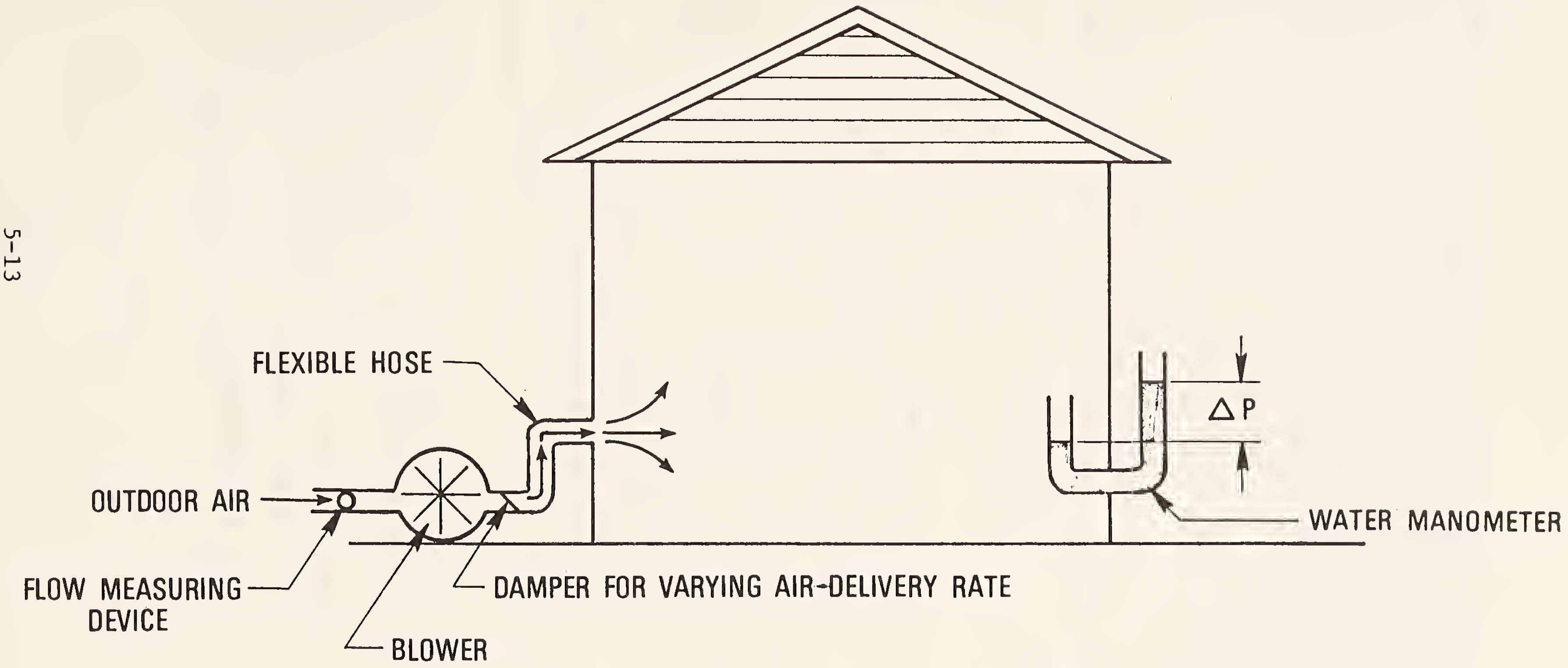

Figure 5-5. Schematic illustrating the pressurization technique.

5.2.2.3 Locating Specific Air-Leakage Paths

Once particular buildings have been identified as having comparatively high air-tightness figures, it may be desirable to perform additional measurements to locate the major air-leakage paths for these buildings. Retrofit actions may then be taken to eliminate these air-leakage paths. Two measurement techniques suggested for this purpose are infrared thermography (discussed in section 5.2.1) and the smoke technique.

The procedure for performing the smoke technique is to release colored pyrotechnique smoke in the various rooms (zones) of a building. The smoke is permitted to fill uniformly the interior of the building, after which the building is pressurized using a centrifugal blower as shown in figure 5-5. Major air-leakage paths will be dramatically displayed by effusing smoke. A disadvantage of this method is that many pyrotechnique smokes leave a chaulky dust layer over interior furnishings which may require a substantial clean-up effort. Further information on the smoke technique may be found in reference [4].

The preceding techniques are effective for measuring natural air infiltration rates and air-tightness of buildings, and for locating specific air-leakage paths. They may be useful to determine which buildings are the best targets for retrofit actions. However, two additional factors should be considered in determining the energy-conservation effectiveness of retrofit measures.

- Retrofit actions to reduce air leakage may not be effective in reducing heating and cooling energy requirements for buildings that are of tight construction to begin with.
- Substantial air leakage may still exist after all major air-leakage paths have been eliminated, since a significant amount of air leakage sometimes occurs as a result of air permeation through the ceiling, walls, and floor.

5.2.3 Energy Use Indexes

The Energy Use Index (EUI), defined as the total annual energy (in Btu's) consumed by a building per gross square foot of conditioned floor area, is a single-figure energy-consumption index which may be used to rank buildings according to their energy consumption. Energy-use-index data can also be used to monitor the success or failure of an energy-conservation program.

The following buildings or groups of buildings are presently individually metered on most Air Force Bases:

- family housing, metered as a group
- hospitals
- Base exchanges
- commissaries
- clubs

Energy-use-index data could readily be derived for these buildings. Other buildings on Air Force Bases are generally not individually metered and it would be more difficult to obtain energy use indexes.

If each Air Force Base were to develop EUI data for the foregoing building types, comparison of the energy consumption of these buildings could be obtained throughout the Air Force. A compilation and analysis of EUI data on these types of buildings could lead to the development of guidelines for distinguishing between energy-efficient and energy-wasteful buildings. In addition, the success or failure of Base-level energy conservation programs for these buildings could be readily monitored.

In a recent study [10], the following factors were identified as having a significant effect on the energy use indexes for buildings:

- total building refrigeration capacity
- cooling system operating hours
- type of HVAC system
- thermal transmittance of the gross exterior wall area
- lighting power density
- operation of computer and data systems
- computer-room operating hours.

An analysis of these factors is given in reference [10]. In a strict sense, EUI's should be compared for buildings which have approximately the same seven factors.

Further information on energy use indexes may be found in references [10, 11].

5.2.4 Detecting Moisture in Built-up Roofing Systems

When the top membrane of a built-up roofing system cracks due to weathering or is inadvertently punctured, rain water is able to penetrate the roof system. Since the bulk of the top, vapor-impervious membrane is intact, the escape of water is a very slow process. The accumulation of moisture may wet the insulation and reduce its insulating properties, thereby causing high heat transmission to occur through those portions of the roof having wet insulation. If the regions having wet insulation could be located by some means, local repairs could be made on those defective portions instead of replacing the complete roof system.

The following methods are available for investigating the moisture content of built-up roofing systems:

- infrared thermography (discussed in section 5.2.1)
- nuclear moisture meters
- resistivity measurements
- dielectric moisture meters.

Determining the moisture content of a built-up roofing system using a resistivity meter is based on the principle that the resistivity of roofing materials varies as a function of moisture content. The measurement procedure is to drive two closely separated probes into a built-up roof. An electric potential is applied between the probes, and the resulting electric current is measured. This technique has the disadvantage that it does require the roof to be punctured.

The dielectric moisture meter is based on the principle that the dielectric properties of roofing materials vary with the moisture content of the materials.

A nuclear moisture meter contains a radioactive material (such as radium, beryllium, etc.) which emits fast-neutrons to surfaces placed underneath the device. The operation of the device is based on the principle that the number of hydrogen atoms present in the material below determines the fraction of the neutrons which are reflected. The device contains a neutron-counting detector. Since each atom of water contains two hydrogen atoms, the fraction of neutrons that are reflected over the surface of a built-up roof system will vary with respect to the amount of accumulated moisture therein. It should be noted, however, that variations in the thicknesses of hydrocarbon-based materials (such as asphalt) will also produce changes in the instrument response.

The procedure in using a nuclear moisture meter is to set up a grid network for the roof system. A measurement is performed at the center of each sub-area of the grid network. Since the nuclear moisture meter provides only a relative response which varies with respect to moisture content of the roofing material, it is suggested that samples be cored at locations having the highest and lowest responses. The moisture content of these samples may be determined by an oven-drying technique and subsequently used to calibrate the readings of the nuclear moisture meter.

Depending on the type of radioactive isotope used, licensing by the Nuclear Regulatory Commission in agreement States and/or registration in non-agreement States may or may not be required. It is suggested that customers check with equipment suppliers or their State health department. Insomuch as these devices contain radioactive materials, some reasonable precautions must be observed in handling, transporting, and use to avoid any unnecessary or excess exposure to the operator. Further information on the use of nuclear moisture meters may be found in references [12-16].

5.3 Assessing the Thermal Performance of Building Equipment

5.3.1 Outside Make-Up Air

Often the most significant reductions in cooling and heating energy requirements can be achieved by reducing the volume of outside make-up air (that is introduced into the building via the mechanical system) to Air Force criteria values [17] which are given in table 3-1 (see Chapter 3).

The following methods may be used to measure the volume of outside make-up air:

- direct method
- indirect temperature-measurement method.

For the direct method, the average flow rate through the outside make-up-air opening(s) is measured using a flow measuring device. The volumetric flow rate of outside air through each outside make-up air opening is equal to the product of the mean intake velocity through the opening and its net open area. If more than one intake opening exists for the mechanical system, the gross volume of outside make-up air is equal to the sum of the volumetric flow rates through the separate openings.

For some HVAC Systems, the flow rate of the mixture of the return and outside air may conveniently be measured (see figure 5-6). In such instances, the volume of outside makeup air may be determined by measuring the temperatures of the return air, outside air, and the mixture. This technique for measuring the volume of outside make-up air is called the indirect temperature-measurement method. The volumetric flow rate ($\dot{V}_0$) of outside air is determined from the relation:

$$\dot{V}_o = \dot{V}_m \cdot \frac{t_m - t_r}{t_o - t_r} \qquad (3)$$

where $\dot{V}_m$ = volumetric flow of mixture of outside air and return air

t_m = temperature of mixture

t_r = temperature of return air

t_o = temperature of outside air.

Once the volume of outside make-up air is measured by one of the foregoing methods, the value is adjusted to the Air Force criteria values given in table 3-1 (see Chapter 3).

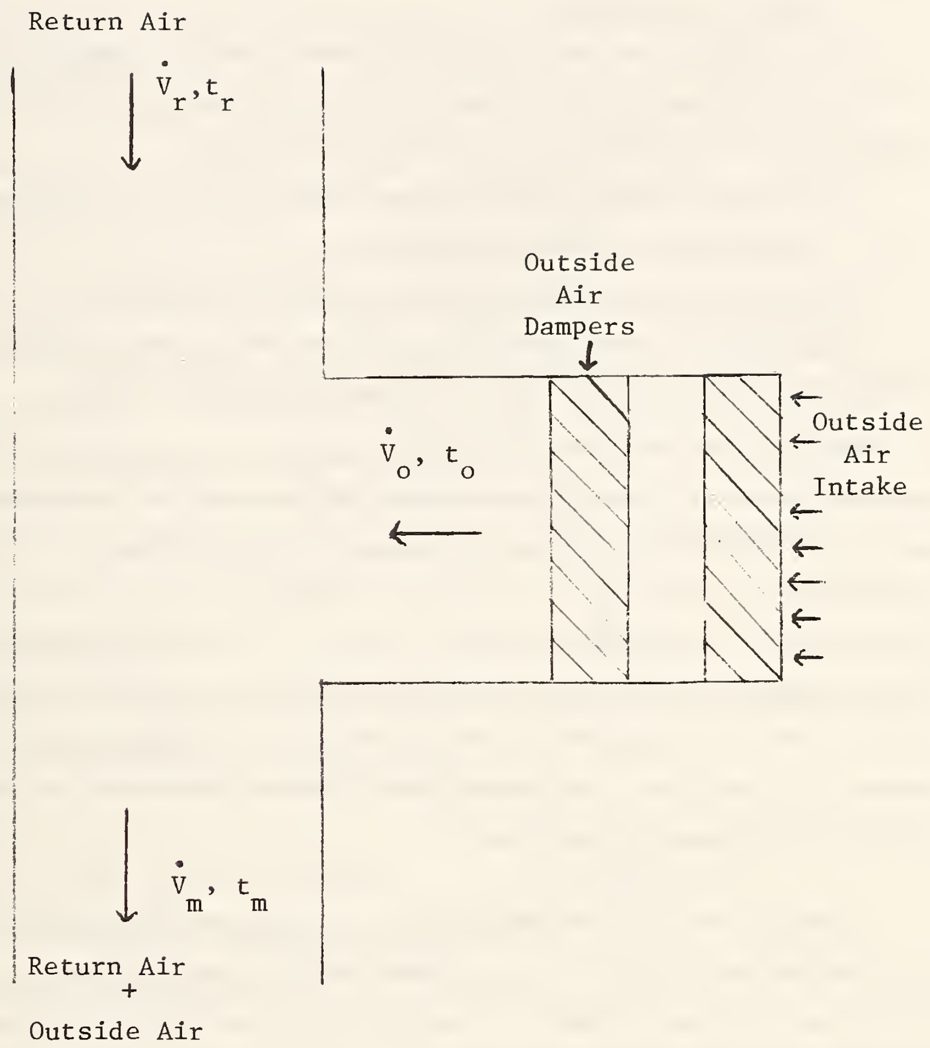

Figure 5-6. Schematic illustrating the indirect temperature-measurement method for determining the volume of outside make-up air.

5.3.2 Indoor Environment

A team from the Base Civil Engineer's Office should be established to survey the indoor environment of the buildings located on every Air Force Base to identify the following conditions:

- non-compliance with end-use restrictions on temperature
- overheating and overcooling
- excessive humidification or dehumidification.

The team should be equipped with a sling psychrometer for measuring indoor dry-bulb temperature and relative humidity (rh).

During the winter the dry-bulb temperature should not exceed 68^{o}F and the indoor rh should not exceed 30 percent.* A winter indoor rh exceeding 30 percent indicates excessive humidification. Human comfort studies have shown that raising the indoor rh above 30 percent does not provide much improvement in the comfort of the indoor environment. Since energy is required to evaporate water for any humidification process, maintaining indoor rh levels above 30 percent will cause energy wastes. In addition, indoor rh levels above 30 percent may contribute to moisture problems in buildings.

Perhaps some of the most blatant energy wastes on Air Force Bases are heating buildings in the winter and cooling buildings in the summer with their windows opened. Usually this condition is a result of inadequate or malfunctioning temperature controls, which causes overheating or overcooling of work and living spaces. The occupants, unable to perform adjustments on the heating or cooling system of their building, have no alternative but to open windows to try to restore a comfortable indoor condition.

During every heating and cooling season, the Base civil engineer should conduct a survey of the buildings on the Base to identify buildings having overheating and overcooling conditions. Elimination of these conditions should receive top priority in the Base energy conservation program, since they represent gross energy wastes.

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*An exception to this requirement is rooms containing special purpose electronic equipment, such as computer rooms.

During the summer, indoor temperatures of air-conditioned spaces should not be kept below 76°F. For buildings located in humid climates, indoor rh levels maintained below 50 percent may represent excessive dehumidification. If the cooling equipment is maintaining these conditions through sensible heat addition (terminal reheat) to the conditioned air, a very wasteful situation exists. In such instances, it may be possible to effect substantial energy savings by eliminating a part or all of the reheating process. For buildings located in extremely humid climates, however, one must be careful when eliminating sensible heat addition so that excessive indoor humidity levels are not created. If the indoor dewpoint temperature is permitted to reach the temperature of any indoor surface, then condensation will occur at that surface.

5.3.3 Building Equipment

A few simple measurements for identifying energy conservation actions for building equipment are summarized in table 5-1. The equipment needed for these measurements includes:

- voltmeter
- portable (clip-on type) ammeter
  } or portable watt recorder
- bulb thermometer (or digital thermometer)
- slant gage for measuring pressure drops.

These measurements in most cases are easy to perform and will identify potentials for energy conservation.

In buildings having complex HVAC systems, energy waste may exist due to:

- excessive pumping power
- excessive humidification and dehumidification.

TABLE 5-1. MEASUREMENTS ON BUILDING EQUIPMENT

Measurement	Recommended Level	Remedial Action
Hot water supply temperature	Not higher than 120 °F	Adjust thermostat on hot-water heating equipment
Measure temperature drop between hot water immediately leaving hot-water-heating equipment and hot water leaving faucets (after water has been running for sufficient period to reach steady-condition).	Should not exceed 10 °F drop	Install insulation to hot water pipes
Measure pressure drop across filters	Should not exceed values for a clean filter by more than 30%.	Replace or clean dirty filters
Measure the temperature difference between air immediately leaving the plant and air being delivered to the zones of the building.	Should not exceed $0.1\cdot(T_{supply} - T_{return})$	Insulate ducts

It is quite common for many existing buildings to have their HVAC systems altered during the years without comprehensive plans. The existing equipment and systems may be operating at quite different conditions than originally designed. This frequently occurs where space usage has been changed from that originally designed and balanced. Such conditions may often be remedied by measuring, adjusting, and balancing the HVAC system. This involves measuring the various operating parameters (temperatures, air-delivery rates, etc.) of the system and comparing these values with design values for the system. Modifications of the system are then performed to adjust its operation to an efficient condition which usually will be consistent with the original design. Detailed procedures for testing, adjusting, and balancing of HVAC systems are given in references [18, 20]. There are also many consulting firms which specialize in balancing HVAC systems.

5.4 Energy Audit for Central Plants

5.4.1 Steam and Hot-Water Generating Plants

For central-heating plants, there are two very important measurements which should be performed once a month to provide an indication of the operating performance of the plants:

- plant efficiency
- boiler efficiency.

(1) Plant efficiency

The thermal plant efficiency (η) is defined by the relation:

$$\eta = \frac{\text{Btu delivered per month}}{\text{Btu fuel input to generators per month}} \quad (4)$$

Since the plant efficiency depends somewhat on the load, it is recommended that monthly totals for Btu delivered and fuel-input values be used to calculate efficiency. Most Air Force plants are equipped with fuel-consumption meters that register the total amount of input fuel consumed. It is important that the fuel supplied only to the steam or high-temperature-hot-water (HTW) generators be metered. For steam, the Btu delivered per month is the product of the pounds of steam delivered per month by the enthalphy of the steam leaving the plant. The enthalpy of the steam leaving the plant is readily determined from the steam tables [21]. Many Air Force Bases are not equipped with totalizing steam-consumption meters, but it is recommended that such meters be installed. Information on the measurement of steam is available in references [22,23]. For HTW plants, the Btu delivered per month is the integrated product of the flow rate and the temperature differential between supply and return. This is usually measured with a Btu meter. Information on the various types of Btu meters is given in reference [23].

It is suggested that monthly plant efficiencies be tabulated in a log book. A central heating plant should have an efficiency of at least 70 percent. If a plant has an efficiency significantly lower than this figure, a strong need exists to take a look at energy conservation measures for the central plant, outlined in chapter 2 of this report. A record of monthly plant efficiencies makes it possible to compare plant performance from one year to the next. A need for improved maintenance procedures is readily identified.

(2) Boiler Combustion Efficiency

The most important component of a thermal plant is the boiler(s). The single factor that best characterizes the performance of a boiler is its combustion efficiency--defined as the fraction of the Btu fuel input which is transferred as heat to the working fluid.

In general, to determine the combustion efficiency when operating under full-load steady-state conditions, the following parameters are measured:

CO_2 and CO content of flue gas;

stack temperature.

It is important that the foregoing quantities be measured after heat-recovery devices. It is recommended that prior to performing these measurements, the boiler be operated at full-load conditions long enough to ensure that steady boiler operation has been achieved. Once the foregoing quantities have been measured, the boiler efficiency may be determined from a series of nomographs given in reference [24].

It is suggested that the boiler efficiency be determined once a month. Typical boiler efficiencies for a wide variety of boilers are given in table 5-2. Note that two values are cited in the table. The maximum attainable value is the theoretical maximum value based on the thermodynamic Carnot efficiency. In practice, the theoretical maximum value can only be approached. A boiler, if it is an efficient one, should be slightly higher than the "as found" value. When a boiler has an efficiency substantially lower than the "as found" value, examine the energy conservation measures for the central plant outlined in chapter 2 of this report.

5.4.2 Chilled-Water Plants

The performance of a chilled-water plant can be characterized by a plant coefficient of performance (η) which is defined as:

$$\eta = \frac{\text{Btu removed from distribution system at plant exit}}{\text{energy input to refrigeration equipment}} \tag{5}$$

The energy removed from the chilled-water distribution system is best measured by a Btu meter. Such a meter senses the increase in water temperature observed between the return and supply chilled-water temperatures at the same time that the weight rate of water flow is measured. The Btu meter integrates the product of the measured temperature difference and the mass rate of flow. Since both the chilled-water flow rate and the temperature drop may vary as a function of time, it is not recommended that the heat removed be determined from a separate measurement of flow rate and temperature difference. Information on Btu meters may be found in reference [23].

TABLE 5-2 INDUSTRIAL BOILER EFFICIENCY (PERCENT)
AVERAGE "AS FOUND"* AND MAXIMUM ATTAINABLE

RATED CAPACITY RANGE (10^6 Btu/h)	10-16		16-100		100-250		250-500	
Watertube - w/o air preheat								
Gas	77.5	85	79.4	86	76.0	87		
Oil			81.6	88	80.[illegible]	88		
Coal			79.4	88			80.8	88
Watertube - w/air preheat								
Gas			83.4	86	83.4	87	82.3	87
Oil			86.4	88	84.7	88	85.8	88
Coal			75.9	88	81.8	88	85.8	88
Firetube								
Gas	81.6	85	81.9	85				
Oil	85.7	87	85.0	88				

FEA/EPA/AGA

*Based on typical field surveys.

In the case of electrically-driven refrigeration compression equipment, the energy input would be the electrical energy delivered to the equipment over a specified period of time. The electrical energy is conveniently measured with a watthour meter. In the case of absorption-refrigeration equipment, the energy input would be the heating energy supplied to the generator of the absorption machine over a specified period of time.

It is recommended that plant coefficient of performances be determined from measurements over a 24-hour period. Once a suitable instrumentation system is installed, daily coefficient of performance could be derived from daily readings of fuel consumption and Btu's removed. The daily plant coefficient of performance is dependent on the cooling load.

Unfortunately, part-load coefficient of performance data for chilled-water generating plants are not well known. However, a chilled-water plant having vapor-compression refrigeration equipment should have a coefficient of performance of approximately 3.1; a chilled-water generating plant using an absorption-refrigeration plant should have a coefficient of performance of 0.6 (single effect) and 1.1 (double effect).

Measurements on thermal plants are no better than the accuracy of the instrumentation. Measuring devices should be calibrated at least once a year. Calibration services are available on most Air Force Bases.

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CHAPTER 6. ECONOMIC ANALYSIS

CONTENTS

CHAPTER 6. ECONOMIC ANALYSIS

6.1 Introduction

Energy inputs needed to achieve required energy-related services in Base facilities, including thermal comfort, illumination, hot water, and other functional services, can be significantly reduced through improvements in facility design (retrofit) and in operational procedures. A considerable amount of data is available in this handbook and elsewhere to help identify both the potential methods for reducing energy requirements and the potential energy savings that can be achieved through their use.

In some cases, significant improvements can be made at little or no cost and are therefore immediately attractive. In general, however, physical improvements to Base facilities which reduce energy consumption without reducing the quality of related services are costly enough to require some economic justification. Moreover, where some of these improvements can be utilized at two or more distinct levels of application (such as insulation), an analysis is required to determine which of these utilization levels is the most economically justified.

As energy prices increase relative to the cost of energy conservation improvements, greater levels of investment in energy conservation should be considered to offset these higher energy costs. The purpose of this section of the handbook is to provide a framework for making such economic considerations on a systematic basis. This framework is generally referred to as life-cycle benefit-cost analysis. It requires that all benefits and costs incurred throughout the economic life of energy-related improvements to Base facilities be compared on a consistent, time-equivalent basis. The method used to make all these benefits and costs consistent is explained in section 6.2.

The objective of such an analysis is to determine not only which improvements are "worth" doing, but the extent to which they are worth doing. In most conventional benefit-cost studies, the former is stressed. Discussion of this conventional approach to benefit-cost analysis takes place in section 6.3. In this handbook, however, equal weight is given to the use of benefit-cost analysis in choosing the most economical investment level when more than one level is feasible. This is the subject of section 6.4.

6.2 Present-Worth Analysis

Most physical energy conservation improvements in Base facilities require some immediate outlay of military funds. This outlay is made in anticipation of reduced operational expenditures over the remaining economic life of the facility. These reduced expenditures, or savings, are generally expected to more than pay back the initial investment.

However, there are competing uses for military funds other than energy conservation improvements. Some of these competing uses may pay higher "dividends" than some energy conservation improvements. Thus, in general, energy conservation investments should be made when their rate of return is at least as great, or greater than that for the next best alternative investment.

DoD has suggested that the minimum acceptable rate of return (representing that next best alternative) is 10 percent. For this reason, a discount rate of 10 percent is required to transform savings accrued in some future time period to their present value in a way that implicitly includes a 10 percent annual rate of return. For example, $1000 invested at 10 percent will be worth $1100 at the end of one year. Thus the present value of $1100 received one year from now is $1000 if a discount rate of 10 percent is appropriate. Present Value is determined by the formula:

$$PV = FV_{(t)} \times \frac{1}{(1+D)^t} \qquad (1)$$

where PV is the Present Value, $FV_{(t)}$ is the Future Value in t years, and D is the Discount rate. In our example, $PV = 1100 \times \frac{1}{(1.1)^1} = 1000$.

Benefits and costs so adjusted can then be compared on a time-equivalent basis; i.e., adjusted for foregone investment opportunities over the life of the facility.

At the same time that future savings are discounted to present value, it must be recognized that these savings may gradually increase over time. While the physical reduction in energy use may remain relatively constant from year to year, an increase in energy prices will result in higher annual dollar-value savings. These escalating energy prices are only partly due to general inflation. Increasing world demand for energy resources, more costly extraction processes, greater transportation distances, and stricter environmental controls will all result in higher energy prices unless a major breakthrough in energy production technology is achieved. Energy conservation investments are made not only to reduce energy expenditures today but throughout the life of the facility. Thus the effects of increasing energy prices should be fully incorporated in the decision analysis.

The discount rate and rate of energy price increase are both essential to the determination of present-value life-cycle savings. Equally important is the number of years over which the savings will accrue, i.e., the economic life of the investment. DoD generally recommends that 25 years be used as the economic life of Base buildings and utilities. Thus this recommendation should be followed whenever the expected economic life of an improvement to Base facilities is expected to be at least 25 years. If the estimated life of an energy conservation improvement is less than 25 years, the lower estimate should be used.

When the discount rate (D) and the appropriate economic life (L) are known, and the average annual rate of energy price increase (P) over the relevant lifetime is estimated, a present-worth factor (PWF) can be quickly calculated which can be used in estimating the present value of life-cycle energy savings. Given that physical energy savings from an energy conservation improvement are approximately constant from year to year, the PWF will relate the present value of savings over the entire life of the investment to the average annual savings, estimated at current energy prices:

Present-value life-cycle savings = average annual savings at current prices x PWF

Table 6-1 gives present-worth factors for a range of economic lives and for several different assumptions as to the rate of future energy price

TABLE 6-1. PRESENT-WORTH FACTORS

P = Average Annual Rate of Fuel Price Increase	Economic Life, Years				
	5	10	15	20	25
0%	3.79	6.14	7.61	8.51	9.08
2%	4.01	6.76	8.64	9.93	10.82
4%	4.24	7.44	9.86	11.69	13.07
6%	4.48	8.20	11.30	13.87	16.00
8%	4.73	9.05	12.99	16.59	19.87
10%	5.00	10.00	15.00	20.00	25.00
12%	5.28	11.06	17.38	24.30	31.87

increases. Each PWF is based on a discount rate of 10 percent. Alternatively, the PWF can be calculated as follows:

$$\text{PWF} = \left(\frac{1+P}{D-P}\right)\left(1-\left(\frac{1+P}{1+D}\right)^{L}\right) \qquad D \neq P \qquad (2)$$

$$\text{PWF} = L \qquad D = P \qquad (3)$$

Example #1. Present value of annual energy savings.

Replacement of an electric furnace in a Base housing unit with a heat pump reduces annual kWh consumption by an average of 7000 kWh. Electricity currently costs 3¢ per kWh, but is expected to increase in price by about 6 percent per year over the next 15 years, the estimated economic life of the heat pump. What is the present value of the life-cycle savings attributable to an investment in such a heat pump system?

First year savings ($.03/kWh x 7000 kWh)	=	$210.00
Present Worth Factor (15 years; 6%)	=	11.3
Present-value life-cycle savings	=	$2373.00

In example #1, energy savings are realized only during the heating season. For many energy conservation investments in buildings, energy savings may be accrued in both heating and cooling seasons. These savings should both be considered in estimating life-cycle savings. In some cases, it is possible that reductions in heating energy requirements will be partially offset by increased cooling energy requirements, or vice versa. In such cases, life-cycle savings should represent net savings. Where heating and cooling operations require different conversion efficiencies, the differences in energy costs per unit of output should be recognized. In addition, where estimates of energy price increases differ by type utilized, present-value savings (or losses) should be evaluated for each fuel type and then summed in present-value terms.

Most costs incurred as a result of energy conservation improvements are initial costs; i.e., they represent an immediate outlay of funds. As such, they are already expressed in present-value terms. However, some improvements may require increased operation and/or maintenance (O&M) expenditures, recurring at regular or irregular intervals throughout their lifetime. In order to evaluate these costs on a consistent basis with initial costs, they must be expressed in present-value terms as well.

If similar O&M requirements are expected to occur at periodic intervals, they can be evaluated in present-value, life-cycle terms using the average annual O&M requirements at current costs and the PWF appropriate for the economic life and the expected annual rate of increase in recurring O&M costs (due primarily to inflation) over this life.

If increased O&M costs are expected to occur at non-regular intervals, or require different expenditures at different times, these must be evaluated individually in order to reduce future costs to present value and then summed in present-value terms. This evaluation can be accomplished using the Present Value formula (eq. 1):

$$\text{Present value of O\&M cost incurred in } t^{th} \text{ year} =$$

$$\text{O\&M cost in } t^{th} \text{ year} \times \frac{1}{(1+D)^t} = \frac{\text{O\&M cost in } t^{th} \text{ year}}{(1.1)^t} \qquad (4)$$

where t = the number of years into the future when the cost is expected to be incurred.

<u>Example #2</u>. Present value of increased O&M costs.

Replacement of an electric furnace with a heat pump, while significantly reducing annual kWh consumption for space heating, may increase annual O&M requirements related to the heating and cooling system. For a given installation, this increase is estimated to cost $25.00 annually at current prices and to increase 4 percent annually over the 15-year life of the system. In addition, after eight years a new compressor will be needed, at an estimated future installed cost of $200.00. What is the additional present-value, life-cycle cost of the system due to increased O&M costs?

Average annual O&M cost at current prices	=	\$25.00
Present Worth Factor (15 years; P=4%)	=	9.86
Present-value, life-cycle recurring costs	=	\$246.50

Present-value compressor replacement cost:

$\$200.00/(1.1)^8$	=	\$93.30
Total present value increased O&M costs		\$339.80

We now have a systematic method for expressing life-cycle, dollar-value energy savings and related costs on a time-equivalent basis. A benefit-cost analysis is now required to determine the economic priority of the alternative energy conservation projects considered.

Energy conservation projects can be classified into two general groups:

(1) Fixed-sized projects: those which are relatively non-variable in application--either they are utilized or they are not; e.g., storm doors, storm windows for a given-size window.

(2) Variable-size projects: those which can be utilized at two or more distinct levels of application; e.g., building insulation.

Although benefit-cost analysis is used in assessing the economic priorities of both of these classifications, it is approached differently in each case. For this reason, the following discussion of benefit-cost analysis is divided into two sections.

6.3 Benefit-Cost Analysis: Fixed-Size Projects

Fixed-size energy conservation projects are generally considered to be economically justified only if their total present-value, life cycle benefits (i.e., the dollar value of reduced energy requirements) equals or exceeds their total present-value, life-cycle costs. This economic criterion requires that the benefit-cost ratio, the ratio of total benefits (TB) to total costs (TC), be equal to, or greater than, one:

$$\frac{TB}{TC} \geq 1 \qquad (5)$$

Alternatively, this same criterion can be restated to require that the payback period be less than or equal to the economic life of the project in question. The payback period is defined here as the time required to pay back the investment in full, including allowances for increasing energy costs and the required annual rate of return, as represented by the discount rate.

Note that the payback period, as used here, requires that it be referenced with respect to the economic life of the project. For this reason the payback period is not adequate for determining the relative economic priority of alternative projects with different economic lives. Arbitrarily determined payback period requirements applied to a group of projects being considered simultaneously, all with different lives, will usually result in the suboptimal allocation of energy conservation funds.

Economic priorities, where funds for energy conservation projects are limited, are best determined by the benefit-cost (B/C) ratio. Those projects with the highest B/C ratio should be given highest priority, and those with the lowest B/C ratio (even if greater than one) should be given lowest priority. An exception to this allocation rule may occur when the project with a high priority B/C ratio requires more funding than is available, in which case the next highest priority project that can be funded should be implemented. However, as more funding becomes available, the former project should be again given highest priority.

Example #3. Storm windows and doors are considered for installation in housing.

Prime windows are of two basic sizes, 3'x3' and 3'x5'. Storm windows on the smaller window are estimated to save nine gallons of fuel oil and on the larger window 15 gallons of fuel oil per year per window. Storm windows for prime windows of both sizes cost $25.00 each, installed, and are expected to last for 15 years. The storm door is expected to save 10 gallons of oil per year, costs $40.00 installed, and is expected to last 10 years. Oil currently sells for 40¢ per gallon and is expected to increase in cost at 6 percent annually. No additional maintenance is expected. Which of these modifications are economically justified and what priority should be given to them?

	Storm Windows		Storm Doors
	3'x3'	3'x5'	
Annual savings at current oil cost	$3.60	$6.00	$4.00
Present-Worth Factor (Table 1)	11.30	11.30	8.20
Present-value life-cycle savings	$40.68	$67.80	$32.80
Total cost	$25.00	$25.00	$40.00
Benefit-cost ratio	1.63	2.71	0.82
Economically justified?	YES	YES	NO
Priority	2	1	(3)

The payback period for any of these projects can be approximated as follows, using table 6-1. Divide the total cost by annual savings at current energy cost. Then locate the result (as nearly as possible) in the row of PWF's in table 6-1 appropriate to the rate of energy price increases used. The payback period will be the economic life relative to that PWF. Interpolation will give more accurate results.

Example #4. What is the payback period for the storm windows and storm door in example #3?

	Storm Windows		Storm Doors
	3'x3'	3'x5'	
(1) Total cost	$25.00	$25.00	$40.00
(2) Annual savings at current oil cost	$ 3.60	$ 6.00	$ 4.00
(3) (1)/(2)	6.94	4.17	10.00
(4) Payback in years (interpolated from table 6-1, 6% row)	8 years	5 years	13 years

Again, it is stressed that the payback period is not adequate for determining economic priorities if the projects considered have different economic lives.

It should be recognized that the calculation of benefits and costs to this point has been based on an assumption of no change in the level of energy-related services provided. In fact, improvements to building or equipment design, either in initial construction or by retrofit, will often result in increased occupant comfort, and at the same time reduce peak-load power requirements and dependence on fluctuating energy supplies. These additional benefits are somewhat intangible in that it is difficult to assign dollar values to them. However, where the economic justification of an energy conservation investment may appear to be a borderline case if energy savings alone are considered, these intangible factors may assist in making the final decision.

6.4 Benefit-Cost Analysis: Variable-Size Projects

The economic evaluation of variable-sized energy conservation projects requires not only the determination of economic justification, but the determination of the most economically justified level of utilization of such projects.

The best example of a variable-size energy conservation project is that of installing thermal insulation in buildings. As more and more insulation is added to a building, annual energy requirements decrease, but at a decreasing rate, so that each additional unit of insulation saves less energy than the preceding unit. However, as energy prices have increased relative to insulation prices, the present value of these energy savings has increased and more insulation has generally become economically justified than in the past.

This section presents a simplified approach to the determination of how much of a variable-size energy conservation technique is economically justified. This approach is called marginal, or incremental, analysis, because the benefit-cost analysis is applied to successive increments of the project rather than to the total project. The need for such an incremental approach will become clear with an inspection of table 6-2. This table shows the total benefits and costs associated with the use of several values of insulation resistance (R) in an attic of a Base housing unit. Present-value life-cycle savings are based on a 25-year life and a 6 percent annual energy price escalation factor (PWF=16).

While the benefit-cost ratios of table 6-2 decrease as resistance is increased, they remain well above 1, so that total benefits exceed total costs at all levels of application. In addition, the payback periods are all well below the economic life of the insulation. What then is

TABLE 6-2. REPRESENTATIVE TOTAL DOLLAR COST AND SAVINGS FOR SEVERAL VALUES OF THERMAL RESISTANCE (R)

R-value	Total Cost	Total Savings	B/C Ratio	Payback (P=6%)
0	0	0	--	--
10	$200	$3200	16.0	1 year
20	350	3610	10.3	1.5 years
30	500	3770	7.5	2.2 years
40	650	3850	5.9	3 years
50	800	3900	4.9	3.5 years

TABLE 6-3. REPRESENTATIVE INCREMENTAL DOLLAR COSTS AND SAVINGS FOR SEVERAL VALUES OF THERMAL RESISTANCE (R)

R-value	Incremental Cost	Incremental Savings	Incremental B/C Ratio	Incremental Payback Period
0	0	0	--	--
10	$200	$3200	16.0	1 year
20	150	410	2.73	7 years
30	150	160	1.07	23 years
40	150	80	0.53	∞
50	150	50	0.33	∞

the most economical level of insulation resistance? The B/C ratio and payback period criteria appear to have failed because the less insulation is utilized, the better these criteria appear to be.

Table 6-3 will help to resolve this problem. In table 6-3, incremental costs and benefits are shown for the same resistance levels shown in table 6-2. Incremental costs and benefits are the difference between the total costs or total benefits of two different levels of resistance. For example, the incremental cost between R-10 and R-20 insulation is \$350 - \$200 = \$150. Benefit-cost ratios and payback periods are computed for incremental savings and costs. Using the same criteria for economic justification as outlined in section 6.3, applied instead to the successive increments of a variable-size energy conservation project, one can now determine the extent to which a given increment is economically justified and what its economic priority is relative to increments of other variable projects and to fixed-size projects.

In table 6-3, the third increment of resistance, R-30 (R-10 added to R-20) is the last increment economically justifiable. Note that at this level total savings exceed total costs by the greatest amount (\$3770 - \$500 = \$3270) of any of the levels considered. For this reason, R-30 is considered to be the most economically justified level of insulation in this example. While this third increment takes approximately 23 years to pay back, it includes a 10 percent rate of return during that time period which is by assumption greater than the next best alternative investment. While this entire investment pays back in only 2.2 years, we see now that the total payback period for a variable-size investment is irrelevant to its real economic status.

In the example shown in tables 6-2 and 6-3, increments of 10 resistance units were arbitrarily chosen for ease of presentation. In actual practice, the levels of utilization chosen might better represent available options (e.g., R-11, R-19, R-22, R-30, R-38, etc.). Where the level of some energy conservation technique is continuously variable (e.g., blown-in attic insulation), the most economical level can be determined by using small increments near the most economical level (which can be approximated using larger increments) or by using more sophisticated mathematical techniques.

In some cases, incremental B/C ratios may fall below the economically justified level and then rise above it again. If physical conditions require that increments with lower B/C ratios be utilized before an increment with a higher B/C ratio can be utilized, the increments with the lower B/C ratios should be combined with that of the higher B/C ratio and evaluated as one large increment in order to determine if it is economically justified.

When a variable-size energy conservation technique is utilized up to the point that includes the last economically justified increment, the most economically justified level of investment has been determined. As in the case of fixed-size projects, when investment funds are limited so that not all projects or increments with B/C ratios greater than or equal to one can be utilized, those with the highest B/C ratios should be utilized first.

In determining actual energy savings attributable to the various energy conservation projects considered in a given facility, significant interdependence among projects must be recognized. For example, replacement of an electric furnace with a heat pump and the simultaneous installation of additional insulation in the same building are interdependent actions. The heat pump lowers the effective cost of energy needed to heat the building, thus reducing the savings attributable to the additional insulation. At the same time the additional insulation reduces the heating load on the heat pump, which, in turn, reduces the energy savings attributable to the heat pump.

In a sophisticated economic analysis, the annual energy savings credited to one project should be based on the assumption that all other energy conservation techniques are employed at their most economical level. Such an analysis may require several iterations, in which each sucessive iteration comes closer to the determination of an optimal combination of energy conservation projects for a facility.

APPENDIX A. CALCULATION OF ENERGY REQUIREMENTS FOR SPACE HEATING AND SPACE COOLING OF BUILDINGS

CONTENTS

APPENDIX A. CALCULATION OF ENERGY REQUIREMENTS FOR SPACE HEATING AND SPACE COOLING OF BUILDINGS*

1. INTRODUCTION

The calculation of heating and cooling energy requirements for buildings is necessary in order to evaluate energy savings resulting from energy conservation modifications to existing buildings. The calculation methodology is presented in four principle sections.

(1) <u>Design heating load</u>

The design heating load is the maximum heat delivery rate of a heating plant required to maintain the indoor environment at comfort conditions, when the outdoor environment is at design (cold weather) conditions. The calculation of design heating load makes it possible to estimate the required capacity for the heating plant.

(2) <u>Design cooling load</u>

The design cooling load is the maximum rate at which heat must be removed at the cooling coil of the air conditioning equipment of a building in order to maintain the indoor environment at comfort conditions when the outdoor environment is at design (hot weather) conditions.

(3) <u>Energy requirement for space heating</u>

This section presents a methodology for the calculation of the net heating energy requirement for a building over the entire heating season. Such predictions are necessary in order to quantify the annual reduction in heating energy requirement resulting from energy conservation modification to a building.

(4) <u>Energy requirement for space cooling</u>

This section presents a methodology for the calculation of seasonal energy required by the cooling equipment to remove sensible and latent heat gains from the occupied space of a building during the cooling season.

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* Many of the equations for this chapter were taken from reference [2].

Items (1) and (2) are essentially based on steady-state heat transfer between indoor and outdoor design conditions and are mostly taken from chapters 21 and 22 of the 1972 ASHRAE Handbook of Fundamentals [2] with certain simplifications. Even with these simplifications, certain fundamental heat-transfer concepts must be adhered to in the determination of heating and cooling loads. It is expected that those using these calculation procedures are familiar with the fundamentals.

Items (3) and (4) above are based on the bin method which is probably the most accurate method without the use of a computer for the determination of seasonal energy requirements. For more accurate techniques for predicting heating and cooling energy requirements, the reader is referred to Appendix D which presents a survey of computer programs for evaluating building and system performance.

2. DESIGN PARAMETERS

The following design parameters shall be used in heating and cooling load determinations. (If for special cases such as hospitals, laboratories, computer room, etc., other special indoor design conditions are maintained, heating and cooling load calculations shall be based on these conditions).

2.1 Outdoor Design Conditions

Winter and summer outdoor design conditions shall be the 97-1/2 percent and 2-1/2 (dry bulb) and 5 (wet bulb) percent values, respectively, as given in Chapter 6, Engineering Weather Data of AFM 88-8 [1]*.

2.2 Indoor Design Conditions

Winter indoor design conditions shall be 70° F dry-bulb temperature with humidification, if provided, to not more than 30 percent rh. Summer indoor design conditions shall be 78° F dry-bulb temperature, with controlled humidification, if provided, to not less than 60 percent rh.

* Numbers in brackets indicate references cited at the end of chapter.

2.3 General Ventilation

For load design of all building types except one- and two-family dwelling units, mechanical ventilation with outside air shall be taken as 5 cfm (standard) outdoor air per person for occupied spaces and not to exceed 2 air changes per hour for unoccupied spaces such as corridors, foyers, etc.

2.4 Infiltration

Infiltration shall be taken as 0.7 air changes per hour for one- and two-family dwellings and 0.5 air changes per hour for all other buildings. For hangars and warehouses, it is recommended that air infiltration rates of 1.5 air changes per hour be used for energy calculations.

2.5 Heat Gains

Internal heat gains from lights, people, and equipment (appliances) shall be included in the following load and energy requirement calculations. During the summer, internal heat gains represent a sensible load which must be removed by the air-conditioning equipment. In the winter, internal heat gains actually provide some of the required heating energy and therefore need to be included in the calculation of the heating energy requirement.

3. DESIGN HEATING LOAD

The following methodology shall be used for calculation of the heating loads:

3.1 Thermal Transmittance

The thermal transmittance (U_i) for each of the heat-flow paths is determined by adding the thermal resistance for all elements in the path and then taking the reciprocal of the sum. Therefore, for each path,

$$U_i = \frac{1}{R_a + R_b + R_c + \ldots + R_r} \tag{1}$$

Resistance values for air films, air spaces, and insulation and construction materials are found in tables 1, 2 and 3, Chapter 20 of the ASHRAE Handbook of Fundamentals [2]. U-values for windows and doors are found in tables 8 and 9 of reference [2]. R and U values shall pertain to winter heating conditions.

3.2 Overall Thermal Transmittance

An overall thermal transmittance for the exterior envelope of a building or residence is obtained by the relation:

$$U_o = \frac{U_1A_1 + U_2A_2 + U_3A_3 + \dots + U_nA_n}{(A_1 + A_2 + A_3 + \dots + A_n)} \tag{2}$$

Here A denotes the surface area of each heat-flow path, such as exterior windows, walls, ceiling or roof, and floor above crawl spaces. The sum of the heat-transfer areas equals A_o, the total envelope area (excluding slab-on-grade floors, basements and foundations below grade).

3.3 Special Temperature Conditions

In cases for which certain rooms are maintained at a temperature other than prescribed by section 2.2, the thermal transmittance (U_i) for the heat-flow path shall be adjusted by the relation:

$$U_i = U\,\frac{t_a - t_o}{t_i - t_o} \tag{3}$$

where t_i = the design indoor air temperature
t_o = the design outdoor air temperature
t_a = temperature maintained in the room.

3.4 Thermal Transmittance of Unheated Spaces

The thermal transmittance of unheated spaces is based on heat-flow from heated rooms to unheated rooms and to the outside. The thermal transmittance based on the total area of unheated space exposed to the outdoors is:

$$\frac{1}{U_n} = \frac{1}{\Sigma\, U_n a_n} + \frac{1}{\Sigma\, U_r a_r} \tag{4}$$

where U_n = thermal transmittance of heat-flow paths between heated and unheated spaces,

U_r = thermal transmittance of heat-flow paths between unheated space and the outdoors,

and a_n and a_r are the ratios of surface area of the individual heat-flow paths to the total area of unheated space exposed to the outdoors.

3.5 Heat Loss from Below-Grade Walls and Floors

If the below-grade space is heated to a specified temperature, the heat loss should be calculated in the usual manner, based on the proper wall and floor coefficients and the outdoor air and ground temperatures. Heat loss through windows and walls above grade should be based on sections 3.1 and 3.2. Heat loss through below-grade walls and floors should be based on the floor and wall coefficients for surfaces in contact with the soil, and on the proper ground-water temperature. The heat loss rate for below-grade walls and floors may be calculated from the relation $H_f = F \cdot A$, where A is the pertinent surface area. Values of the heat loss factors (F) for various ground-water temperatures are given in table A-1.

TABLE A-1. HEAT LOSS FACTORS[a] (F) FOR BELOW-GRADE WALLS AND FLOORS [2]

Ground-Water Temperature, °F	Basement Floor Loss[b], Btu/h·ft^2	Below-Grade Wall Loss[c], Btu/h·ft^2
40	3.0	6.0
50	2.0	4.0
60	1.0	2.0

[a] These values in a strict sense apply to residential buildings, but they may also be used for other type buildings.

[b] Based on below-grade space temperature of 70° F and thermal transmittance (U) of below-grade wall of 0.1 Btu/h·ft^2·F.

[c] Assumed twice below-grade floor loss.

3.6 Heat Loss to Crawl Space

The temperature in crawl spaces will vary widely depending on the number and size of the ventilation openings, the amount of hot water piping and warm-air ducting present and the amount of pipe and duct insulation. It is necessary, therefore, to evaluate the conditions and to select an appropriate crawl-space temperature. Frequently this is done by assuming that it is midway between the design indoor and outdoor temperatures. After evaluating the thermal transmittance values for the various heat-flow paths for the floor as indicated in section 3.1, the relation of section 3.3 is used to determine the overall U-value for the floor.

3.7 Heat Loss from Floor Slabs

Two types of concrete slab floors are (a) the unheated slab-on-grade, relying for warmth on heat delivered above floor level by the heating system, and (b) the heated slab-on-grade containing heated pipes or ducts that constitute a radiant slab or portion thereof for complete or partial heating of the house. With unheated slabs-on-grade the heat loss (H_f) can be estimated by means of the equation

$$H_f = F \cdot P \tag{5}$$

where F = heat-loss coefficient, Btu/h (per linear foot of exposed edge)

P = perimeter or exposed edge of floor, linear feet.

Values for the heat-loss coefficient factor (F) are given in table A-2 for various thicknesses (R-values) of edge insulation. The insulation is considered to extend under the floor horizontally for 2 ft, or to be located along the foundation vertical wall extending 2 ft below the floor level. Floors containing heating pipes or ducts are now in common use. The heat loss downward into the ground and outward through the edges of the floor slab is called the reverse loss. Heat-loss factors (F) for heated slabs are also found in table A-2. For heated slabs using perimeter heating, (F) values are given in table A-3.

TABLE A-2. HEAT LOSS FACTORS (F) FOR CONCRETE FLOORS AT OR NEAR GRADE LEVEL PER FOOT OF EXPOSED SLAB EDGE, Btu/h·ft [2]

Outdoor Design Temperature, °F	Total length of Insulation, in.	Value of F for Unheated Slab[a]			Value of F for Heated Slab[b]		
		R = 5.0	R = 3.75	R = 2.50	R = 5.0	R = 3.33	R = 2.50
-30 and colder	24	34	51	67	46	69	92
-25 to -29	24	32	48	64	44	66	88
-20 to -24	24	30	45	60	41	61	82
-15 to -19	24	28	43	57	39	59	78
-10 to -14	24	27	40	54	37	55	74
- 5 to - 9	24	25	38	51	35	52	70
0 to - 4	24	24	36	48	32	48	64
+ 5 to + 1	24	22	33	44	30	45	60
+10 to + 6	18	21	31	42	25	38	50
+15 to +11	12	21	31	42	25	38	50
+20 to +16	Edge Only	21	31	42	25	38	50

R = Thermal resistance of insulation.

[a] Where perimeter insulation is not required, use F = 50 for unheated slabs or F = 75 for heated slabs.

[b] Slab floors having heating pipes or ducts under the slab shall be considered as heated slabs.

TABLE A-3. SLAB HEAT LOSS FACTORS (F) IN Btu/h·ft TO BE USED WHEN WARM AIR PERIMETER HEATING DUCTS ARE EMBEDDED IN SLABS[a] [2].

Outdoor Design Temperature °F	Edge Insulation		
	1-in Vertical Extending Down 18 in Below Floor Surface	1-in L-Type Extending at Least 12 in Deep and 12 in Under	2-in L-Type Extending at least 12 in Down and 12 in Under
-20	105	100	85
-10	95	90	75
0	85	80	65
10	75	70	55
20	62	57	45

[a] Factors include loss downward through inner area of slab.

3.8 Infiltration Heat-Loss Rate

The formula for the heat required to warm the outdoor air which enters a building by infiltration to the building indoor air temperature is:

$$H_i = 0.016 \cdot V_s \cdot I \cdot (t_i - t_o) \tag{6}$$

where H_i = infiltration heat-loss rate, Btu/h
V_s = volume of air space or spaces in building, ft^3
I = infiltration air-flow rate (see section 2.4), air changes/h
t_i = indoor-design temperature (see section 2.2), °F
t_o = winter outdoor-design temperature (see section 2.1) °F

3.9 Design Heat-Loss Rate

The total heat-loss rate (H_t) in Btu/h for a building is given by the relation:

$$H_t = A_o \cdot U_o \cdot (t_i - t_o) + H_i + H_f \tag{7}$$

where U_o = overall thermal transmittance of the building envelope (not including slab-on-grade floors and below grade spaces)

A_o = exterior surface area of the building envelope

$t_i - t_o$ = inside-to-outside temperature differences

H_i = heat-loss rate due to air infiltration

H_f = heat-loss rate through slab-on-grade floors or below grade spaces.

3.10 Design Heating Loads

The design heating load (H_{Load}) is simply equal to the differences between the design heat-loss rate (H_t) and the heat-release rate due to occupancy, lighting, and equipment or

$$H_{Load} = H_t - H_{ps} - H_{as} - H_e \tag{8}$$

where H_{ps} = sensible heat-release rate from occupants

H_{as} = sensible heat-release rate from equipment

H_e = heat-release rate from lighting.

Procedures for estimating the foregoing internal heat-release rates are presented in sections 4.5 and 4.6. In some buildings under design conditions the internal heat-release rates (H_{ps}, H_{as}, and H_e) are small in comparison to the design heat-loss rate for the building and may be neglected.

4. DESIGN COOLING LOADS

The following methodology shall be used for the calculation of cooling loads.

4.1 Thermal Transmittance

The thermal transmittance for each series heat-flow path is determined in section 3.1, where R-values should be obtained from tables applicable to summer cooling conditions.

4.2 Heat Gains through Windows

The heat flux through each window or fenestration orientation and configuration shall be calculated by the relation:

$$Q_w = SC \cdot SHFG + U_w \cdot (t_o - t_i) \tag{9}$$

where Q_w = rate of heat flow through window, $Btu/h \cdot ft^2$

SC = shading coefficient

$SHFG$ = solar heat gain factor, $Btu/h \cdot ft^2$

U_w = thermal transmittance of window, $Btu/h \cdot ft^2 \cdot {}^\circ F$

Solar heat gain factors (SHFG) are given in table A-4 for various surface orientations and latitudes. Shading coefficients (SC) for various fenestration configurations are given in tables A-5 to A-8. The total rate of heat gain through the windows is equal to the sum of the products of the rate of heat flow (Q_w) and window area for each of the windows of the building.

TABLE A-4. COOLING LOAD SOLAR HEAT-GAIN FACTORS (SHFG), $Btu/h \cdot ft^2$ [2]

Surface Orientation	LATITUDE-DEGREES NORTH			
	24	32	40	48
N	45	37	30	24
NW	168	167	163	156
W	190	198	204	207
SW	101	114	127	138
S	17	19	20	21
SE	17	18	19	20
E	17	18	19	20
NE	17	18	19	20
HORIZ	72	81	88	92

TABLE A-5. SHADING COEFFICIENTS (SC) FOR SINGLE GLASS AND INSULATING GLASS [2].

A. Single Glass

Type of Glass	Nominal Thickness	Shading Coefficient $h_o = 4.0$	$h_o = 3.0$
Regular Sheet	3/32, 1/3	1.00	1.00
Regular Plate/ Float	1/4	0.95	0.97
Grey Sheet	1/8	0.78	0.80
	1/4	0.86	0.88
Heat-Absorbing	3/16	0.72	0.75
Plate/Float	1/4	0.70	0.74

B. Insulating Glass

Type of Glass	Nominal Thickness	Shading Coefficient $h_o = 4.0$	$h_o = 3.0$
Regular Sheet Out, Regular Sheet In	3/32, 1/8	0.90	0.90
Regular Plate/ Float Out, Regular Plate/ Float In	1/4	0.83	0.83
Heat-Absorbing Plate/Float Out, Regular Plate/ Float In	1/4	0.56	0.58

TABLE A-6. SHADING COEFFICIENTS (SC) FOR SINGLE GLASS WITH INDOOR SHADING BY VENETIAN BLINDS AND ROLLER SHADES [2]

Type of Glass	Type of Shading				
	Venetian Blinds		Roller Shade		
			Opaque		Translucent
	Medium	Light	Dark	White	Light
Regular Sheet Regular Plate/Float Regular Pattern Heat-Absorbing Pattern Grey Sheet	0.64	0.55	0.59	0.25	0.39
Heat-Absorbing Plate/ Float[a] Heat-Absorbing Pattern Grey Sheet	0.57	0.53	0.45	0.30	0.36
Heat-Absorbing Plate/ Float or Pattern Heat-Absorbing Plate/ Float[a]	0.54	0.52	0.40	0.28	0.32
Heat-Absorbing Plate or Pattern	0.42	0.40	0.36	0.28	0.31
Reflective Coated Glass					
S.C.[b] = 0.30	0.25	0.23			
0.40	0.33	0.29			
0.50	0.42	0.38			
0.60	0.50	0.44			

[a] Refers to grey, bronze, and green tinted heat-absorbing plate/float glass.

[b] Shading Coefficient for glass with no shading device.

TABLE A-7. SHADING COEFFICIENTS (SC) FOR INSULATING GLASS WITH INDOOR SHADING BY VENETIAN BLINDS AND ROLLER SHADES [2]

Type of Glass	Type of Shading				
	Venetian Blinds[a]		Roller Shade		
			Opaque		Translucent
	Medium	Light	Dark	White	Light
Regular Sheet Out Regular Sheet In Regular Plate/Float Out Regular Plate/Float In	0.57	0.51	0.60	0.25	0.37
Heat-Absorbing Plate/Float[b] Out Regular Plate/Float In	0.39	0.36	0.40	0.22	0.30
Reflective Coated Glass					
SC = 0.20	0.19	0.18			
0.30	0.27	0.26			
0.40	0.34	0.33			

[a] For vertical blinds with opaque white or beige louvers, tightly closed, SC is approximately the same as for opaque white roller shades.

[b] Refers to bronze or green tinted heat-absorbing plate/float glass.

TABLE A-8. SHADING COEFFICIENTS (SC) FOR DOUBLE GLAZING WITH BETWEEN-GLASS SHADING [2]

Type of Glass	Description of Air Space	Type of Shading: Venetian Blinds		Type of Shading: Louvered Sun Screen
		Light	Medium	
Regular Sheet Out Regular Sheet In	° Shade in contact with glass or shade separated from glass by air space.	0.33	0.36	0.43
Regular Plate Out Regular Plate In	° Shade in contact with glass--voids filled with plastic	---	---	0.49
Heat-Abs. Plate/Float[a] Out	° Shade in contact with glass or shade separated from glass by air space.	0.28	0.30	0.37
Regular Plate In	° Shade in contact with glass--voids filled with plastic.	---	---	0.41

[a] Refers to grey, bronze, and green tinted heat-absorbing plate/float glass.

4.3 Rate of Heat Gain through Opaque Walls

The rate of heat gain (Q_q) in Btu/h·ft^2 through opaque walls shall be calculated from the relation:

$$Q_q = U_q \cdot TETD \tag{10}$$

Here U_q is the thermal transmittance of opaque walls as calculated from section 4.1 and TETD is the total equivalent temperature difference specified in table A-9. The procedure is to identify the wall construction type of table A-10 which is closest to the actual wall construction type. Identify the group number for the wall construction from table A-10 and then find the total equivalent temperature difference from table A-9. The total heat gain through opaque walls is equal to the sum of products of Q_q by their respective opaque wall area for each orientation.

4.4 Rate of Heat Gain through Roof Areas

The rate of heat flux (Q_r) in Btu/h·ft^2 through opaque roof areas shall be calculated from the relation:

$$Q_r = U_r \cdot ETD \tag{11}$$

Here U_r is the thermal transmittance of the roof area as calculated from section 4.1 and ETD is the equivalent temperature difference, as specified in table A-11 for selected type roofs. The net heat gain through a roof is equal to $H_r = Q_r \cdot A_r$, where A_r is the ceiling area below the roof.

TABLE A-9. TOTAL EQUIVALENT TEMPERATURE DIFFERENCE (TETD) FOR CALCULATING HEAT GAIN THROUGH SUNLIT WALLS* [2]

North Latitude Wall Facing	Exterior Color of Wall--D = dark, L = light							
	D	L	D	L	D	L	D	L
	Group A		Group B		Group C		Group D	
NE	23	17	39	20	24	15	21	14
E	26	19	36	24	30	19	27	17
SE	26	18	37	24	39	18	25	16
S	25	18	36	24	23	15	20	13
SW	41	26	42	26	23	15	22	15
W	48	30	40	25	22	15	23	15
NW	38	25	30	20	18	12	19	13
N	21	16	22	16	15	11	14	11

*The values in the table were calculated for an inside temperature of 75° F and an outdoor maximum temperature of 95° F with an outdoor daily range of 21° F. The table remains approximately correct for other maximums (93°-103° F) and the other outdoor ranges (16°-34° F) providing the outdoor daily average temperature remains approximately 85° F. If the room temperature is different from 75° F and/or the outdoor daily average temperature is different from 85° F, apply the following corrections:

- For room air temperatures less than 75° F, add the difference between 75° F and the room air temperature; if greater than 75° F, subtract the difference.
- For outdoor daily average temperature less than 85° F, subtract the daily average temperature; if greater than 85° F, add the difference.

TABLE A-10. DESCRIPTION OF WALL CONSTRUCTIONS [2]

Group	Components	Wt., lb per sq ft	U-Value $Btu/h \cdot ft^2 \cdot °F$
A	1" stucco + 4" l.w.[1/] concrete block + air space	28.6	0.267
	1" stucco + air space + 2" insulation	16.3	0.106
B	4" face brick + 4" l.w. concrete block + 1" insulation	62.5	0.158
	1" stucco + 4" h.w.[2/] concrete + 2" insulation	62.9	0.114
C	4" face brick + 8" clay tile + 1" insulation	96.4	0.137
	4" face brick + 8" common brick	129.6	0.280
	1" stucco + 12" h.w. concrete	155.9	0.365
	4" face brick + 2" insulation + 4" common brick	89.8	0.106
	4" face brick + 2" insulation + 4" h.w. concrete	96.5	0.111
	4" face brick + 2" insulation + 8" h.w. concrete block	90.6	0.102
D	4" face brick + air space + 8" clay tile	96.2	0.200
	4" face brick + 2" insulation + 8" common brick	129.9	0.098
	4" face brick + 2" insulation + 8" h.w. concrete	143.3	0.107

[1/] l.w. denotes light weight

[2/] h.w. denotes heavy weight

TABLE A-11. TOTAL EQUIVALENT TEMPERATURE DIFFERENTIALS (ETD) FOR CALCULATING HEAT GAIN THROUGH FLAT ROOFS [2]

Description of Roof Construction	Wt lb/ft^2	U-value $Btu/h \cdot ft^2 \cdot °F$	ETD Roof Color Dark	ETD Roof Color Light
Light Construction Roofs--Exposed to Sun				
1" insulation + steel siding	7.4	0.213	78	45
2" insulation + steel siding	7.8	0.125	81	46
1" insulation + 1" wood	8.4	0.206	86	48
2" insulation + 1" wood	8.5	0.122	88	49
1" insulation + 2.5" wood	12.7	0.193	79	42
2" insulation + 2.5" wood	13.1	0.117	76	41
Medium Construction Roofs--Exposed to Sun				
1" insulation + 4" wood	17.3	0.183	62	32
2" insulation + 4" wood	17.8	0.113	58	30
1" insulation + 2" h.w. concrete	28.3	0.206	81	44
2" insulation + 2" h.w. concrete	28.8	0.122	79	43
4" l.w. 1/ concrete	17.8	0.213	88	48
6" l.w. concrete	24.5	0.157	72	38
8" l.w. concrete	31.2	0.125	49	24
Heavy Construction Roofs--Exposed to Sun				
1" insulation + 4" h.w. 2/ concrete	51.6	0.199	61	32
2" insulation + 4" h.w. concrete	52.1	0.120	58	30
1" insulation + 6" h.w. concrete	75.0	0.193	48	25
2" insulation + 6" h.w. concrete	75.4	0.117	46	23

1/ l.w. denotes light weight

2/ h.w. denotes heavy weight

4.5 Heat Gain from Occupants

The rates at which heat and moisture are given off by human occupants under different states of activity are given in table A-12. The sensible heat gain (H_{ps}) is:

$$H_{ps} = P_1 \cdot G_1 + P_2 \cdot G_2 + P_3 \cdot G_3 + \ldots + P_n \cdot G_n \tag{12}$$

where P_n = number of persons engaged in specific activity of table A-12

G_n = sensible heat for activity of table A-12, Btu/h per person.

The latent heat gain (H_{pl}) is:

$$H_{pl} = P_1 \cdot L_1 + P_2 \cdot L_2 + P_3 \cdot L_3 + \ldots + P_n \cdot L_n \tag{13}$$

where L_n is the latent heat for activity of table A-12, Btu/h per person

4.6 Heat Gain from Heat-Producing Processes

Heat gain from heat-producing processes includes heat from lighting, electric motors and office, kitchen and cleaning appliances. Heat gains from these sources must be released within the conditioned space in order to be considered as heat gain to the building. Heat released to an exhaust air stream or refrigeration circuit which is in turn rejected to an outdoor environment must not be considered as heat gain. Heat gain from electric lighting (H_e) is calculated from the relation:

$$H_e = 3.412 \cdot W \cdot M \cdot N \tag{14}$$

where W = total installed lighting power, watts

M = special allowance factor, 1.2 for fluorescent fixtures, 1.0 for incandescent

N = use factor

= 0.9 for commercial

= 0.8 for residential

TABLE A-12. RATES OF HEAT GAIN FROM OCCUPANTS OF CONDITIONED SPACES [2]

Degree of Activity	Sensible Heat (G) Btu/h	Latent Heat (L) Btu/h
Seated at rest	225	105
Seated; very light work	245	105
Moderately active office work	245	155
Standing; light work; or walking slowly	250	200
Walking; seated Standing; walking slowly	250	250
Sedentary work	250	275
Light bench work	275	475
Moderate dancing	305	545
Walking 3 mph; moderately heavy work	375	625
Heavy work	580	870

The heat gain from motors is given in table 30, Chapter 22 of the 1972 ASHRAE Handbook of Fundamentals [2]. The heat gain from all office appliances must be calculated in a manner similar to that for lighting, where the use factor must be based on engineering judgment. For kitchen appliances, both the sensible and latent heat gains must be considered. Tables 31 and 32 of the same source contain recommended rate of heat gain values for kitchen and miscellaneous appliances.

4.7 Heat Gain Due to Ventilation and Infiltration

In section 2.3, mechanical ventilation with outside air shall be taken as 5 cfm (standard) per person for occupied spaces and not to exceed 2 air changes per hour for occupied spaces such as corridors, foyers, etc., for all building types except one- and two-family dwelling units. The sensible heat gain (H_{vs}) from mechanical ventilation is:

$$H_{vs} = 5.5 \cdot P \cdot (t_{od} - 78) \tag{15}$$

where P = number of occupants

and t_{od} = summer outdoor design temperature (section 2.1).

The latent heat gain (H_{vl}) from this source is

$$H_{vl} = 24{,}210 \cdot P \cdot (\omega_{od} - 0.012) \qquad (16)$$

where ω_{od} is the humidity ratio of the outdoor air at the design wet-bulb and dry-bulb condition specified in Ref. [1]. If ω_{od} is less than 0.012, H_{vl} should be set to zero. Sensible heat gain (H_{is}) due to air infiltration is:

$$H_{is} = 0.0183 \cdot V \cdot I \cdot (t_{od} - 78) \qquad (17)$$

where V = volume of conditioned space of building, ft^3

and I = rate of air infiltration, air changes/h.

Latent heat gain (H_{il}) due to infiltration is:

$$H_{il} = 80.7 \cdot V \cdot I \cdot (\omega_{od} - .012) \qquad (18)$$

If ω_{od} is less than or equal to 0.012, H_{il} should be taken to be zero.

4.8 Design Sensible and Latent Cooling Loads

As defined in sections 4.2 through 4.7, the design sensible heat gain (H_{ds}) in Btu/h is:

$$H_{ds} = H_w + H_q + H_r + H_{ps} + H_e + H_{as} + H_{vs} + H_{is} \qquad (19)$$

and the design latent heat gain (H_{dl}) is:

$$H_{dl} = H_{pl} + H_{al} + H_{vl} + H_{il} \qquad (20)$$

where H_w = heat gain through window

H_q = heat gain through walls

H_r = heat gain through roof

H_{ps} = sensible heat gain from occupant

H_e = heat gain from electric lighting
H_{as} = sensible heat gain from appliances
H_{vs} = sensible heat gain from mechanical ventilation
H_{is} = sensible heat gain from natural air infiltration
H_{pl} = latent heat gain from occupants
H_{al} = latent heat gain from appliances
H_{vl} = latent load due to mechanical ventilation
H_{il} = latent load due to natural air infiltration

The sum of the heat gains H_{ds} and H_{dl} is the total instantaneous heat gain to the building at design conditions. For the simplified calculation procedures presented herein, it is recommended that a conservative estimate for the design cooling load be taken as the instantaneous heat gain (H_{ds} + H_{dl}). The actual cooling load for the building will be lower than this estimated figure, because radiated heat gains (such as solar radiation through a window that strikes a floor surface) are not instantaneously converted into cooling loads. The storage of radiation heat gains causes some leveling off in the peak cooling load.

A simplified method for relating the instantaneous heat gain to instantaneous cooling load is outlined as follows:

Separate the instantaneous heat gain into convective and radiant heat gain by the values given in Table A-13. Consider the convective portion of the instantaneous heat gain as instantaneous cooling load. Consider the radiant portion of the instantaneous heat gain as reduced or averaged over a period of time by the thermal storage of the building. For lightweight construction, consider the instantaneous cooling as an average of the radiant instantaneous heat gain over a 2- to 3-h period up to and including the time of maximum load conditions. For heavy construction, consider the instantaneous cooling load as an average of the radiant instantaneous heat gain over a 6- to 8-h period up to and including the time of maximum load conditions. The total instantaneous cooling load is the sum of the convective portion and the time-integrated average of the radiant portion.

TABLE A-13. CONVECTIVE AND RADIANT HEAT GAIN TO COOLING LOAD [2]

Heat Gain Source	Radiant Heat, %	Convective Heat, %
Solar, without inside blinds	100	--
Solar, with inside blinds	58	42
Fluorescent lights	50	50
Incandescent lights	80	20
People	40	20
Transmission through walls and roof	60	40
Infiltration and ventilation	--	100
Machinery or appliances*	20-80	80-20

* The load from machinery or appliances varies, depending upon the temperature of the surface. The higher the surface temperature, the greater the radiant heat load.

5. ENERGY REQUIREMENT FOR SPACE HEATING

A bin method is outlined in the following sections for predicting the seasonal energy requirement for space heating. In general, steady-state heating loads for 5° F temperature increments (bins) are calculated. The cumulative energy requirement for each bin is then obtained by multiplying the heating load for the bin by the number of heating hours for the bin. The seasonal heating energy requirement is equal to the sum of the energy requirements for the separate bins. All outdoor air temperatures above 65° F are considered to have no heating season energy requirements. When the calculated heat generated within the building exceeds the building heat loss and raises the indoor temperature above 78° F, the calculation for the heating energy requirement shall be set equal to zero.

5.1 Heating Hours (T_h)

Total heating hours for 5° F temperature increments below 65° F are found in sections C and D, Chapter 6 of AFM 88-8 [1]. For locations other than those given in AFM 88-8 [1], weather data should be obtained from USAF/ETAC, Scott AFB, IL 62225.

5.2 Building Heat-Loss Rate (H)

Building heat-loss rate includes heat-loss rate through windows and opaque walls, floors, and roofs and heat loss due to infiltration and forced ventilation. Values for the 5° F temperature increments below 65° F are determined for the relation:

$$H = H_t \frac{(t_i - t_o)}{(t_i - t_{od})} \qquad (21)$$

H_t = design heat-loss rate from section 2.9, Btu/h

t_i = design indoor temperature (section 2.2), °F

t_{od} = design outdoor temperature (section 2.1), °F

t_o = mean of temperature increment (62, 57, 52, etc.), °F

5.3 Heat Generation Rate (H_g) within a Building

The heat generation rate within a building is due to heat release from lights, people, office, kitchen and cleaning appliances, motors, and heat loss from hot water heaters and piping. Since these sources of heat release are not operated on a continuous basis, the average heat release rate must be determined by using a diversity factor which is multiplied by the heat generation rate of the source. Diversity factors for deriving average heat release rates are presented below:

Lights	.25
Office appliances	.05
Kitchen appliances	.08
Cleaning appliances	.04
Hot water heaters and piping	.05
Motors	.1
People	.33

These diversity factors are presented only as guidelines. They were not derived from measured data nor are they based on a rigorous engineering analysis. They should be used only when more accurate information on building operation is unavailable. The diversity factors to be used in actual applications are for the most part a matter of judgment and depend on the occupancy and operation of a particular building. It is recommended that for calculating energy requirement for space heating, an average figure for the metabolic heat-release rate of 225 Btu/h per person be used.

5.4 Efficiency of Heating Plant (e_h)

Approximate part-load efficiencies as a function of the part-load fraction for a gas-fired forced-air furnace and a gas-fired boiler are given in figure A-1. Here the term "part-load fraction" is defined as the ratio of heating demand to rated heating plant output. The part-load efficiency for fuel-fired heating plants is seen to become less as the part-load fraction decreases. At low part-load fractions, the amount of time for which the heating plant does not operate and the number of furnace cycles increase. Both of these factors increase the amount of stored heat that is lost (to the flue) from the heat exchanger during periods when the heating plant is not operating.*

A procedure for calculating seasonal performance factors for unitary heat pumps is discussed in the Unitary Heat Pump Specification for Military Family Housing [6]. Part-load efficiency data for other types of heating plants are presently not available. A strong need exists to develop this data.

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* The present discussion assumes that heating plants are not equipped with automatic flue dampers which close when the burner is not operating.

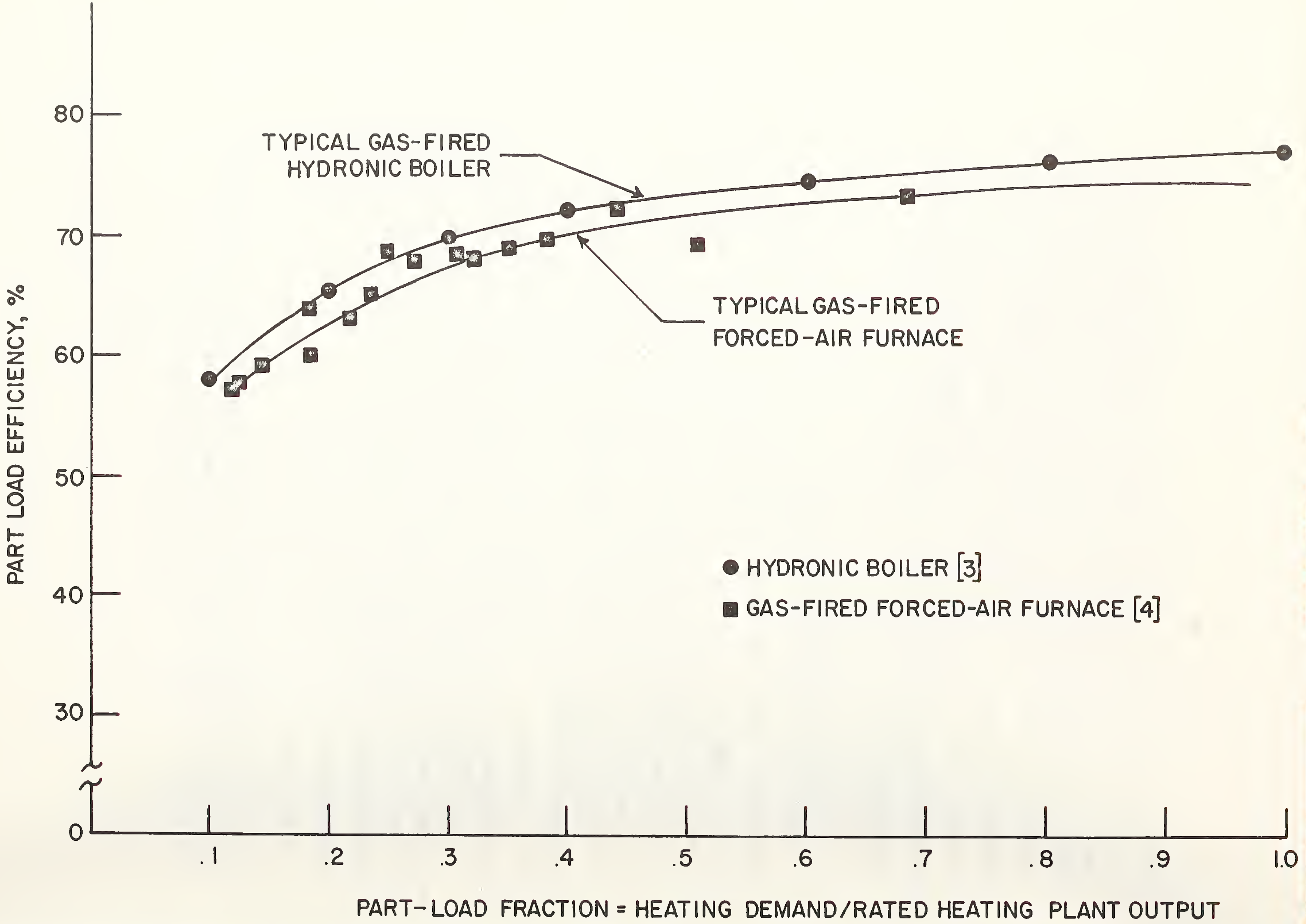

Figure A-1. Part-load efficiencies of various types of heating plants.

5.5 Correction (E) for Nighttime Setback of Thermostat

If the thermostat is set back from the indoor design temperature of 70° F for a period during the night, reductions in heating energy re-requirement are realized. For the various 5° F temperature increments, correction factors (E) are given in table A-14 for the variables of outdoor temperature, amount of setback, and duration of setback. With no nighttime setback, E = 1.

TABLE A-14. CORRECTION FACTORS (E) FOR NIGHTTIME SETBACK*

AVERAGE TEMP. OF 5° F INCREMENT	Duration, h					
	6		8		10	
	Setback, °F		Setback, °F		Setback, °F	
	5	10	5	10	5	10
62	.91	.88	.88	.85	.86	.82
57	.91	.88	.89	.85	.87	.82
52	.92	.89	.90	.86	.88	.84
47	.93	.90	.91	.87	.89	.85
42	.93	.91	.92	.88	.90	.86
37	.94	.92	.93	.89	.91	.87
32	.95	.95	.94	.90	.92	.88
27	.96	.93	.95	.91	.92	.89
22	.96	.94	.95	.92	.93	.90
17	.97	.95	.96	.93	.94	.91
12	.97	.96	.97	.94	.95	.92
7	.97	.96	.97	.95	.95	.93
2	.98	.96	.97	.95	.96	.94
-3	.98	.97	.98	.95	.97	.94
-8	.98	.97	.98	.96	.97	.95

* These values may not be applicable to buildings having complex HVAC system high internal heat generation.

5.6 Winter Solar Heat Gain (H_s) through Windows

The winter solar heat gain for each orientation is the product of window area, shading coefficient (see section 4.2), and winter solar heat gain factor as given in table A-15. The total solar heat gain through windows is the sum of the computations for each orientation multiplied by the ratio Y/1550 which is a function of geographic location. Values for Y are given in figure A-2.

TABLE A-15. WINTER SOLAR HEAT GAIN FACTORS, $Btu/h{\cdot}ft^2$

Orientation	Latitude 24	32	40	48
N	9	8	7	6
NE	16	14	11	9
E	37	33	29	25
SE	51	51	50	46
S	55	59	62	59
SW	51	51	50	46
W	37	33	29	25
HW	16	14	11	9
Horiz	66	56	45	34

5.7 Heating Energy Requirement

For the parameters defined in sections 5.1 through 5.4, the heating energy requirement (Q_i) in Btu for a particular 5° F temperature increment is given by the relation:

$$Q_i = E \cdot T_h \cdot (H - H_g - H_s)/e_h \qquad (22)$$

where E = correction factor for nighttime setback

T_h = number of heating hours for temperature increment

H = heat-loss rate at mean temperature of the temperature increment

Figure A-2. Values of Y [5].

H_g = internal heat released by lights, equipment, and occupants

H_s = solar heat gain through windows

e_h = efficiency of the heating plant

For each 5° F temperature increment, the total winter energy requirement (Q_o) in Btu for a building is equal to the energy needed for space heating plus the energy needed to operate the lighting and equipment, or

$$Q_o = Q_i + (H_g - H_p)\cdot T_h \tag{23}$$

Here H_p is the heat released by the occupants. To obtain the total energy requirement over the entire winter season, equation (23) is evaluated for each of the 5° F temperature increments. The resulting values are then summed to give the total winter season energy requirement. The foregoing procedure for calculating the winter energy requirement is illustrated with the following sample problem:

Sample Problem

For a residence located in Columbus, Ohio (latitude 40°) the design heat-loss rate at 4° F (Chapter 6 of AFM 88-8 [1]) is H_t = 80,000 Btu/h. It is required to find the heating energy requirement, the energy needed to operate lighting and appliances, and the total energy consumption by the building during the winter heating season.

The calculation is carried out in table A-16. The 5° F temperature increments, the average temperature of the increments, and the total heating hours for the increments for a typical winter at Columbus, Ohio are given in columns 1, 2 and 3, respectively. These data were obtained from Chapter 6 of AFM 88-8 [1]. Equation (21) was used to calculate the heat-loss rate for the building at the mean outdoor air temperature for each 5° F temperature increment. These heat-loss rates are tabulated in column 4. The sum of internal heat generation (H_g) and solar heat gain through windows (H_s) is given in column 5. The net heat demands for the increments ($H - H_g - H_s$) are tabulated in column 6. The calculation of internal heat generation (H_g) and winter solar gain through windows (H_s) is carried out in tables A-17 and A-18. The part-load fractions of the heating plant for

TABLE A-16. CALCULATION OF WINTER ENERGY REQUIREMENTS

1	2	3	4	5	6	7	8	9	10
Temp. Increment °F	Mean Temp. °F	Heating Hours	Heat Loss Rate 1000 Btu/h	Internal Heat Gains 1/ 1000 Btu/h	Heating Demand 2/ 1000 Btu/h	Part Load Fraction	Efficiency 3/ of Heating Plant	Correction Factor for Nighttime Setback	Heating Energy Requirements 10^6 Btu/h
60/64	62	804	9.70	11.71	-	-	-	-	-
55/59	57	792	15.76	11.71	4.05	.04	.41	.85	6.650
50/54	52	668	21.82	11.71	10.11	.10	.51	.86	11.388
45/49	47	650	27.88	11.71	16.17	.16	.56	.87	16.329
40/44	42	672	33.94	11.71	22.23	.22	.61	.88	21.551
35/39	37	705	40.00	11.71	28.29	.28	.64	.89	27.735
30/34	32	689	46.06	11.71	34.35	.34	.66	.90	32.273
25/29	27	477	52.12	11.71	40.41	.40	.68	.91	25.795
20/24	22	230	58.18	11.71	46.47	.46	.68	.92	14.460
15/19	17	98	64.24	11.71	52.53	.53	.70	.93	6.839
10/14	12	51	70.30	11.71	58.59	.59	.71	.93	3.914
5/9	7	18	76.36	11.71	64.65	.65	.72	.94	1.519
0/4	2	10	82.42	11.71	70.71	.71	.73	.95	.920
-5/-1	-3	9	88.48	11.71	76.77	.77	.74	.95	.982
-10/-6	-8	3	94.54	11.71	82.83	.83	.74	.95	.319
-15/-11	-13	2	100.61	11.71	88.89	.89	.75	.96	.228
Total		5878						Total	170.902×10^6

Seasonal Energy for Heating Device = 170.902×10^6

Heat generation Energy = $5.878 \times 10^3 \times (7.83-.37)$ 4/ $\times 10^3$ = 43.850×10^6

Total Energy Required = 214.752×10^6 Btu

1/ Includes heat gains from lighting, equipment, occupants, and transmitted solar radiation.

2/ Does not include effect of nighttime setback of the thermostat which is taken into account by correction factor given in column 9.

3/ Values derived from measured part-load efficiency data.

4/ Values taken from table A-17.

the 5° F temperature increments are tabulated in column 7. Part-load efficiencies (e_h) and correction factors (E) for nighttime setback of the thermostat are given in columns 8 and 9, respectively. Equation (22) was used to calculate the cumulative heating energy requirements for each 5° F temperature increment. Heating energy requirement for each 5° F temperature increment is given in column 10. And finally, the net annual heating energy requirement is found by summing the values of column 10.

The total energy consumed by the building during the heating season is equal to the sum of the energy required for space heating and the energy required for lighting and equipment (see calculation given under the tabulated values).

TABLE A-17. CALCULATION OF INTERNAL HEAT GENERATION (H_g), Btu/h

1.	Lights, 340 watts x 3.412 x .25	= 3275.5
2.	Kitchen Appliances 6000 watts x 3.412 x .08	= 1637.8
3.	Cleaning Appliances 4000 watts x 3.412 x .04	= 546.
4.	Hot water Heater and Pipes 400,000 x .05	= 2000.
5.	Metabolism 5 persons x 225 x .33	= 371.
		7830. Btu/h

TABLE A-18. CALCULATION OF WINTER SOLAR GAIN THROUGH WINDOWS (H_s), Btu/h

	Area	SHGF	Shading Coefficient	
N	80	7	.56	313.6
E	70	29	.59	1197.7
S	90	62	.83	4631.4
W	70	29	.59	1197.7
				7340.4

From figure A-2, $Y = 820$

$$H_s = 7340.4 \times 820/1550 = 3883.3 \text{ Btu/h}$$

$$H_g + H_s = 11.713 \times 10^3 \text{ Btu/h}$$

6. ENERGY REQUIREMENT FOR SPACE COOLING

The methodology for determining the cooling season energy requirement includes the building sensible and latent heat gains, heat generated within the building, efficiency of cooling plant, and solar heat gains through walls, roofs and fenestration areas. Energy requirement for space cooling will be taken to be zero, when the outdoor air temperature is lower than 65° F. In addition, the cooling energy requirement is taken to be zero when the indoor temperature falls below the design indoor temperature level, i.e., no heating or cooling is called for in the indoor temperature range from 71° to 77° F.

6.1 Cooling Hours (T_c)

Total cooling hours for 5° F temperature increments, as well as coincident wet-bulb temperatures, may be found in AFM 88-8 [1]. For locations not found in the above, weather data should be obtained from USAF/ETAC , Scott AFB, IL 62225.

6.2 Conductive Heat Gain (H_c)

Exclusive of the solar heat gains, the conductive heat gain (H_c) through the exterior envelope of a building due to air-to-air temperature difference may be determined from the relation:

$$H_c = (A_w \cdot U_w + A_q \cdot U_q + A_r \cdot U_r + M \cdot A_f \cdot U_f) \cdot (t_o - t_i) \qquad (24)$$

where A = area, ft^2

U = thermal transmittance, $Btu/h \cdot ft^2 \cdot °F$

t_o = mean temperature of increment, °F

t_i = indoor design temperature, °F

and subscripts w, q, r, f, denote windows, opaque walls, roof, and floor, respectively. For floors, values for the factor (M) are given below:

slab-on-grade floors,	M = 0.2
floor over crawl space,	M = 0.8
non-cooled basements,	M = 0.3
cooled basements,	M - 0.7

6.3 Summer Solar Heat Gain through Windows and Walls (H_w)

The solar heat gain through windows and walls (H_w) defined for each orientation by the relation:

$$H_w = Y \cdot SHG \cdot [SC \cdot A_w + 0.18 \cdot A_q \cdot U_q] \tag{25}$$

SHG = summer solar heat gain from table A-19, Btu/h·ft^2

SC = shading coefficient (from tables A-5 through A-8)

Y = $Y'/2{,}400$ (from figure A-3)

where A_w, A_q and U_q are defined in section 6.2. The total solar heat gain (H_s) is the sum of the values calculated for each wall orientation of the building.

6.4 Solar Heat Gain through Roofs (H_r)

The solar heat gain (H_r) through roofs is defined by

$$H_r = Y' \cdot \cos^2(\phi)\ [0.18 \cdot SHG_h - 7] \cdot A_r \cdot U_r \tag{26}$$

where SHG_h is solar heat gain for a horizontal surface as given in table A-19, ϕ is the roof-pitch angle, Y' is as defined in figure A-3, and A_r and U_r the surface area and thermal transmittance of the roof, respectively.

TABLE A-19. SUMMER SOLAR HEAT GAIN (SHG) FOR VERTICAL AND HORIZONTAL SURFACES, Btu/h·ft^2

	LATITUDE				
ORIENTATION	24	32	40	48	56
N	20	18	17	17	18
NE	37	35	33	32	32
E	46	47	48	50	51
SE	33	39	45	51	56
S	17	23	31	41	51
SW	33	39	45	51	56
W	46	47	48	50	51
NW	37	35	33	32	32
HORIZ	89	89	87	84	78

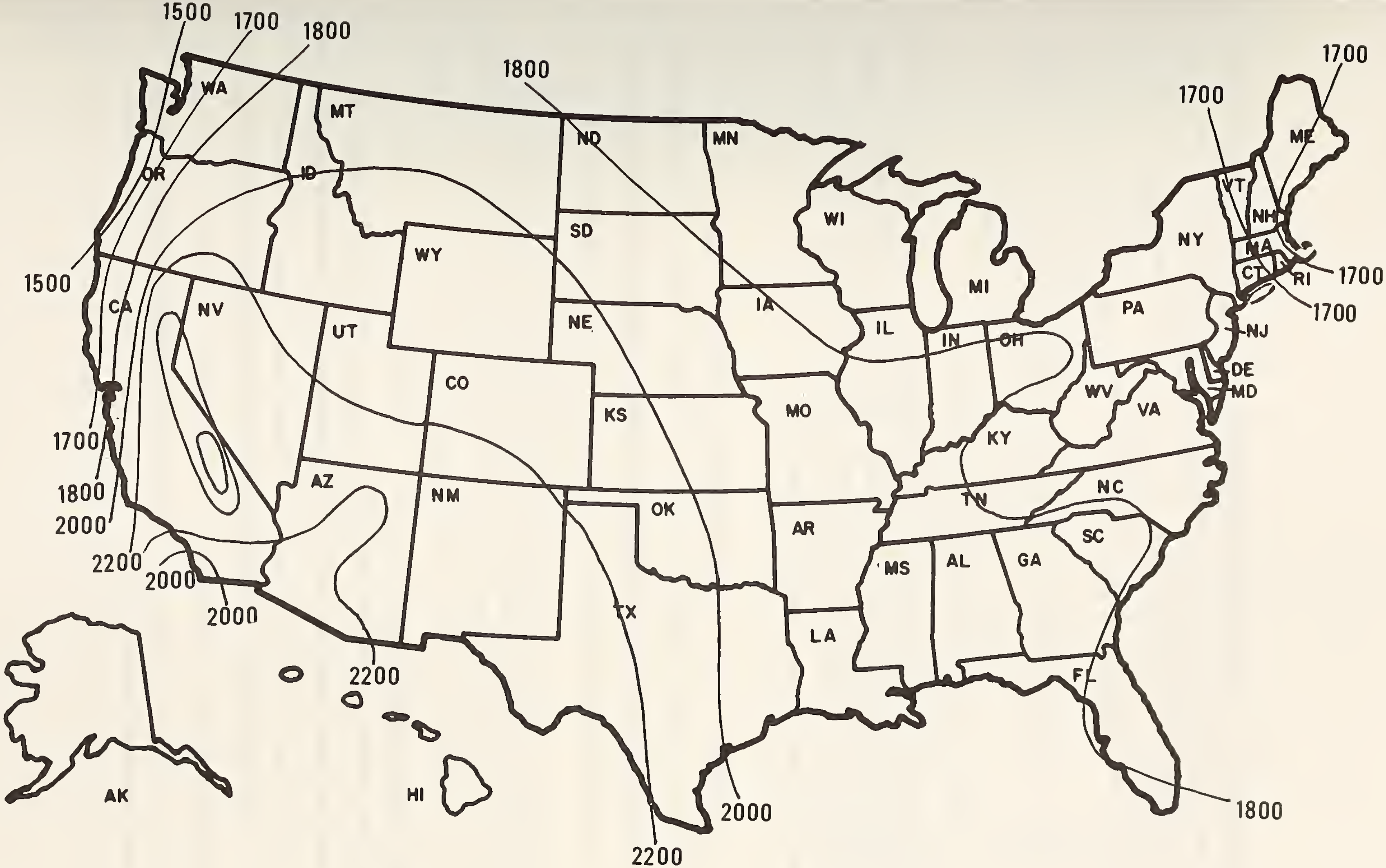

Figure A-3. Values of Y' [5].

6.5 Heat Gain due to Air Infiltration and Mechanical Ventilation (H_v)

The heat gain due to air infiltration and outside air introduced by mechanical ventilation is separated into the sensible and latent loads. The sensible load (H_{vs}) from these sources is:

$$H_{vs} = (0.0183 \cdot V \cdot I + 5.5 \cdot U_p \cdot P) \cdot (t_o - t_i) \tag{27}$$

V = volume of conditioned space of building, ft^3

I = rate of air infiltration, air changes/h

P = number of occupants, $P = 0$ if no mechanical ventilation

U_p = diversity factor for people (see table A-20 of section 6.6).

The latent load (H_{vl}) from these sources is

$$H_{vl} = (80.7 \cdot V \cdot I + 24{,}210 \cdot U_p \cdot P) \cdot (\omega_o - .012) \tag{28}$$

ω_o = humidity ratio at coincident wet-bulb temperature for the 5° F temperature increment (see section 6.1), lb water/lb dry air

When the humidity ratio is less than 0.012, the latent load is set equal to zero.

6.6 Internal Heat Gains (H_g)

The heat generated within buildings is due to lights, people, office, kitchen and cleaning appliances, motors, and from hot water heaters and piping. The heat generated from these sources does not occur on a continuous basis. To obtain the average heat release rate for an internal source, multiply the heat release rate for the source by a diversity factor. Diversity factors for various sources of heat release are given in table A-20.

TABLE A-20. DIVERSITY FACTORS (U_p)

	Residential	Office
People	.5	.25
Lights	.2	.2
Office equipment	-	.05
Kitchen equipment	.1	.08
Cleaning equipment	.06	.04
Hot water heater and piping	.05	.05
Motors	.02	.05

These diversity factors are approximate values based on engineering judgment and should be used only when more accurate estimates are not available. Diversity factors are a matter of judgment and depend significantly on the type of occupancy and the operation of a particular building.

For calculating the energy requirement for space cooling, it is recommended that average figures for the metabolic rates be taken as 225 Btu/h for sensible heat and 105 Btu/h for latent heat.

6.7 Efficiency of Cooling Systems (E)

The efficiency of cooling systems depends upon the coefficient of performance (COP) of the cooling plant, the energy required to pump working fluids, the method of temperature control to conditioned spaces and the method of heat rejection. The efficiency of air conditioning systems varies over a wide range and depends upon the system type, the outdoor temperature conditions, and the way in which it is operated. It is recommended that manufacturers' seasonal-efficiency values be used unless measured performance data are available.

6.8 Cooling Energy Requirement

The sensible cooling load (H_s) for a particular 5° F temperature increment in equal to the sum of the heat gains to the occupied space, or

$$H_s = H_c + H_w + H_r + H_{vs} + H_{gs} \tag{30}$$

where H_c = conductive heat gain through exterior building envelope due to air-to-air temperature difference

H_w = solar heat gain through walls and windows

H_r = solar heat gain through roof

H_{vs} = sensible load due to air-exchange between indoor and outdoor air (includes both natural air infiltration and mechanical ventilation).

H_{gs} = sensible heat release by internal sources such as lights, occupants, and equipment.

The latent load (H_l) for a particular 5° F temperature increment is equal to the sum of the latent load due to the air exchange between indoor and outdoor air and the latent load due to moisture release from internal sources, or

$$H_l = H_{vl} + H_{gl} \tag{31}$$

where H_{vl} = latent load due to the influx of outdoor air (includes both natural air infiltration and mechanical ventilation)

H_{gl} = latent load due to moisture release from internal sources.

For a particular 5° F temperature increment, the total heat gain is equal to:

$$Q = T_c \cdot (H_s + H_l) \tag{32}$$

where T_c = number of cooling hours for the temperature increment.
The total heat gain during the summer cooling season is equal to the sum of the heat gains for the various 5° F temperature increments.

The total energy consumed by the building for space cooling during the summer cooling season is equal to the total heat gain divided by the seasonal efficiency (E) of the cooling plant. The total energy consumed by the building is equal to the energy consumed by the cooling plant, plus the energy consumed by the lights and equipment $[T_c \cdot (H_g - H_p)]$, where $(H_g - H_p)$ is the rate of energy consumption for lighting and equipment and T_c is the number of cooling hours for the summer cooling season.

Sample Problem

It is required to find the energy requirement for space cooling and energy consumption by the lighting and equipment during the cooling season for a three-story office building located at Robins AFB, Georgia. The dimensions of the building are 80 ft long, 40 ft wide, and 33 ft high. The thermal transmittances of the windows, opaque walls, roof, and floor are 1.06, 0.15, 0.15, and 0.08 Btu/h·ft^2·°F, respectively. Normal occupancy is 100 persons. The mechanical cooling system has seasonal efficiency of 2.09. The calculation is carried out below.

A. From Chapter 6 of AFM 88-8 [1], the total cooling hours for the applicable 5° F temperature increments for Robin AFB, Ga. were obtained and inserted in column 2 of table A-21. The average temperature and the mean coincident wet-bulb temperatures for the various 5° F temperature increments are given in columns 1 and 3, respectively.

B. Based on the methodology presented in section 6.3, the calculation for solar heat gain (H_w) through the various walls and windows is carried out below:

Orientation	Type	Area ft^2	SC or .18 U_q	SHG	Heat Gain, Btu/h
N	window	660	.80	18	10466
	wall	1980	.027		
E	window	132	.51	47	4672
	wall	1188	.027		
S	window	660	.51	23	8971
	wall	1980	.027		
W	window	132	.51	47	4672
	wall	1188	.027		
					28,781 Btu/h

$$H_w = \frac{1800}{2400} \times 28,781 = 21,586 \text{ Btu/h}$$

The windows on the north and south exposures were assumed to be 25 percent of the wall area, and 10 percent of the wall area for the east and west exposures of the building.

TABLE A-21. CALCULATION OF TOTAL HEAT GAIN TO THE BUILDING OVER THE SUMMER COOLING SEASON

1	2	3	4	5	6	7	8	9	10
Properties for Temp. Increments									
Mean DB Temp. °F	Cooling Hours	Mean WB Temp. °F	Total Heat Transmission Btu/h	H_v Btu/h	Rate of Total Sensible Heat Gain Btu/h	$(\omega_o - 0.012)$ lb water per lb dry air	Rate of Total Latent Heat Gain Btu/h	Rate of Total Heat Gain Btu/h	Total Heat Gain 10^6 Btu
102	3	76	91,404	25,272	143,142	.0013	8,651	151,799	.455
97	59	76	77,084	20,007	123,557	.0025	14,226	137,783	8.129
92	274	74	62,764	14,742	103,972	.0019	11,441	115,413	31.623
87	524	71	48,444	9,477	84,387	.0007	5,873	90,260	47.296
82	651	69	34,124	4,212	64,802	.0002	3,553	68,355	44.499
77	1003	67	19,804	-1,053	45,217	0	2,625	47,842	47.985
72	1300	65	5,484	-6,318	25,632	0	2,625	28,257	36.734
67	932	62	-8,836	-11,583	6,047	0	2,625	8,672	8.082
	4746								224.803 x 10^6 Btu

C. Using the methodology of section 6.4, the solar heat gain (H_r) through the roof is

$$H_r = \frac{1800}{2400} (.18 \times 89 - 7) \times 3200 \times .05 = 1,082 \text{ Btu/h}.$$

D. Using the methodology of section 6.2, the total conduction heat gain through the building envelope is found to be

$$H_c = 2,864 \cdot (t_o - t_i).$$

Here 2,864 is the sum of the U·A products for the various elements of the exterior building envelope. The total heat transmitted (H) through the building envelope is equal to the conduction heat gain due to air-to-air temperature difference and solar heat gain through walls, windows, and the roof, or

$$H = H_c + H_w + H_r$$
$$= 22,668 + 2,864 \cdot (t_o - t_i)$$

Values for H were calculated for the various temperature increments and are tabulated in column 4 of table A-21.

E. From section 6.5, the sensible heat gain due to air exchange between the indoor and outdoor air is given by the relation:

$$H_{vs} = 1,053 \cdot (t_o - t_i)$$

Values are computed and entered in column 5 of table A-21.

F. The sensible heat gain (H_g) from internal sources is

People	100 x .25 x 225	5,625
Lights	2.5 watts/ft^2 x 889^2 .2 x 3.412	15,169
Office Appliances	5000 watts x .05 x 3.412	853
Cleaning Appliances	6000 watts x .04 x 3.412	819
Hot water heat and piping losses	80,000 x .05	4,000
		26,466 Btu/h

The total sensible heat gain from internal sources is seen to be 26,466 Btu/h. The total sensible heat gain to the occupied space is equal to the sum of the heat gains from internal sources (26,466 Btu/h), the heat gain due to total heat transmission (column 4), and the heat gain due to air exchange between indoor and outdoor air (column 5). The total sensible heat gains for the various temperature increments are tabulated in column 6 of table A-21.

G. Using the psychrometric chart, the humidity ratio (ω_o) for the dry-bulb temperature (column 1) and the wet-bulb temperature (column 3) was determined for each 5°F temperature increment. The difference in humidity ratio (ω_o -.012) was subsequently determined and entered in column 7 of table A-21. When the difference (ω_o -.012) was less than zero, a value of zero was entered in the table. Equation (28) was then used to calculate the total latent load, or

$$H_L = 4.64 \times 10^6 \cdot (\omega_o - .012) + (100) \cdot (0.25) \cdot (105)$$

The first term is due to moisture entering the building by way of air exchange between the indoor and outdoor air, whereas the second term is due to moisture released by the occupants. Values of total latent load for the 5° F temperature increments are entered in column 8 of table A-21. The rate of total heat gain for each 5° F temperature increment is equal to the sum of the rate of total sensible heat gain (column 6) and the rate of total latent heat gain (column 8). Values for the rate of total heat gain are tabulated in column 9.

H. The total heat gain for each 5° F temperature increment is obtained by multiplying the number of cooling hours (column 2) by the rate of total heat gain (column 9). These values are tabulated in column 10. The total heat gain to the building during the summer cooling season is obtained by summing the values of column 10 and is found to be 224.8×10^6 Btu.

I. The total energy requirement for space cooling is

$$Q = \frac{224.803 \times 10^6}{2.09} = 107.6 \times 10^6 \text{ Btu}$$

J. The energy consumed by internal lighting and equipment over the summer cooling season is

$$T_c \cdot (H_g - H_p) = 4746 \cdot (20{,}841) = 98.9 \times 10^6 \text{ BTU}$$

K. The total energy consumption by the building over the summer cooling season is simply equal to the sum of the energy requirement for space cooling and for lighting and equipment, or 206.5×10^6 Btu. This is equivalent to 60,500 kilowatt hours.

REFERENCES

1. Engineering Weather Data, Air Force Manual 88-8, Chapter 6, Department of the Air Force, June 15, 1967.

2. ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1972.

3. Computer Simulation of Fossil-Fuel-Fired Hydronic Boilers, J. Chi, Proc. 2nd ASHRAE/NBS Conference on HVAC Equipment and Systems, Purdue University, Lafayette, Indiana, April, 1976.

4. The Thermal Performance of a Two-Bedroom Mobile Home, G. T. Teitsma and B. A. Peavy, NBSIR 76-1182, National Bureau of Standards, January, 1977.

5. Intermediate Minimum Property Standards for Solar Heating and Domestic Hot Water Systems, NBSIR 77-1226, National Bureau of Standards, 1977.

6. Unitary Heat Pump Specification for Military Family Housing, C. W. Phillips, B. A. Peavy, and W. J. Mulroy, NBSIR 76-1029, National Bureau of Standards, 1976.

APPENDIX B. SOLAR ENERGY SYSTEMS FOR AIR FORCE APPLICATIONS

CONTENTS

APPENDIX B. SOLAR ENERGY SYSTEMS FOR AIR FORCE APPLICATIONS

1. INTRODUCTION

In the previous chapters of this handbook, steps have been outlined where more or less conventional methods can be adopted to conserve energy in existing buildings on Air Force installations. This appendix has been included to provide guidelines for when consideration should be given to the use of solar energy systems in buildings.

The sun is the source of practically unlimited energy, most of which is now wasted. For example, the total solar energy reaching the ground in the continental 48 States averages over 500 times the present United States energy consumption. In the early 1970's, with the emphasis on finding solutions to the nation's impending energy problems, a special panel of experts met at the request of the President's Office of Science and Technology and identified three broad applications for solar energy as most promising [1]*:

a. The heating and cooling of residential and commercial buildings,

b. The generation of electricity, and

c. The chemical and biological conversion of organic materials to liquid, solid, and gaseous fuels.

The amount of research being done on these applications is growing at an ever increasing rate both in the public and private sectors. The first application, the space heating and cooling of buildings including heating of domestic hot water, is the one closest to being used on a wide scale, and the only one that should be considered at the present time for use in and around Air Force buildings.

The principal obstacle to the use of solar energy in buildings is an economic one. There can be substantial energy and dollar savings through the use of the kinds of systems to be described in this chapter; however, there is also a substantial investment involved in purchasing

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* Numbers in brackets indicate references listed at the end of this appendix.

and installing the equipment. The decision to be made, then, is one of deciding whether the savings in fuel costs over a specified lifetime will justify the initial investment.

Another secondary obstacle to the use of solar heating and cooling is the uncertainty associated with the durability and maintenance problems that are likely to occur. Considerable experience has been gained over the last several years in the construction and operation of these systems. Reference [2] describes 220 buildings that have been built or are currently under construction, which use some form of solar heating and/or cooling. Most of the buildings in reference [2] have been built in the United States and in many cases reports are available describing the operating experience. As early as 1923, solar energy systems were used for heating domestic hot water in Florida and southern California. Reference [3] describes an analysis of the solar water heating industry in Florida during the period 1923-1974. It includes a thorough description of the failures and successes that occurred through the use of various kinds of components. As a result of the vast amount of experience obtained and documented in recent years, the design and installation practice is developing so that it is possible to construct durable systems having a lifetime equal with the conventional systems.

The remainder of this chapter is divided into three sections. In section 2, a description is given of the typical kinds of systems to be considered, as well as their major components. In section 3, methods are outlined for estimating what kind of fuel savings to expect from these systems. In section 4, a simplified method is given for performing an economic analysis and using the expected fuel savings to determine whether or not a system is likely to be economical. If more detailed information is required than is supplied in this chapter, it is recommended that references [4-6] be consulted.

2. DESCRIPTION OF TYPICAL SOLAR ENERGY SYSTEMS AND COMPONENTS

Systems for converting the solar energy into useful thermal energy for space heating, cooling, or heating of domestic hot water can be classified as either active or passive. In an active system, the flow of thermal energy into the building is controlled through the use of auxiliary equipment such as pumps, valves, and heat exchangers. In the passive systems, the energy flow is automatic, requiring little or no auxiliary equipment as in the case of the active systems. In most of the passive systems, either a south wall or a flat roof is designed so that the structural unit (or a part of it) collects and stores the thermal energy. Examples are a massive concrete wall, painted black and covered with a transparent surface; or stationary roof ponds/water beds covered with movable insulation. Experiments have been conducted on these types of systems and similar concepts and it appears that they are the most promising from an economical viewpoint in certain regions of the country. However, in almost all cases, the passive system must be designed as an integral part of the building. Therefore, only the active systems will be addressed in this document since they are in many cases easily retrofitted into existing buildings.

Figure B-1 is a schematic representation of an active solar space heating and domestic hot water system. A fluid is pumped through a solar collector where the incident solar radiation heats the fluid, and the thermal energy is transferred to a storage unit through a heat exchanger. In turn, the resulting stored energy is transferred through additional heat exchangers either to heat domestic hot water or to heat the building space. The system can be made more complicated by the inclusion of additional equipment that would also use the thermal energy in the storage tank to provide either space cooling or space heating by some other means than through the heat exchangers as shown.

Methods of using the thermal energy to provide space cooling include the use of absorption and Rankine-cycle refrigeration units. In the absorption units, the energy from thermal storage is used as a heat source to boil refrigerant from a solution. The refrigerant subsequently absorbs energy from the space to be cooled. In the Rankine-cycle units, the thermal energy is used to drive a heat engine which is coupled to a conventional vapor-compression-cycle refrigeration unit. Detailed information can be obtained from references [7, 8] on these and other methods of solar cooling. Systems for solar cooling have been put into operation, but in general, they are even less competive with conventional equipment (on an economic basis) than either solar domestic water heating or space heating.

An alternate method of providing space heating using solar energy is to integrate a heat pump into the system of figure B-1. This can be done in several ways and can involve air-to-air, air-to-water, or water-to-water vapor-compression-cycle heat pumps. If the building is heated entirely by the heat pump and the source side of the heat pump is exclusively the thermal storage unit, it is called a solar-boosted heat pump. If a more flexible yet more complicated system is designed, where the heat exchanger shown in figure B-1 is used as well as the heat pump, and the source side of the heat pump can be either the thermal storage unit or the outside air, depending upon operating conditions, the system is designated as a solar-assisted heat pump. More detailed information on these as well as other heat pump concepts is included in reference [9].

The remainder of this chapter will discuss the applicability of the type of system in figure B-1. In general, one should find that for most localities the heating of domestic hot water will be the most economically competitive use of solar energy, followed by space heating. If space heating, using the concept in figure B-1, appears to be worth considering for certain types of buildings, and a detailed analysis is subsequently conducted, consideration should also be given to systems employing heat pumps.

2.1 Active Systems

The types of active systems commercially available today are divided into those that use water or a water-antifreeze solution as the basic heat transfer fluid, and those that use air.

For the systems that use water or a water-antifreeze solution, they are configured basically as shown in figure B-1. The storage unit is usually one or more large water tanks. The heat exchanger between the collector and storage tank could be eliminated if the system were located in an area where freezing would not occur, or if provisions were made to drain the solar collector at night. There is usually a pump in each loop of the system and the pumps are controlled depending upon the available solar radiation and the demand for either hot water or space heating within the building. A special case of this system is one which only heats domestic hot water. Since this is generally the system that is most economical and should be considered first, it will be discussed in more detail in section 2.2.

Figure B-2 shows a schematic diagram of a system which uses air as the basic working fluid. Only one fan is used; the air flow is through ducts, and it is controlled and directed to the various components through the use of motorized dampers. For example, when solar energy is available for collection and there is a space heating load, room-temperature air is drawn through the solar collectors, heated, and returned to the building. When solar energy is available for collection and there is no space heating load, air is drawn from the bottom of the storage unit (which is most often a packed bed of rocks or pebble bed), heated and returned to the top of the storage unit. When there is a space heating load and no solar energy can be collected, hot air is drawn from the top of the pebble bed and blown into the house while room temperature air is returned to the bottom of the pebble bed. In contrast to the water-system, there is no need for a heat exchanger between the collector and the storage unit and between the storage unit and the building space. An air-to-water heat exchanger is usually used for heating the domestic hot water.

In both types of systems, the auxiliary energy source for both space heating and domestic water heating would in most cases be the conventional equipment now installed. The components would be compatible with almost any type of energy distribution system installed in the existing buildings; however, depending upon the type of installed system, one might tend to select specific types of solar components. For example, an installed system using radiators or fan-coil units for space heating would be more compatible with a water-type system than with one using air collectors.

The primary consideration in comparing air and liquid systems involves their life-cycle costs. Specifically, this is the annual total cost of owning and operating the system. The cost of ownership includes the sum of annual depreciation of the equipment based on its total useful life and an interest charge on the undepreciated equipment value. The operating costs include maintenance, repairs, and the cost of auxiliary energy for both heating and cooling as well as circulating the system fluids.

Although complete information on total installed costs of liquid and air systems are not yet definitive, there do not appear to be large differences. Since both use only small amounts of auxiliary energy for circulating fluids and have comparable overall thermal efficiencies, the major differences appear to be in durability and maintenance. Since the air system is permanently dry, no significant corrosion of the collector absorber occurs. There is no possibility of freezing the collector fluid, so decomposable antifreeze compounds or other organic liquids are not required. Boiling cannot occur, so replacement of collector fluid is never needed. Fluid leakage, although not precluded, involves minimal corrective cost and no secondary damage. Some long-term operating experience with an air system shows that maintenance is not a significant expense and that the equipment can be expected to have a life comparable to that of the building itself [10].

In October of 1975, the Energy Research and Development Administration (ERDA) published the results of a survey made to determine the availability of solar heating and cooling products [11]. The report is simply a compilation of responses to questionnaires that were sent to all known manufacturers of solar components. Since the publication of reference [11], ERDA has conducted a somewhat comprehensive analysis of the information that 85 different organizations voluntarily submitted on 130 systems. Through the analysis, 36 different systems were found technically acceptable for use in federally-sponsored demonstration programs [12]. The basis for the evaluation included adequacy of design, operating experience and verification of performance, compliance with applicable interim performance criteria, warranty, and manufacturing capability. It is expected that this evaluation will take place periodically and that the results will be a good source of information to be used in selecting possible components/systems.

2.2 Domestic Hot Water Heating Systems

The heating of domestic hot water with solar energy is the most economically viable application and the one that should be considered for all locations within the United States. There are several reasons for this. One is the low capital cost relative to solar space heating or cooling systems and the others relate to the unique characteristics of domestic water heating.

The demand for hot water is relatively constant throughout the year. This means that the collector and other parts of the system produce energy for long periods of time and consequently can result in large savings that eventually must pay for the higher initial cost of the system. This is in contrast to a solar space heating system which is fully operational only during the coldest months of the heating season.

A solar water heating system can also be sized more closely ɔ the demand. While space heating systems have to handle extreme loads only a few days of the year, they still have to be sized large enough (including the auxiliary back-up) to meet those extremes. A water heating system, on the other hand, will have roughly the same load day in and day out. Except in unusual applications, the design load (expected maximum load) will be close to the normal daily load. By avoiding the problems of widely fluctuating loads, the solar water heating systems can be much more economical.

A problem common to all types of solar heating is the variable nature of sunshine. Solar water heaters, however, can often have an additional advantage over space heating systems because the requirements for hot water are less rigid than those for space heating. If there is an extended sunless period and the user is extremely conscious of energy and cost savings, some uses of hot water can be temporarily postponed. This, of course, is not possible with space heating systems.

This curtailment of hot water use is of course not necessary due to the way the systems are normally designed. Figure B-3 shows a schematic of a typical system [13]. The collector, heat exchanger, and pumps are used to heat water in a storage tank, which in turn is fed to the conventional or already existing water heater in the building. The already existing heater only has to make up the difference between the incoming water temperature from the solar storage tank and the constant supply temperature to the user. The controls in such a system are very much simpler than those required by most solar space heating systems. The thermostat in the already existing heater controls the auxiliary energy supply, as always. The only other controls needed are the sensor and differential thermostat shown to turn the pumps on and off.

More will be said concerning the economics of these systems in section 4. However, it should be noted that many manufacturers are now offering solar water heating packages for residential buildings which include everything shown in figure B-3, except the conventional water heater, for approximately $1000 [14].

2.3 Solar Collectors

The cross-section of a typical flat-plate solar collector used to heat a liquid is shown in figure B-4 [15]. It consists of a dark-painted metal absorbing plate through which the fluid passes, one or two sheets of some transparent material on the top through which the solar energy passes, and several inches of insulation on the back. The transparent covers and the insulation both serve to eliminate heat loss from the metal absorbing plate and to maximize the amount of the energy being carried away by the fluid. The collector is usually facing south ($\pm$ 20°) and tilted at an angle designed to optimize the amount of solar energy received over a complete heating season*.

Typical metal absorbing plates are made of copper, aluminum, and steel. Generally speaking, for liquid systems copper is preferred, because of superior corrosion resistance and durability. However, it is expensive and usually results in a heavy collector. Aluminum has been used a great deal over the past several years because of its relative low cost, light weight, and ease of fabrication. However, use of aluminum absorbers in a liquid system requires special precautions to prevent corrosion. Steel has had some limited use because of its low cost but it is difficult to fabricate, is heavy, and like aluminum requires corrosion protection. Plastics have been used in isolated cases for the absorber, but usually only for relatively low temperature applications such as heating swimming pools. Whenever air collectors are being used, the primary criteria for material selection for the absorber is cost, since there is no corrosion. Figure B-5 [6] shows the cross-section of three different types of air collectors.

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* This is approximately the latitude + 10°.

The primary source of energy loss from the solar collector is convection and radiation heat transfer from the top of the absorber to the cover plate and subsequent heat transfer away to the environment. A variety of techniques are used to limit this heat loss. The use of double glazing is the most straightforward and widely used technique and can reduce the heat loss as much as 40-50 percent when the collector is operating at a temperature as high as 190° F above the ambient and the absorber is painted with an ordinary black paint. Other techniques include the use of a selective coating on the absorber to reduce re-radiation; a transparent honeycomb insert between the absorber and the cover plate assembly to limit air motion and decrease convection heat transfer; some form of concentrator incorporated into the design to concentrate the energy onto the absorber; evacuating the space between the absorber and the cover-plate assembly to eliminate convection altogether; or some combination of the above. The collectors being built today which have an evacuated space usually are of tubular construction where the absorber is a metal or glass tube and the cover plate is a glass cylinder surrounding the absorber. Reference [16] should be consulted for a thorough description of the various types of collector configurations being designed and built.

Although some very encouraging results have been obtained in the last few years in the solar collector field, it is very difficult to economically justify any collector other than the simple single- or double-glazed flat-plate collector for domestic water heating or space heating. Most of the new designs result in significant increases in efficiency at the high temperatures required for air conditioning (190-240° F); however, the improvement at the lower operating temperatures for domestic water heating and space heating (120-150° F) in most cases cannot justify the additional investment. Figure B-6 is a plot of solar collector efficiency versus the difference in temperature between the absorber and the ambient air divided by the incoming radiation flux for some typical collectors [6]. The efficiency is defined as the percentage of the incoming radiation normal to the collector surface that can be converted to thermal energy and carried away by the heat transfer fluid. As can be seen, for operating conditions typical of domestic water heating and space heating ($\Delta t/I \leq 0.3$), the efficiency of all the collectors shown is of comparable magnitude. It isn't until the operating temperature of the collector gets significantly higher than the ambient ($\Delta t/I \geq 0.5$) typical of air conditioning operation that the differences become significant. Figure B-6 shows that air collectors will generally be less efficient than water collectors <u>for the same operating conditions</u>. However, air collectors operating in conjunction with pebble-bed storage units usually operate at lower temperatures than liquid-type collectors in systems performing the same functions. Consequently, the systems using air and those using a liquid usually have comparable overall efficiencies.

2.4 Thermal Storage

The majority of all systems built to date have used either water tanks or pebble beds for storing of thermal energy, the water tanks being used primarily in systems employing water-heating collectors and the pebble beds with air collectors. The primary reasons are proven reliability, low cost, and ease of installation and maintenance. Other methods of storing thermal energy that have been proposed and are being used primarily in research and demonstration projects involve the use of materials which absorb significant amounts of energy upon change of phase and consequently result in much larger energy storage quantities per unit volume than through the use of water or rocks. The major work in recent years has involved the use of paraffins and eutectic mixtures of salts and salt hydrates. The reader is referred to reference [21] for a thorough discussion of these and other means of storing thermal energy. The two basic materials described above (water and rock) are the only ones recommended for consideration at this time due to unproven reliability and durability over any reasonable period of time.

3. ESTIMATE OF SOLAR SYSTEM PERFORMANCE

The techniques available for estimating the amount of energy that could be obtained from a solar energy system range from a very approximate and crude estimate to using a detailed computer program that allows the system performance to be calculated hour-by-hour for an entire year or more, using actual weather data. Fortunately, techniques are now being developed that are based on complete system studies using the sophisticated computer programs, yet are relatively easy to use and are quite accurate provided the type of system being considered is similar to the one which was simulated. The major portion of this section will be devoted to describing such a technique for estimating the performance of the types of systems shown in either figure B-1 or figure B-2. It is recommended that the methods outlined herein be used to determine whether or not the use of solar energy systems appears feasible. If so, and the system deviates to any great extent from the ones shown in figures B-1 and B-2 and described in more detail in section 3.2, serious consideration should be given to having a detailed study conducted using computerized analysis techniques such as those in reference [22].

3.1 Approximate Technique

Perhaps the easiest way of determining what kind of performance to expect from a solar energy system when only an estimate is required is to:

a. estimate the amount of energy that could be collected under "average" conditions each month,

b. calculate the space heating, space cooling and/or domestic hot water load for the building each month, and

c. if for the amount of collector assumed to be used, there is a deficit, calculate the amount of fuel to be used in the installed equipment in order to satisfy the deficit each month.

The error using this technique could be large since a solar energy system operates under transient conditions at almost all times and the dynamic interaction between the variable solar radiation and the collector, the collector and thermal storage, and thermal storage and the building, governs the actual performance of the system. However, this technique is useful in determining whether or not to even consider using such a system.

In order to perform the calculations outlined above, the following steps are required:

a. monthly energy collected

1. Estimate the monthly radiation incident on a collector of specified area, orientation, and tilt using techniques outlined in section 3.2.
2. Using an efficiency curve such as figure B-6 for the collector being considered, estimate the amount of energy extracted from the collector. Frequently those curves are available from the manufacturer. In order to use a curve such as plotted in figure B-6, it will be necessary to know the monthly average daytime ambient temperature for the location under consideration (t_a), the daily average radiation incident on the tilted collector surface (I) (see section 3.2), and the operating temperature of the collector (t_e). The operating temperature can be assumed to be 140° F for space heating and domestic water heating and 200° F for space cooling in lieu of detailed knowledge of the system under consideration.

b. monthly building load

The building space heating, space cooling, and domestic water heating load can be calculated by using conventional techniques outlined elsewhere in this document or in publications of the American Society of Heating, Refrigerating, and Air Conditioning Engineers [23, 24].

c. monthly auxiliary energy required

Once the deficit between a. and b. has been determined, the amount of energy required by the conventional system can be calculated by a simple subtraction and then dividing the result by the efficiency of the installed system (expressed as a decimal equivalent).

3.2 Calculation Procedure for the Systems of Figures B-1 and B-2

Klein, Beckman, and Duffie [25, 26] have recently published a simplified design procedure for estimating the fraction of the space heating and hot water load in a building that can be supplied by the solar energy system shown in figures B-1 and B-2 on a month-by-month basis. They used the computer program TRNSYS [22] to conduct over 300 simulations where the design parameters for the system were changed to include a variety of flat-plate collector designs, sizes, tilt angles, building space heating demands, and building hot water demands. The results of the simulations and subsequent analyses were presented in the form of correlation curves shown in figures B-7 and B-8 for water and air systems, respectively. In order to use the curves, one determines the value of the dimensionless variables on the y and x axes on a month-by-month basis for the building and system under consideration, and f is the fraction of the monthly space heating and hot water load that is supplied by the solar energy system. The remaining fraction is then met by the auxiliary or presently installed system.

The validity of the method was checked by comparing the predicted fraction using the curves of figures B-7 and B-8 with the results of detailed half-hour by half-hour simulations of four differently sized systems over eight-year periods in Madison, Wisonsin. In addition, the systems were also simulated in Massachusetts, South Carolina, New Mexico, and Colorado, and results compared. It was found in both cases, that the standard error in the fraction of the load met by solar (expressed as a decimal) for the entire year was less than 0.017. Significant differences did occur for specific monthly periods (generally for months in which the load was small relative to the yearly total load) and consequently the procedure should only be used for predicting seasonal performance. The results using the method were compared with experimental results for a residential building in Massachusetts and the differences were small.

In order to use figures B-7 and B-8, one must know how to determine the value of the two dimensionless variables. Most of the parameters involved are easily explained; however, one must first understand the basic performance equation for a solar collector.

The performance of flat-plate collectors operating under steady-state conditions can be described by the following relationship [5]:

$$\frac{q_u}{A_c} = F_R\, I(\tau\alpha) - F_R\, U_L\, (t_{f,i} - t_a) \tag{1}$$

where q_u = rate of useful energy extraction from the solar collector, Btu/h

A_c = cross-sectional area, ft^2

F_R = solar collector heat removal factor

I = total solar energy incident upon the plane of the solar collector per unit time per unit area, $Btu/(h \cdot ft^2)$

$(\tau\alpha)$ = transmission-absorptance product for the cover plate-absorber combination

U_L = heat transfer loss coefficient for the solar collector, $Bth/(h \cdot ft^2 \cdot °F)$

$t_{f,i}$ = temperature of the fluid entering the solar collector, °F

t_a = ambient air temperature, °F

Equation (1) expresses the fact that the useful energy extracted is equal to the amount that passes through the cover-plate assembly and is absorbed (F_R I $(\tau\alpha)$) minus what is lost by heat transfer back to the environment (F_R U_L $(t_{f,i} - t_a)$).

If the efficiency of the collector can be defined as the useful energy extracted divided by the energy incident upon the collector, then

$$\eta_c = \frac{q_u/A}{I} \tag{2}$$

or

$$\eta_c = F_R(\tau\alpha) - F_R U_L \frac{(t_{f,i} - t_a)}{I} \tag{3}$$

If a manufacturer's performance data can be plotted as η_c vs. $\frac{(t_{f,i} - t_a)}{I}$ as shown in figure B-9 [27], then the performance factors F_R $(\tau\alpha)$, and F_R U_L can be determined directly as the y intercept and slope of the curve, respectively.

The additional parameters not introduced above that appear in the two dimensionless variables of figures B-7 and B-8 are defined as follows:

$$F_R' = F_R \left\{ 1 + \left[\frac{F_R U_L A_c}{(\dot{m} C_p)_c} \right] \left[\frac{(\dot{m} C_p)_c}{\varepsilon_c (\dot{m} C_p)_{min}} - 1 \right] \right\}^{-1}$$

where $\dot{m}$ = mass rate of flow of fluid through the solar collector, lb/h

C_p = specific heat of the fluid flowing through the solar collector, Btu/(lb·°F)

ε_c = effectiveness of the heat exchanger in the collector-storage flow loop

$(\dot{m} C_p)_{min}$ = the smallest value of the two $\dot{m} C_p$ products for the two fluid streams passing through the collector-storage flow loop heat exchanger, Btu/(h·°F)

The quantity in the braces raised to the -1 power can be considered as a penalty factor for the use of a heat exchanger in the collector-storage flow loop. If no heat exchanger is used such as in the air system of figure B-2, $F_R' = F_R$.

$(\overline{\tau\alpha})$ = the average value of $(\tau\alpha)$ for the solar collector operating throughout the entire day. It is usually equal to 90-95 percent of the value determined at normal incidence and obtained from a plot of solar collector efficiency such as figure B-9. For collectors oriented directly towards the equator at a slope from 0 to 15° greater than the latitude, $(\overline{\tau\alpha})$ should be assumed to be 93 percent and 91 percent of the normal incidence value for 1 and 2 glass cover collectors, respectively.

S = monthly value of solar radiation incident on the solar collector array, Btu/(month·ft^2)

L = monthly building space heating and hot water load, Btu/month

t_{ref} = 212 °F

$\bar{t}_a$ = monthly average daytime ambient air temperature

$\Delta\tau$ = number of hours in the month, h

The following procedure is recommended in the use of figures B-7 and B-8:

a. Calculate the monthly space heating and hot water load for the building under consideration using conventional techniques outlined elsewhere in this document or in publications of the American Society of Heating, Refrigerating, and Air Conditioning Engineers [23, 24].

b. Calculate the monthly incident solar radiation on the collector array (to be explained later in this section).

c. Determine all the component parameters necessary to evaluate the dimensionless variables of figures B-7 and B-8. It is suggested that the calculation be done as follows:

y axis

$$\frac{A_c F_R' \overline{(\tau\alpha)} S}{L} = [A_c] \times \overbrace{[F_R (\tau\alpha)_n]}^{\text{from plot of manufacturers's data, ie., Fig. B-9}} \times \left[\frac{\overline{(\tau\alpha)}}{(\tau\alpha)_n}\right] \times \left[\frac{F_R'}{F_R}\right] \times [S] \times \left[\frac{1}{L}\right]$$

x axis

$$\frac{A_c F_R' U_L (t_{ref} - \bar{t}_a) \Delta\tau}{L} = [A_c] \times \overbrace{[F_R U_L]}^{\text{from plot of manufacturers' data, ie., Fig. B-9}} \times \left[\frac{F_R'}{F_R}\right] \times [t_{ref} - \bar{t}_a]$$

$$\times [\Delta\tau] \times \left[\frac{1}{L}\right]$$

d. Using figure B-7 or B-8, determine the fraction of the load each month of the season that is carried by the solar system.

e. Determine the fraction of the yearly load that is carried by the solar system. The user is cautioned that the yearly fraction is not simply the average of all of the monthly fractions but that each monthly fraction must be weighted according to the fraction of the yearly load that occurs that month.

The technique is rather simple to use and if one examines the variables in figures B-7 and B-8, it can be seen that the performance of the system depends upon three quantities:

a. The amount of solar radiation that is absorbed by the collector array over the month, $A_c\ F_R'\ (\tau\alpha)\ S$.

b. The amount of heat loss that occurs from the collector array during the month with a reference temperature as the basis, $A_c\ F_R'\ U_L\ (t_{ref} - t_a)\ \Delta\tau$.

c. The monthly building space heating and hot water load, L.

In developing the f-chart of figures B-7 and B-8, certain assumptions were made concerning the design of the systems in figures B-1 and B-2. For the water system of figure B-1, the capacity of the thermal storage units used for both the space heating and domestic hot water processes were assumed to total 1.85 gallons per ft^2 of solar collector. This is equivalent to 15 Btu of energy storage per °F temperature rise in the thermal storage unit per ft^2 of solar collector. If a different size storage unit is chosen, the yearly fraction of the load carried by solar can be multiplied by the correction fraction in figure B-10. As can be seen, there is marginal improvement in the system performance if the storage unit is sized larger than what was assumed originally; however, the system effectiveness can be significantly affected if the storage unit is undersized.

An additional assumption used in the development of figure B-7 was that the heat exchanger between the thermal storage unit and the building space for heating was sized so that:

$$\frac{\varepsilon_L\ (\dot{m}\ C_p)_{min}}{UA} = 2$$

where ε_L = effectiveness of the heat exchanger in the storage-building flow loop

$(\dot{m}\ C_p)_{min}$ = the smallest value of the fluid capacitance rates for the two fluid streams passing through the storage-building flow loop heat exchanger, Btu/(h·°F)

UA = heat loss factor for the building space, Btu/(h·°F)

If the heat exchanger is sized greatly different from this, a second correction factor in figure B-11 can be applied the same way as in figure B-10. Again, an undersized heat exchanger has a much more pronounced effect on the system performance than does an oversized one.

The f-chart for air heating systems (figure B-8) was generated assuming a flow rate times specific heat divided by F_R (capacitance rate) of air equal to 2.87 Btu/(h·°F) per ft^2 of collector area. Since the performance of air collectors can vary signficantly with flow rate, the estimated system performance using figure B-8 should be corrected for flow rates that differ from this value. For collector capacitance rates ($\dot{m}\, C_p/F_R\, A_c$) between 1.47 and 5.88 Btu/(h·ft^2.°F), the system performance should be estimated after multiplying the value of the x axis parameter by

$$\left[\frac{(\dot{m}\, C_p)_c}{F_R\, A_c} \quad \frac{1}{2.87}\right]^{0.28}$$

The capacity of the pebble bed used in establishing figure B-8 was 19.6 Btu/°F per ft^2 of collector area. The performance of systems with other storage capacities between 9.8 and 78.3 Btu/(ft^2·°F) can be determined by multiplying the x axis parameter by

$$\left[\frac{V\, \rho_{app}\, C_r}{F_R\, A_c} \cdot \frac{1}{19.6}\right]^{-0.3}$$

where V = volume of storage, ft^3

ρ_{app} = apparent density of storage medium, lb/ft^3

C_r = specific heat of storage medium, Btu/(lb·°F)

The only remaining step to be explained in the use of the present technique is the calculation of monthly solar radiation incident on the solar collector array. The recommended procedure is based on the original work of Liu and Jordan [28] which has been adopted and published most recently by EI and I Associates [29].

Table B-1 shows the average daily extraterrestrial (outside the earth's atmosphere) solar energy falling on a horizontal surface for the various months of the year and as a function of latitude. Table B-2 gives the average daily terrestrial solar energy falling on a horizontal surface for the various months of the year for a large number of weather stations in the United States. In order to obtain the average daily terrestrial solar energy falling on south-facing solar collectors tilted up at a specified angle, the following equation should be used:

$$\overline{I}_t = \overline{I}_h \overline{R} \tag{4}$$

where $\overline{I}_t$ = average daily terrestrial solar energy incident on a south-facing tilted surface, ly/day

$\overline{I}_h$ = average daily terrestrial solar energy incident on a horizontal surface from table B-2, ly/day

$\overline{R}$ = ratio of average daily terrestrial solar energy incident on a tilted surface to that on a horizontal surface. This ratio is determined from figures B-12 through B-18 knowing the latitude, tilt angle of the collector, month and the ratio of $\overline{I}_t/\overline{I}_o$

$\overline{I}_o$ = average daily extraterrestrial solar energy incident on a horizontal surface from table B-1, ly/day.

The unit of langley (ly) is one frequently used in the recording of solar radiation data and is equivalent to a gram calorie per cm^2. It is equal to 3.687 Btu/ft^2. Once the average daily radiation on the tilted collector surface has been determined using equation (4), the monthly value required in the use of either the approximate technique of section 3.1 or the more accurate procedure of section 3.2 is determined by simply multiplying by the number of days in the month

4. SIMPLIFIED ECONOMIC ANALYSIS

There are a number of ways to perform an economic analysis to determine whether or not a solar energy system is competitive with conventional heating and whether consideration should be given to retrofitting a building with such a system. Reference [30] gives an in-depth review and explanation of the various ways it can be done.

The procedure recommended for use here is one where the cost of owning a solar energy system over a specified lifetime is computed. The technique is a slightly simplified form of an annual cost model. All costs are forecasted over the assumed lifetime of the system and then are converted into uniform annual costs by discounting.

The basic formula to be used in computing the annual dollar savings over the lifetime of the system is:

$$DS = f \frac{L}{\eta_a} \overline{C}_F - i(C_c A_c + C_{ST} + C_E) \qquad (5)$$

where DS = dollar savings in using the solar system over the use of the installed conventional system, \$/year

f = fraction of the building space heating and hot water load carried by the solar system and computed by the techniques in section 3

L = yearly building space heating and domestic hot water load, 10^6 Btu

η_a = overall efficiency of the conventional installed space heating and/or domestic hot water heating system expressed as a decimal

$\overline{C}_F$ = average annual costs of conventional fuels used in the presently installed system over the lifetime used in the analysis, $\$/10^6$ Btu

i = capital recovery factor for a specified lifetime and interest rate taken from table 3

C_c = solar collector installed cost, $\$/ft^2$

A_c = solar collector area, ft^2

C_{ST} = installed cost of thermal storage unit, \$

C_E = installed cost of everything else associated with the installation of the solar energy system including pumps, piping, controls, heat exchanger, etc.

The procedure to be followed in analyzing the economic trade-offs associated with the use of the solar energy system is to first make estimates of the fraction of the building load that might be carried by a system having different amounts of solar collector area using the techniques in section 3. Then using equation (4), determine if there are likely to be any dollar savings through the use of the system.

Example Problem

Determine the relationship between the yearly savings, the initial investment, and fuel costs for retrofitting a townhouse in the Washington, D.C. area with a solar space heating and domestic hot water system. The house has a floor area of approximately 1200 ft^2 and a heat loss factor of 307 Btu/(h·°F) (UA for the house). The house is now heated with a gas furnace in a forced-air distribution system and the domestic hot water is heated by a conventional 80-gallon electric water heater. The use of hot water can be assumed to be approximately 90 gallons/day. Assume that the house faces south but that due to the shape of the roof, the maximum amount of solar collector that can be used is 485 ft^2, approximately 60 percent of it tilted at an angle of 11° from the horizon and the remainder at 55°.

Solution

Based on heating-degree days for the Washington, D.C. area as tabulated in reference [24], the seasonal heating load was found to be 31×10^6 Btu. Based on a daily average temperature rise of 75° F for the hot water for the year, the yearly domestic hot water load was computed to be 20×10^6 Btu. The technique of section 3.2 was used to predict that approximately 75 percent of the space heating and hot water load could be carried by using double-glazed liquid, flat-plate solar collectors with a black-painted absorber, a correctly sized water-to-air heat exchanger in the air distribution system for space heating, and a 1000-gallon water tank for thermal storage.

Using equation (5), the yearly savings for the system was computed for various values of total initial investment and four different fuel prices for both gas and electricity (competing energy sources in this case). Assumed efficiencies of 60 percent and 100 percent were used for the gas-fired furnace and electric water heater, respectively. A lifetime of 15 years was assumed with an interest rate of 10 percent (capital recovery factor of 0.13147). Figure B-19 shows the results of the calculations. The lower curve represents the approximate cost of natural gas and electricity in the Washington, D.C. area in early 1976. The estimate of the installed cost of the system is as follows:

1.	485 ft^{2}2 of solar collector, not installed	\$3,000.00
2.	1000-gallon water storage tank	1,000.00
3.	heat exchanger for collector-storage fluid loop	200.00
4.	heat exchanger and extra 80-gallon tank for heating of domestic hot water	500.00
5.	piping, hoses, pumps, valves, pipe and tank insulation, 40 gallons of antifreeze, controls, and installation	2,300.00
		\$7,000.00

As can be seen from figure B-19, the average fuel prices over the next 15 years would have to be between 3 and 4 times the present price before a decision to install such a system today could be made.

It is estimated by using the technique of section 3.1 that approximately 72 ft^2 of solar collector could be installed along with the exchanger, extra 80-gallon tank, pump and controls for heating 75 percent of the domestic hot water alone at a total installed cost of \$1,200.00. The plot in figure B-20 for domestic water heating alone in this house indicates, based on the assumptions made, that the system is economical, even if the average electricity prices over the next 15 years remain unchanged.

Nationwide Implications

The staff of the University of Wisconsin's Solar Energy Laboratory has recently published the results of an economic study done for a residential building very near the size of the one in the sample above [31]. The f-chart technique was used to design an economically optimized solar space heating and hot water system for the house and then the net annual dollar savings was computed for the house and system in 87 locations in North America. Figure B-21 was taken from their study and shows net annual savings represented on a map by dots of increasing sizes for four classes of savings, i.e., 0-100, 100-200, 200-300, and over 300 $/year. Assumptions were made that resulted in a net savings in the 0-100 $/year class for the mid-Atlantic region, in agreement with the above example for average fuel prices over the next 15 years of three times present prices. As can be seen, under the same conditions, the system would be even more economically feasible in other parts of the country such as in the upper midwest and Rocky Mountain States.

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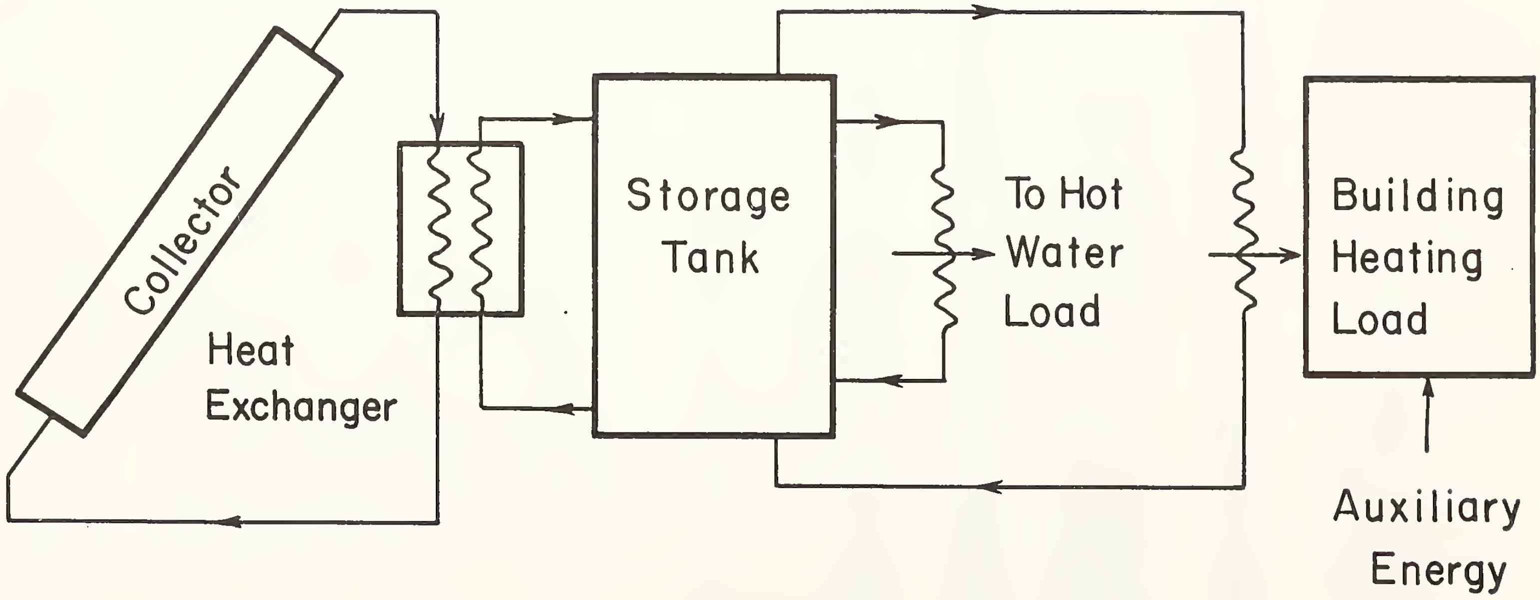


Figure B-1. **SCHEMATIC DIAGRAM OF A LIQUID SOLAR HEATING AND DOMESTIC HOT WATER SYSTEM [25].**

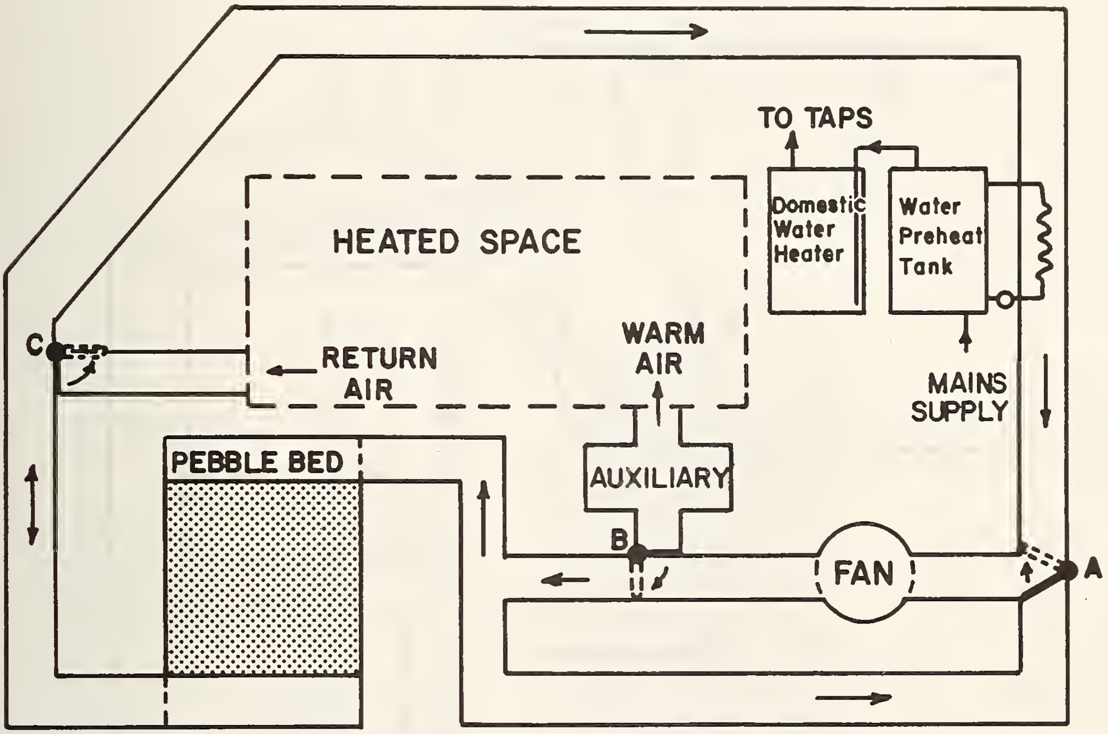


Figure B-2. Schematic Diagram of a Solar Air Heating and Domestic Hot Water System [26]

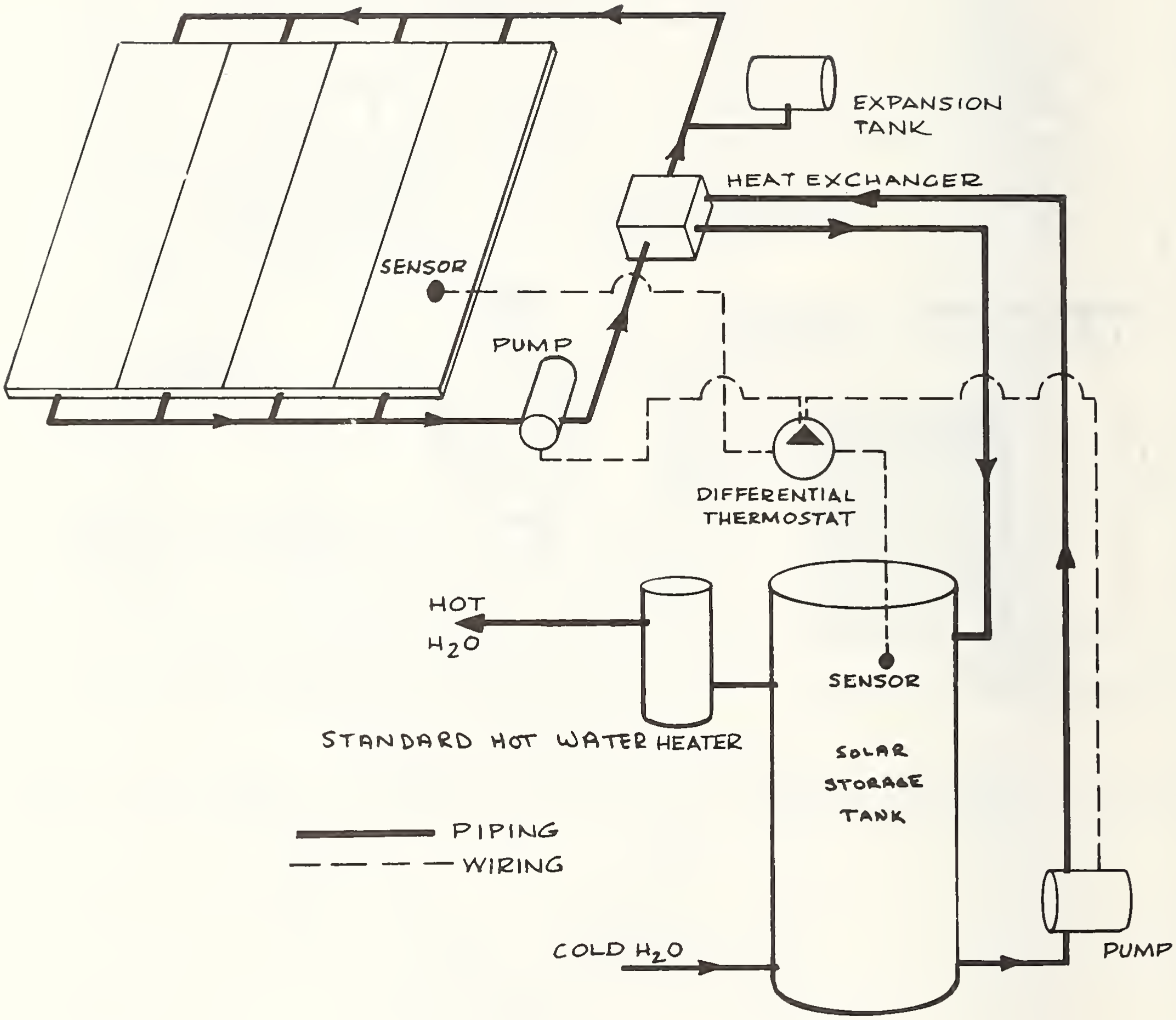


Figure B-3. Schematic Diagram of a Solar Domestic Hot Water System [13]

1" = 0.0254 m

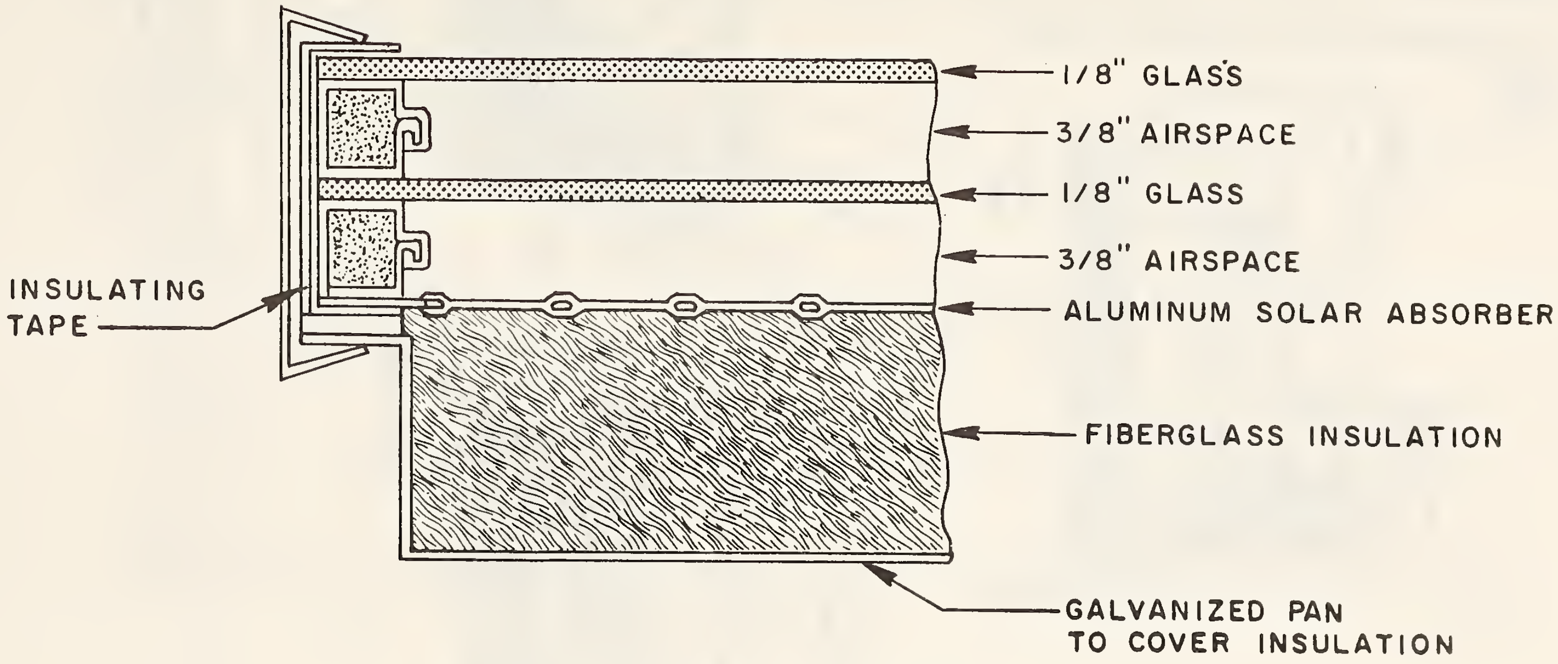


Figure B-4. Schematic Diagram of Flat-Plate Liquid-Heating Solar Collector [15]

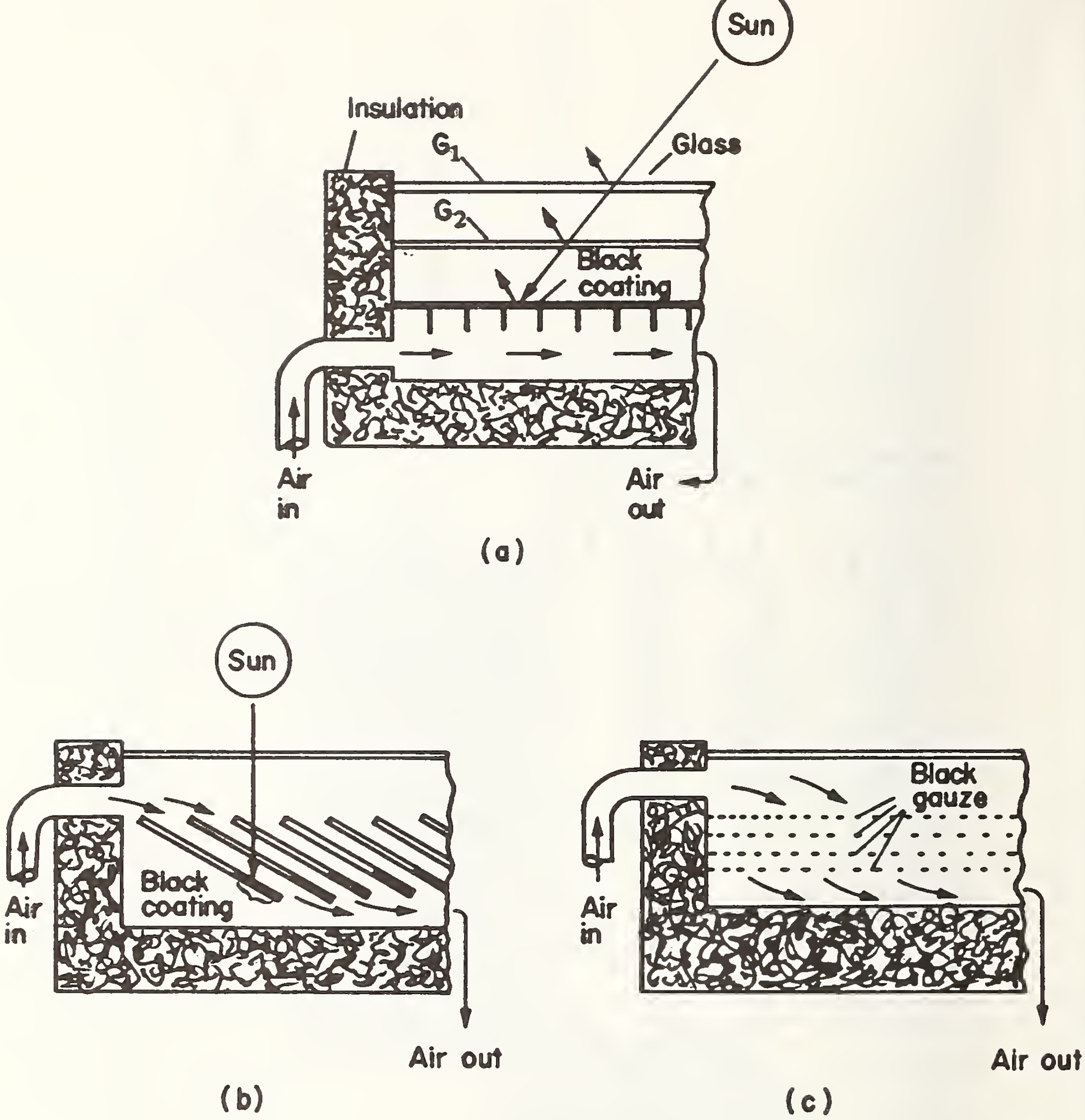


Figure B-5. Air-cooled Solar Collectors. (a) Double-glazed, finned absorber plate; (b) Single-glazed, overlapping glass plate; (c) Single-glazed, porous absorber [6]

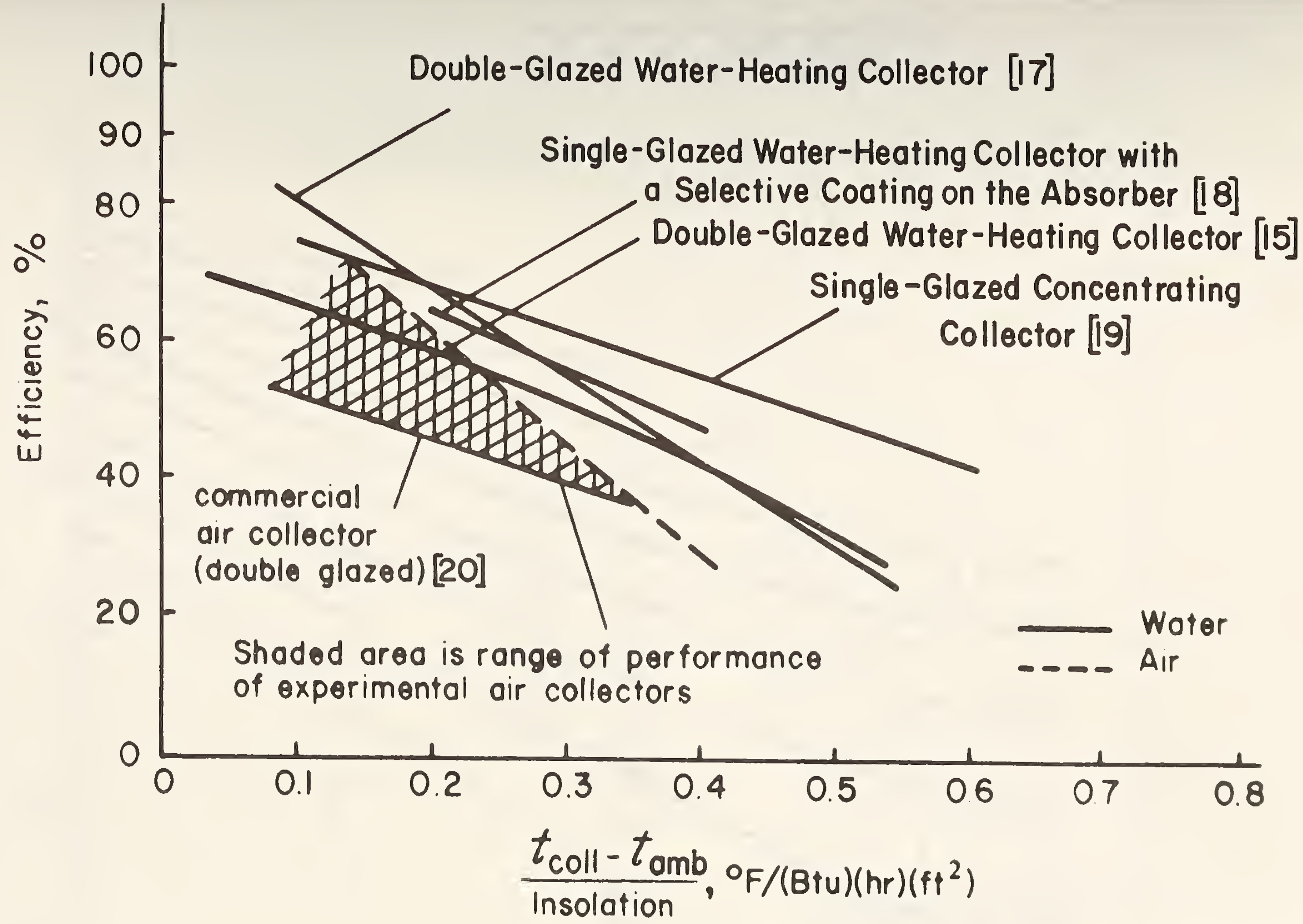
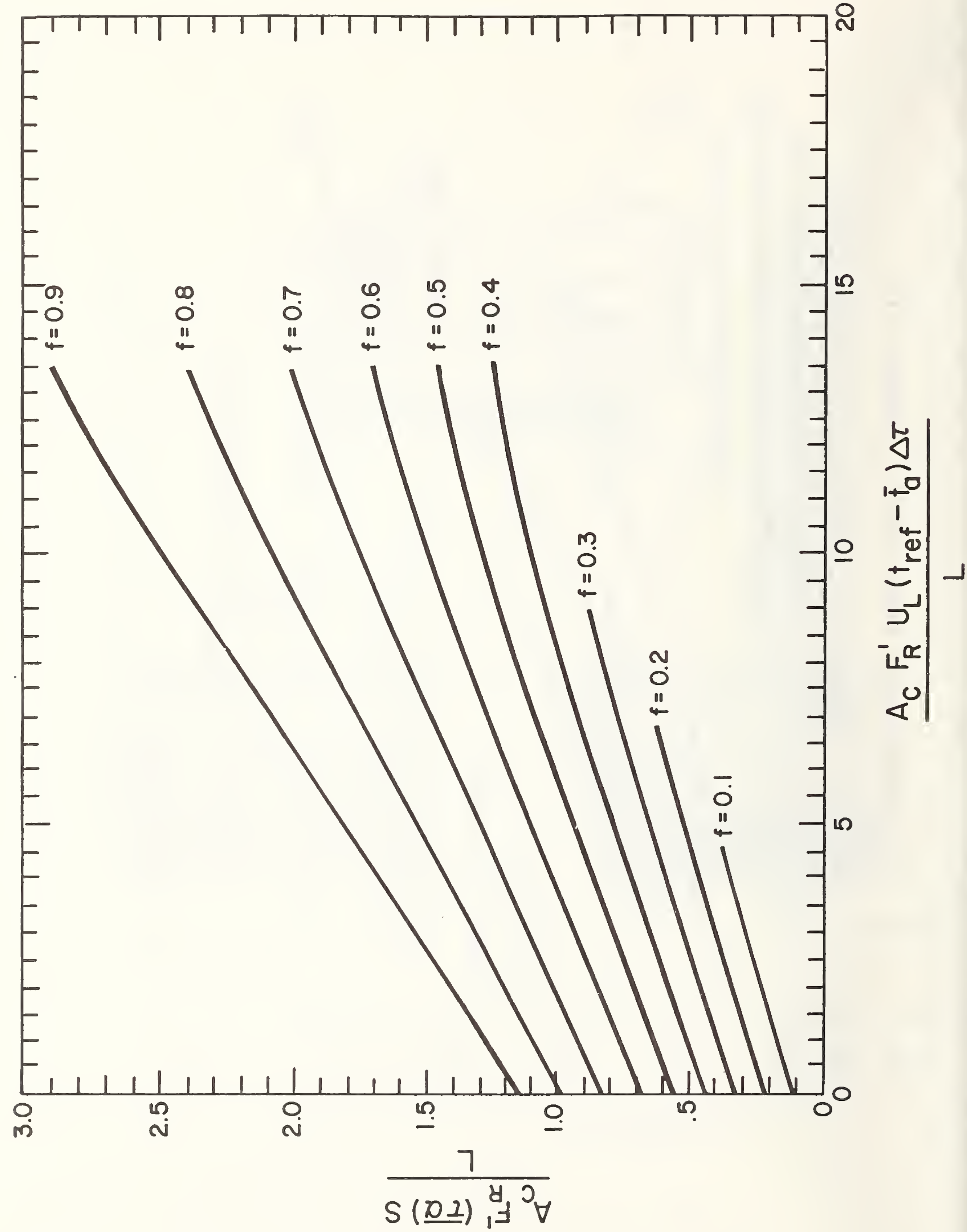


Figure B-6. Performance comparison of several commercial and experimental solar collectors of practical design [6].


f = 0.9
f = 0.8
f = 0.7
f = 0.6
f = 0.5
f = 0.4
f = 0.3
f = 0.2
f = 0.1
3.0
2.5
2.0
1.5
1.0
.5
0
0
5
10
15
20
$\frac{A_c F_R' (\overline{\tau\alpha}) S}{L}$
$\frac{A_c F_R' U_L (t_{ref} - \bar{t}_a) \Delta\tau}{L}$

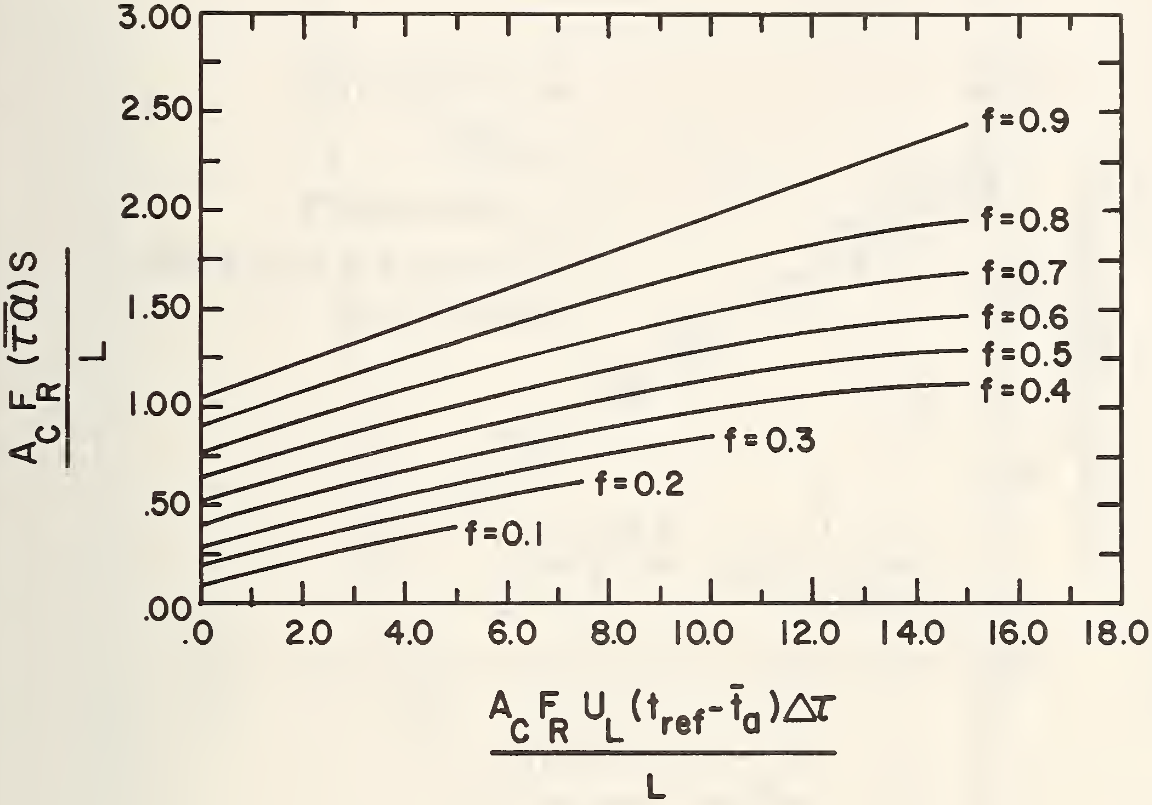


Figure B-8. f- Chart for Solar Air Heating Systems [26]

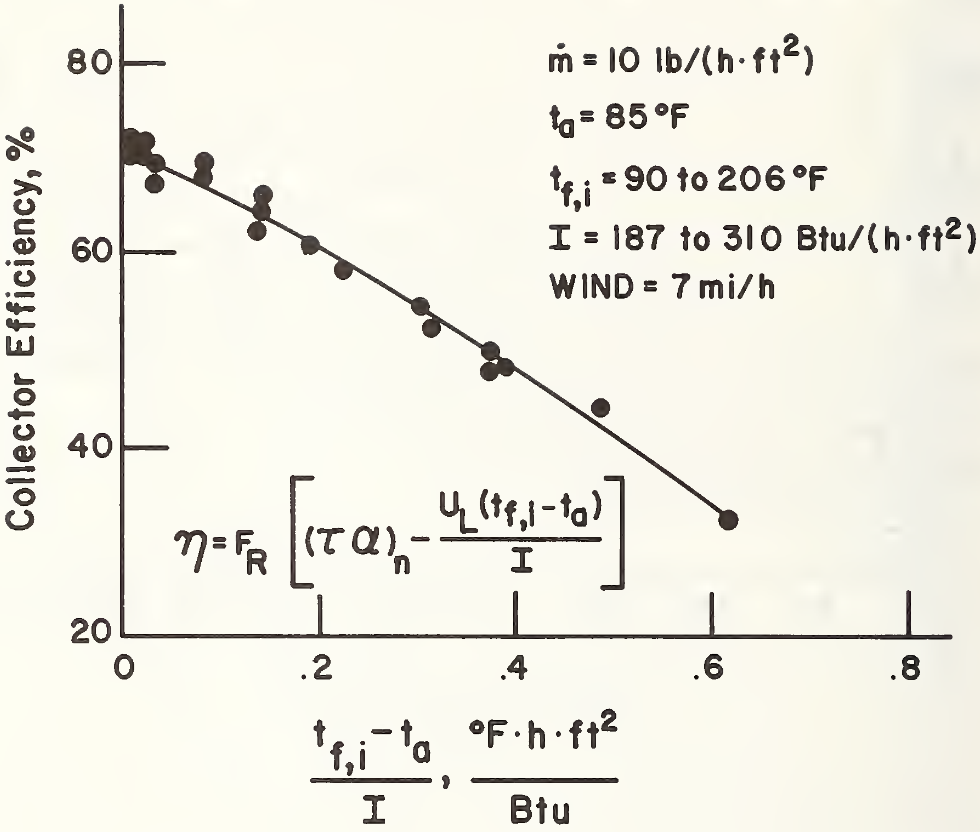


Figure B-9. Efficiency Curve for a Double-Glazed Flat-Plate Liquid-Heating Solar Collector with a Selective Coating on the Absorber [27]

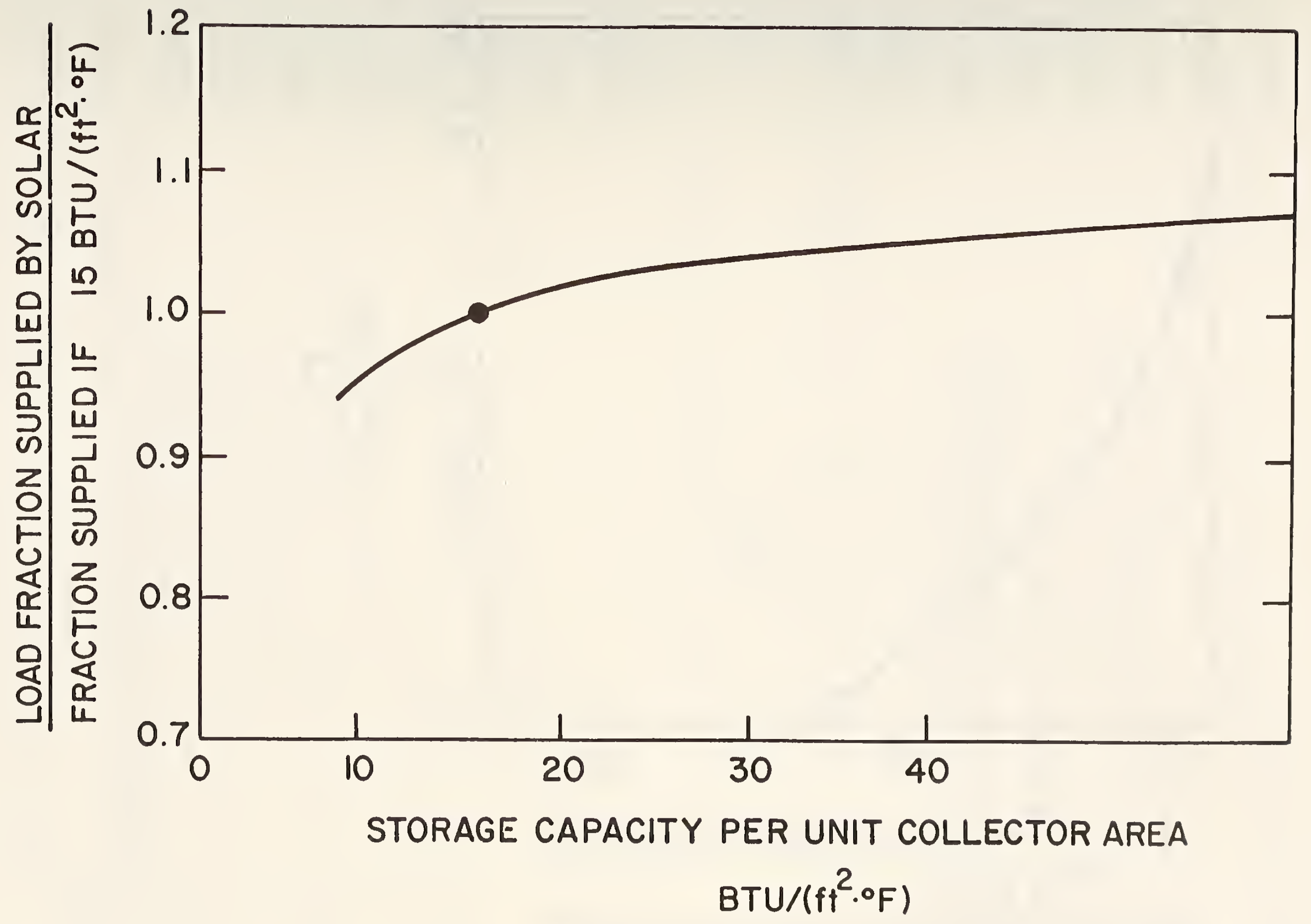


Figure B-10. Correction Factor for Storage Capacities Other than 15 BTU/ (ft^2·°F) (Liquid Solar Heating Systems) [25].

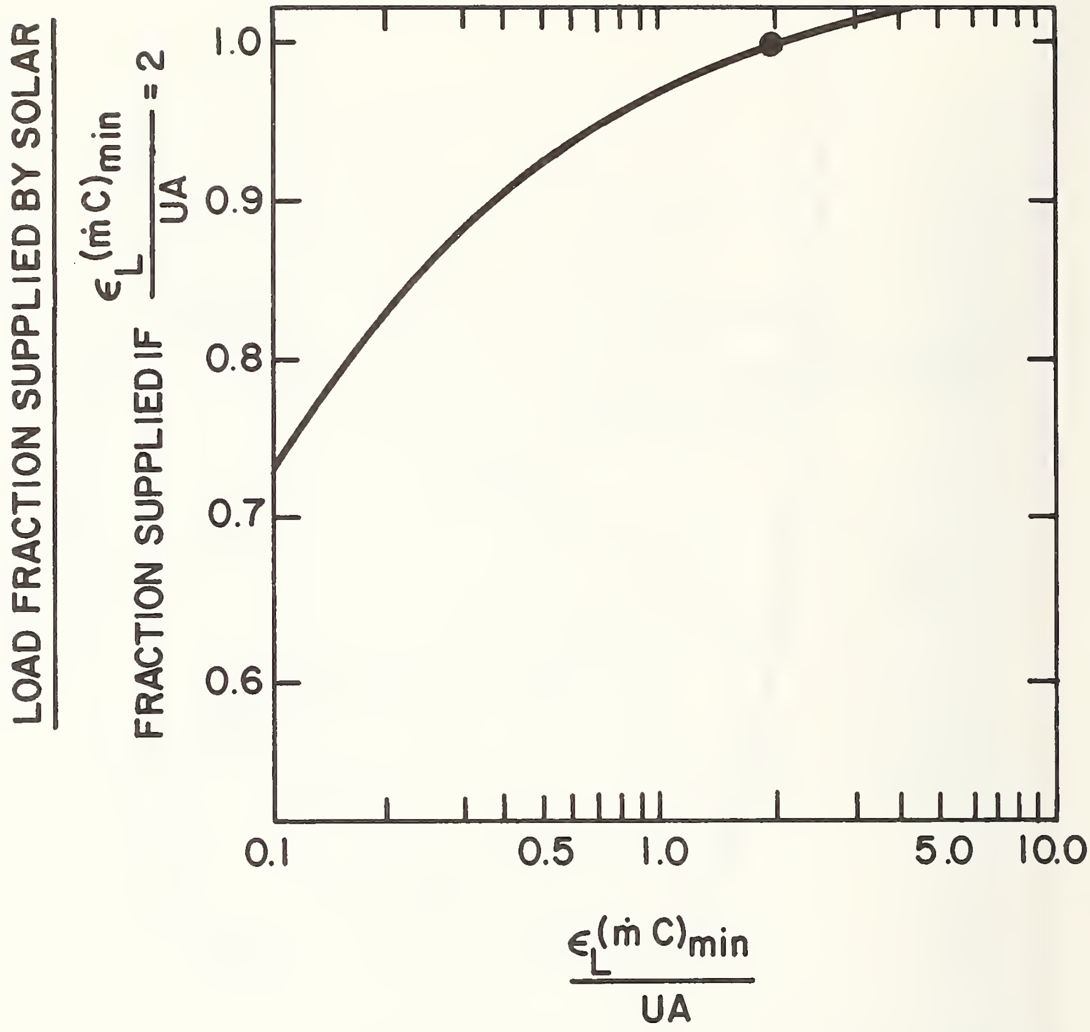


Figure B-11 Correction Factor for Various Values of $\varepsilon_L (mC_p)_{min}/(UA)$ (Liquid Solar Heating systems [25]

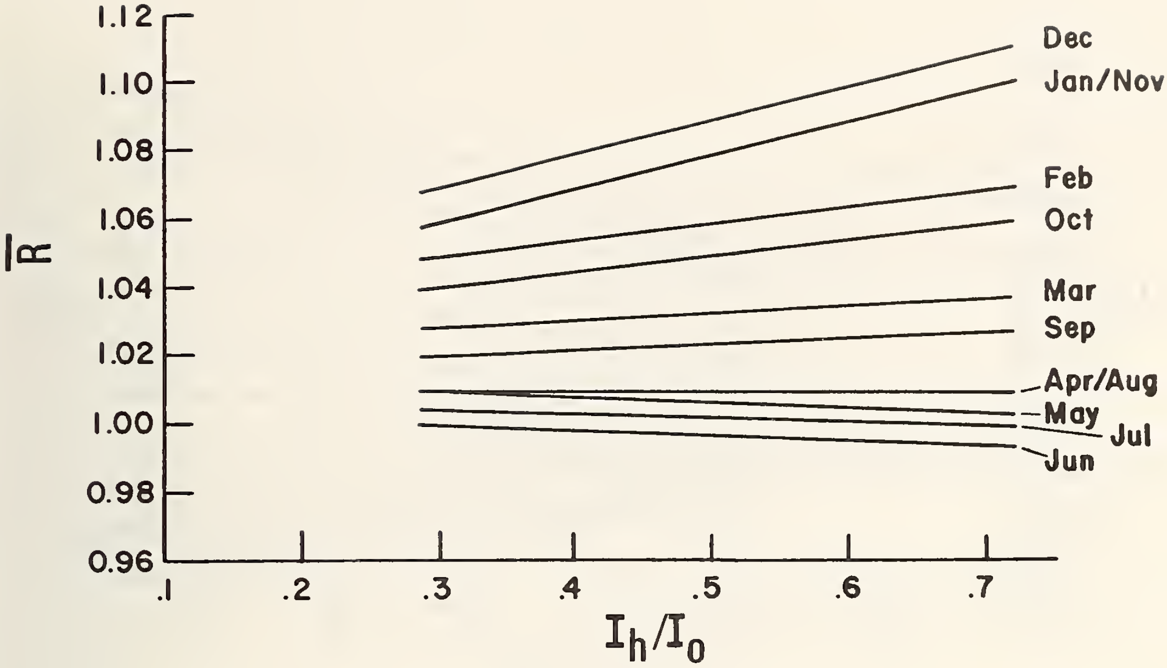


Figure B-12a 20° North Latitude; 5° Collector Tilt

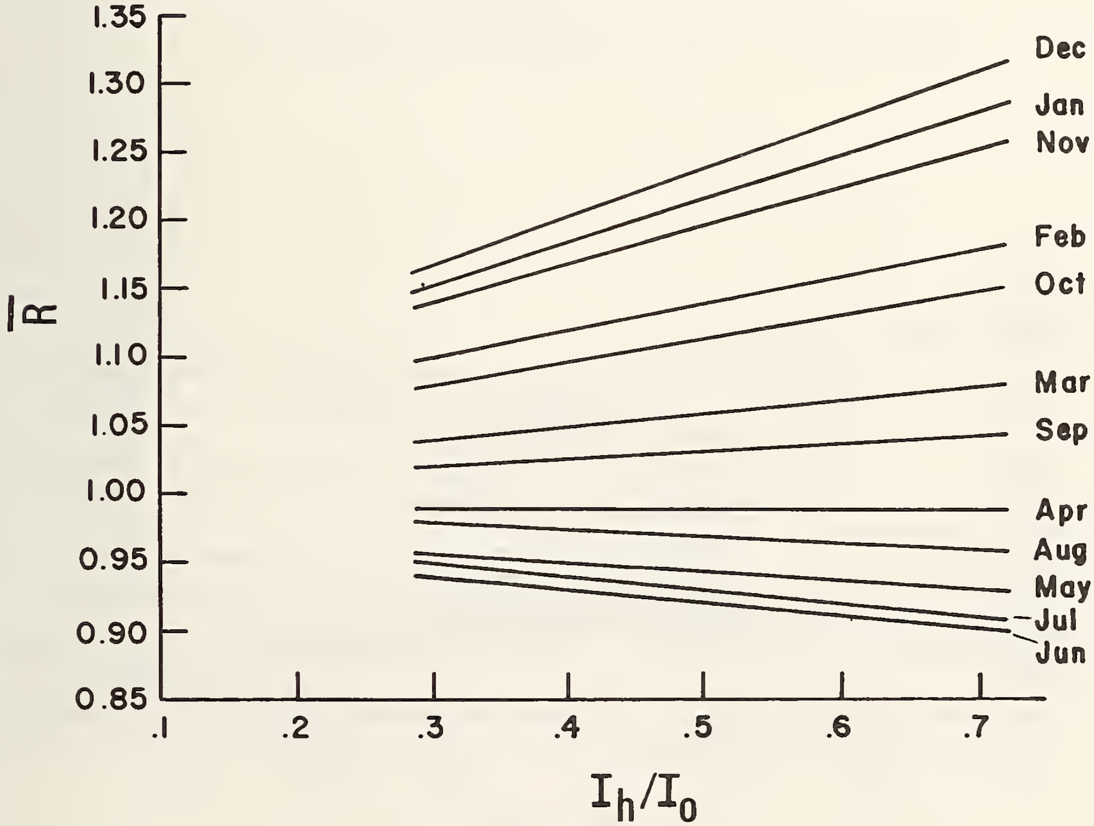


Figure B-12b 20° North Latitude; 20° Collector Tilt

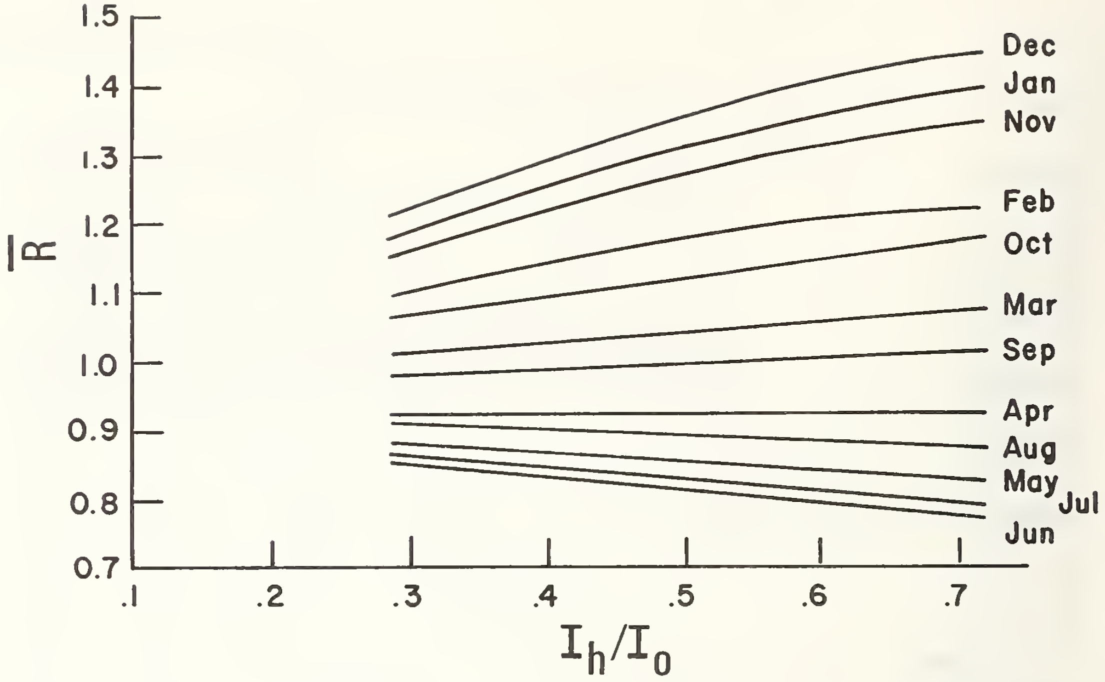


Figure B-12c 20° North Latitude; 35° Collector Tilt

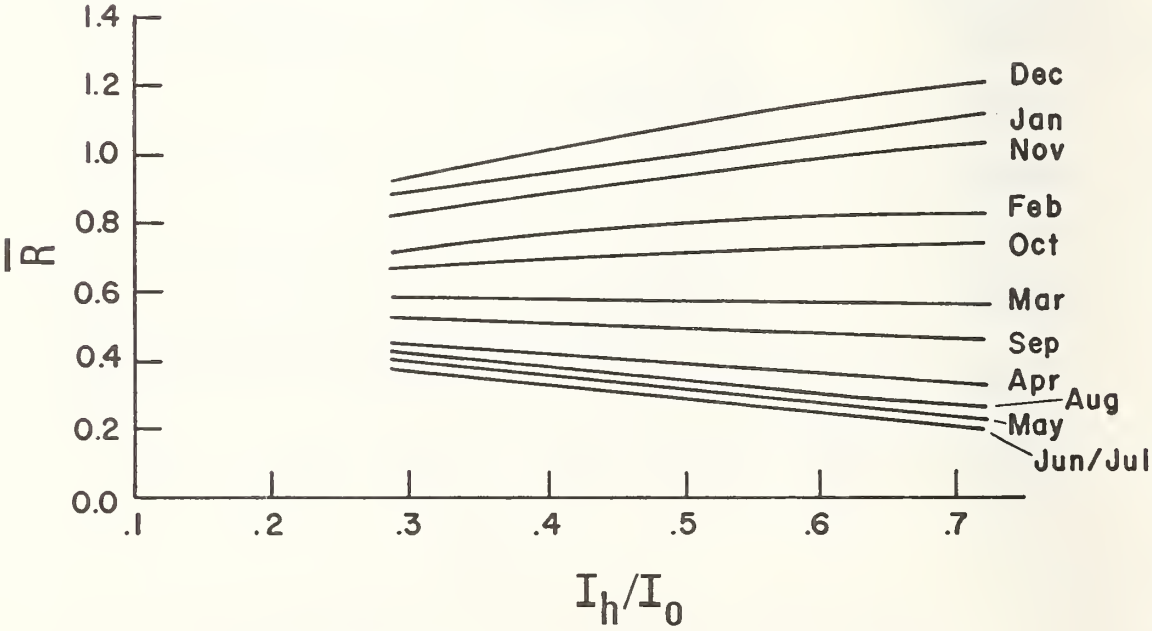


Figure B-12d 20° North Latitude; 90° Collector Tilt

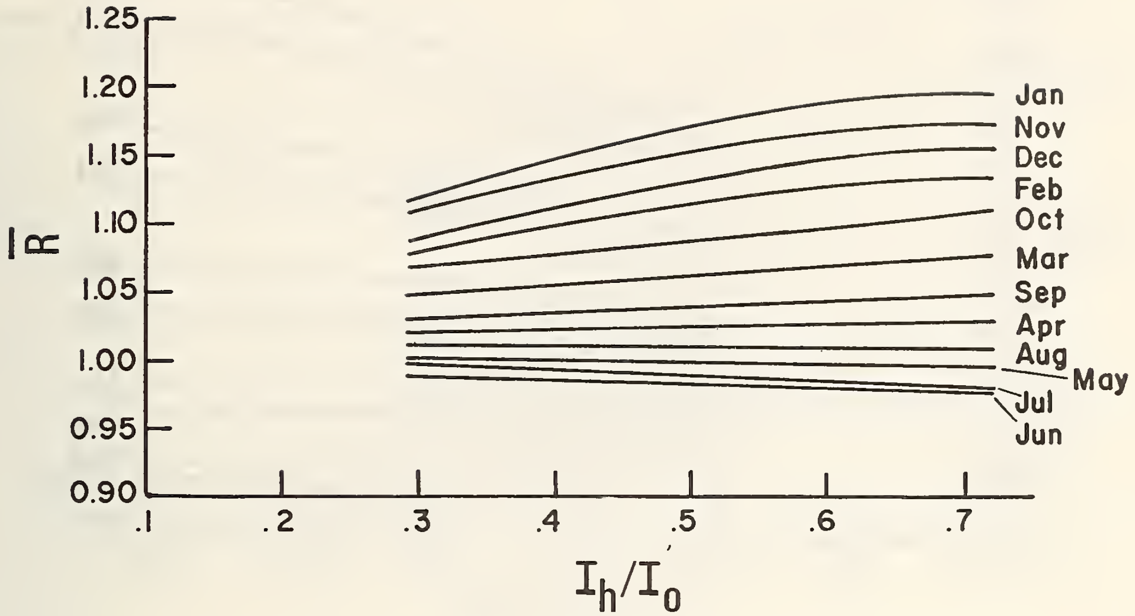


Figure B-13a 25° North Latitude; 10° Collector Tilt

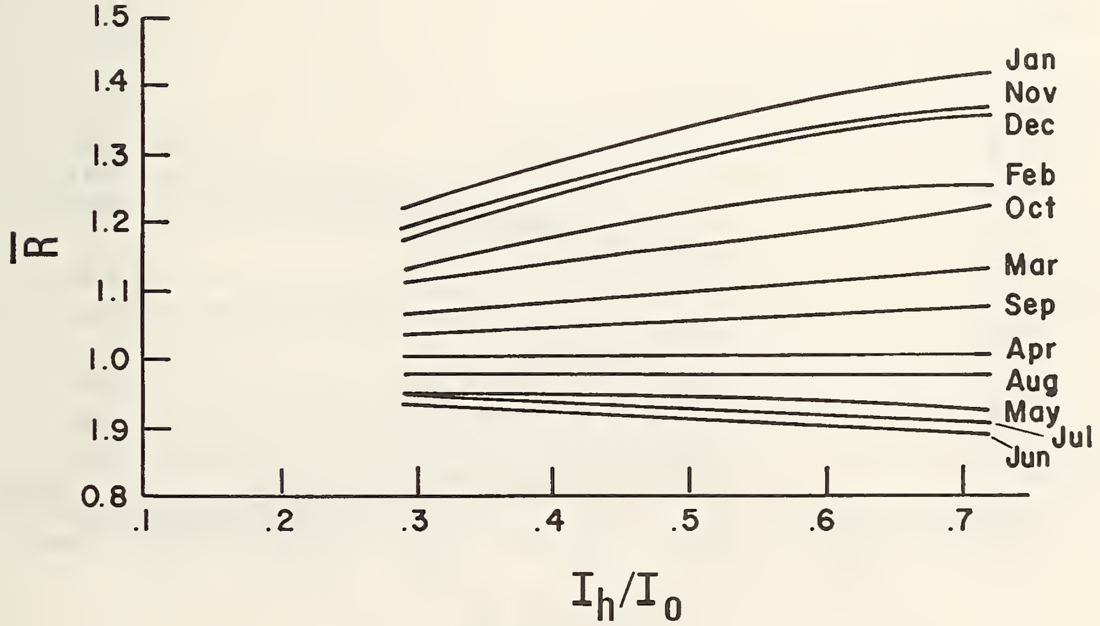


Figure B-13b 25° North Latitude; 25° Collector Tilt

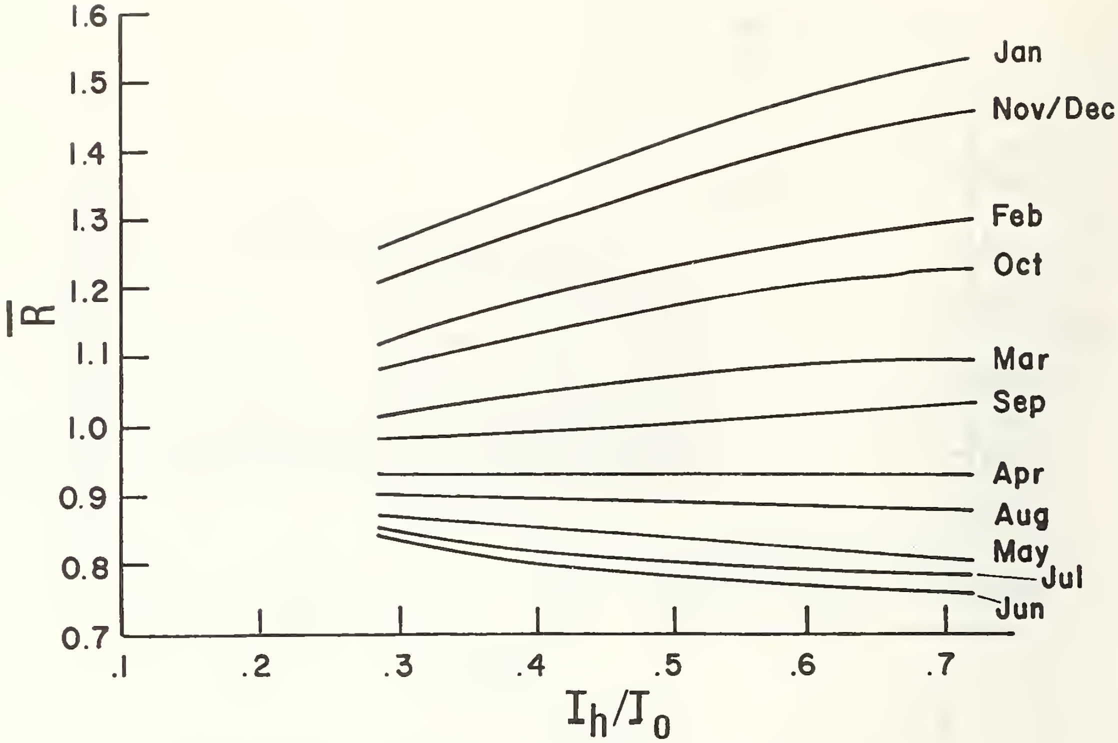


Figure B-13c 25° North Latitude; 40° Collector Tilt

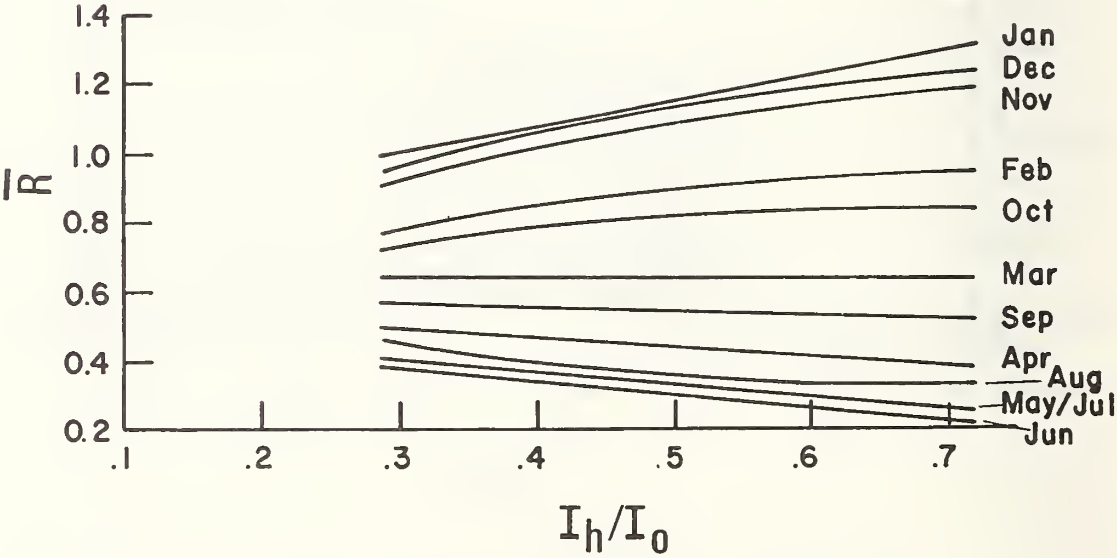


Figure B-13d 25° North Latitude; 90° Collector Tilt

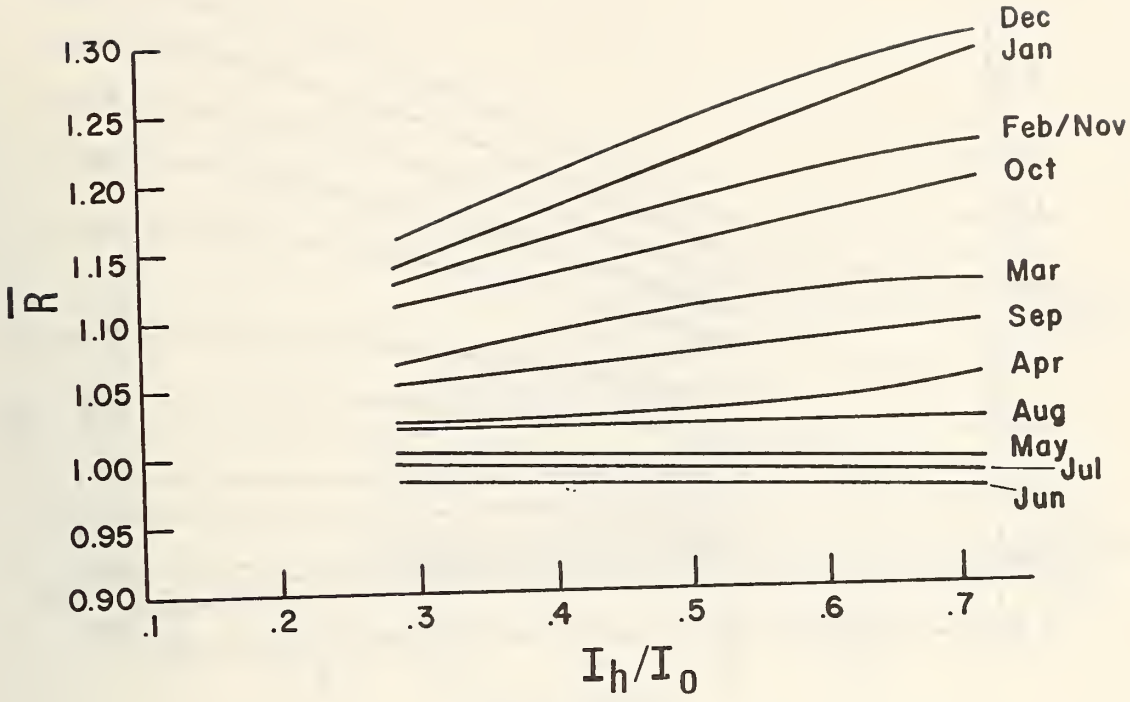


Figure B-14a 30° North Latitude; 15° Collector Tilt

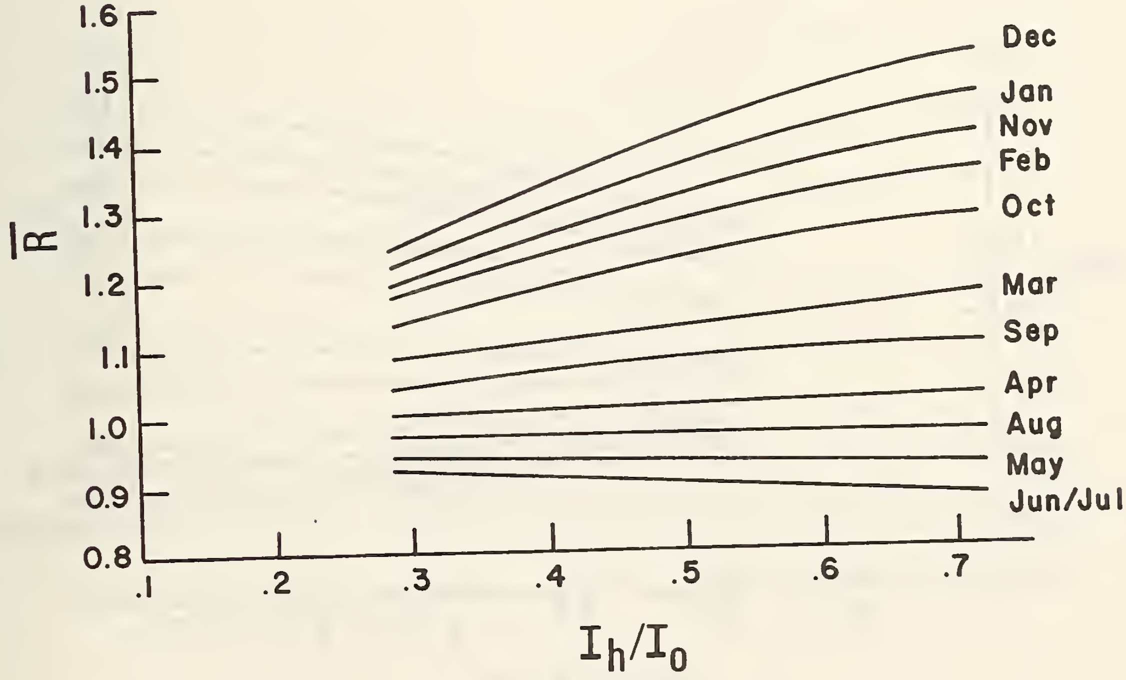


Figure B-14b 30° North Latitude; 30° Collector Tilt

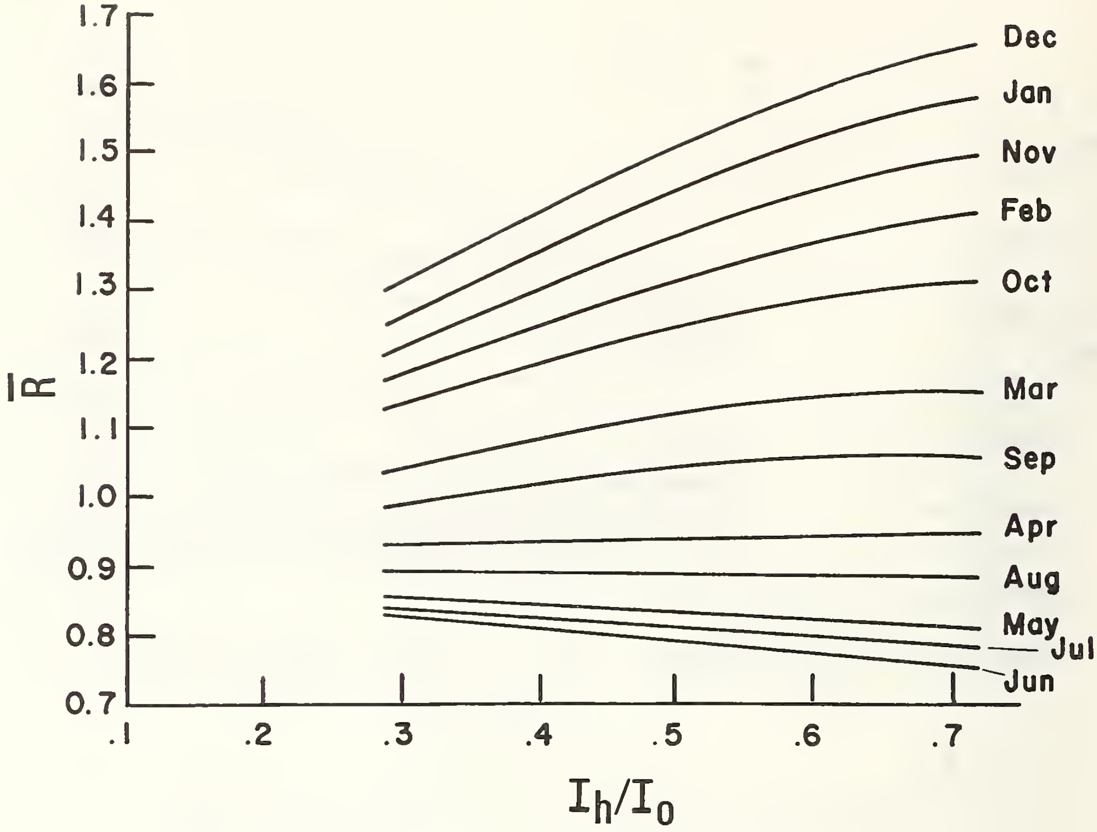


Figure B-14c 30° North Latitude; 45° Collector Tilt

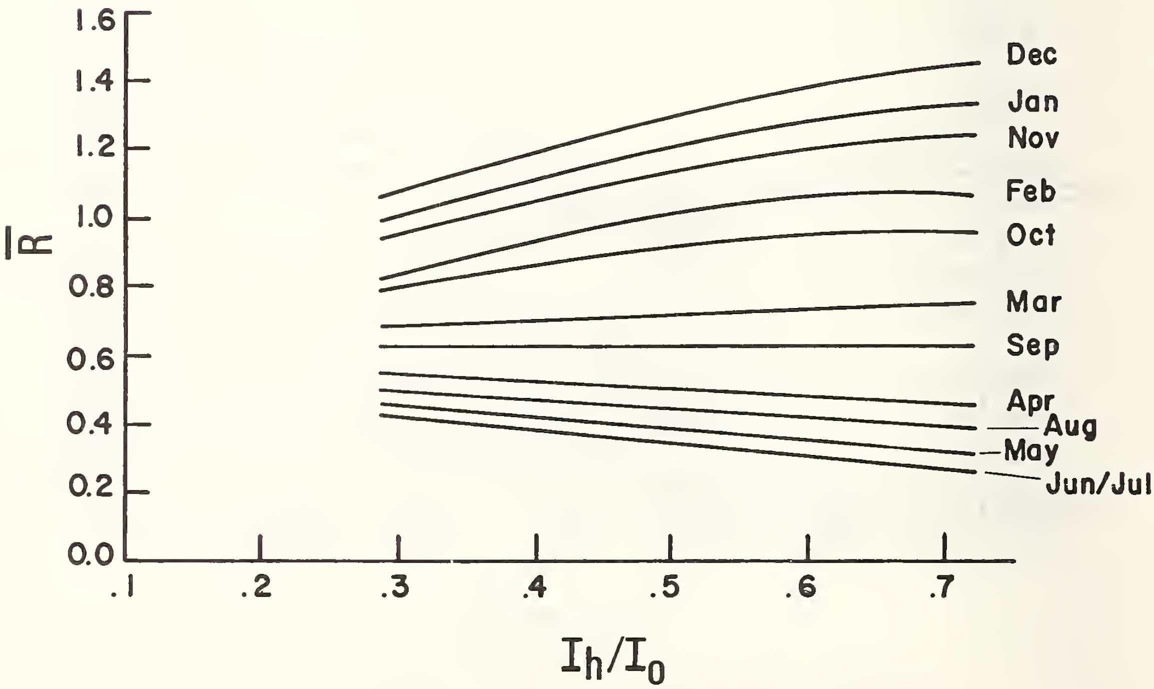


Figure B-14d 30° North Latitude; 90° Collector Tilt

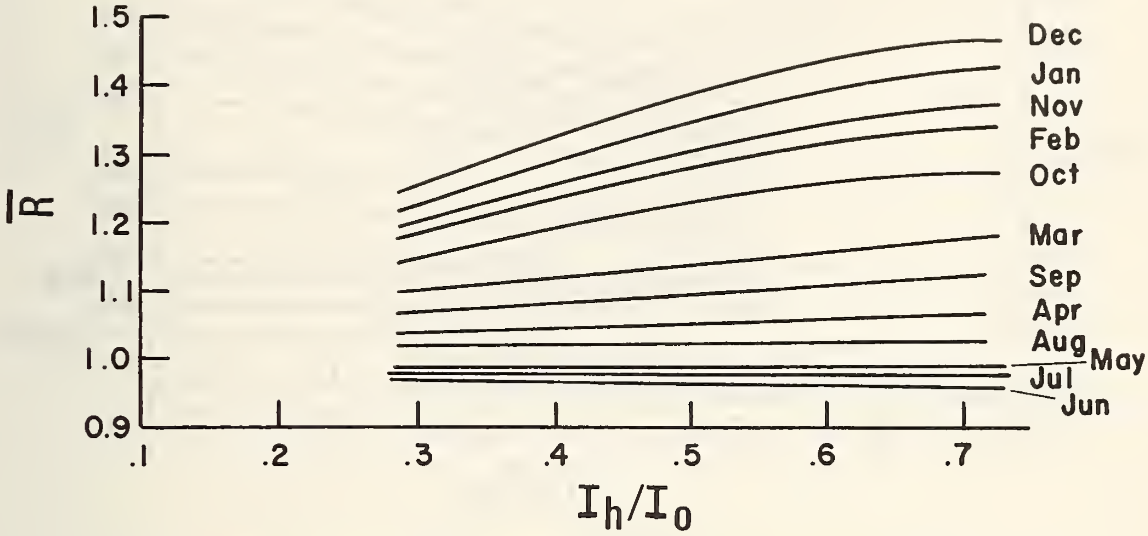


Figure B-15a 35° North Latitude; 20° Collector Tilt

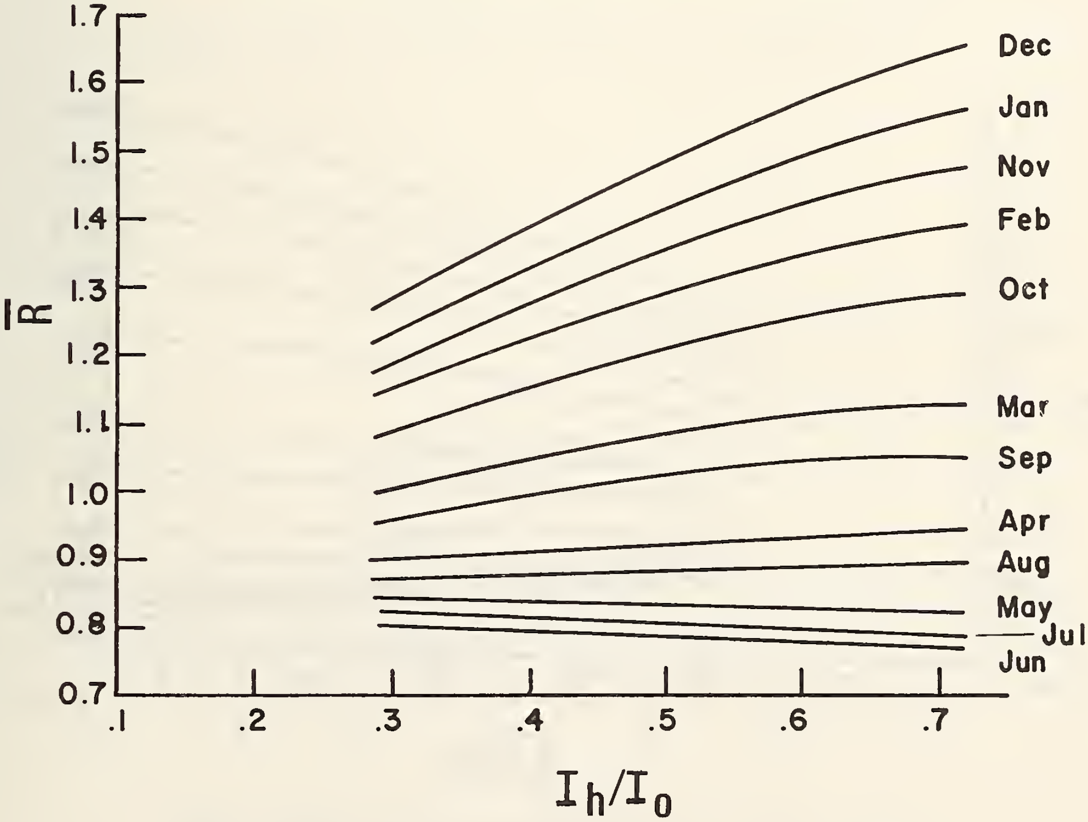


Figure B-15b 35° North Latitude; 35° Collector Tilt

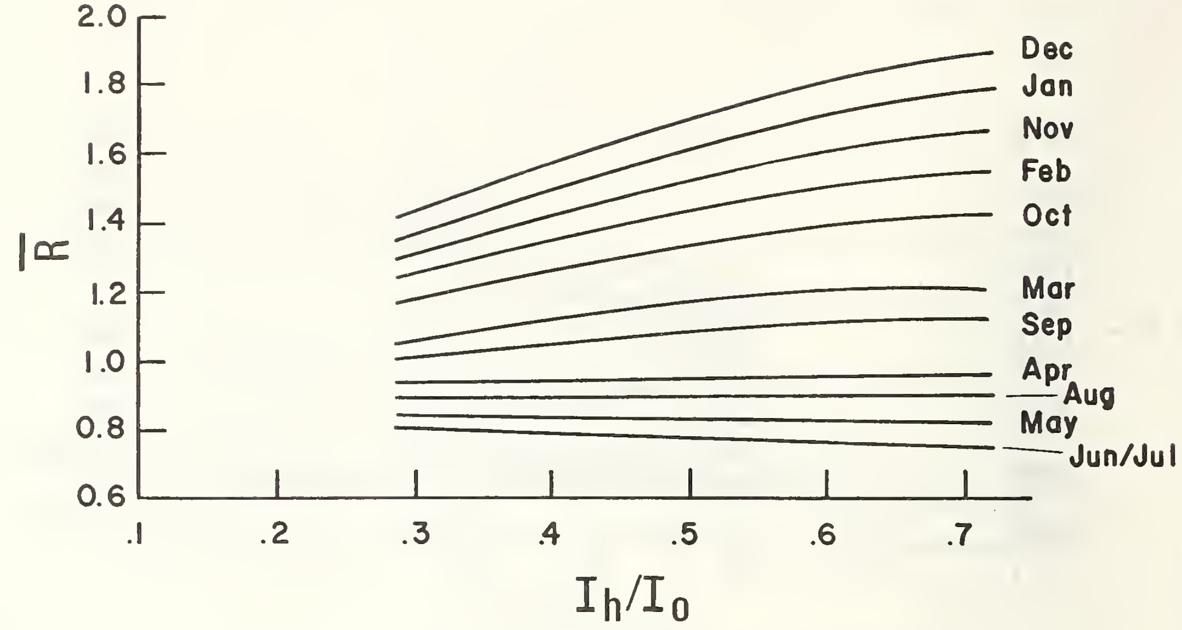


Figure B-15c 35^{o} North Latitude; 50^{o} Collector Tilt

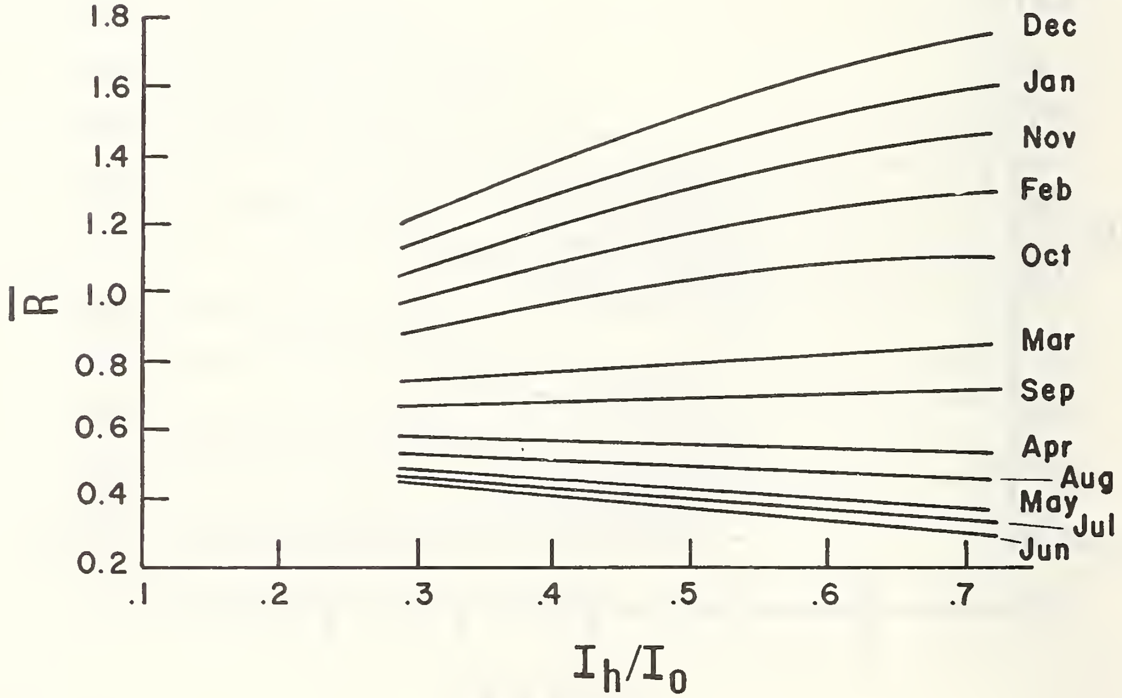


Figure B-15d 35^{o} North Latitude; 90^{o} Collector Tilt

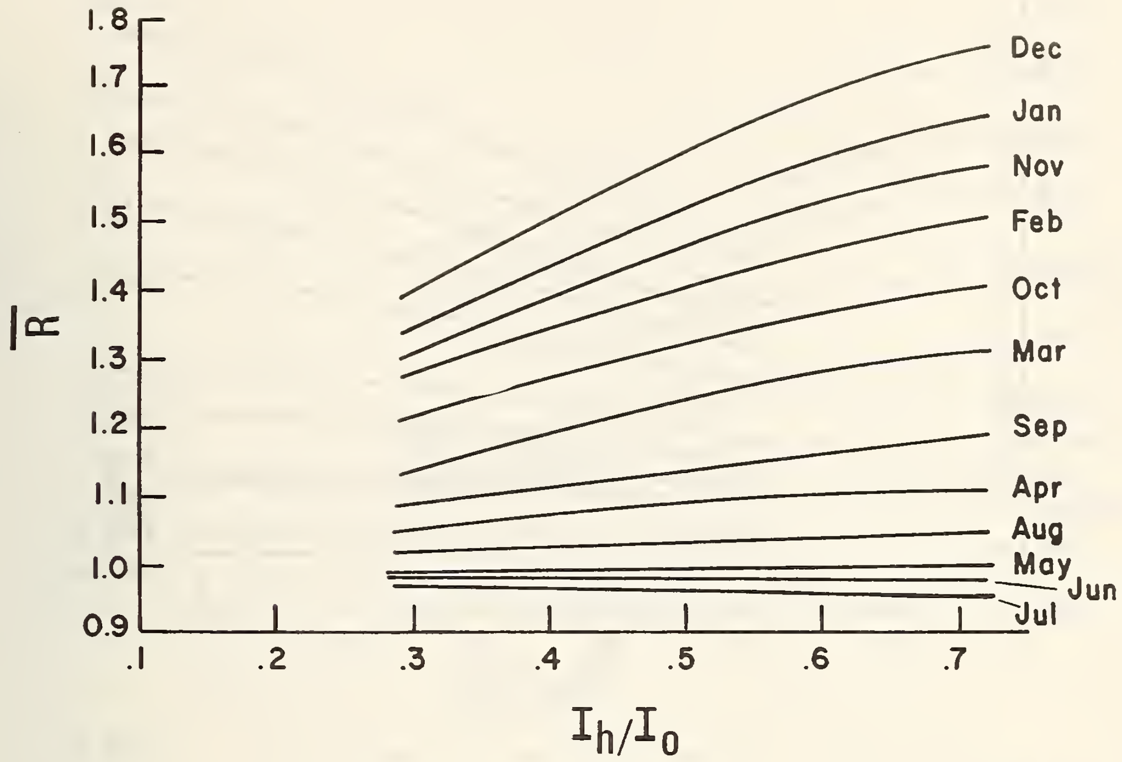


Figure B-16a 40° North Latitude; 25° Collector Tilt

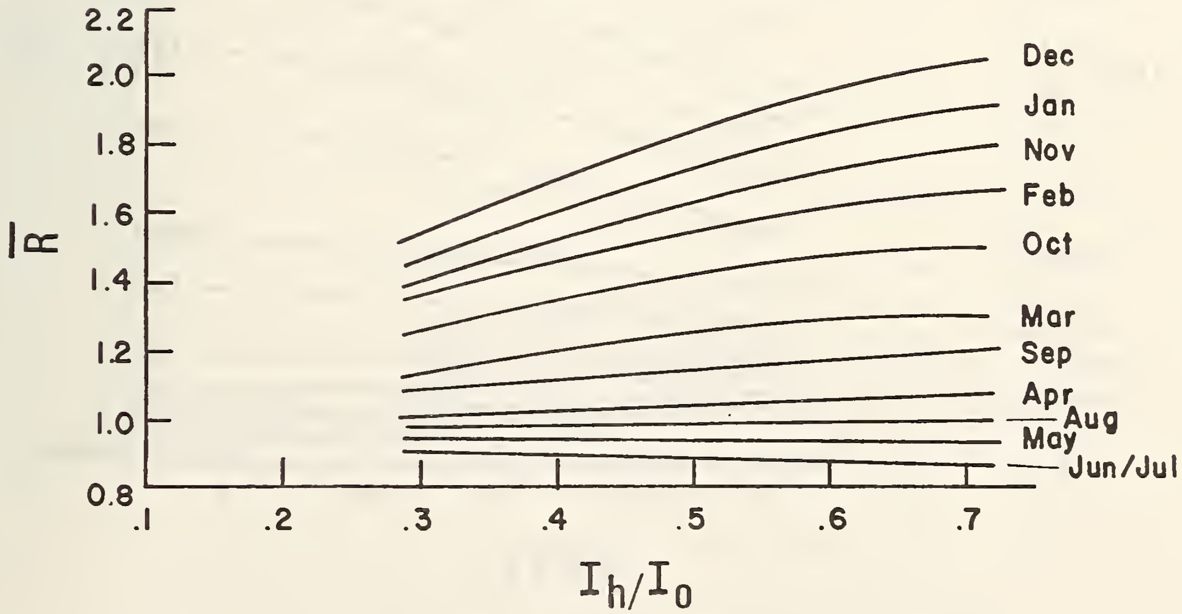


Figure B-16b 40° North Latitude; 40° Collector Tilt

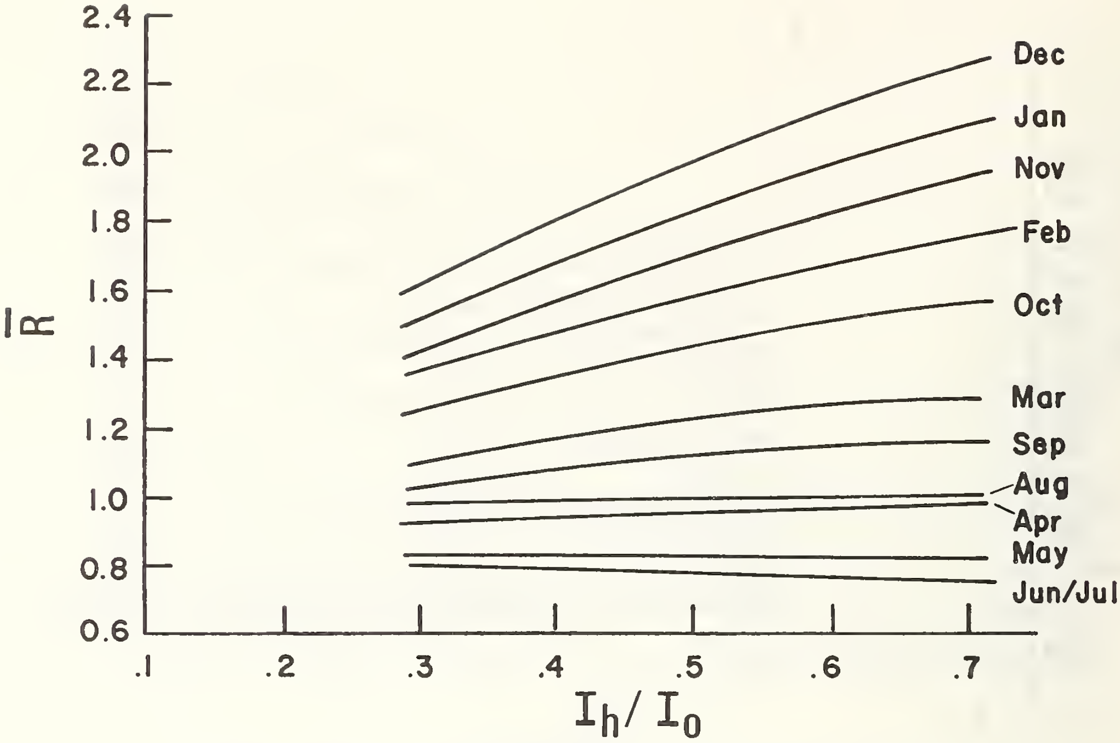


Figure B-16c 40° North Latitude; 55° Collector Tilt

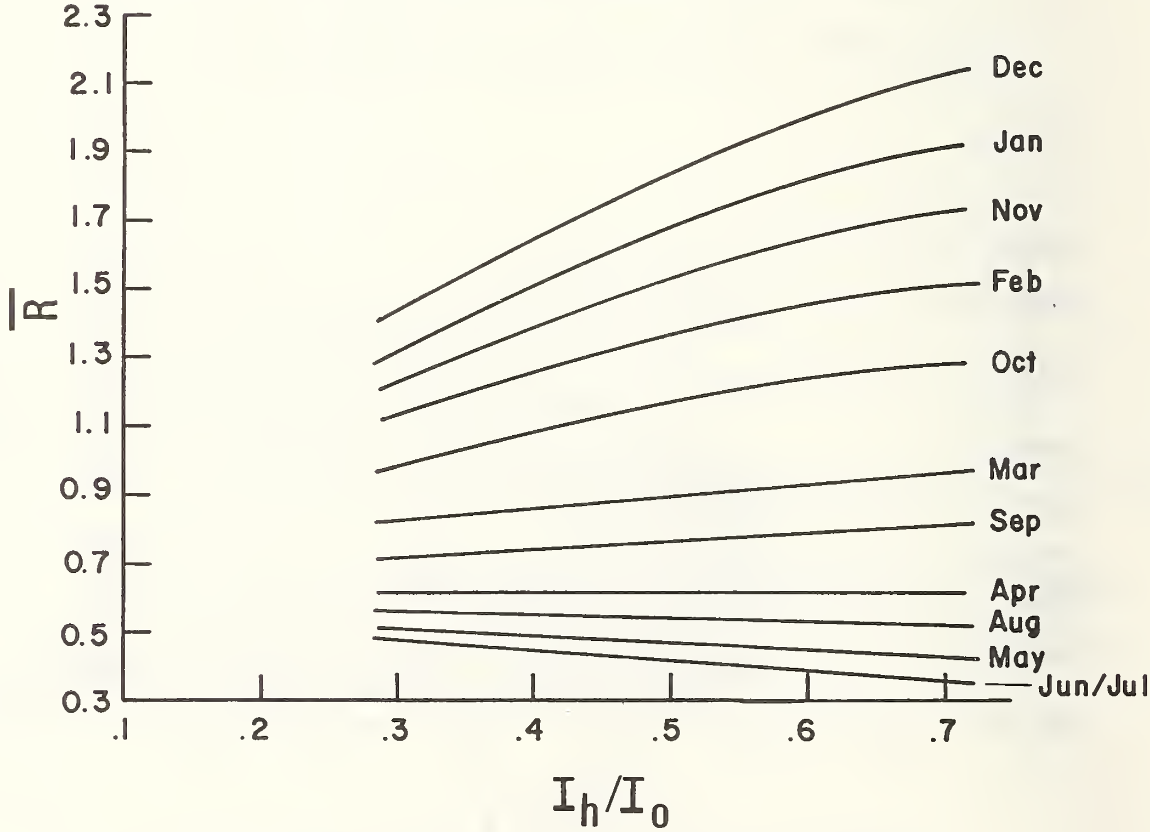


Figure B-16d 40° North Latitude; 90° Collector Tilt

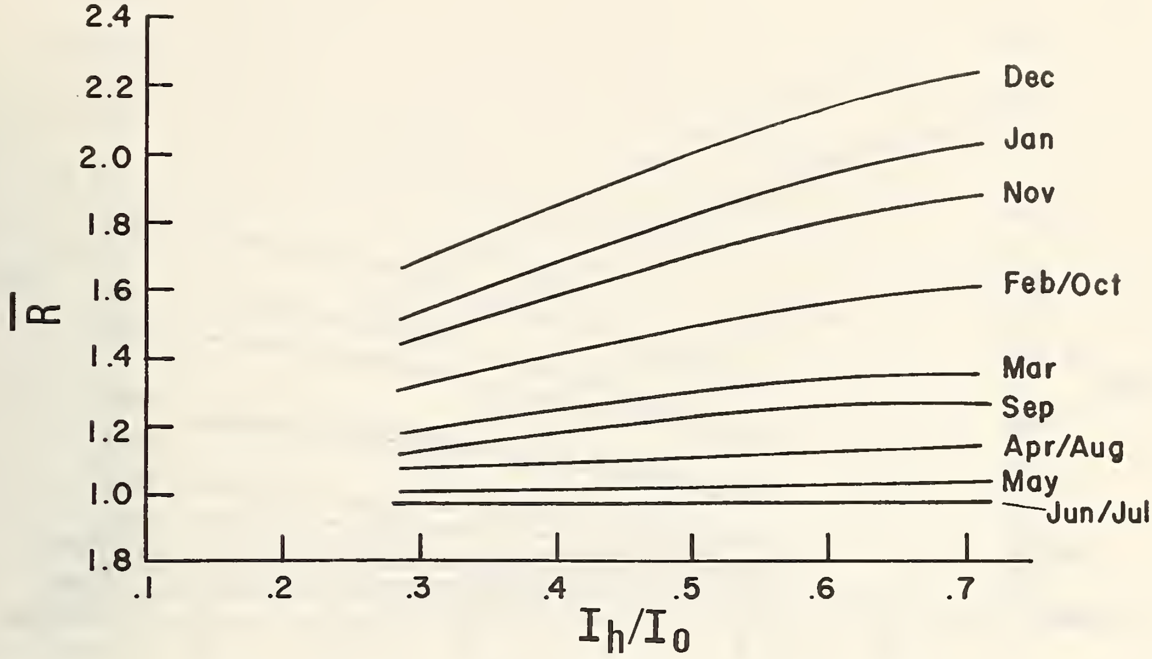


Figure B-17a 45° North Latitude; 30° Collector Tilt

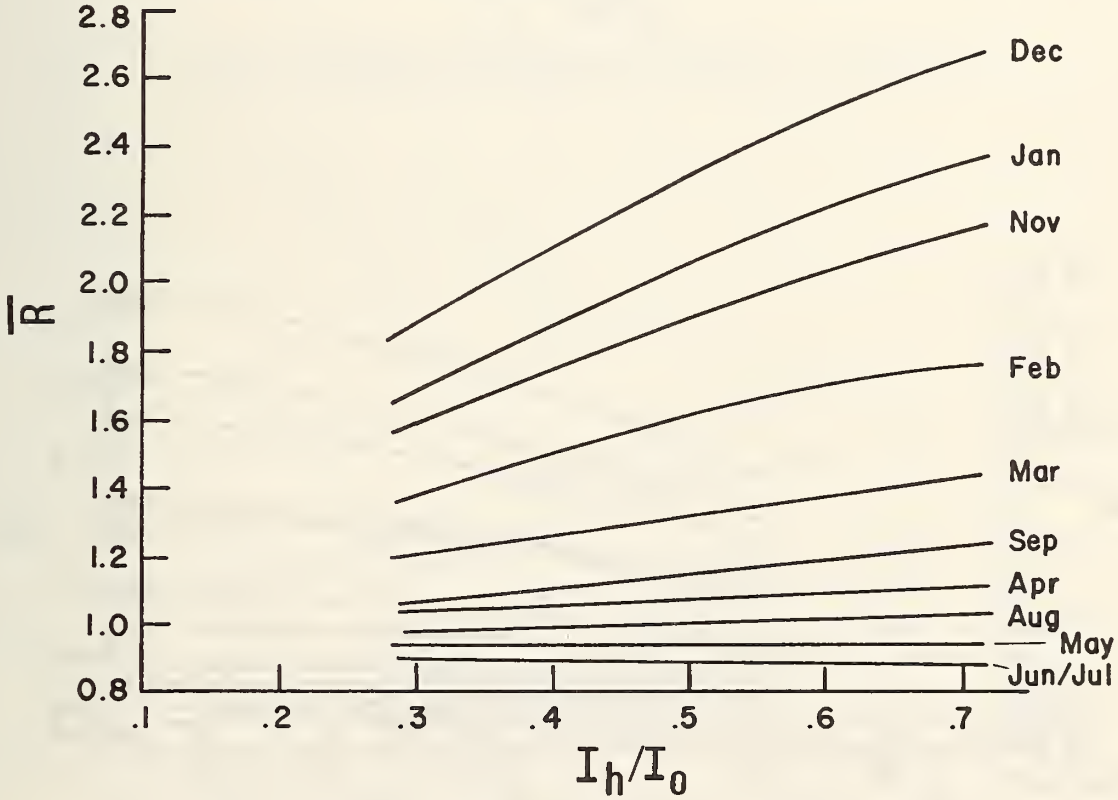


Figure B-17b 45° North Latitude; 45° Collector Tilt

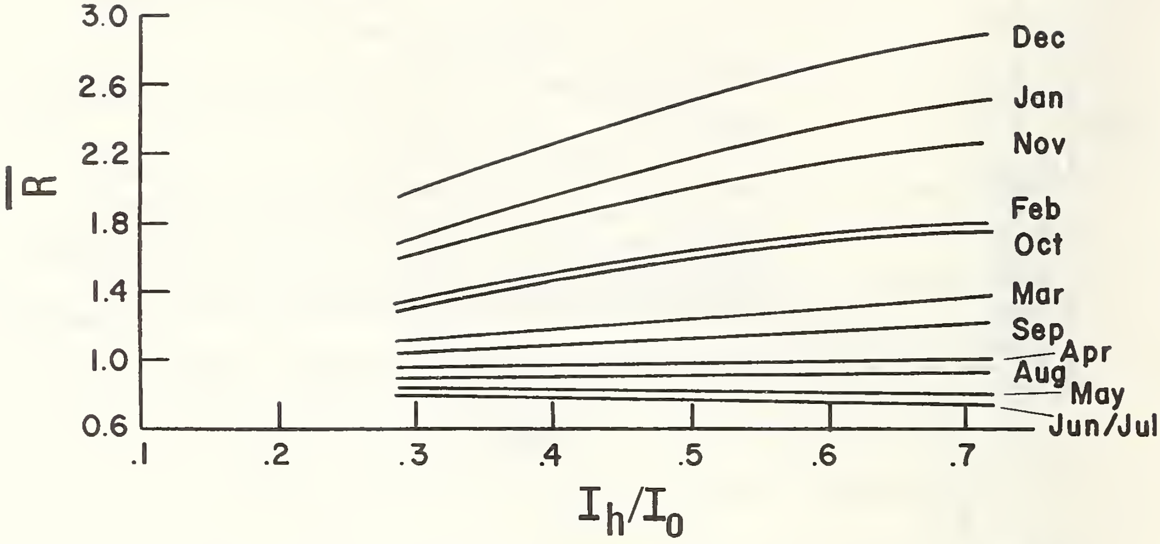


Figure B-17c 45° North Latitude; 60° Collector Tilt

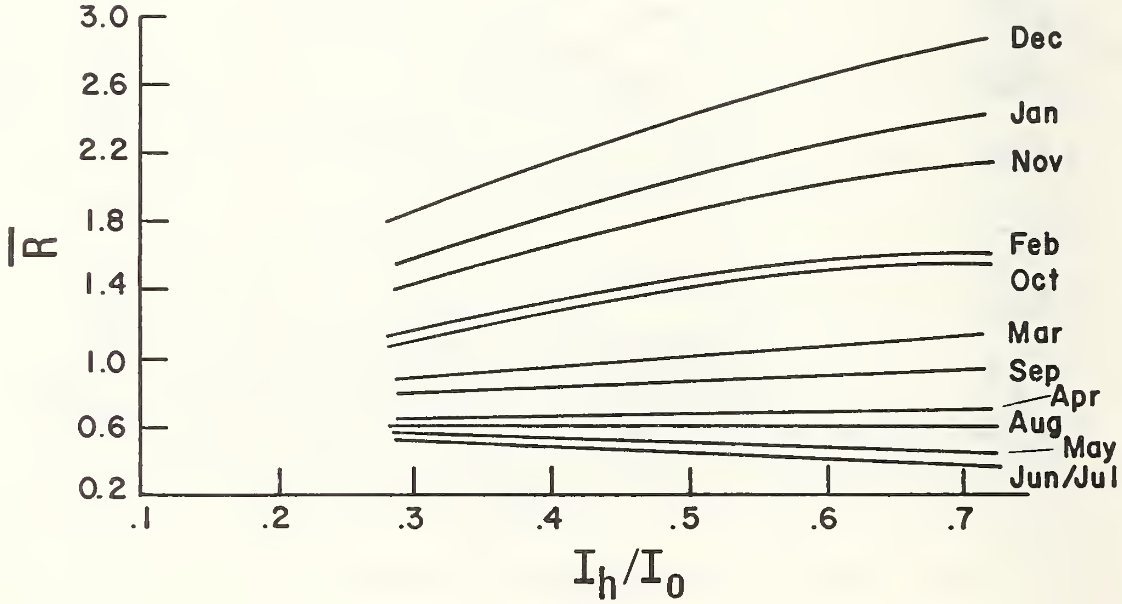


Figure B-17d 45° North Latitude; 90° Collector Tilt

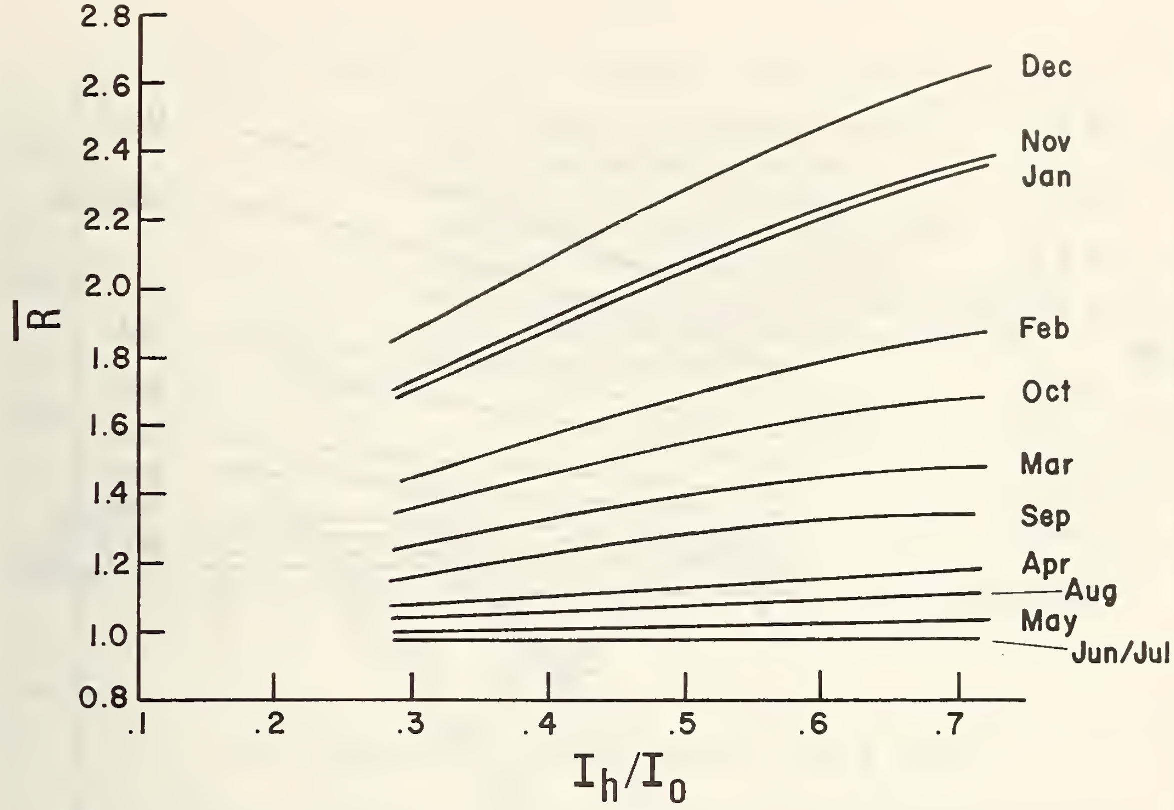


Figure B-18a 50° North Latitude; 35° Collector Tilt

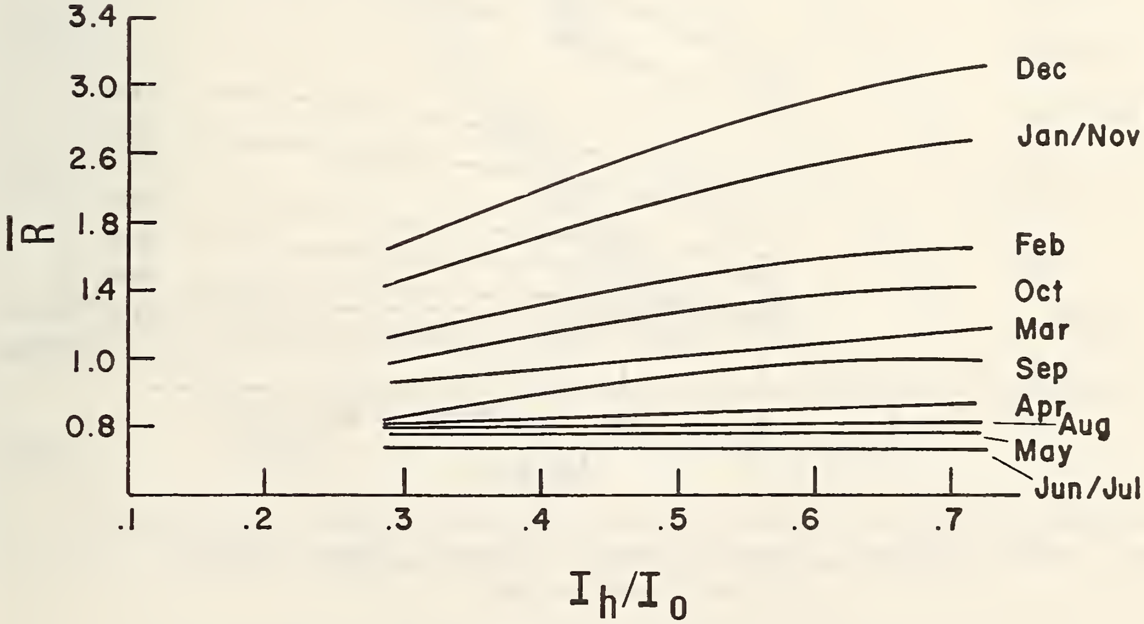


Figure B-18b 50° North Latitude; 50° Collector Tilt

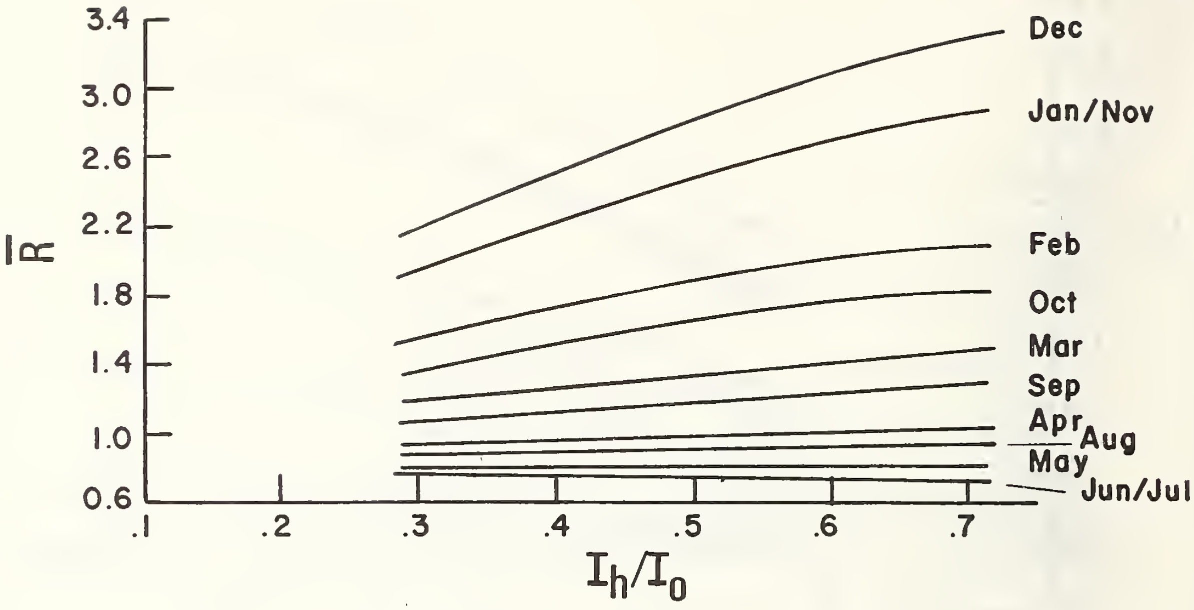


Figure B-18c 50° North Latitude; 65° Collector Tilt

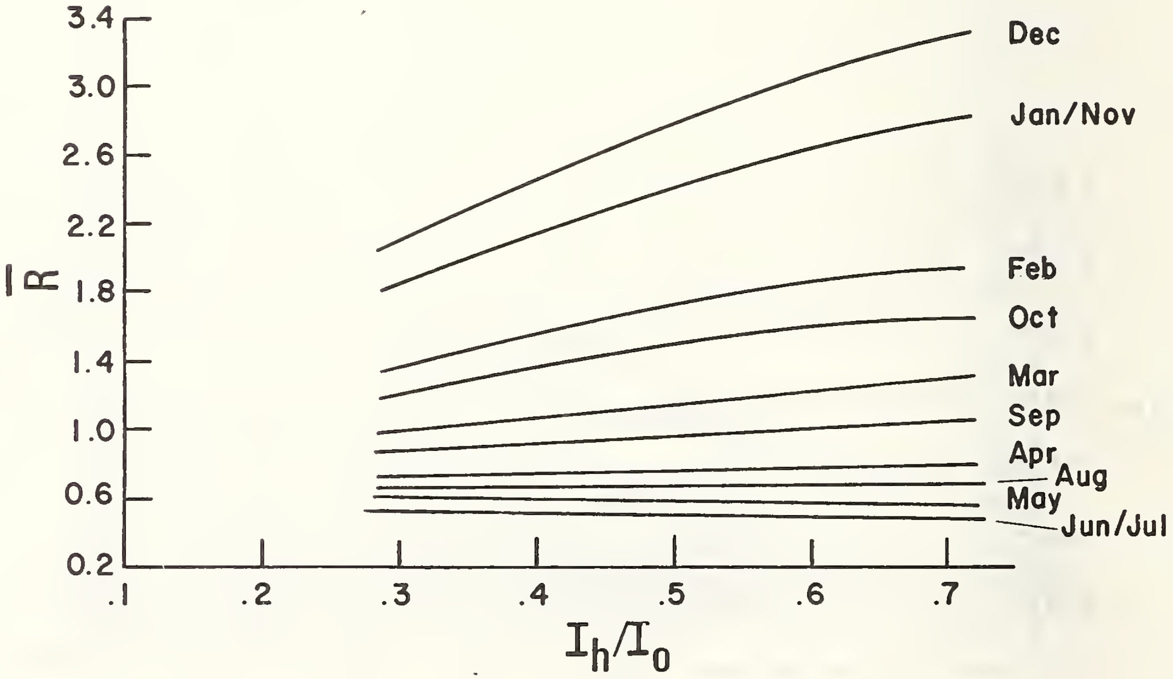


Figure B-18d 50° North Latitude; 90° Collector Tilt

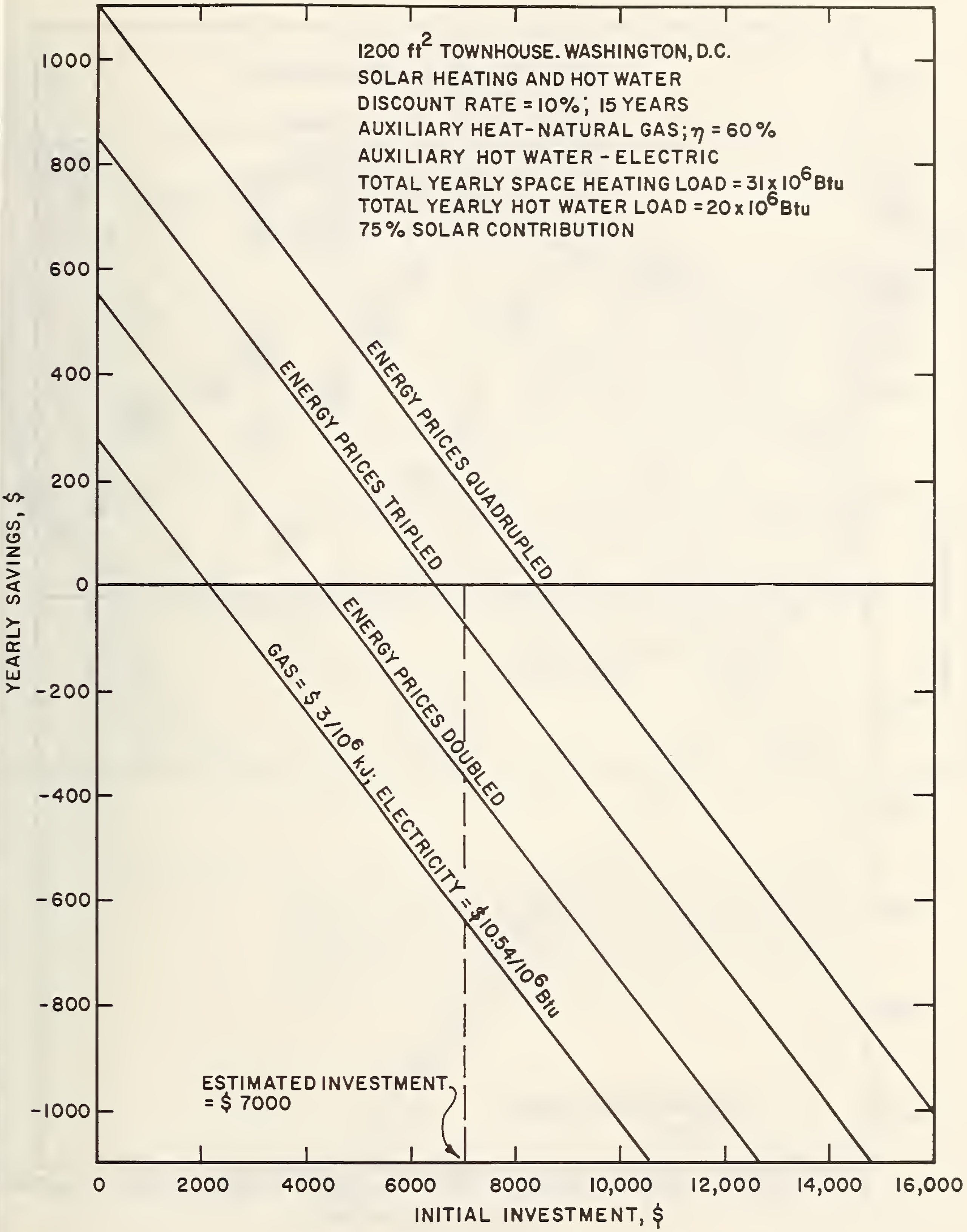


Figure B-19 Results of Cost Analysis for Example Problem, Solar Heating and Hot Water

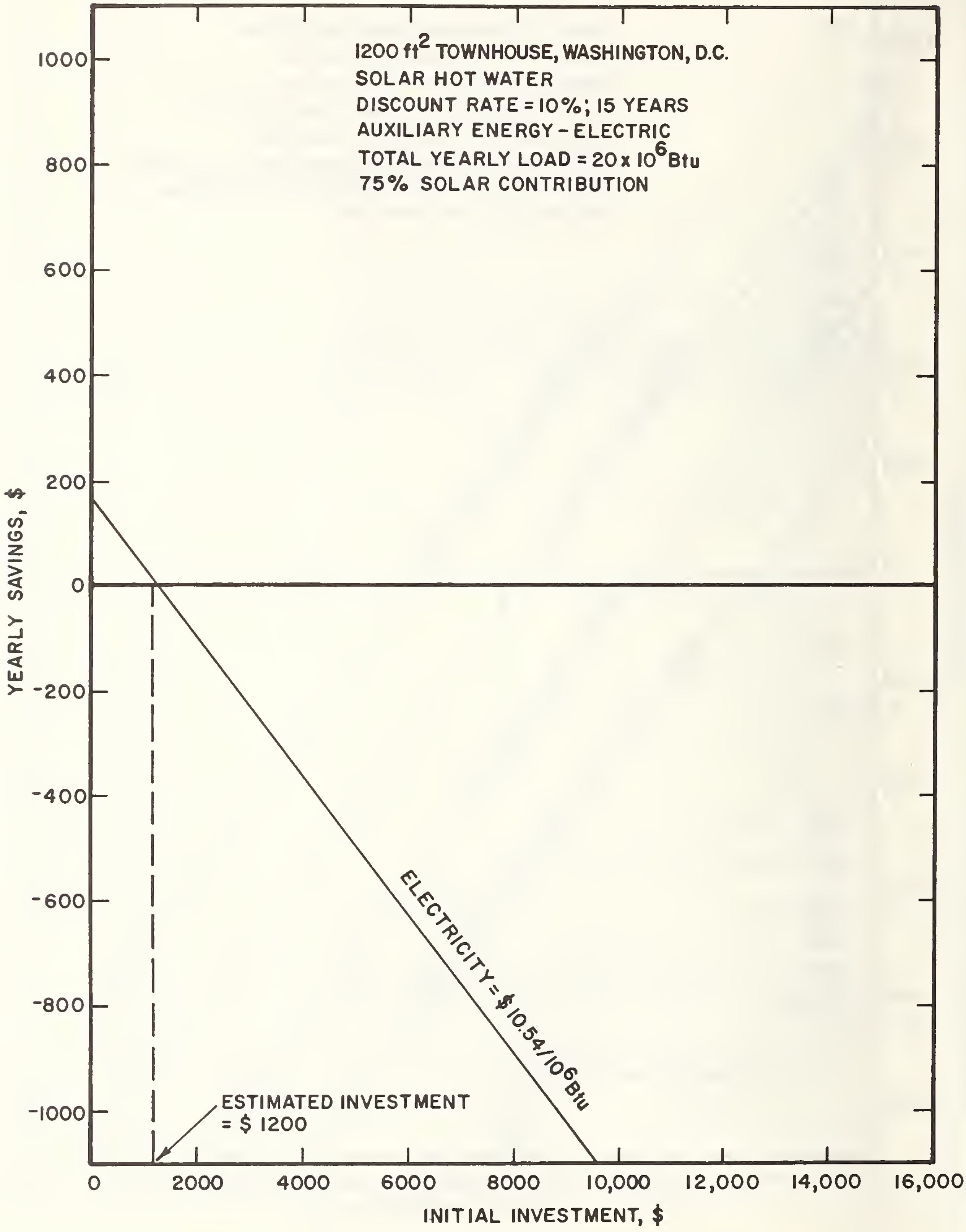


Figure B-20 Results of Cost Analysis for Example Problem, Solar Hot Water

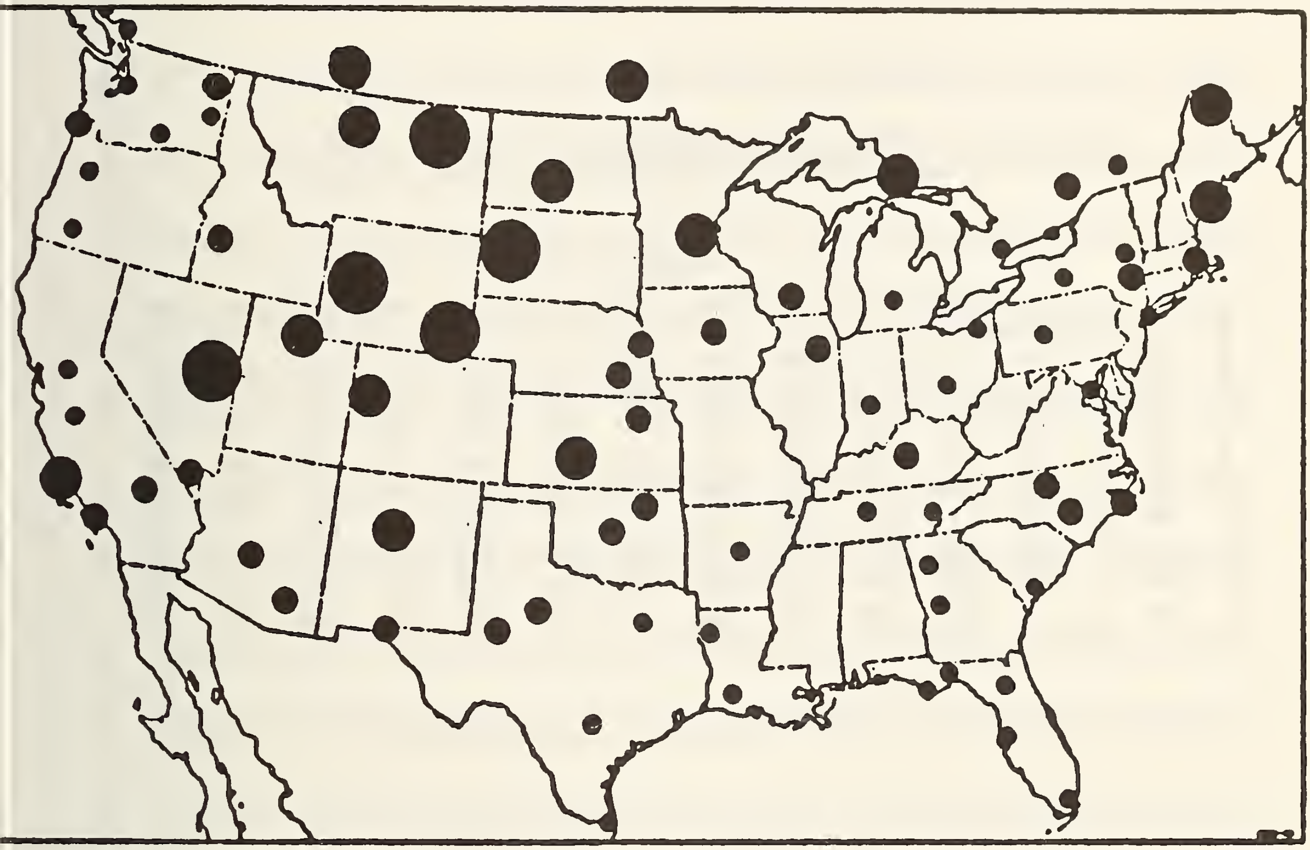


Figure B-21 Map of Net Annual Dollar Savings for Space and Domestic Water Heating with Solar Energy in a Residential Building. Small to Large Dots Represent Net Annual Savings of 0-100, 100-200, 200-300, and over 300 $/year [31]

Table B-1 AVERAGE DAILY EXTRATERRESTRIAL SOLAR ENERGY RECEIVED ON A HORIZONTAL SURFACE (ly/day)

North Latitude, degrees	Months of Year											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
20	636	725	819	896	929	935	929	904	843	750	657	609
25	571	671	785	806	940	956	946	904	818	702	585	541
30	502	613	744	870	945	972	957	896	786	650	529	471
35	432	551	697	848	944	982	962	883	749	593	461	399
40	359	485	646	819	937	987	962	864	706	532	390	325
45	287	417	589	785	925	987	957	839	657	467	319	253
50	215	346	529	745	908	983	947	808	604	399	247	182
55	145	274	464	701	887	976	934	774	546	329	177	114
60	81	203	396	652	864	969	920	735	484	258	110	54

Table B-2 AVERAGE DAILY TERRESTRIAL SOLAR ENERGY RECEIVED ON A HORIZONTAL SURFACE (ly/day)

LOCATION	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	ANNUAL
Annette, Ak	63	113	231	360	457	466	481	352	266	122	59	40	251
Barrow, Ak	0	38	180	380	513	528	429	255	115	41	0	0	206
Bethel, Ak	38	108	282	444	457	454	376	252	202	115	44	22	233
Fairbanks, Ak	16	71	213	376	461	504	434	317	180	82	26	6	224
Matanuska, Ak	32	92	242	356	436	462	409	314	198	100	38	15	224
Little Rock, Ar	188	260	353	446	523	559	556	518	439	343	244	187	385
Page, Az	294	367	516	618	695	707	680	596	516	402	310	243	495
Phoenix, Az	301	409	526	638	724	739	658	613	566	449	344	281	520
Tucson, Az	315	391	540	655	729	699	626	588	570	442	356	305	518
Yuma, Az	305	401	517	633	703	705	652	587	530	442	330	271	506
Davis, Ca	158	256	402	528	636	702	690	611	498	348	216	148	433
Eureka, Ca	146	194	306	399	471	494	462	406	342	249	159	131	313
Fresno, Ca	186	296	438	545	637	697	668	606	503	375	241	160	446
Inyokern, Ca	312	419	578	701	789	836	784	738	648	484	366	295	579
La Jolla, Ca	244	302	397	457	506	487	497	464	389	320	277	221	380
Los Angeles-WBAS, Ca	248	331	470	515	572	596	641	581	503	373	289	241	447
Los Angeles-WBO, Ca	243	327	436	483	555	584	651	581	500	362	281	234	436
Pasadena, Ca	251	333	439	509	569	580	634	599	482	366	271	236	439
Riverside, Ca	271	362	468	526	608	666	652	603	521	400	309	260	470
San Mateo, Ca	195	282	409	512	577	598	540	477	425	332	229	176	396
Santa Maria, Ca	263	346	482	552	635	694	680	613	524	419	313	252	481
Soda Springs, Ca	223	316	374	551	615	691	760	681	510	357	248	182	459
Boulder, Co	201	268	401	460	460	525	520	439	412	310	222	182	367
Grand Junction, Co	227	324	434	546	615	708	676	595	514	373	260	212	457
Grand Lake, Co	212	313	423	512	552	632	600	505	476	361	234	184	416
Washington, DC	158	231	322	398	467	510	496	440	364	278	192	141	333
Aplachicola, Fl	298	367	441	535	603	578	529	511	456	413	332	262	444
Belle Isle, Fl	297	330	412	463	483	464	488	461	400	366	313	291	397
Gainesville, Fl	278	367	445	539	586	544	520	508	444	368	318	254	431
Jacksonville, Fl	267	346	423	514	556	525	522	476	383	331	274	230	404
Key West, Fl	327	410	490	572	579	543	534	501	445	394	332	292	452
Miami, Fl	343	416	491	544	552	531	537	508	447	389	354	319	453
Pensacola, Fl	250	321	405	509	562	568	537	509	430	394	278	224	416
Tallahassee, Fl	274	311	423	483	548	476	544	537	424	353	364	260	416
Tampa, Fl	327	391	474	539	596	574	534	494	454	400	356	300	453
Atlanta, Ga	228	284	377	484	535	554	538	502	412	350	265	201	394
Griffin, Ga	238	302	388	519	577	580	559	523	437	372	288	210	416
Honolulu, Hi	363	422	516	559	617	615	615	612	573	507	426	371	516
Pearl Harbor, Hi	355	404	438	536	577	562	610	575	536	466	393	349	483
Boise, Id	138	236	342	485	585	636	670	576	460	301	182	124	395
Twin Falls, Id	163	240	355	462	552	592	602	540	432	286	176	131	378

Table B-2 (cont) AVERAGE DAILY TERRESTRIAL SOLAR ENERGY RECEIVED ON A HORIZONTAL SURFACE (ly/day)

LOCATION	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	ANNUAL
Chicago, Il	96	147	227	331	424	458	473	403	313	207	120	76	273
Lemont, Il	171	232	326	390	497	553	527	486	384	265	157	131	343
Moline, Il	159	220	317	402	493	558	565	498	407	290	176	134	352
Indianapolis, In	147	214	312	393	491	547	542	486	405	293	176	130	345
Ames, Ia	174	253	326	403	480	541	436	460	367	274	187	143	345
Dodge City, Ks	255	316	418	528	568	650	642	592	493	380	285	234	447
Kansas City, Ks	182	251	342	441	522	589	579	525	426	327	215	164	380
Manhattan, Ks	192	264	345	433	527	551	531	526	410	292	227	156	371
Topeka, Ks	192	249	337	430	505	554	552	512	424	320	214	165	371
Lexington, Ky	172	263	357	480	581	628	617	563	494	357	245	175	411
Louisville, Ky	164	231	325	420	515	560	550	498	408	303	190	150	360
Lake Charles, La	239	304	396	483	554	582	521	506	448	402	296	232	414
New Orleans, La	237	296	393	479	539	549	502	491	418	389	269	220	399
Shreveport, La	232	292	384	446	558	557	578	528	414	354	254	205	400
Blue Hill, Ma	153	228	319	389	469	510	502	449	354	266	162	135	328
Boston, Ma	139	198	293	364	472	499	496	425	341	238	145	119	311
Cambridge, Ma	153	235	323	400	420	476	482	464	367	253	164	124	322
East Wareham, Ma	140	218	305	385	452	508	495	436	365	258	163	140	322
Lynn, Ma	118	209	300	394	454	549	528	432	341	241	135	107	317
Annapolis, Md	175	243	340	419	488	557	542	469	383	294	189	155	355
Silver Hill, Md	182	244	340	438	513	555	516	459	397	295	202	163	359
Caribou, Me	133	231	364	400	476	470	508	448	336	212	111	107	316
Portland, Me	157	237	359	406	513	541	561	482	383	273	157	138	351
East Lansing, Mi	121	210	309	359	483	547	540	466	373	255	136	108	311
Sault Ste Marie, Mi	130	225	356	416	523	557	573	472	322	216	105	96	333
St. Cloud, Mn	170	251	366	423	499	541	555	491	360	241	146	123	348
Columbia, Mo	173	251	340	434	530	574	574	522	453	322	225	158	380
Glasgow, Mt	154	258	385	466	568	605	645	531	410	267	154	116	388
Great Falls, Mt	140	232	366	434	528	583	639	532	407	264	154	112	366
Summit, Mt	122	162	268	414	462	493	560	510	354	216	102	76	312
Lincoln, Ne	188	259	350	416	494	544	568	484	396	296	199	159	363
North Omaha, Ne	193	229	365	463	516	546	568	519	410	298	204	170	379
North Platte, Ne	200	266	358	475	523	599	598	540	432	322	220	178	393
Bismarck, N.D.	157	250	356	447	550	590	617	516	390	272	161	124	369
Cape Hatteras, NC	244	317	432	571	635	645	629	557	472	361	284	216	447
Greensboro, NC	200	276	354	469	531	564	544	485	406	322	243	197	383
Sea Brook, NJ	157	227	318	403	478	522	518	457	385	285	192	139	340
Trenton, NJ	173	244	343	424	501	546	540	469	389	294	195	155	355
Albuquerque, NM	303	386	511	618	686	726	683	626	554	438	334	276	512
Ely, Nv	238	333	464	564	624	708	648	608	519	393	287	220	467
Las Vegas, Nv	277	384	519	621	702	748	675	627	551	429	318	258	509
Reno, Nv	234	324	449	592	664	714	707	646	532	395	277	209	479
Ithaca, NY	116	194	272	334	440	501	515	453	346	231	120	96	302

Table B-2 (cont) AVERAGE DAILY TERRESTRIAL SOLAR ENERGY RECEIVED ON A HORIZONTAL SURFACE (ly/day)

LOCATION	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	ANNUAL
New York, NY	146	210	312	378	455	526	518	492	361	262	160	128	324
Sayville, NY	160	249	335	415	494	565	543	462	385	289	186	142	352
Schenectady, NY	130	200	273	338	413	448	441	397	299	218	128	104	282
Upton, NY	155	232	339	428	502	573	543	475	391	293	182	146	355
Cleveland, Oh	125	183	303	286	502	562	562	494	278	289	141	115	320
Columbus, Oh	128	200	297	391	471	562	542	477	422	286	176	129	340
Put in Bay, Oh	126	204	302	386	468	544	561	487	382	275	144	109	332
Oklahoma City, Ok	255	317	407	498	540	623	610	588	484	379	284	237	435
Stillwater, Ok	205	289	390	454	504	600	596	545	455	354	269	209	405
Astoria, Or	90	162	270	375	492	469	539	461	354	209	111	79	301
Medford, Or	116	169	216	317	429	491	497	409	339	207	118	77	282
Philadelphia, Pa	175	242	347	425	493	554	538	465	388	293	191	152	355
State College, Pa	139	202	297	373	467	544	528	454	361	275	155	120	335
Newport, RI	115	231	330	395	489	538	517	449	380	273	175	141	339
Charleston, SC	250	308	393	517	553	556	523	495	417	349	281	228	406
Rapid City, SD	183	277	400	482	532	585	590	541	435	315	204	158	392
Oak Ridge, Tn	161	239	331	450	518	551	526	478	416	318	213	163	364
Memphis, Tn	192	267	359	470	554	589	583	535	442	354	238	184	397
Nashville, Tn	163	240	329	450	517	567	553	494	428	327	217	161	370
Brownsville, Tx	287	336	402	458	556	604	619	555	465	406	284	253	435
Corpus Cristi, Tx	262	330	413	474	561	604	629	558	470	408	285	240	436
Dallas, Tx	231	307	394	454	521	595	588	538	458	363	261	221	411
El Paso, Tx	331	432	549	655	715	730	670	639	575	462	367	313	536
Fort Worth, Tx	250	320	427	488	562	651	613	593	503	403	306	245	477
Midland, Tx	283	258	476	550	611	617	608	574	552	396	325	275	466
San Antonio, Tx	279	347	417	455	541	612	639	585	493	398	295	256	442
Flaming Gorge, Ut	238	298	443	522	565	650	599	538	425	352	262	215	426
Salt Lake City, Ut	163	256	354	479	570	621	620	551	446	316	204	146	394
Norfolk, Va	208	270	372	477	540	572	550	481	398	310	223	184	382
Burlington, Vt	129	198	300	367	495	530	532	455	343	231	124	103	317
Friday Harbor, Wa	87	157	274	418	514	578	586	507	351	194	102	75	320
Pullman, Wa	111	205	304	462	558	653	699	562	410	245	146	96	372
Prosser, Wa	117	222	351	521	616	680	707	604	458	274	136	100	399
Seattle, Wa	70	124	244	360	446	471	501	431	310	174	90	59	273
Spokane, Wa	119	204	321	474	563	596	665	556	404	225	131	75	361
Tacoma, Wa	75	139	265	403	503	511	566	452	324	188	104	64	300
Greenbay, Wi	137	210	312	385	490	542	539	462	353	240	139	110	327
Madison, Wi	148	220	313	394	466	514	531	452	348	241	145	115	324
Milwaukee, Wi	149	210	312	403	509	565	562	485	392	267	161	120	345
Lander, Wy	226	324	452	548	587	678	651	586	472	354	239	196	443
Laramie, Wy	216	295	424	508	554	643	606	536	438	324	229	186	408

Table B-3 CAPITAL RECOVERY FACTORS FOR INTEREST RATES FROM 0% to 25%

n \ i	0%	2%	4%	6%	8%	10%	12%	15%	20%	25%
1	1.00000	1.02000	1.04000	1.06000	1.08000	1.10000	1.12000	1.15000	1.20000	1.25000
2	0.50000	0.51505	0.53020	0.54544	0.56077	0.57619	0.59170	0.61512	0.65455	0.69444
3	0.33333	0.34675	0.36035	0.37411	0.38803	0.40211	0.41635	0.43798	0.47473	0.51230
4	0.25000	0.26262	0.27549	0.28859	0.30192	0.31547	0.32923	0.35027	0.38629	0.42344
5	0.20000	0.21216	0.22463	0.23740	0.25046	0.26380	0.27741	0.29832	0.33438	0.37184
6	0.16667	0.17853	0.19076	0.20336	0.21632	0.22961	0.24323	0.26424	0.30071	0.33882
7	0.14286	0.15451	0.16661	0.17914	0.19207	0.20541	0.21912	0.24036	0.27742	0.31634
8	0.12500	0.13651	0.14853	0.16104	0.17401	0.18744	0.20130	0.22285	0.26061	0.30040
9	0.11111	0.12252	0.13449	0.14702	0.16008	0.17364	0.18768	0.20957	0.24808	0.28876
10	0.10000	0.11133	0.123 29	0.13587	0.14903	0.16275	0.17698	0.19925	0.23852	0.28007
11	0.09091	0.10218	0.11415	0.12679	0.14008	0.15396	0.16842	0.19107	0.23110	0.27349
12	0.08333	0.09456	0.10655	0.11928	0.13270	0.14676	0.16144	0.18448	0.22526	0.26845
13	0.07692	0.08812	0.10014	0.11296	0.12652	0.14078	0.15568	0.17911	0.22062	0.26454
14	0.07143	0.08260	0.09467	0.10758	0.12130	0.13575	0.15087	0.17469	0.21689	0.26150
15	0.06667	0.07783	0.08994	0.10296	0.11683	0.13147	0.14682	0.17102	0.21388	0.25912
16	0.06250	0.07365	0.08582	0.09895	0.11298	0.12782	0.14339	0.16795	0.21144	0.25724
17	C.05882	0.06997	0.08220	0.09544	0.10963	0.12466	0.14046	0.16537	0.20944	0.25576
18	0.05556	0.06670	0.07899	0.09236	0.10670	0.12193	0.13794	0.16319	0.20781	0.25459
19	0.05263	0.06378	0.07614	0.08962	0.10413	0.11955	0.13576	0.16134	0.20646	0.25366
20	0.05000	0.06116	0.07358	0.08718	0.10185	0.11746	0.13388	0.15976	0.20536	0.25292
25	0.04000	0.05122	0.06401	0.07823	0.09368	0.11017	0.12750	0.15470	0.20212	0.25095
30	0.03333	0.04465	0.05783	0.07265	0.08883	0.10608	0.12414	0.15230	0.20085	0.25031
40	0.02500	0.03656	0.05052	0.06646	0.08386	0.10226	0.12130	0.15056	0.20014	0.25003
50	0.02000	0.03182	0.04655	0.06344	0.08174	0.10086	0.12042	0.15014	0.20002	0.25000
100	0.01000	0.02320	0.04081	0.06018	0.08004	0.10001	0.12000	0.15000	0.20000	0.25000
∞		0.02000	0.04000	0.06000	0.08000	0.10000	0.12000	0.15000	0.20000	0.25000

APPENDIX C. HEAT AND CHILLED WATER DISTRIBUTION SYSTEMS

CONTENTS

APPENDIX C. HEAT AND CHILLED WATER DISTRIBUTION SYSTEMS

1. Introduction

Many Air Force Bases utilize heat distribution systems (both underground and aboveground) to carry heat and/or chilled water to outlying buildings from a central power plant. Most of these systems were, however, constructed immediately following World War II. The underground systems were usually selected on the following basis:

(1) It had been established by careful economic analysis that the central heating and cooling plant with underground heat and chilled water distribution network is most cost effective for heating and cooling of sprawling industrial complexes, campus-type housing projects, shopping centers, university campuses and Air Force Bases.

(2) Better utilization of expensive ground-surface area was possible as compared to the aboveground piping network, which tends to crowd, clutter and obstruct the valuable surface area.

(3) The heat loss is less, due to milder ground-temperature conditions around the pipe than for aboveground pipes which are exposed to severe ambient temperature fluctuations. This is especially true for the chilled-water distribution systems.

(4) If properly installed, piping systems can be protected to minimize the exposure to severe weather as well as to traffic abuse. (Some of the underground systems have been in operation for more than 20 years with no appreciable reduction in their thermal performance.)

Many systems were, however, installed hastily without thorough engineering analysis, field experience, and technical data, which has resulted in numerous failures manifested in the leakage of steam and heat. In some installations, pipe heat loss amounts to as much as 10 percent of the total steam heat generated at the plant.

Most common cause of underground system failure is the seepage of ground water into the thermal insulation through poorly constructed and/or installed insulation envelopes. Examples:

- ° Thin black-iron casings, sometimes painted with a light bituminous material, were used as a conduit. Low-melt asphalt materials were poured around the insulated pipe.
- ° Insulating concretes containing large amounts of construction water were utilized.
- ° Prefabricated metal conduits with little, if any, space for drainage and drying, utilized a loose-fill insulation material.
- ° Pipe supports which consisted only of the stub end of a bolt were being used to carry piping weight.
- ° Shredded rubber made from used tires mixed with a cement binder was installed, and with little supervision.
- ° Some of the insulating materials used were hygroscopic (absorbing moisture instead of hydrophobic (repelling ground water).

\- Careful evaluation of the thermal performance of existing underground systems, therefore, should be conducted to determine the thermal distribution efficiency of the plant before deciding whether to replace or to repair either all or part of an underground distribution system.

Aboveground systems are also very common in many Air Force Bases, especially in localities where the ground water table is high and where the ground surface is not congested. The aboveground heat-distribution systems have several advantages over underground systems in that they are easy to install, inspect, maintain, repair and replace. However, certain shortcomings inherent in the aboveground systems must be recognized.

1. They are unsightly and occupy precious and expensive surface areas, often causing inconvenience and unwanted obstruction to surface traffic.
2. They are easily damaged by external mechanical shocks, vandalism and accidental damage from passing vehicles.
3. In some localities, they are subject to much more severe weathering cycles than underground systems, such as excessively

large temperature changes. They are also exposed to vibrations, corrosive salt air, rain and snow loading.

4. The freeze-up of the system due to the failure of heat supply is also of serious consequence.

In order to assist the Air Force Base engineers in evaluating their underground systems, this section is prepared to briefly describe the various types of systems used, methods of determining the heat loss characteristics of existing systems, and methods of arriving at economic optimal pipe insulation thicknesses for the retrofit design. A comprehensive bibliography is provided at the end of this chapter for those who are interested in further information regarding underground heat distribution technology.

2. TYPICAL HEAT DISTRIBUTION SYSTEMS

2.1 Concrete Tunnel/Trench System

Concrete trench/tunnel system is a utility tunnel or a poured-in-place reinforced concrete box with immovable concrete slab. Pipes are insulated with preformed insulation and are supported on steel rollers of some type which are embedded in the sides of the concrete walls. The concrete trench tunnel is extremely durable, almost indestructible, but it is quite expensive to construct compared to other types of systems. The tunnel system is extremely well constructed and provides for operation, maintenance and checking. It should be well ventilated to keep its temperature below 100° F for personnel entry. The trench cover is, however, very difficult to make watertight. In fact, when the ground water level is high, concrete trenches frequently leak profusely. In order to ensure satisfactory operation, means must be provided to remove the water which drains into the trench. A modified version of the basic design is the half-round clay tile system (fig. C-1) and concrete conduit system (fig. C-2). These systems are no longer popular in this country.

2.2 Air-Space Conduit System

One of the most widely used underground systems is the prefabricated pressure-testable, drainable, driable system (fig. C-3). This type of

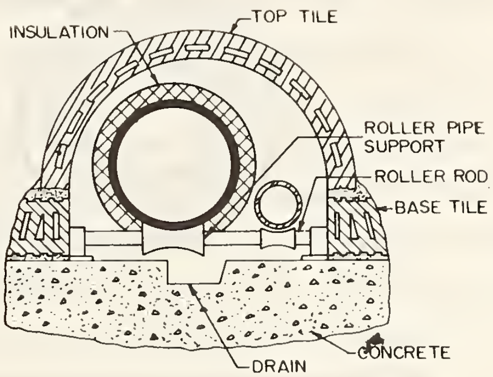


Figure C-1. Half-round clay tile trench system.

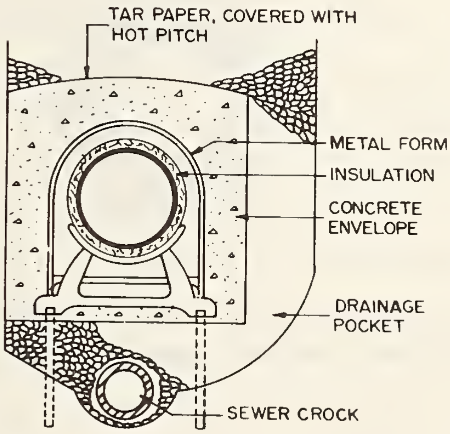


Figure C-2. Concrete conduit system.

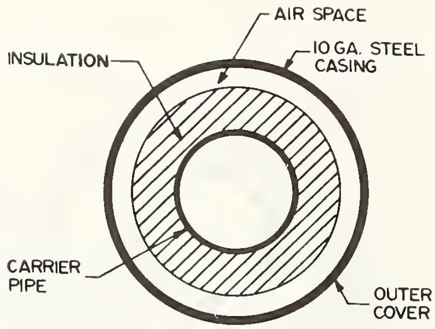


Figure C-3. Air-space conduit system (drainable, driable and testable).

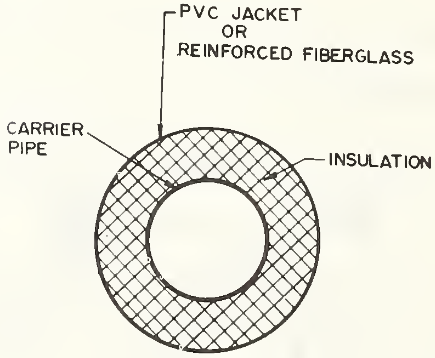


Figure C-4. Sealed non-air space conduit system.

system is sold by several manufacturers, each of whom has incorporated certain proprietary features in their product. The system consists of a steel outer casing which may be either smooth or corrugated, either galvanized or ungalvanized. One or more interior pipes are insulated with a preformed pipe insulation such as calcium silicate. Pipe support is usually achieved with a thermal insulating material. Various materials used to protect the outside casing from corrosion include cold tar enamel, glass-fiber reinforced epoxies, and glass-fiber reinforced asphalt compounds with a felt wrap. The system is prefabricated in the factory in 20- or 40-foot lengths which are welded together in the field. The completed system is intended to be pressure tight and hence water tight. Ordinarily, water tightness is verified by applying air pressure to the interior of the casing after welding and before back filling. Unless properly protected, however, its metallic outer casing is subject to corrosion. Even if water does seep into the casing during or after a rain storm it can be drained and the casing dried by removing drain and vent plugs at the end of the conduit.

2.3 Sealed Non-Air Space Conduit System

Figure C-4 shows a sealed non-air space conduit system, which is commonly used for chilled water or low temperature hot-water distribution systems. The system is available in a variety of configurations, but basically all systems of this type consist of a single carrier pipe enclosed with pipe insulation having a tight outer casing around the insulation. The most commonly used insulation for such systems is urethane foam, provided that the foam is not exposed to temperatures higher than 200° F. Calcium silicate and cellular glass insulation are used for high-temperature systems. The most popular outer jacket is glass-fiber reinforced epoxy or polyester. However, plastic materials such as PVC and asphalt-coated felt wraps are also used. Generally, such systems are prefabricated. In one version of this system, the jacket around the insulation is bonded to the carrier pipe at both ends of the carrier pipe so that each segment of the system is in effect a sealed unit. In most systems, however, insulation and jacket are continuous for the whole run of the system. Since many of these systems are not air pressure testable, neither drainable nor driable, it is absolutely essential that the insulation be kept dry under all conditions. This is rather difficult to achieve in practice.

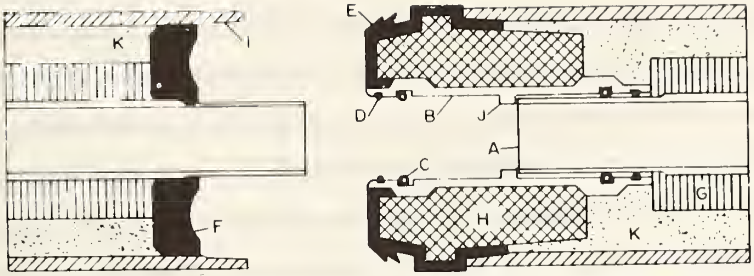
2.4 Sealed-Conduit System with Non-Welded Joint

Figure C-5 shows a sealed-conduit system comprised of a single carrier pipe surrounded by annular regions of calcium-silicate and urethane-foam insulation. This system is presently not used by the Air Force. Cellular glass insulation could be used instead of calcium silicate. The outer casing consists of an asbestos cement or plastic sheet. The system is prefabricated in 30-foot sections. Each section is sealed at both ends so that infiltrating water cannot migrate into other sections. In the field, sections are connected with holding seals and specially designed gaskets. This system is highly resistant to water infiltration, but the leakage of the carrier pipe fluid into the insulation is possible when the sealing rings fail. Its outer casing, being asbestos cement, is not subject to corrosion. The system usually is easier to install than the previously mentioned systems.

2.5 Insulating Concrete Envelope System

This system consists of a concrete-slab base which supports carrier piping surrounded by an envelope of insulating concrete. The envelope is wrapped in a water-resistant sheet material such as polyvinyl chloride. Drain holes are placed in the concrete envelope to permit the water originally present in insulating concrete during its construction to be drained off and discharged into a manhole. Various types of insulating concretes are used, but the most prevalent is portland cement/ vermiculite aggregate.

During the past few years, a new type of insulating concrete employing a polystyrene bead aggregate has been developed. The polystyrene bead is highly water resistant, but the vermiculite aggregate concrete is not. Therefore, a good water-resistant coating is essential for the satisfactory performance of the system using vermiculite concrete.



A. CARRIER PIPE
B. METALIC COUPLING
C. SEAL GASKET
D. SECONDARY SEAL - O - RING
E. COUPLING SEALING RING
F. PIPE END SEALING RING
G. INSULATION - THERMO BESTOS
H. LOCK BLOCK
I. CEMENT ASBESTOS CASING
J. THRUST SAFETY SHOULDER
K. INSULATION - POLYURETHANE

Figure C-5. Sealed conduit system with non-welded joint.

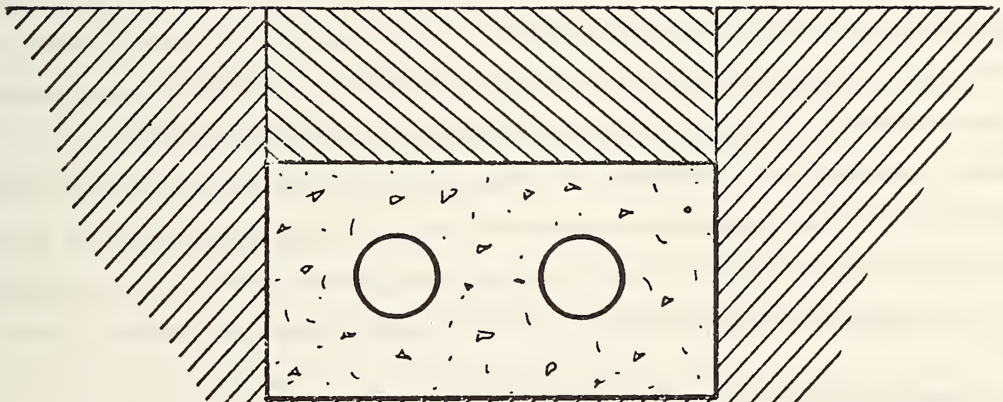


Figure C-6. Poured-in-place insulation envelope system.

2.6 Poured-In-Place Insulation Envelope System

Figure C-6 shows the poured-in-place insulation envelope system, which is found at several Air Force Bases. This system consists of nothing more than an envelope of loose-fill insulation material poured around the carrier pipe in the trench. Several loose-fill materials have been used with this system over the years. Currently two materials are in widespread use: powdered hydrocarbon material and a chemically treated (oleic acid) powdered chalk material. Both are said to be resistant to water infiltration. Systems are constructed by suspending carrier pipe in a trench, building a form around the pipe to establish a surrounding envelope, and then pouring insulation uniformly around the pipe. In some cases a plastic film is laid over the complete envelope. The water tightness of the system depends on the water resistance of the insulation material. If water should get into the material, it is assumed that the heat of the pipe will drive it out and/or the water penetration resistance of the material will confine damage to a relatively small section of the system. Such systems are probably significantly less expensive than other systems. This system is also relatively easy to install and repair. Since the resistance of the insulation material to water penetration diminishes as the water pressure increases, such systems have to be evaluated carefully when the ground-water level is of such height that it imposes an excessive head on the system. Ingress of roots, insects, moles and earthworms should be prevented from these systems. This type of system, however, should not be used for chilled-water distribution systems because the water vapor permeance of the insulation is very high. Water vapor that passes through the insulation in such instances will condense on the interior chilled water pipe.

2.7 Glass-Fiber-Reinforced Plastic Pipe System

Glass-fiber-reinforced plastic pipe is sometimes found in the steam condensate return lines because of their extremely high resistance to corrosion. Being used as return lines, these pipes are not usually insulated. There appear to be no major problems connected with these plastic pipes, except when the field joints have been made without appropriate supervision of preparing the epoxy resin.

3. OPERATION AND MAINTENANCE OF EXISTING SYSTEMS

Any underground system performs satisfactorily as long as the pipe insulation is kept dry. The pipe insulation becomes wet when ground water seeps into the insulation or when water or steam in the carrier pipe breaks through the seal. The drainable, driable and air-testable system (commonly referred to as DDT System) permits the Base Civil Engineer to inspect the system, perform preventative maintenance, and locate areas where corrective repairs are necessary, even though the system is buried and out of sight. The Base Civil Engineer should utilize these features for establishing a sensible inspection program which includes regular examination of the conduit pipe openings for signs of water. Drain caps located at the bottom of the conduit or end plate should be removed to ascertain if water is present in the conduit system. Manholes should be kept dry or ventilated, and all pipes and fittings should be insulated. Many conduit systems have been saturated by a combination of flooded manholes and missing drain and/or vent plugs.

A provision for avoiding the problem of missing drain and/or vent plugs is to extend the conduit vent piping to a point above the manhole top, ending with a 180° gooseneck. Also, drain valves could be installed on the drain-pipe nipples. Surface water may be prevented from entering the system through manholes by raising the level of manholes at least 6 inches above ground level. If this is not feasible, the manhole cover should be sealed tight to prevent entrance of surface water, but still permitting access for maintenance. Adequate ventilation of manholes should be provided in accordance with Air Force specifications, since it is dangerous for workmen to enter manholes that are too hot. Unventilated manholes may be hotter than manholes equipped with two vent pipes by as much as 50° F.

In the event that water or moisture is detected in the conduit system, the water should be drained and the conduit system completely dried by blowing air through the conduit.

Other non-conduit systems which have been developed in recent years also have this driable and air-testable feature although they may not be drainable. The dryness of a system may be determined by holding a cool mirror near the vent pipe opening, allowing forced air to strike the mirror. When the mirror no longer fogs, the system is dry. After all precautions have been taken and the conduit system has been completely dried out as described above and a subsequent inspection indicates the presence of water or moisture, a more serious problem exists.

In general, the successful operation of a heat-distribution system hinges upon a good preventive maintenance program as prescribed by the manufacturer of the specific system and periodic monitoring of its operational performance. Suggested procedures are the following:

(1) The heat-distribution system may be isolated from the site load during weekends or holidays by shutting off the supply valves to each of the load-using facilities. In this manner heat loss/gain characteristics of the distribution system may be ascertained provided the load for the plant itself is known or separately measured. By keeping this type of record, it is possible to detect significant changes in pipe performance due to pipe failures.

(2) Manholes and the building terminals should be checked weekly for water leakage, wisps of steam, corrosion or any other signs of developing troubles. The system should be "walked" occasionally to look for evidence of potential trouble, particularly noise, water puddling, shallow cave-ins, melted snow cover, or wisps of steam from drains and vents. Auxiliary external drainage facilities should be checked at least monthly to ascertain that the proper operating conditions exist.

(3) The failure of insulation in underground heat-distribution systems creates elevated ground-surface temperatures above the pipe which are accompanied by dead brown grass or melted snow. Recent NBS studies show that the ground-surface temperature over a failed pipe system could be as much as 16° F higher than the undisturbed surrounding ground. A temperature rise of 4-6° F may be considered normal for a system with a 6- to 8-foot ground cover.

Since surface temperatures are very difficult to measure and fluctuate rapidly due to wind and sun conditions, it is recommended that a 6-inch ground temperature probe be used. Figure C-7 illustrates 6-inch depth ground temperature profiles measured over pipe systems with good and failing insulation. An inexpensive ground temperature probe consisting of a mercury thermometer within a protective glass sheath (figure C-8) could be used to measure the ground-temperature profile. The procedure for obtaining these profiles is to measure the earth temperature with the ground temperature probe over a distance of approximately 15 feet from the pipe in both directions normal to the pipe. This method assumes that the ground temperature at 15 ft away from the pipe represents the undisturbed earth temperature. If the thermal conductivity (k_s) of the ground, depth of the pipe (δ), and the pipe heat loss (Q) are known, the ground temperature excess (θ) over the undisturbed earth temperture due to the presence of the pipe can be calculated by the relation:

$$Q = 2\pi \cdot k_s \cdot \theta / \ell n \frac{y + \delta}{y - \delta} \qquad (1)$$

Here y is the distance from the ground level to the center of the pipe. The quantity θ in the foregoing equation is the difference between the soil temperature measured at the depth (δ) directly above the pipes and the undisturbed earth temperature of the ground.

Equation (1) was used to calculate the 6-inch-depth ground-temperature excess as a function of pipe heat-transfer factor (k_p) for various pipe depths (y). Here the pipe heat-transfer factor is the pipe heat loss/gain per foot of pipe per hour per temperature difference between the pipe and the undisturbed earth. The results of these calculation are given in figure C-9. When the measured

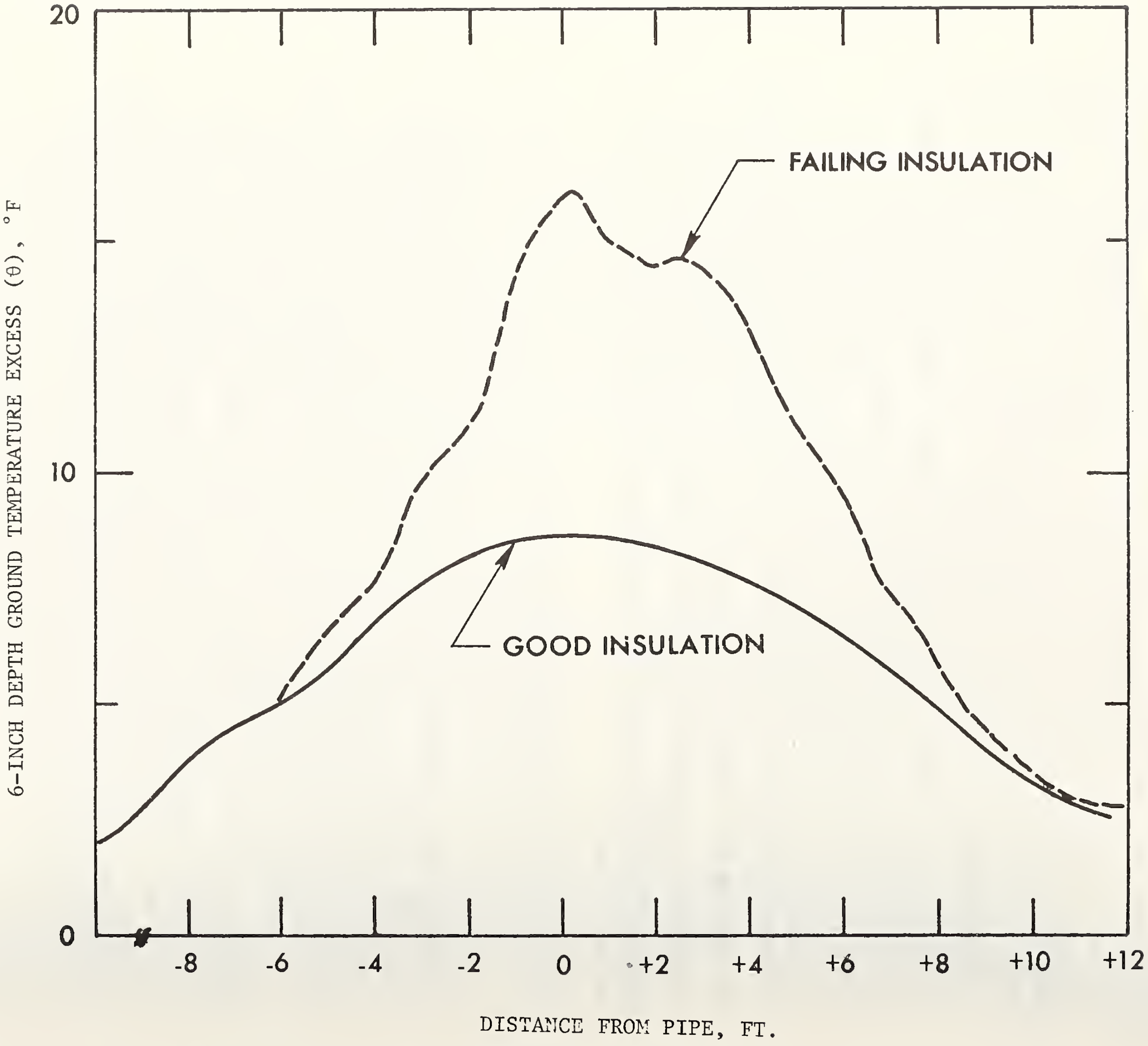


Figure C-7. Measured 6-inch depth temperatures above underground heat distribution systems with good and failing pipe insulation.

Figure C-8. Ground-temperature probe.

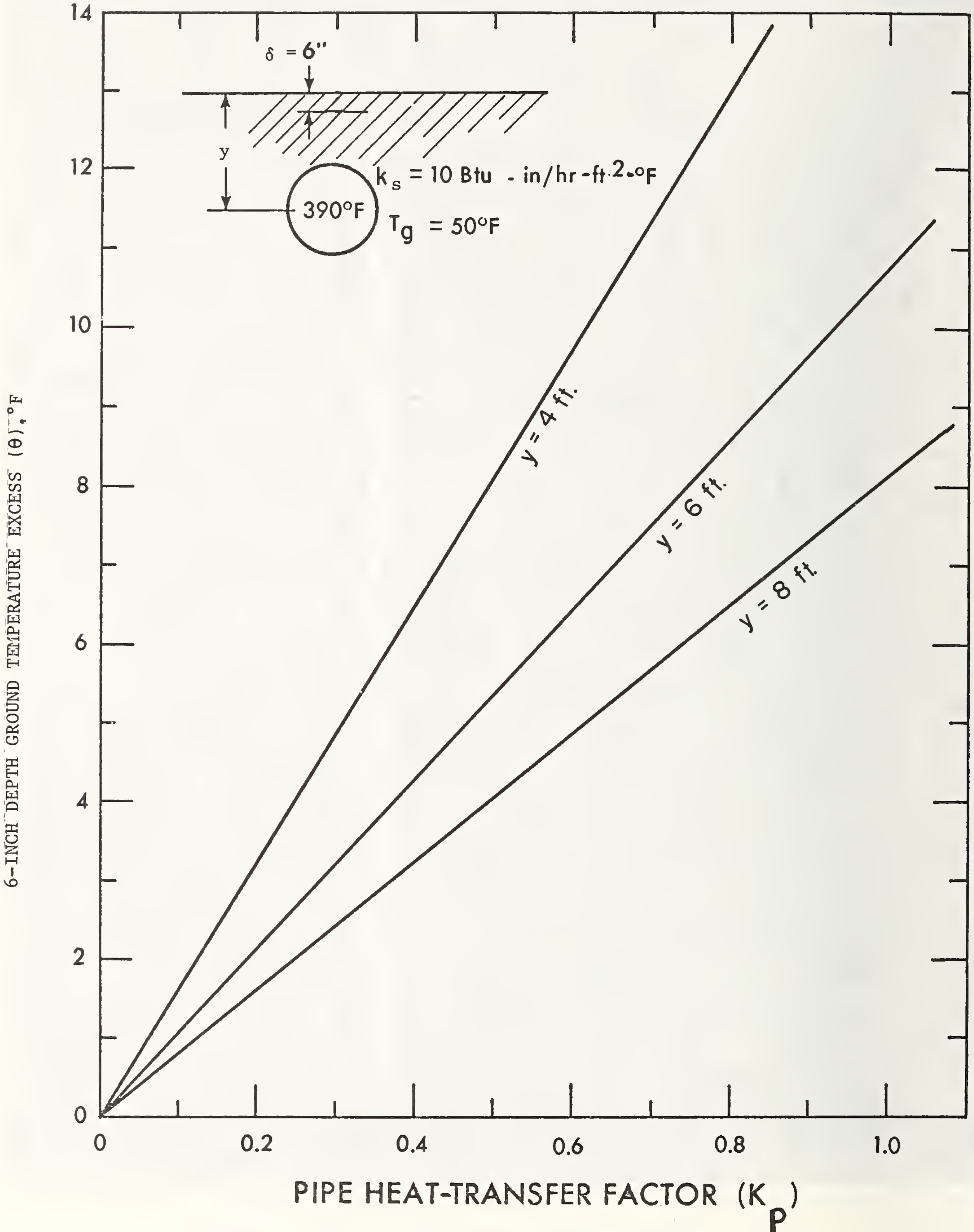


Figure C-9. Calculated values of the 6-inch depth ground-temperature excess.

ground-temperature excess is significantly greater than the predicted values given in figure C-9, the pipe insulation has probably failed.

(4) The thermal conductivities for most pipe insulations range between 0.3 and 0.4 Btu·in/h·ft^2·°F. According to the Building Research Advisory Board (BRAB), optimum insulation thicknesses for typical pipe sizes range between 1 and 3 inches even including pipe sizes as large as 14 inches in diameter [11]. Based on these results, figure C-10 was constructed to determine a recommended "pipe insulation factor" which is defined on the figure. The concept of pipe insulation factor is illustrated in the example below:

<u>Example</u> - A 4-inch pipe is insulated with 2-inch-thick insulation, yet its conduit surface temperature was measured and found to be 110° F. when the ambient temperature was 70° F. The question is, "Is the insulation failing?" From table C-1 the pipe temperature, which is usually assumed to be the same as the steam temperature, at 150 psig is 366° F. The pipe insulation factor under these conditions is

$$\text{Pipe ins. factor} = \frac{110 - 70}{366 - 70} \times 100 = 13.5\%$$

On the other hand, from figure C-10, the predicted insulation factor for a 4-inch pipe with 2-inch insulation is 6.25. The fact that the measured pipe insulation factor is significantly greater than the predicted value of figure C-10 indicates that the insulation of the subject pipe is not functioning as it should be.

Had the insulation been adequate, the conduit surface temperature should not have exceeded

$$\text{Conduit surface temp.} = 70 + \frac{6.25 \times (366 - 70)}{100} = 88.5°\text{F}$$

For large installations, the performance of aboveground pipe insulation may be evaluated by scanning the conduit surface temperature using an infrared television system.

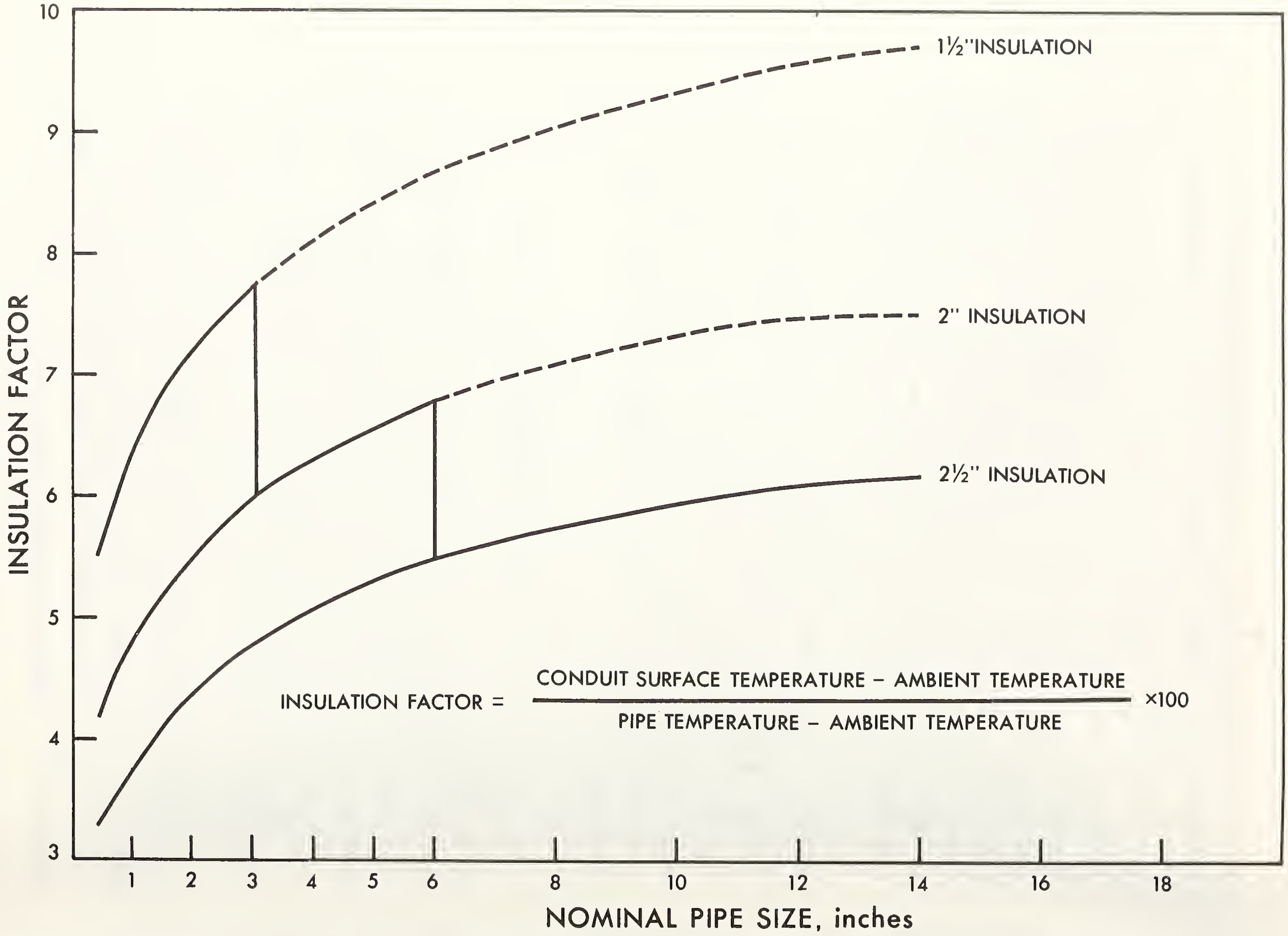


Figure C-10. Pipe insulation factors for standard pipes.

TABLE C-1. PROPERTIES OF SATURATED STEAM

Gage Pressure psig	Absolute Pressure psia	Temp. °F	Gage Pressure psig	Absolute Pressure psia	Temp. °F
0	14.696	212.00			
10	24.696	239.39	260	274.696	409.35
20	34.696	258.77	270	284.696	412.57
30	44.696	274.02	280	294.696	415.72
40	54.696	286.72	290	304.696	418.78
50	64.696	297.67	300	314.696	421.75
60	74.696	307.33	310	324.696	424.66
70	84.696	316.01	320	334.696	427.50
80	94.696	323.90	330	344.696	430.28
90	104.696	331.16	340	354.696	432.99
100	114.696	337.88	350	364.696	435.65
110	124.696	344.16	360	374.696	438.25
120	134.696	350.05	370	384.696	440.79
130	144.696	355.61	380	394.696	443.29
140	154.696	360.86	390	404.696	445.74
150	164.696	365.87	400	414.696	448.14
160	174.696	370.63	410	424.696	450.50
170	184.696	375.19	420	434.696	452.82
180	194.696	379.56	430	444.696	455.09
190	204.696	383.75	440	454.696	457.32
200	214.696	387.77	450	464.696	459.52
210	224.696	391.68	460	474.696	461.68
220	234.696	395.45	470	484.696	463.81
230	244.696	399.08	480	494.696	465.91
240	254.696	402.61	490	504.696	467.97
250	264.696	406.03	500	514.696	470.00

(5) Simple repair techniques, short of complete system rehabilii-tation, are often available from the manufacturer of the heat-distribution system. Excavation for repair, however, should be done with great care to avoid further damage to pipe sections. Usually, it is recommended that the last 6 inches of earth be re-moved by hand. Protective measures should be applied immediately to preserve the integrity of the undamaged portion of the system in the vicinity of the excavation.

(6) For aboveground systems, frequent system inspection to detect conduit surface failure or wet insulation should be conducted. Particular attention must be given to moisture seepage into the joints and fittings, where pipe insulation usually is broken. Al-though rain water which seeps into the insulation can be dried out for hot-pipe systems, conduit failure is much more critical for chilled-water systems. Many chilled-water pipe insulations, including the moisture-impervious polyurethane systems, may be found waterlogged due to localized failures permitting the ingress of water vapor. Condensed moisture can literally seep along the pipe through the narrow space between the pipe and its insulation.

4. FINDING LEAKS IN DISTRIBUTION SYSTEMS

There is no easy way to locate leaks in underground distribution systems.

The "hit-or-miss" method consists of observing melted snow above the system, or of burned grass, or maybe just the feel of the ground that is dug up where it is suspected the leak is located. This is a tedious and often a frustrating method.

In a pressure-testable system, leaks in the casing may be found by applying 15 pounds of air pressure. In some of the modern systems, an automatic monitoring device consisting of an elaborate indicator console, nitrogen pressurization mechanism and alarm mechanism is available. If the pressure drops, leaks are indicated. The task then is to locate the leaks in the casing. This could be done by pressure testing each section between manholes. If the problem can be narrowed to a specific section, the exact location of the leak can be ascertained by one of several methods.

A soap bubble leak detector may be employed to detect pressurized air leaks from the conduit or steam leaks from steam or condensate lines.

Another method of locating leaks in conduit casing is the use of a halogenated-hydrocarbon refrigerant gas as a tracer. The conduit system is plugged at the terminal ends, a pressure of 5 pounds is applied, and a small amount of refrigerant gas is introduced. Special probes are positioned along the conduit at about 10-foot intervals. A leak detector will show the presence of refrigerant gas in the vicinity of leaks. Instead of refrigerant, helium or sulfur hexafluoride (SF_6) have been used in conjunction with their special purpose detectors. The SF_6 leak detection is especially attractive for a large system because of its extremely sensitive chromatographic detection by the use of the ion-capture principle. This means that a very small amount of tracer gas is needed for a large-scale leak detection operation.

Leaks also can be located by use of an odorant called mercaptan, introduced into the conduit through an open vent with 15 psig air applied. All drains and vents, of course, must be closed except ones at the end seals in manholes. After the odorant is introduced, the line should be walked until the leak is located. The length of time required and success of the detection depend on the depth of the conduit, porosity of the soil, potency of the odorant, and length of the conduit.

If tests indicate a tight conduit system, there must be leaks in the steam or condensate lines. These leaks may be isolated by excavating midway between manholes. An opening cut through the conduit at this location will reveal the direction and nature of flow. Keeping in mind that pipe sections are often 21 feet long, additional excavations and openings may be made until the leak(s) are located. Intermittent flow indicates a leak in a condensate line, whereas continuous vapor flow usually indicates a leak in a steam line.

Another method of locating a conduit leak is to pressurize the conduit to approximately 15 psig and lay down a 2- to 4-inch thick layer of fire-extinguishing foam on the ground above the path of the distribution system. Air bubbles will appear in the foam above the point where the leak is located.

A radioactive isotope also may be used for the purpose of leak detection, but its use requires a license from the Nuclear Regulatory Commission.

5. ASSESSING THE THERMAL PERFORMANCE OF EXISTING SYSTEMS

It is recognized that there is no simple method to measure heat loss from underground steam lines. At installation, where there is excess boiler capacity at the steam generating plant, distribution-system losses are often ignored unless the users complain of insufficient steam or hot water. Although not always easily measured, heat loss from the underground mains can be estimated using several procedures:

(1) Calorimetric method. Fluid-flow meters and temperature measurements across a given segment of a steam line may be used to determine the change in fluid enthalpy which in turn equals the pipe heat loss. If the pipe fluid is saturated steam, the measurement of condensate accumulation along the test section by means of condensate collectors and traps is also necessary. This method, however, may entail the installation of new transducers, which requires temporary shutdown of the system.

(2) Heat-Flux Meter Method. In the case of tunnel systems, it is possible to wrap several heat-flux meters (figure C-11) around the pipe insulation surface at selected spots. The heat-flux-meter output signal should be integrated over a period of at least one hour. For the non-tunnel system, use of the heat-flux readings of the back-filled system should be monitored for at least two weeks to permit a stablized heat loss/gain value to be reached.

In many cases, especially in the air conduit system, the measured heat loss by the heat-flux meter strapped around the

Figure C-11. Heat-flux meter for heat-loss measurement.

exterior surface of the piping system can be much different from the total heat loss from the carrier pipe. This situation occurs when water is boiling in the air space and the generated steam is exhausted to the manhole.*

Measurement of heat loss/gain of aboveground heat-distribution systems can be more readily performed than for underground systems because the surface heat-flux transducers can be readily installed without excavation. The heat-flux meter method has been used successfully in the past to monitor the heat-flow rate at the exterior surface of pipes. For the aboveground system, however, it is recommended that these transducers be installed at not less than three circumferential places (i.e., the top, bottom, and side of the conduit). This is because the heat-flux value tends to vary considerably from the top to the bottom. It is also recommended that the transducer output be integrated for approximately one hour to reduce uncertainty due to fluctuations in conduit surface temperature and pipe temperature.

(3) Pipe Heat-Loss Gain/Calculation Methods. Although the measurement of existing underground pipe heat loss/gain is rather difficult, it can be estimated by the pipe heat-transfer factor (K_p). Typical values of K_p are given in figure C-12 for various sizes of pipes buried at a depth of 6 ft in relatively moist soil. The lineal heat loss in Btu/h per foot of pipe is determined by multiplying this heat-transfer factor (K_p) by the temperature difference between the pipe and the undisturbed ground temperature. For this calculation, the pipe temperature is usually assumed to be equivalent to the fluid temperature within the pipe. Precise ground temperatures are usually not available. Approximate values for selected cities in the United States are given in Appendix C-2.

*According to a recent test conducted at NBS, the heat loss from water boiling in the conduit could be as much as 30,000 Btu/h, an equivalent of 30 lb of steam per hour per foot of pipe, which is almost 300 times larger than the heat-loss rate for a well insulated system.

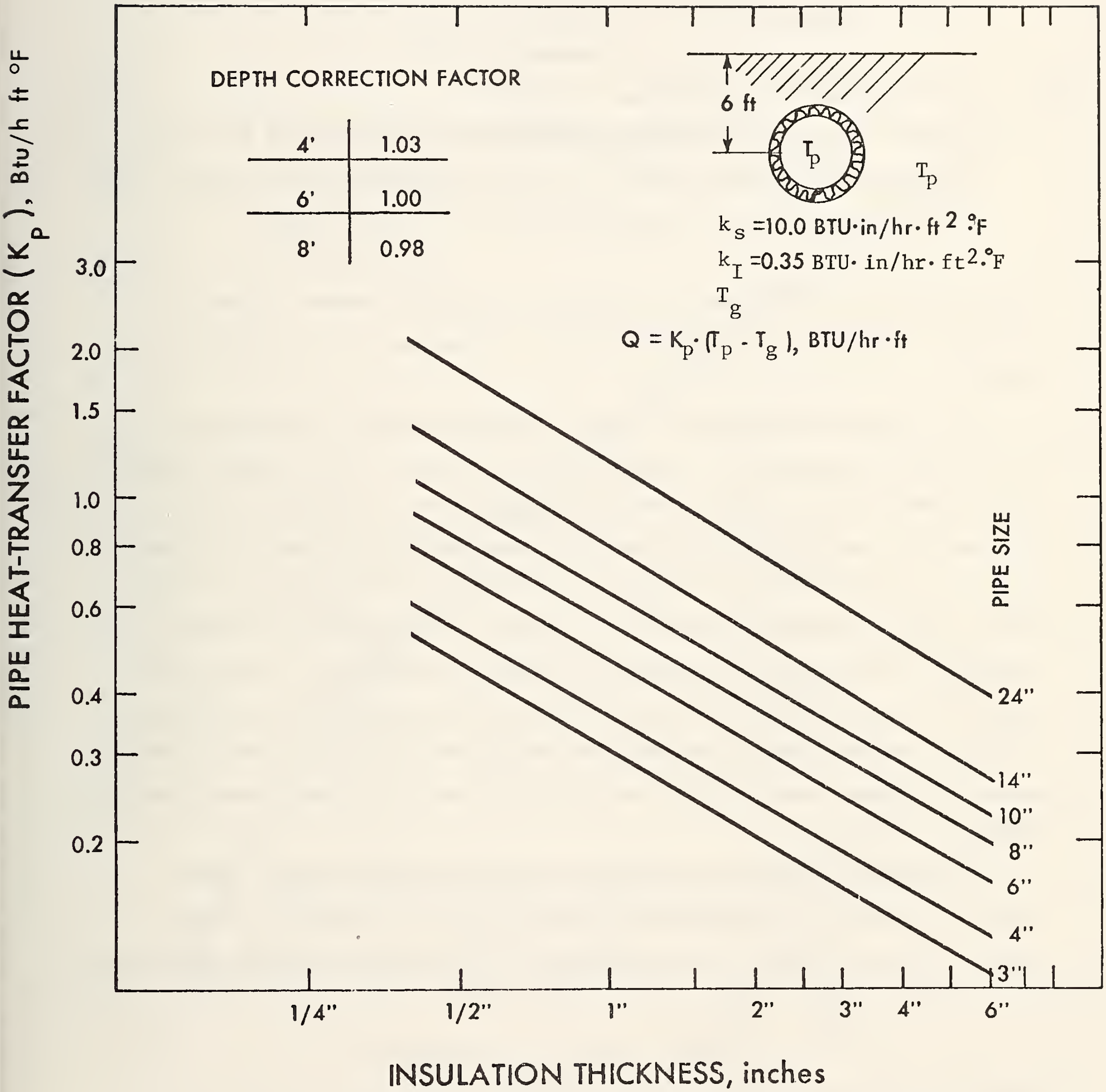


Figure C-12. Heat-transfer factors (K_p) for insulated underground pipes.

As an example, the heat-loss rate from a 14-inch pipe in a sealed insulation conduit with a 6-foot earth cover in Washington, D.C. is calculated below. Assumptions for the calculation are given in the following:

pipe temperature: 350° F

ground temperature: 47° F (winter ground temperature given in appendix C-2).

pipe insulation thickness: 1.5 inch

thermal conductivity of pipe insulation: $k_I = 0.35$ Btu·in/h.ft^2·°F

pipe heat-transfer factor: $K_p = 0.78$ Btu/h·ft·°F (from fig. C-12)

the pipe heat loss is found to be: $Q = 0.78 \cdot (350-47)$

$= 236$ Btu/h·ft

For underground distribution systems having air spaces in the conduit, calculation procedures given in appendix C-1 should be used.

The pipe heat-transfer factor is very strongly dependent on the thermal resistance of the pipe insulation. It is a weak function of the depth and the soil thermal conductivity, provided the pipe is well insulated. One inch of calcium silicate installed around a 6-inch pipe can reduce its heat loss by a factor of 3 as compared to the pipe with no thermal insulation. (Figure C-13 depicts the heat-transfer factors for pipes with no thermal insulation.) When the measured or estimated heat loss by the method described in previous sections substantially exceeds the calculated heat-loss value (based upon the design data), it is very likely that the pipe system has dramatic failures.

(4) Pipe Heat-Loss Calculations for Aboveground Systems. Heat-loss calculation for aboveground pipes can be obtained by using table C-2 and figure C-14 in conjunction with the following equation:

$$Q = (T_p - T_a)/R_t \qquad (2)$$

where $R_t = R_a + R_c + R_I$

Q = pipe heat loss, Btu/h·ft

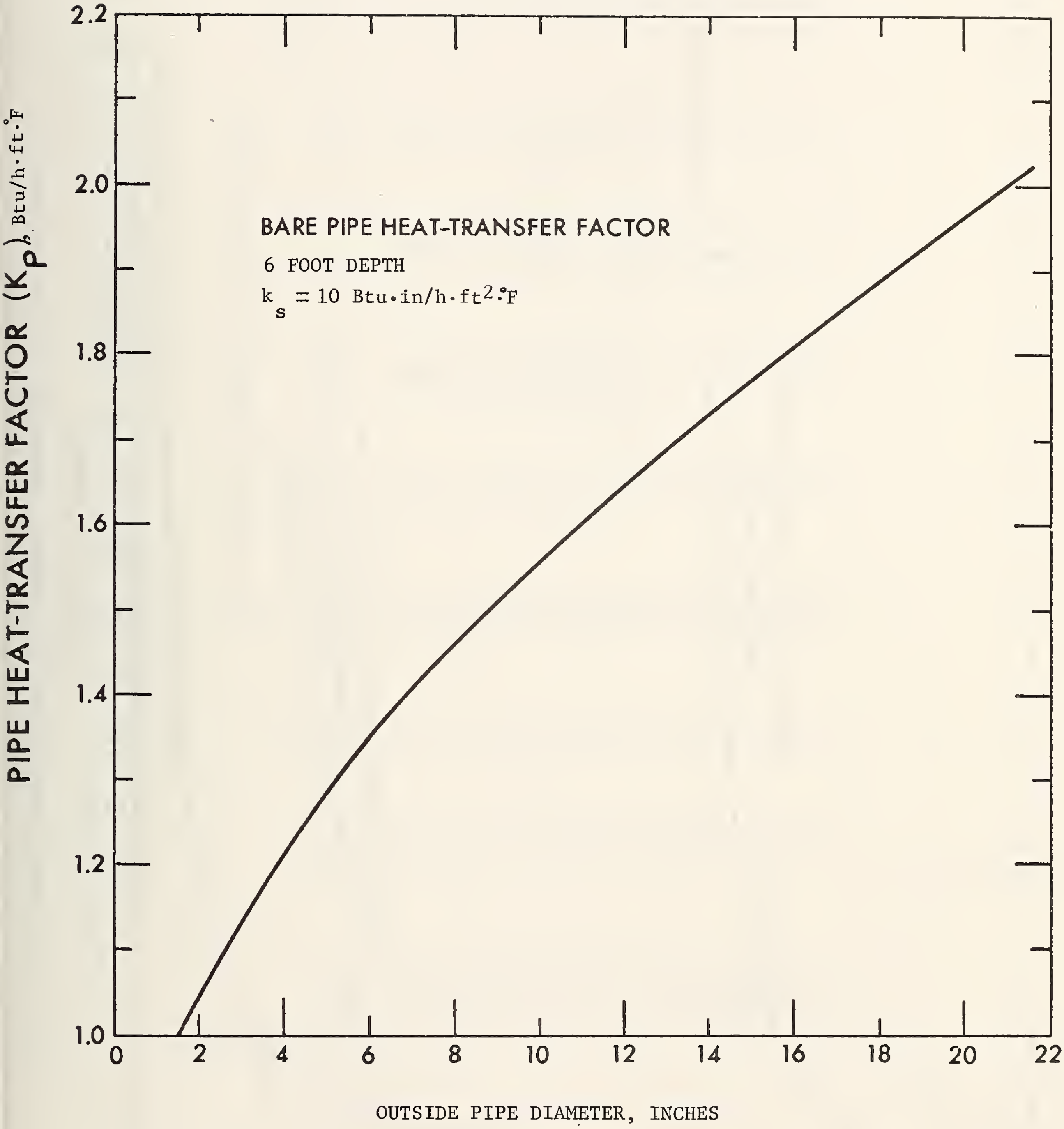


Figure C-13. Heat-transfer factors for uninsulated underground pipes.

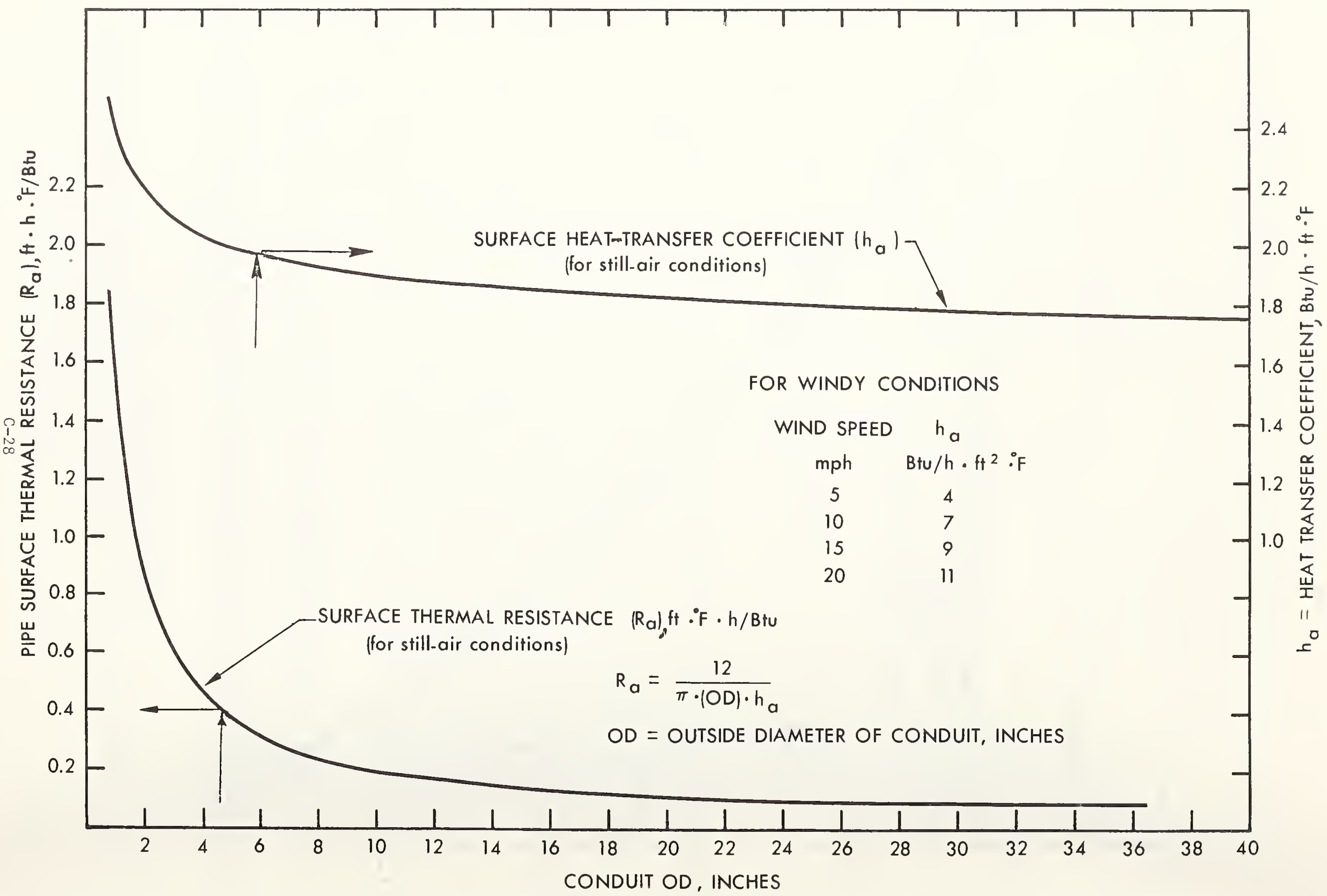


Figure C-14. Surface thermal resistance of aboveground heat-distribution conduits.

TABLE C-2. THERMAL RESISTANCES FOR VARIOUS THICKNESSES OF PIPE INSULATION

$$R_I = F/k_I$$

where F = pipe thermal resistance factor, in/ft

k_I = thermal conductivity for pipe insulation, Btu·in/h·ft^2·°F

Nominal Pipe Size	OD Inches	Pipe Thermal Resistance Factor (F), in/ft — INSULATION THICKNESS, INCHES: 1/2	1	1 1/2	2	2 1/2	3	3 1/2	Air Space Resistance
1/2	.840	1.498	2.327	2.903	3.345	3.703	4.005	4.266	1.52
3/4	1.050	1.278	2.037	2.578	3.000	3.345	3.637	3.890	1.21
1	1.315	1.080	1.766	2.269	2.667	2.997	3.277	3.522	0.97
1 1/4	1.660	.901	1.510	1.971	2.343	2.653	2.921	3.155	0.77
1 1/2	1.900	.808	1.373	1.809	2.164	2.463	2.722	2.949	0.67
2	2.375	.671	1.167	1.560	1.886	2.164	2.407	2.622	0.54
2 1/2	2.875	.570	1.009	1.365	1.665	1.924	2.153	2.357	0.44
3	3.500	.480	.863	1.182	1.456	1.695	1.907	2.098	0.36
3 1/2	4.000	.426	.774	1.069	1.324	1.549	1.750	1.932	0.32
4	4.500	.383	.702	.976	1.215	1.427	1.618	1.792	0.28
5	5.563	.316	.587	.824	1.035	1.225	1.397	1.556	0.23
6	6.625	.268	.504	.713	.902	1.074	1.232	1.377	0.19
8	8.625	.210	.398	.570	.728	.873	1.009	1.135	0.15
10	10.750	.170	.326	.470	.604	.729	.847	.958	0.12
12	12.750	.144	.278	.404	.521	.632	.737	.836	0.10
14	14.000	.132	.255	.371	.480	.583	.681	.774	0.09
16	16.000	.166	.225	.328	.426	.519	.608	.693	0.08
18	18.000	.103	.201	.294	.383	.468	.549	.627	0.07

Typical values of "k" @ 200° F	
Glass fiber	0.3
Asbestos	0.4
Magnesium	0.45
Calcium silicate	0.37
Cellular glass	0.48

T_p = pipe temperature which is usually approximated by the pipe fluid temperature, °F

T_a = ambient temperature, °F

R_t = total thermal resistance, h·ft·°F/Btu

R_a = pipe surface thermal resistance as found in figure C-14

R_c = conduit air space (if any) thermal resistance as found in table C-2 h·ft·°F/Btu

R_I = thermal resistance of pipe insulation as found in table C-2,h·ft·°F/Btu

As an example, the heat loss from a nominal 4-inch pipe with 2-inch-thick fibrous glass insulation (thermal conductivity 0.3 Btu·in/h·ft^2·°F in an 8-inch-diameter steel conduit is calculated The pipe is exposed to still-air conditions. The detail of the calculation are given in the following:

R_a	for the 8-inch diameter conduit from fig. C-14	0.24
R_c	conduit air-space thermal resistance from table C-2	0.28
R_I	thermal insulation resistance from table C-2	4.05
R_t	total thermal resistance	4.57

Thus, the heat loss for the system carrying 150 psig saturated steam to the ambient of 50° F, is

$$Q = (366 - 50)/4.57 = 69.1 \text{ Btu/h·ft}$$

For the foregoing calculation, the pipe temperature (T_p) was obtained from table C-1.

If the same system were exposed to a 15-mph wind condition, the surface resistance would be determined by the equation given in figure C-14 as follows:

$$R_a = \frac{12}{\pi \times 8x9} = 0.053$$

In this case R_t becomes 4.38, resulting in a pipe heat-loss rate of 72.1 Btu/h·ft.

This indicates the significance of the wind for pipe heat loss. Moreover, the increase of the heat loss due to wind is more pronounced for poorly insulated pipes than for well insulated pipes.

(5) Heat Loss from Pipe Fittings. Very often, even when the pipe is well insulated, fittings such as flanges, elbows, and tees are left bare because it is difficult to insulate them, and because of the belief that losses from these parts are not large. However, the fact that a pair of 10-inch standard flanges having an area of 5.20 sq ft of surface could lose a heat equivalent of as much as 40,000 cu ft of natural gas per year for a 250-psig steam system shows the necessity for insulating such surfaces. Table C-3 gives approximate surface areas of 250-psig fittings, together with the bare surface per linear foot for various pipe sizes. Unfortunately, similar data for the valves are not available. It is suggested that the surface area equivalent of linear foot of pipe be used for estimating the heat-transfer surface area for valves, unless better data are available.

6. ECONOMIC CONSIDERATIONS FOR ESTIMATING OPTIMUM THERMAL INSULATION OF UNDERGROUND DISTRIBUTION SYSTEMS

When a portion of an underground system is to be replaced by a new system, the cost of pipe materials, the installation costs, and the cost of energy expenditures must be considered in order to estimate the economically optimum amount of thermal insulation.

A suggested procedure to determine the optimum insulation thickness is described below:

Step #1 -- Determine pipe size by taking into account the heat-carrying capacity and pressure-drop requirement of the heat distribution system.

Step #2 -- Determine

a) total pipe length
b) number of expansion loops needed
c) number of anchors to be used
d) number of couplings, fittings and other accessories to be used in the system.

TABLE C-3. SURFACE AREA OF PIPE FITTINGS*

Nominal pipe size inch	pipe surface/ linear ft	flanged joint	250 psig 90°	Long Radius	Tee	Cross
1	0.344	0.438	1.015	1.083	1.575	2.07
1 1/4	0.435	0.510	1.098	1.340	1.926	2.53
1 1/2	0.498	0.727	1.332	1.874	2.66	3.54
2	0.622	0.848	2.01	2.16	3.09	4.06
2 1/2	0.753	1.107	2.57	2.76	4.05	5.17
3	0.917	1.484	3.49	3.74	5.33	6.95
4	1.178	1.914	4.64	4.99	7.07	9.24
5	1.456	7.18	5.47	6.02	8.52	10.97
6	1.738	2.78	6.99	7.76	10.64	13.75
8	2.257	3.77	9.76	11.09	14.74	18.97
10	2.817	5.20	13.58	15.60	20.41	26.26
12	3.338	6.71	17.73	18.76	26.65	34.11
14	3.663	8.30	22.31	25.70	33.63	43.15
16	4.188	10.05	27.18	31.73	40.94	52.35

Note: For 100-psig system fittings, the surface areas are approximately 27% less than those shown in this table.

*This table excerpted from Piping Handbook, Sabin Crocker, pg. 698, McGraw Hill, 1945.

Step #3 -- Obtain from manufacturers the cost of piping system including freight, supervision, and installation labor for the five insulation thicknesses given in the table below.

Insulation Thermal Conductivity (k_I), Btu·in/h·ft^2·°F

Insulation thickness to be studied, inches	.15	.25	.35	.45	.65
	1/2				
	3/4	3/4			
	1	1	1		
	1 1/2	1 1/2	1 1/2	1 1/2	
	2	2	2	2	
		2 1/2	2 1/2	2 1/2	
			3	3	3
				3 1/2	4
					5
					6
					7

These five insulation thicknesses should be analyzed to determine which one is most economical.

Step #4 -- Determine amortization period for the distribution system and discount rate for the capital investment.

Step #5 -- Determine from these economic factors the annual owning cost ($\$_o$) of the system for a unit pipe length from the following equation:

$$\$_o = \frac{\text{Total pipe cost as installed}}{\text{present worth factor x pipe length}}$$

Present worth factors are given in the following table:

Amortization period Years	Discount rate, % 6	7	8	10
10	7.36	7.02	6.71	6.14
15	9.71	9.11	8.56	7.60
20	11.47	10.59	9.82	8.51

Step #6 -- a) Obtain from manufacturer or calculate using formulas given in reference [8] the pipe heat-transfer factors (K_p) and pipe heat loss per unit length (Q).

b) Repeat (a) for the five insulation thicknesses under consideration.

c) Obtain five sets of cost figures for the five insulation thicknesses from the following formula:

$$\$_H = \frac{Q \cdot HR \cdot C_H}{10^6}$$

Where $\$_H$ = cost of heat loss per year per ft of pipe

Q = pipe heat loss per foot of pipe per hour

HR = total hours of operation per year

C_H = cost of heat/10^6 Btu

Step #7 -- Summarize the foregoing calculation in a table as shown below. The numbers in parenthesis denote the five insulation thicknesses.

Insulation type or thickness	Heat-transfer factors	Heat loss	Annual pipe cost as installed	Annual heat cost	Total cost
t_I (1)	K_p (1)	Q (1)	$\$_o$ (1)	$\$_H$ (1)	$\$_T$ (1)
t_I (2)	K_p (2)	Q (2)	$\$_o$ (2)	$\$_H$ (2)	$\$_T$ (2)
t_I (3)	K_p (3)	Q (3)	$\$_o$ (3)	$\$_H$ (3)	$\$_T$ (3)
t_I (4)	K_p (4)	Q (4)	$\$_o$ (4)	$\$_H$ (4)	$\$_T$ (4)
t_I (5)	K_p (5)	Q (5)	$\$_o$ (5)	$\$_H$ (5)	$\$_T$ (5)

Here $\$_T$ = total cost = $\$_o + \$_H$

Step #8 -- Choose the insulation thickness or the type of insulation which yields the smallest total costs ($\$_T$).

7. SUMMARY

In summary, the following measures are suggested for the improvement of thermal performance of existing heat-distribution systems.

1. Conduct frequent inspections along the system to determine possible insulation failures by observing telltale signs of steam or pipe-fluid leakage. Measurements of the ground surface or the conduit surface (in case of the aboveground system) could indicate excessive pipe heat loss or failure of the insulation.

2. Pipe heat loss can be minimized by shielding the pipe surface from high wind and water.

3. Advanced techniques such as the use of nitrogen pressurization for the air-testable system, the SF_6 tracer-gas method, and use of thermographics may be applicable to expedite failure detection for large installations.

4. Insulate pipe fittings, since heat loss from them can be significant. Data are provided to estimate the heat loss from uninsulated fittings.

5. Construct fences or other barriers to protect aboveground pipes from vandalism or surface traffic.

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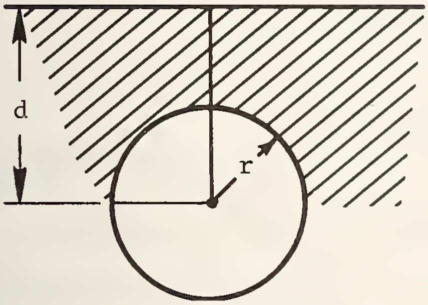
APPENDIX C-1. CALCULATION OF UNDERGROUND PIPE HEAT-TRANSFER FACTORS

1. Definitions



d = distance from ground surface to center of pipe, inches

r = radius of conduit, inches

C = thermal conductance of pipe and conduit, $Btu/h \cdot ft^2 \cdot {}^\circ F$

T_g = undisturbed earth temperature

T_p = temperature of fluid in pipe, °F

k_s = thermal conductivity of undisturbed earth, $Btu \cdot in/h \cdot ft^2 \cdot {}^\circ F$ (table C-1.1)

2. Circular Systems

2.1 Heat-Transfer Factors

$$\frac{1}{K_p} = \frac{1}{C} + \frac{6}{\pi k_s} \ell n \left\{ \frac{d}{r} + \sqrt{\left(\frac{d}{r}\right)^2 - 1} \right\}$$

when $\frac{d}{r} \gg 1$

$$\frac{1}{K_p} = \frac{1}{C} + \frac{6}{\pi k_s} \ell n \left(\frac{2d}{r}\right)$$

Pipe heat-loss rate:

$$Q = K_p \cdot (T_p - T_g) \quad Btu/h \cdot ft$$

Note: Heat-transfer-factor calculation procedures for more complex piping systems are found in the references cited in the list of references for the main text.

TABLE C-1.1 THERMAL CONDUCTIVITY OF SOIL (k_s)*/, Btu·in/h·ft^2·°F

Moisture Content	Type Soil		
	Sandy	Silty	Clay
Dry	2	1	1
Medium	13	9	7
Wet	15	15	15

*/ These data are approximate values and should be used only when precise data are unavailable from the site or from the manufacturer.

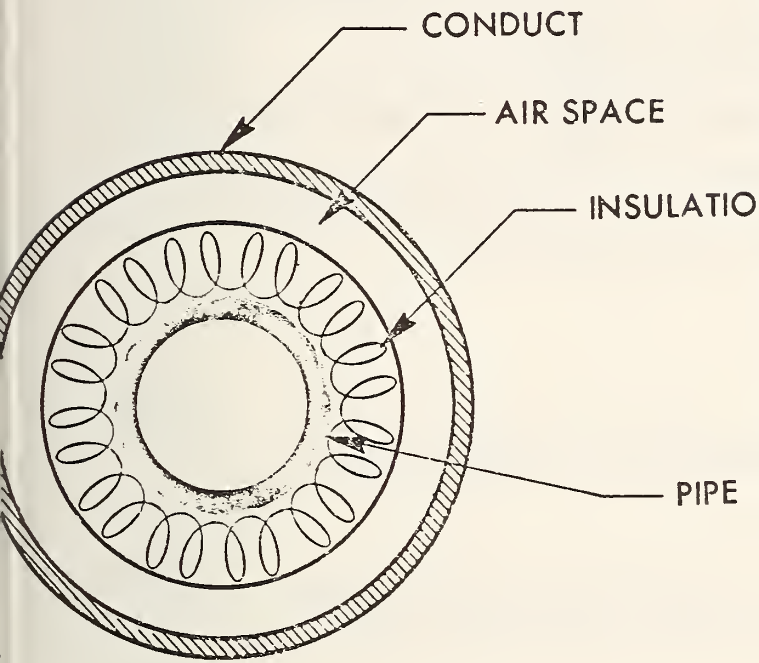
TABLE C-1.2 THERMAL CONDUCTIVITY OF PIPE INSULATION (k_T)*/

Insulation Type	Btu·in/h·ft^2·°F
Asbestos	1.3
Calcareous loose fill	0.8
Calcium silicate	0.3
Cellular glass	0.4
Cork	0.3
Hydrocarbon loose fill	0.7
Polyurethane	0.2

*/ These data are to be used only when no precise data are available from the site or from the manufacturer.

2.2 Pipe Conductances

Define the following parameters for all the pipes involved in the system. If the pipe is in a conduit, exclude the conduit.



Insulation

Thickness, in. t_I = ______

Thermal conductivity, Btu·in/h·ft^2·°F (table C-1.2) k_I = ______

Pipe

Thickness, in. t_w = ______

Pipe thermal conductivity*, Btu·in/h·ft^2· k_w = ______

Outside diameter, in. D = ______

$$\frac{1}{C} = \frac{6}{\pi} \left\{ \frac{1}{k_w} \cdot \ln \left(\frac{D}{D-2t_w} \right) + \frac{1}{k_I} \cdot \ln \left(\frac{D + 2 \cdot t_I}{D} \right) \right\}$$

*For steel k_w = 360 Btu·in/h·ft^2·°F
plastic k_w = 2-3 Btu·in/h·ft^2·°F
concrete k_w = 12 Btu·in/h·ft^2·°F
cement asbestos k_w = 16 Btu·in/h·ft^2·°F.

If the single pipe is in a conduit with air space, adjust the C value of the pipe as follows:

$$\frac{1}{C} = \left(\frac{1}{C}\right)_{\text{of pipe}} + \frac{6}{\pi}\left\{\frac{2}{3D_p} + \frac{1}{k_c} \cdot \ln\left(\frac{D}{D_c - 2t_c}\right)\right\}$$

where D_p = outside diameter of the insulation, in

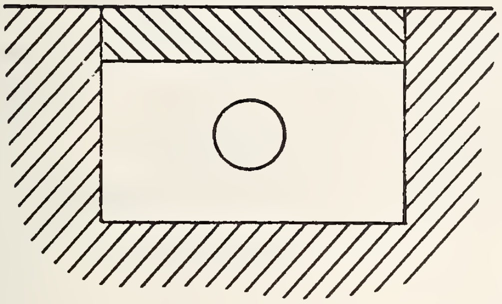
k_c = thermal conductivity of conduit, Btu·in/h·ft^2·°F

t_c = thickness of conduit, in

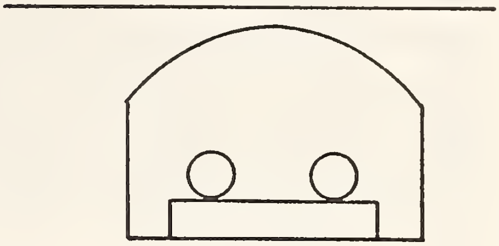
D_c = outside diameter of the conduit, in

3. Non-circular systems

For non-circular underground distribution systems, such as shown below, the manufacturer should provide one of the following heat-transfer factors for specified soil conductivity, temperature, and depth.



Case 1



Case 2

Case 1

$$K_p = \frac{\text{Heat loss per foot of pipe}}{\text{pipe temperature - soil temperature}}$$

Case 2

$$K_p = \frac{\text{heat loss per foot of pipe}}{\text{Average temperature of pipe - soil temperature}}$$

When the manufacturer provides calculated values for K_p, he should be required to submit the calculation procedure used to derive the values.

APPENDIX C-2. EARTH TEMPERATURE TABLES FOR UNDERGROUND HEAT-DISTRIBUTION-SYSTEM DESIGN

Tables TG-1 through TG-11 were developed by using monthly average temperatures prepared by the U.S. Weather Bureau for many localities in the United States which include cities next to U.S. Air Force Bases. These tables were generated by using the procedure outlined in a 1965 ASHRAE technical paper entitled "Earth Temperature and Thermal Diffusivity at Selected Stations in the United States" by T. Kusuda and P. R. Achenbach.

10-FT AVERAGE EARTH TEMPERATURE FOR UNDERGROUND PIPE HEAT TRANSFER

TG-1

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Alabama	Anniston AP*/	55.	58.	70.	67.	63.
	Birmingham AP	54.	58.	71.	68.	63.
	Mobile AP	61.	63.	74.	71.	67.
	Mobile CO	61.	64.	75.	72.	68.
	Montgomery AP	58.	61.	73.	70.	65.
	Montgomery CO	59.	62.	74.	71.	66.
Arizona	Bisbee COOP#/	55.	58.	70.	67.	62.
	Flagstaff AP	35.	39.	54.	50.	45.
	Ft Huachuca Prvg. Grn.	55.	58.	71.	68.	63.
	Phoenix AP	60.	64.	79.	75.	69.
	Phoenix CO	61.	65.	80.	76.	70.
	Prescott AP	46.	49.	65.	61.	55.
	Tucson AP	59.	62.	76.	73.	68.
	Winslow AP	45.	49.	65.	61.	55.
	Yuma AP	65.	69.	84.	80.	75.
Arkansas	Fort Smith AP	52.	56.	72.	68.	62.
	Little Rock AP	53.	57.	72.	68.	62.
	Texarkana AP	56.	60.	74.	71.	65.
California	Bakersfield AP	56.	60.	74.	70.	65.
	Beaumont CO	53.	56.	67.	64.	60.
	Bishop	47.	51.	65.	61.	56.
	Blue Canyon AP	43.	46.	58.	55.	50.
	Burbank AP	58.	60.	68.	66.	63.
	Eureka CO	50.	51.	54.	54.	52.
	Fresno AP	54.	58.	72.	68.	63.
	Los Angeles AP	58.	59.	64.	63.	61.
	Los Angeles CO	60.	61.	68.	66.	64.
	Mount Shasta CO	41.	44.	57.	54.	49.
	Oakland AP	53.	54.	60.	59.	56.
	Red Bluff AP	54.	58.	72.	69.	63.
	Sacramento AP	53.	56.	67.	64.	60.
	Sacramento CO	54.	57.	68.	65.	61.
	Sandberg CO	47.	50.	63.	60.	55.
	San Diego AP	59.	60.	66.	65.	62.
	San Francisco AP	53.	54.	59.	57.	56.
	San Francisco CO	55.	55.	59.	58.	57.
	San Jose COOP	55.	57.	64.	62.	59.
	Santa Catalina AP	57.	58.	64.	62.	60.
	Santa Maria AP	54.	55.	60.	59.	57.

*/ AP = Airport Data
+/ CO = City Office Data
#/ COOP = Cooperative Weather Station

10-FT AVERAGE EARTH TEMPERATURE FOR UNDERGROUND PIPE HEAT TRANSFER

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Colorado	Alamosa AP	30.	35.	52.	48.	41.
	Colorado Sprgs. AP	39.	43.	59.	55.	49.
	Denver AP	39.	43.	60.	56.	50.
	Denver CO	41.	45.	61.	58.	51.
	Grand Junction AP	39.	44.	65.	60.	52.
	Pueblo AP	41.	45.	62.	58.	51.
Connecticut	Bridgeport AP	40.	44.	61.	57.	50.
	Hartford AP	39.	43.	61.	57.	50.
	Hartford AP Brainer	39.	43.	60.	56.	50.
	New Haven AP	40.	44.	60.	56.	50.
Delaware	Wilmington AP	44.	48.	64.	60.	54.
D.C.	Washington AP	47.	51.	66.	63.	56.
	Washington CO	47.	51.	66.	63.	57.
	Silver Hill OBS	46.	50.	65.	61.	55.
Florida	Apalachicola CO	63.	65.	75.	73.	69.
	Daytona Beach AP	65.	67.	75.	74.	70.
	Fort Myers AP	70.	71.	78.	76.	74.
	Jacksonville AP	63.	66.	75.	73.	69.
	Jacksonville CO	64.	66.	76.	73.	70.
	Key West AP	74.	75.	80.	79.	77.
	Key West CO	75.	76.	81.	79.	78.
	Lakeland CO	68.	69.	77.	75.	72.
	Melbourne AP	68.	70.	77.	75.	72.
	Miami AP	72.	74.	79.	78.	76.
	Miami CO	72.	73.	78.	77.	75.
	Miami Beach COOP	74.	75.	80.	78.	77.
	Orlando AP	68.	70.	77.	75.	72.
	Pensacola CO	62.	64.	74.	72.	68.
	Tallahassee AP	61.	64.	74.	72.	68.
	Tampa AP	68.	69.	77.	75.	72.
	West Palm Beach	71.	73.	79.	77.	75.
Georgia	Albany AP	60.	63.	75.	72.	67.
	Athens AP	54.	58.	71.	68.	63.
	Atlanta AP	54.	57.	70.	67.	62.
	Atlanta CO	54.	57.	70.	67.	62.
	Augusta AP	56.	59.	72.	69.	64.
	Columbus AP	56.	59.	72.	69.	64.
	Macon AP	58.	61.	74.	71.	66.
	Rome AP	53.	56.	70.	67.	61.
	Savannah AP	60.	63.	74.	71.	67.
	Thomasville CO	62.	64.	74.	72.	68.
	Valdosta AP	61.	64.	74.	72.	68.

10-FT AVERAGE EARTH TEMPERATURE FOR UNDERGROUND PIPE HEAT TRANSFER

TG-3

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Idaho	Boise AP	40.	44.	62.	58.	51.
	Idaho Falls 46 W	30.	35.	55.	50.	42.
	Idaho Falls 42 NW	28.	33.	54.	49.	41.
	Lewiston AP	42.	46.	63.	59.	52.
	Pocatello AP	35.	40.	59.	55.	47.
	Salmon CO	32.	37.	56.	52.	44.
Illinois	Cario CO	49.	53.	70.	66.	60.
	Chicato AP	38.	43.	62.	57.	50.
	Joilet AP	37.	42.	61.	56.	49.
	Moline AP	38.	43.	62.	58.	50.
	Peoria AP	39.	44.	63.	58.	51.
	Springfield AP	41.	45.	64.	60.	52.
	Springfield CO	43.	47.	66.	62.	54.
Indiana	Evansville AP	47.	51.	67.	63.	57.
	Fort Wayne AP	39.	43.	61.	57.	50.
	Indianapolis AP	41.	46.	64.	59.	52.
	Indianapolis CO	43.	48.	65.	61.	54.
	South Bend AP	38.	42.	61.	56.	49.
	Terre Haute AP	42.	47.	65.	60.	53.
Iowa	Burlington AP	39.	44.	64.	59.	51.
	Charles City CO	33.	38.	60.	55.	46.
	Davenport CO	39.	44.	64.	59.	51.
	Des Moines AP	37.	42.	63.	58.	50.
	Des Moines CO	38.	43.	64.	59.	51.
	Dubuque AP	34.	39.	60.	55.	47.
	Sioux City AP	35.	40.	62.	57.	49.
	Waterloo AP	35.	40.	61.	56.	48.
Kansas	Concordia CO	42.	47.	67.	62.	54.
	Dodge City AP	43.	48.	67.	62.	55.
	Goodland AP	38.	43.	62.	57.	50.
	Topeka AP	43.	47.	66.	62.	55.
	Topeka CO	44.	49.	68.	63.	56.
	Wichita AP	45.	50.	68.	64.	57.
Kentucky	Bowling Green AP	47.	51.	67.	63.	57.
	Lexington AP	44.	48.	65.	61.	54.
	Louisville AP	46.	50.	67.	63.	56.
	Louisville CO	47.	51.	67.	64.	57.

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Louisianna	Baton Rouge AP	61.	63.	74.	72.	67.
	Burrwood CO	65.	67.	77.	74.	71.
	Lake Charles AP	61.	64.	75.	73.	68.
	New Orleans AP	63.	65.	75.	73.	69.
	New Orleans CO	64.	66.	77.	74.	70.
	Shreveport AP	58.	61.	75.	72.	66.
Maine	Caribou AP	24.	29.	50.	45.	37.
	Eastport CO	33.	37.	51.	48.	42.
	Portland AP	33.	38.	56.	51.	44.
Maryland	Baltimore AP	45.	49.	65.	61.	55.
	Baltimore CO	47.	51.	67.	63.	57.
	Frederick AP	44.	48.	65.	61.	55.
Massachusetts	Boston AP	41.	44.	61.	57.	51.
	Nantucket AP	41.	44.	57.	54.	49.
	Pittsfield AP	34.	38.	55.	51.	44.
	Worchester AP	36.	40.	58.	54.	47.
Michigan	Alpena CO	33.	37.	54.	50.	43.
	Detroit - Willow Run A	38.	42.	60.	56.	49.
	Detroit City AP	38.	43.	60.	56.	49.
	Escanaba CO	30.	35.	53.	49.	42.
	Flint AP	36.	40.	58.	54.	47.
	Grand Rapids AP	36.	40.	58.	54.	47.
	Grand Rapids CO	38.	42.	60.	56.	49.
	East Lansing CO	36.	40.	58.	54.	47.
	Marquette CO	31.	35.	53.	49.	42.
	Muskegon AP	36.	40.	57.	53.	47.
	Sault Ste Marie AP	28.	32.	51.	47.	39.
Minnesota	Crookston COOP	25.	31.	55.	49.	40.
	Duluth AP	25.	30.	52.	47.	38.
	Duluth CO	26.	31.	52.	47.	39.
	International Falls	22.	27.	51.	45.	36.
	Minneapolis AP	32.	37.	60.	54.	46.
	Rochester AP	31.	36.	58.	53.	44.
	Saint Cloud AP	28.	33.	56.	51.	42.
	Saint Paul AP	32.	37.	60.	54.	46.

10-FT AVERAGE EARTH TEMPERATURE FOR UNDERGROUND PIPE HEAT TRANSFER

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Mississippi	Jackson AP	57.	61.	73.	70.	65.
	Meridian AP	57.	60.	72.	69.	64.
	Vicksburg CO	58.	61.	74.	71.	66.
Missouri	Columbia AP	43.	48.	66.	62.	55.
	Kansas City AP	44.	49.	68.	64.	56.
	Saint Joseph AP	42.	47.	67.	62.	54
	Saint Louis AP	45.	49.	67.	63.	56.
	Saint Louis CO	46.	50.	68.	64.	57.
	Springfield AP	45.	49.	66.	62.	56.
Montana	Billings AP	35.	40.	59.	55.	47.
	Butte AP	27.	31.	50.	45.	38.
	Glasgow AP	27.	33.	56.	51.	42.
	Glascow CO	28.	34.	57.	52.	43.
	Great Falls AP	34.	38.	56.	52.	45.
	Harve CP	31.	36.	57.	52.	44.
	Helena AP	31.	36.	55.	50.	43.
	Helena CO	32.	36.	55.	50.	43.
	Kalispell AP	32.	37.	54.	50.	43.
	Miles City AP	32.	37.	59.	54.	45.
	Missoula AP	33.	37.	56.	51.	44.
Nebraska	Grand Island AP	38.	43.	64.	59.	51.
	Lincoln AP	39.	44.	64.	60.	52.
	Lincoln CO UNI	40.	45.	65.	61.	53.
	Norfolk AP	35.	40.	62.	57.	48.
	North Platte AP	37.	42.	62.	57.	49.
	Omaha AP	39.	44.	65.	60.	52.
	Scottbluff AP	36.	41.	60.	56.	48.
	Valentine CO	35.	40.	61.	56.	48.
Nevada	Elko AP	34.	39.	57.	53.	46.
	Ely AP	35.	39.	56.	52.	45.
	Las Vegas AP	56.	60.	78.	74.	67.
	Reno AP	40.	44.	58.	55.	49.
	Tonopah	41.	45.	61.	57.	51.
	Winnemucca AP	38.	42.	60.	56.	49.
New Hampshire	Concord AP	33.	38.	56.	52.	45.
	Mt Washington COOP	17.	21.	37.	33.	27.
New Jersey	Atlantic City CO	45.	49.	63.	60.	54.
	Newark AP	43.	47.	63.	59.	53.
	Trenton CO	43.	47.	64.	60.	53.

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
New Mexico	Albuquerque AP	46.	50.	67.	63.	57.
	Clayton AP	43.	47.	63.	59.	53.
	Raton AP	38.	42.	58.	54.	48.
	Roswell AP	51.	54.	69.	66.	60.
New York	Albany AP	36.	40.	59.	54.	47.
	Albany CO	38.	43.	61.	56.	49.
	Bear Mountain CO	38.	42.	59.	55.	48.
	Binghampton AP	34.	38.	56.	52.	45.
	Binghampton CO	38.	42.	59.	55.	48.
	Buffalo AP	37.	41.	58.	54.	47.
	New York La Guardia	44.	48.	64.	60.	54.
	New York CO	44.	47.	63.	59.	53.
	New York Central Par	44.	48.	64.	60.	54.
	Oswego CO	36.	40.	58.	54.	47.
	Rochester AP	37.	41.	58.	54.	47.
	Schenectady COOP	35.	40.	59.	55.	47.
	Syracuse AP	38.	42.	60.	56.	49.
North Carolina	Asheville CO	48.	51.	64.	61.	56.
	Charlotte AP	52.	55.	69.	66.	60.
	Greensboro AP	49.	53.	67.	64.	58.
	Hatteras CO	56.	59.	70.	68.	63.
	Raleigh AP	51.	55.	69.	65.	60.
	Raleigh CO	52.	56.	70.	66.	61.
	Wilmington AP	56.	59.	71.	69.	64.
	Winston Salem AP	50.	53.	67.	64.	58.
North Dakota	Bismarck AP	27.	33.	56.	51.	42.
	Devils Lake CO	24.	29.	54.	48.	39.
	Fargo AP	26.	32.	56.	50.	41.
	Minot AP	25.	31.	54.	49.	39.
	Williston CO	27.	33.	56.	50.	41.
Ohio	Akron-Canton	39.	43.	60.	56.	50.
	Cincinnati AP	43.	47.	64.	60.	54.
	Cincinnati CO	46.	50.	66.	63.	56.
	Cincinnati ABBE OBS	45.	49.	65.	61.	55.
	Cleveland AP	40.	44.	61.	57.	51.
	Cleveland CO	41.	45.	62.	58.	51.
	Columbus AP	41.	46.	62.	59.	52.
	Columbus CO	43.	47.	64.	60.	53.
	Davton AP	42.	46.	63.	59.	52.

10-FT AVERAGE EARTH TEMPERATURE FOR UNDERGROUND PIPE HEAT TRANSFER

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Ohio (Continued)	Sandusky CO	41.	45.	62.	58.	51.
	Toledo AP	38.	43.	60.	56.	49.
	Youngstown AP	39.	43.	60.	56.	50.
Oklahoma	Oklahoma City AP	50.	54.	71.	67.	60.
	Oklahoma City CO	50.	55.	71.	68.	61.
	Tulsa AP	50.	54.	71.	67.	61.
Oregon	Astoria AP	47.	48.	56.	54.	51.
	Baker co	36.	40.	56.	52.	46.
	Burns CO	36.	40.	58.	54.	47.
	Eugene AP	46.	48.	59.	57.	52.
	Meacham AP	34.	38.	52.	49.	43.
	Medford AP	46.	49.	62.	59.	54.
	Pendelton AP	42.	46.	63.	59.	53.
	Portland AP	46.	49.	60.	57.	53.
	Portland CO	48.	50.	61.	59.	55.
	Roseburg AP	47.	49.	60.	57.	53.
	Roseburg CO	48.	51.	61.	59.	55.
	Salem AP	46.	49.	60.	57.	53.
	Sexton Summit	42.	44.	55.	52.	48.
	Troutdale AP	45.	48.	59.	57.	52.
Pennsylvania	Allentown AP	40.	44.	62.	58.	51.
	Erie AP	38.	42.	58.	55.	48.
	Erie CO	40.	44.	60.	56.	50.
	Harrisburg AP	43.	47.	63.	59.	53.
	Park Place CO	36.	40.	57.	53.	46.
	Philadelphia AP	44.	48.	64.	61.	54.
	Philadelphia CO	46.	50.	66.	62.	56.
	Pittsburgh Allegheny	42.	46.	62.	58.	52.
	Pittsburgh Grtr Pitt	40.	44.	61.	57.	51.
	Pittsburgh CO	44.	48.	64.	60.	54.
	Reading CO	43.	47.	64.	60.	54.
	Scranton CO	40.	44.	61.	57.	50.
	Wilkes Barre-Scranton	39.	43.	60.	56.	49.
	Williamsport AP	40.	44.	61.	57.	51.
Rhode Island	Block Island AP	41.	45.	59.	55.	50.
	Providence AP	39.	43.	59.	56.	49.
	Providence CO	41.	45.	62.	58.	51.

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
South Carolina	Charleston AP	58.	61.	72.	70.	65.
	Charleston CO	60.	62.	74.	71.	67.
	Columbia AP	56.	59.	72.	69.	64.
	Columbia CO	57.	60.	72.	69.	64.
	Florence AP	55.	59.	72.	69.	64.
	Greenville AP	53.	56.	69.	66.	61.
	Spartanburg AP	53.	56.	70.	66.	61.
South Dakota	Huron AP	31.	37.	60.	55.	46.
	Rapid City AP	34.	39.	58.	54.	46.
	Sioux Falls AP	32.	37.	60.	55.	46.
Tennessee	Bristol AP	48.	51.	65.	62.	56.
	Chattanooga AP	51.	55.	69.	65.	60.
	Knoxville AP	50.	54.	68.	65.	59.
	Memphis AP	52.	56.	71.	68.	62.
	Memphis CO	53.	57.	72.	68.	62.
	Nashville AP	51.	54.	69.	66.	60.
	Oak Ridge CO	49.	52.	67.	64.	58.
	Oak Ridge 8 S	49.	52.	67.	64.	58.
Texas	Abilene AP	55.	58.	73.	70.	64.
	Amarillo AP	47.	50.	67.	63.	57.
	Austin AP	60.	63.	76.	73.	68.
	Big Springs AP	56.	59.	74.	70.	65.
	Brownsville AP	68.	70.	79.	77.	74.
	Corpus Christi AP	65.	68.	78.	76.	72.
	Dallas AP	57.	61.	76.	72.	66.
	Del Rio AP	62.	65.	77.	75.	70.
	El Paso AP	54.	58.	72.	69.	63.
	Fort Worth AP	57.	60.	75.	72.	66.
	Forth Worth Amon Cart	57.	60.	75.	72.	66.
	Galveston AP	63.	66.	77.	74.	70.
	Galveston CO	63.	66.	77.	74.	70.
	Houston AP	62.	65.	76.	73.	69.
	Houston CO	63.	66.	77.	74.	70.
	Laredo AP	67.	70.	81.	79.	74.
	Lubbock AP	50.	54.	69.	65.	59.
	Midland AP	55.	59.	73.	70.	64.
	Palestine CO	58.	62.	74.	71.	65.
	Port Arthur AP	61.	64.	75.	72.	68.
	Port Arthur CO	63.	65.	76.	74.	69.
	San Angelo AP	58.	61.	74.	71.	66.
	San Antonio AP	61.	64.	77.	74.	69.
	Victoria AP	64.	67.	78.	76.	71.

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STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Texas (Continued)	Waco AP	58.	62.	76.	73.	67.
	Wichita Falls AP	53.	57.	73.	69.	63.
Utah	Blanding CO	39.	43.	60.	56.	50.
	Milford AP	37.	42.	61.	56.	49.
	Salt Lake City AP	40.	44.	63.	59.	51.
	Salt Lake City CO	41.	46.	65.	60.	53.
Vermont	Burlington AP	32.	37.	57.	52.	44.
Virginia	Cape Henry CO	51.	55.	68.	65.	60.
	Lynchburg AP	48.	51.	66.	62.	57.
	Norfolk AP	51.	54.	68.	64.	59.
	Norfolk CO	52.	56.	69.	66.	61.
	Richmond AP	48.	52.	67.	63.	58.
	Richmond CO	50.	53.	68.	64.	59.
	Roanoke AP	48.	51.	66.	62.	57.
Washington	Ellensburg AP	37.	41.	59.	55.	48.
	Kelso AP	45.	47.	57.	54.	51.
	North Head L H Resvn	47.	49.	54.	53.	51.
	Olympia AP	44.	46.	56.	54.	50.
	Omak 2NW	36.	40.	59.	55.	47.
	Port Angeles AP	45.	46.	53.	52.	49.
	Seattle AP Boeing FI	46.	48.	58.	56.	52.
	Seattle CO	47.	50.	59.	57.	53.
	Seattle-Tacoma AP	44.	47.	57.	55.	51.
	Spokane AP	37.	41.	58.	54.	47.
	Stampede Pass	32.	35.	48.	45.	40.
	Tacoma CO	46.	48.	58.	55.	52.
	Tattosh Island CO	46.	47.	52.	51.	49.
	Walla Walla CO	44.	48.	65.	61.	54.
	Yakima AP	40.	44.	61.	57.	50.
West Virginia	Charleston AP	47.	50.	65.	61.	56.
	Elkins AP	41.	45.	59.	56.	50.
	Huntington CO	48.	52.	67.	63.	57.
	Parkersburg CO	45.	49.	65.	61.	55.
	Petersburg CO	44.	48.	63.	60.	54.

STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Wisconsin	Green Bay AP	31.	36.	56.	51.	44.
	La Crosse AP	32.	38.	60.	55.	46.
	Madison AP	34.	39.	59.	54.	47.
	Madison CO	34.	39.	60.	55.	47.
	Milwaukee AP	35.	40.	58.	54.	47.
	Milwaukee CO	36.	41.	59.	55.	48.
Wyoming	Casper AP	34.	38.	57.	52.	45.
	Cheyenne AP	35.	39.	55.	51.	45.
	Lander AP	31.	35.	56.	51.	43.
	Rock Springs AP	31.	35.	54.	50.	42.
	Sheridan AP	33.	37.	56.	52.	44.
Hawaii	Hilo AP	72.	72.	74.	74.	73.
	Honolulu AP	74.	75.	77.	77.	76.
	Honolulu CO	74.	74.	77.	76.	75.
	Lihue AP	72.	73.	76.	75.	74.
Alaska	Anchorage AP	25.	29.	46.	42.	35.
	Annette AP	40.	42.	51.	49.	46.
	Barrow AP	4.	7.	16.	14.	10.
	Bethel AP	18.	23.	41.	37.	30.
	Cold Bay AP	33.	35.	43.	41.	38.
	Cordova AP	32.	35.	45.	43.	39.
	Fairbanks AP	14.	19.	38.	34.	26.
	Galena AP	13.	18.	37.	33.	25.
	Gambell AP	15.	19.	34.	30.	24.
	Juneau AP	34.	36.	47.	45.	41.
	Juneau CO	36.	39.	49.	46.	42.
	King Salmon AP	25.	28.	44.	40.	34.
	Kotzebue AP	10.	14.	31.	27.	21.
	McGrath AP	14.	18.	37.	33.	25.
	Nome AP	16.	20.	37.	33.	26.
	Northway AP	12.	16.	32.	29.	22.
	Saint Paul Island AP	31.	32.	40.	38.	35.
	Yakutat AP	33.	36.	45.	43.	39.
West Indies	Ponce Santa Isabel AP	75.	76.	78.	78.	77.
	San Juan AP	77.	77.	79.	79.	78.
	San Juan CO	77.	77.	79.	79.	78.
	Swan Island	80.	80.	82.	81.	81.
	St. Croix V I AP	78.	78.	81.	80.	79.

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STATE	CITY	WINTER	SPRING	SUMMER	AUTUMN	ANNUAL
Pacific Islands	Canton Island AP	83.	84.	84.	84.	84.
	Koror	81.	81.	81.	81.	81.
	Ponape Island AP	81.	81.	81.	81.	81.
	Truk Moen Island	81.	81.	81.	81.	81.
	Wake Island AP	79.	79.	81.	81.	80.
	Yap	81.	81.	82.	82.	82.

APPENDIX D. A SURVEY OF COMPUTER PROGRAMS FOR EVALUATING BUILDING AND SYSTEM PERFORMANCE

CONTENTS

APPENDIX D. SURVEY OF COMPUTER PROGRAMS FOR EVALUATING BUILDING AND SYSTEM PERFORMANCE

1. INTRODUCTION AND BACKGROUND

The use of computers for building thermal loads and energy consumption analysis has seen a rapid increase, from almost no activity, in the last ten years. Appendix A of this report identified those factors which contribute to the heating and cooling loads of buildings, and described in detail the mechanisms by which they do so. Methods for calculating the amount of energy required by building HVAC systems and equipment to satisfy those loads were also described. The complexity of the load and system energy consumption rate calculations at any given time, and the repetitive nature of these calculations when determining cumulative energy consumption over some extended period of time, (particularly annual consumptions) make it difficult, if not impossible, to accomplish detailed energy consumption analyses unless the calculation procedures are computerized. Recent (and still developing) calculation procedures recommended by ASHRAE have been allowed to become as complex in the pursuit of accuracy as they have only because of the explicit recognition that they would be computerized. In other than the simplest buildings with the simplest HVAC systems, it is not feasible to employ such procedures manually. The speed of the computer allows greater accuracy because all processes that contribute to energy consumption in a building can be modeled in great detail, using as few simplifying assumptions as possible. In addition, repetitive calculations can be made at specific intervals (usually on an hourly basis) to allow for the real-time variation of parameters such as weather, sun location and occupant activity, and related equipment-use rates which affect energy consumption requirements. The feasibility of performing hourly calculations eliminates the need for simplifying shortcuts, such as degree-day, or temperature bin methods, which decrease the accuracy of energy analysis calculations. Computerization also lends itself to the capability to store many different simulation algorithms (particular calculation sequences) representing models of the different construction designs and systems that can occur in buildings, and to link alternative components in system models in many ways. This allows the program to be very comprehensive, and flexible enough to simulate closely the actual energy performance of a diverse range of building designs.

Programs developed for energy analyses tend to be structured alike, and are usually divided into four main segments. The first part calculates, for each thermal zone in a building, the space loads (heating and/or cooling) necessary to maintain specified interior conditions. Input data required include details of the building design, weather conditions, and interior occupancy and equipment use schedules. The second segment simulates the HVAC air-side system (or systems) which are controlled to satisfy the previously calculated space loads. Input to this section of the program includes type of system, performance characteristics, and operation and control procedures. Many of the existing programs contain generic simulations of all or most of the HVAC systems usually encountered, so that the program user has only to specify which ones are to be used in any particular analysis. Some programs are able to simulate more than one HVAC system simultaneously, with different systems satisfying the loads of different thermal zones in the building. The third program segment simulates primary equipment such as chillers, boilers and on-site power generation, and computes the purchased energy requirements necessary for the primary equipment to satisfy the system energy requirements of the HVAC zones in the building. Input requirements include specification of performance characteristics and actual thermal configuration of the equipment. Like the HVAC systems segment, the equipment segment of some programs contains generic simulations of many types of equipment that are usually encountered, and the program user merely specifies the ones which are to be used in any particular analysis. The minimum outputs from this section are usually compilations of monthly and annual energy consumptions for each type of energy to be purchased. In addition, energy consumption/conversion by particular pieces of equipment, monthly and annual operating hours and peak consumption rates can often be output at the discretion of the user. The fourth general program segment consists of an economic analysis of some appropriate type. Such analyses account for the owning and operating cost of the building energy-related systems, and require as input data the cost of the building equipment and purchased energy and/or fuel.

The structure described above is not followed exactly by all programs to be mentioned later, some of which do only part of the analysis sequence. To be considered comprehensive, however, the programs considered should accomplish the majority of the analysis segments described above. For the most part, the proprietary programs to be described are the most comprehensive in aspects of the analysis sequence. They have been set up that way in order to offer a complete analysis sequence in a convenient manner to the potential user. The flexibility of the proprietary programs is also in part due to the engineering services that are provided in conjunction with the use of the program itself.

Both the public domain programs and the proprietary programs to be described have the capability of being applied to either new designs or to effects of retrofit actions for existing buildings. Some of the programs are set up in such a manner that they can analyze several similar alternative building/system configurations automatically during the course of one analysis sequence. This allows direct comparison of energy consumption and/or economic consequences of the alternatives.

The computer programs in this survey have been selected on the basis of their overall comprehensiveness and flexibility, as well as being representative of the accepted state of the art for such analyses. They are generally available in one of three ways.

(1) The program is not considered proprietary and the source code may be purchased (in some cases for a nominal fee) for implementation on the purchaser's computer facilities.

(2) The program is proprietary in nature but can be utilized by means of time-sharing facilities or by leased object language codes.

(3) The program is proprietary and can be utilized only by submitting input data to the program developer who checks the input data and actually makes the program runs.

Outright purchase of a source code is generally most desirable only when significant in-house computer facilities exist. The purchase price of the source code will generally be only a small fraction of the total cost of implementing a program. In addition, the purchaser normally must expend considerable effort in order to assure compatibility with his computer system, to perform debugging and program checkout, and to maintain the program. The problem is compounded by the lack of adequate documentation for a number of the programs being offered. For all of these reasons, acquisition and maintenance of a complex energy analysis program are probably not feasible below the command level. Such a program (to be described subsequently) has been developed by the U.S. Army Construction Engineering Research Laboratory (CERL) for the Air Force.

The problem of initial implementation and day-to-day maintenance of a program can be overcome by utilizing programs that are available on nationwide computer time-sharing networks and service bureaus. The disadvantage of utilizing these programs is that, since the program listings are generally proprietary, the user is frequently unaware of the assumptions and limitations of the program. In addition, the user gives up the flexibility to modify or improve the program which would be possible if he owned the program. These considerations are probably less important at the Base level than they are to the researcher or to active building designers. Most important of all, effective use of a complex program in a time-sharing mode requires that the user be thoroughly trained in both HVAC design and engineering and in correct use of the program itself. This expertise takes some time to be acquired and may not be readily available at the base level.

The third category of program availability, that of submitting input data to a developer for execution, is normally offered by engineering firms, by manufacturers of equipment or architectural materials, and by utilities. The manufacturers and utilities generally offer the service to customers or potential customers at no (or nominal) charge, while the engineering firms' charges vary and are in many instances given as "negotiable". In these cases, the distinction between available computer programs and consulting services which include computer capabilities becomes somewhat unclear. Generally, the user pays more for such a service than if an in-house staff were maintained for such services. This situation can be advantageous, however, if the user requirements for energy analysis are infrequent enough to justify the support of such a staff.

Many of the programs that are subsequently described were at one time available only for batch runs. In the case of proprietary programs, the input data had to be submitted through the program owner who also served as a technical consultant on the proper input of data to the program. The cost of this service is reflected in the rates charged for the use of the programs in this mode. More recently, there has been a gradual and increasing implementation of these programs on time-sharing systems so they are directly available to the user. The benefits of this development mean wider availability of the programs, faster turnaround times, and generally lower costs, since the user requires no technical consulting expertise of the program owner. Royalties for program use are included in the time-sharing fee structure. The primary disadvantage of direct use of the program is the necessity for the user to become familiar with all aspects of the use of the programs through training, which is sometimes extensive.

2. PROGRAM SURVEY

Pursuant to the above discussion, a representative survey of existing programs is presented. The programs are listed in alphabetical order, and their inclusion or discussion does not imply any program endorsement or lack thereof.

The format is consistent for each program and consists of a short description, input requirements, outputs, source code language, systems on which the software is implemented, maintenance of software, its availability, and any additional comments which might be relevant.

ALTERNATE CHOICE COMPARISON FOR ENERGY SYSTEMS SELECTION (Code Name: AXCESS)

DESCRIPTION: AXCESS permits the simultaneous comparison of up to six alternate methods for meeting the energy requirements of a building. The program may be used at various points in the design process. If the structure is still in the conceptual state, the program utilizes routines which approximate the full input information which would normally be available during the later stages of design. The program calculates hourly zone solar and transmission loads for the year (if not input from another program). These loads are utilized along with base energy loads to calculate net zone space conditioning requirements. Terminal system operation is the simulated and equipment energy consumption calculated in one of three general subroutines.

INPUT REQUIREMENTS: Input requirements include weather data and control information along with user-supplied information such as project description, base load profiles, design heating and cooling load, space type data, zone data, HVAC system description, and output desired.

PROGRAM OUTPUTS: Energy consumption and demand for each of up to 36 "meters" for each of the alternative systems. Numerous optional outputs.

PROGRAM LIMITATIONS: Up to 180 energy zones may be calculated per run. A maximum of six schemes or combinations of primary and secondary systems are possible. Each scheme may have up to 12 different types of fan systems and six primary systems.

MAINTAINED BY: S&H Information Systems, Inc., New York, New York, a subsidiary of Syska & Hennessy and the Edison Electric Institute.

AVAILABILITY: AXCESS is available through five different means
(1) Through participating utilities
(2) Time sharing systems (National C.S.S., Inc.)
(3) Source deck may be purchased
(4) Object deck may be purchased
(5) As a service from a participating consulting firm.
For more information contact:

E.S. Douglass
Edison Electric Institute
90 Park Avenue
New York, New York 10017

ADDITIONAL COMMENTS: This program series offers a complete energy analysis of a single or multi-building project, including load determinations, air-side system simulations, plant equipment simulations, and annualized owning and operating costs. The series consists of a complex energy analysis program and an independent short financial analysis program. The period of analysis is usually one year.

The thermal load calculations and complete energy analysis are accomplished each hour before proceeding to the next hour. Therefore the program will not size the plant equipment and base part-load efficiencies on that size. All calculations assume steady-state or equilibrium conditions. Thermal loads are scaled based on design loads and design weather conditions. Building loads based on these calculations are suitable for comparison of alternatives. Response factors may be input (from another program) or default values for light, medium and heavy construction may be utilized.

The air-side and plant systems simulation package is one of the more complete of the program series surveyed. Addition or adaptation of equipment simulations is facilitated by the modular package structure. Estimated equipment size is a required input for part-load performance curves. If the demand exceeds capacity, the design efficiency will be used. The air-handling and plant system simulations available are:

a. Dual Duct
b. Multi-Zone
c. Single Zone Reheat
d. 100% Variable Volume
e. Variable Volume with Reheat
f. Ceiling Induction
g. Heating and Ventilating
h. 2-Pipe Induction
i. 4-Pipe Induction
j. 2-Pipe Fan Coil
k. 2-Pipe Unit Ventilator
l. 4-Pipe Fan Coil
m. 4-Pipe Unit Ventilator
n. Unitary Cooling Units with Separate Heating
o. Unitary Heat Pumps
p. Boilers
q. Furnace
r. Refrigeration
s. Simultaneous Heat Pumps
t. Changeover Heat Pumps
u. On-Site Generation-KW Balance
v. On-Site Generation-Thermal Balance

BUILDING LOAD ANALYSIS AND SYSTEMS THERMODYNAMICS PROGRAM
(Code Name: BLAST)

DESCRIPTION: The Building Load Analysis and Systems Thermodynamics Program was developed by the U.S. Army Construction Engineering Research Laboratory for the Air Force (see Additional Comments). The program contains three simulation segments (loads, air-side systems, and building equipment), a well developed, almost conversational, user-oriented input language and preprocessor for inputting and checking data and setting default values, an output report writer, and provisions for a library of data of building materials and component properties, equipment characteristics, various use schedules and system control procedures. The unique program structure together with sophisticated programming techniques has resulted in a more efficient and faster running program than others using comparable analysis methodology. The library is easily expandable to include a vast amount of data, thereby shortening considerably the amount of data required for input by the user in most cases, while still allowing the user input control if it is desired.

The loads portion is based on algorithms developed from NBSLD, the NBS Loads Determination Program. The systems simulation is in part based on algorithms derived from the NASA Energy and Cost Analysis Program (NECAP). The building equipment simulation segment is a newly developed software package which provides great flexibility in simulating energy configurations of components.

INPUT REQUIREMENTS: Is highly user-oriented and variable in the amount of detail that has to be provided. The user can input a minimum of information and use many built-in default values from the program library, or alternatively provide complete information as input.

PROGRAM OUTPUTS: Input data echo, diagnostic messages plus assigned default values to override erroneous input data, system configurations description, equipment energy consumption analysis (monthly and annual).

PROGRAMMING LANGUAGE: The program is written in CDC FORTRAN Extended, Version 4.

SYSTEM IMPLEMENTATION: Can be used on CDC 6000/7000 series computers without modification.

DOCUMENTATION: The documentation is expected to be very clear and comprehensive. Volume I will provide detailed user instructions and Volume II will contain detailed flowcharts and algorithmic descriptions of the program.

MAINTAINED BY: Dept. of the Army
Construction Engineering Research Laboratory (CERL)

AVAILABILITY: For more information contact:

F. Beason
Air Force Civil Engineering Center (AFCEC)
Tyndall Air Force Base, Florida 32403

ADDITIONAL COMMENTS: This development was conducted at the U.S. Army Construction Engineering Research Laboratory (CERL) for the Air Force Civil Engineering Center, Tyndall AFB, under Work Unit 201021003, Optimization of Energy Usage in Military Facilities. D. Hittle is the principal investigator at CERL.

ENERGY CONSERVATION UTILIZING BETTER ENGINEERING
(Code Name: ECUBE)

DESCRIPTION: The ECUBE series of programs (except for ECUBE III) utilizes design point calculations of peak thermal and electrical loads to make hourly, monthly, and annual estimates of the energy requirements of a building. The energy consumption of various types of systems which may be used to meet these requirements is then calculated and the program compares the total owning and operating costs of the various systems being considered. The program series consists of three basic computer programs: (1) Energy Requirements Program, (2) Equipment Selection and Energy Consumption Program, and (3) Economic Comparison Program.

INPUT REQUIREMENTS: Maximum value and hourly percentage profiles for internal heat gain, electrical load and process load. Transmission and outside air loads as a function of ambient temperature. Maximum solar load on the building. Hourly weather data (available in ready-to-use form for many cities). Heating value of fuel. Part-load equipment performance characteristics. Capital cost differential for alternative systems, salvage values, maintenance costs.

PROGRAM OUTPUTS: Monthly, annual, and peak energy requirements. Annual fuel requirements for alternative systems. Cash flow and discounted rate of return on alternative investments.

COST: Charges based on number and size of runs made.

RATE AND STATUS: The present form of the program which became available in 1975 is technically labeled as Version 2.3 and is marketed under the name ECUBE 75.

MAINTAINED BY: The American Gas Association and Control Data Corporation.

AVAILABILITY: Program can be utilized via Control Data Corporation's Cybernet time sharing network or via leasing agreement. For more information contact:

Mr. K. T. Cuccinelli
Manager--Energy Systems
American Gas Associations
1515 Wilson Boulevard
Arlington, Virginia 22209
Ph. (703) 524-2000

ADDITIONAL COMMENTS: This proprietary program is another which falls in the category of offering a complete energy analysis of a single or multi-building project, from building loads to mechanical systems energy requirement simulation, to utility costs based on those requirements, to economic comparisons between alternate configurations. The period of performance analysis is usually taken to be one year. Thermal load calculations are done on an hourly basis for this entire period. Transmission loads are obtained by scaling, according to weather conditions, from peak-load values and their associated weather conditions, which must be supplied to the program. Loads based on this type of calculation are suitable for determining comparative energy performance of alternate systems. Since the program is available in the time sharing or remote batch mode, user familiarily is necessary for proper utilization. The American Gas Association sponsors regularly scheduled workshops in training for new program users, and publishes a newsletter for all program users. A new and greatly revised version of the program, E-CUBE III is presently under development and should be available shortly. It is expected to have a more sophisticated and complete load calculation sequence, extensive weather data files, and more comprehensive system and equipment simulation.

MERIWETHER ENERGY SYSTEMS ANALYSIS SERIES
(Code Name: ESAS)

DESCRIPTION: The Energy Systems Analysis Series is a library of Computer programs which determine the annual energy consumption of various types of systems and equipment for a typical year of operation, and determine the relationship between these energy costs and other owning and operating costs. The programs normally used in conducting an energy analysis consist of the Energy Requirements Estimate (ERE) which calculates the hour-by-hour thermal and electrical loads for a building, the Equipment Energy Consumption (EEC) which simulates the operation of the various pieces of equipment, and the Economic Comparison of Systems (ECS) which calculates the total owning and operating costs of each system. A variety of auxiliary programs are available to complement the basic analysis series.

INPUT REQUIREMENTS: Engineering design data; description of system design and operation, weather data, utility rate structure, owning and operating cost and financial data.

PROGRAM OUTPUTS: Monthly and annual demand and consumption of energy for systems and equipment; monthly and annual utility costs; life cycle and cash flow analysis.

DOCUMENTATION: No algorithm ever published, since software is proprietary. A user's manual, available only from the program owner, has instructions which are complete, however.

DEVELOPER: Ross F. Meriwether and Associates, Inc., San Antonio, Texas.

AVAILABILITY: The Energy Systems Analysis Series is available via consulting agreements with R. F. Meriwether and Associates, various time sharing systems, or through a special lease agreement. For more information contact:

Ross F. Meriwether and Associates, Inc.
1600 N.E. Loop 410
San Antonio, Texas 78209
Ph. (512) 824-5302

ADDITIONAL COMMENTS: The program series is another of those proprietary systems which offers a complete energy analysis of a single or multi-building project, from load determination to energy requirements simulations of building air-side systems and plant equipments, to economic comparisons between alternative candidate configurations. In this series mid-stream evaluation of results is possible. The period of performance analysis is usually taken to be one year.

Thermal load calculations are done on a hourly basis for this period. Transmission loads are linearly scaled from design values, at design point weather conditions. Both loads and design values are supplied to the program. Building loads based on this type of calculation are suitable for determining comparative energy performance of different mechanical systems, (or the same system controlled in different ways), for the same building.

All simulations of air-side systems and mechancial equipment energy performance are of a steady-state, energy-balance nature. The user needs a comprehensive knowledge of air-side systems, not too easy to use otherwise. It is important that the user understand the computer system simulation.

Equipment and air-handling systems configurations are available because separate types of equipment have separate simulations, which can be limited in many possible ways. The air-handling system simulations available are:

a. No excess cooling or reheating, demand coil leaving temperatures.
b. Terminal reheat with scheduled (or fixed) cold coil discharge temperature during cooling.
c. Terminal reheat with scheduled (or fixed) cold coil discharge temperature during cooling or heating.
d. Introduction or fan-coil type system with scheduled primary air temperature.
e. Terminal reheat with cold coil discharge temperature set by maximum demand of any section.
f. Dual-duct or multi-zone with scheduled (or fixed) hot and cold deck temperatures.
g. Dual-duct or multi-zone with deck temperatures set by greatest demand.
h. Variable volume system for solar and internal loads, with separate single-duct system to offset transmission.
i. Standard variable volume system.

A wide range of mechnical and plant configurations can be simulated by construction from the individual simulations. Although total energy systems are not simulated as a single configuration, some types are constructable from the individual equipment simulations. A simulation of a bulk hot water thermal storage system is available.

APEC HEATING-COOLING CALCULATION PROGRAM
VERSION III
(Code Name: HCC-III)

DESCRIPTION: The program calculates design heating and cooling load utilizing ASHRAE methodology. The room is the basic level of calculation for which the requisite design data is specified. Heating load calculations consist of conventional transmission loss analyses involving areas, U factors, and temperature differences. Infiltration is calculated on an input or master factor basis. The maximum glass solar heat gain for the specified winter month is calculated and its effect on the heat loss for the room is determined along with lighting and occupant heat gain. Cooling load calculations are made on an hourly basis over a 24-hour period for the selected design day, and radiant load components are time-averaged in accordance with the building mass. Glass loads are separated into transmission and solar components, taking into account the glazing material and shading devices. Hourly room loads and cfm values are accumulated and analyzed psychometrically, and peak loads on equipment are determined.

INPUT REQUIREMENTS: Building description, weather data, solar radiation data, zone/room designations, room description, and system description.

PROGRAM OUTPUTS: Room loads and cfm values, zone loads and cfm values, system analysis results, building peak refrigeration and heating load. Optional output available for cooling load analysis for 12 hours of a design day.

PROGRAMMING LANGUAGE: FORTRAN IV

SYSTEM IMPLEMENTATION: IBM 1130 or 360, CED Cybernet System, GE MARK III

MAINTAINED BY: Automated Procedures for Engineering Consultants (APEC), Dayton, Ohio

AVAILABILITY: Program available under a fee for license arrangement. For more information, contact:

APEC Executive Office
Grant-Deneau Tower, Suite M-15
Dayton, Ohio 45402
Ph. (513) 228-2602

ADDITIONAL COMMENTS: HCC-III calculates only peak heating and cooling loads for a building for each thermal zone; it does not have a subsequent systems analysis capability. APEC is presently in the process of developing a more comprehensive energy analysis program based on the NECAP program (see subsequent entry in this survey). APEC membership consists primarily of engineering firms who use the APEC programs as design tools in building design projects. The APEC programs are available, then, along with the services of such firms who as APEC members are knowledgeable in their use.

McDONNELL DOUGLAS AUTOMATION COMPANY'S ANNUAL CONSUMPTION OF ENERGY PROGRAM (Code Name: MACE)

DESCRIPTION: MACE uses methods recommended by ASHRAE wherever possible. The load calculation section utilizes methods outlined in Proposed Procedures for Determining Heating and Cooling Loads for Energy Calculations, ASHRAE 1969. These methods include the use of thermal response factors for calculation of the transient heat transfer through roofs and walls. The component and system simulation section uses procedures outlined in Proposed Procedures for Simulating the Performance of Components and Systems for Energy Calculations, ASHRAE, 1969. The economic section uses the local prevailing utility rates, which are input into the program by the user. The U.S. Weather Bureau is the source of the following hourly weather variables: dry and wet-bulb temperature, wind velocity and direction, cloud cover, cloud type, and barometric pressure. MACE will accept other sources of weather data, once they have been prepared in the proper format.

INPUT REQUIREMENTS: Weather specification, building and system descriptions, months for which calculations are to be made, energy rate structures, master time schedules and appropriate loads (such as occupants, lights, appliances, etc.).

PROGRAM OUTPUT: Hourly space and building loads, electrical usage, fuel consumption, and cost of each on hourly basis and monthly and yearly totals of each.

PROGRAMMING LANGUAGE: FORTRAN IV for IBM system/360 Computer

MAINTAINED BY: McDonnell Douglas Automation Company

AVAILABILITY: Program is available on McDonnell Douglas Time Sharing System. For more information contact.

Mr. Charles E. Whitman
Engineering Services
McDonnell Douglas Automation Company
Box 516
St. Louis, Missouri 63166

MEDSI ANNUAL ENERGY CONSUMPTION

DESCRIPTION: Four programs, model reheat, heat-cool-off, multizone, or double duct, and variable volume systems. Weather files are derived from Air Force Manual 88-8 and U.S. Weather Bureau Climates of States. Calculations are performed for each weather occurrence listed in AFM 88-8 with probabilities of occupancy and solar gain superimposed. Unique features of each system are modeled, and system behavior is a fundamental part of each program. A program checks input data and prints it for documentation. An auxiliary program gives input energy requirements in KWH and 10^6 Btu fuel input, considering equipment efficiencies and part-load profiles for inputs of fuel and electricity to heating and cooling equipment and auxiliaries. Additional programs sum up to four system program runs, model internal source heat pumps, total energy plants, and heat recovery devices.

INPUT REQUIREMENTS: Building design loads, occupancy schedule, temperatures, humidities, enthalpies, air quantities, building gains and losses, internal gains, fan heat.

PROGRAM OUTPUT: Cooling requirements in ton-hours and heating requirements in 10^5 Btu for night, day, and evening periods of each month, plus subtotals and totals. Separate tabulations of monthly use of energy for illumination, fans, and pumps, heating/refrigeration system auxiliaries, domestic hot water system can be output.

MAINTAINED BY: Mechanical Engineering Data Services, Inc. (MEDSI)

AVAILABILITY: Program written in FORTRAN and currently available for time-sharing and batch customers of United Computing Systems. For more information contact:

Mechanical Engineering Data Services, Inc.
30 Kimler Drive
Maryland Heights, Missouri 63043
Ph. (314) 434-2340

or

United Computing Systems, Inc.
2525 Washington Street
Kansas City, Missouri 64108
Ph. (816) 221-9700

ADDITIONAL COMMENTS: This is a series of programs that can find use in certain comparative design and retrofit applications. Training in its proper use by the program owner is required. It is somewhat less expensive to use than the more comprehensive programs.

NATIONAL BUREAU OF STANDARDS LOAD DETERMINATION PROGRAM (Code Name: NBSLD)

DESCRIPTION: NBSLD calculates the hour-by-hour heating and/or cooling load in buildings. The program utilizes the thermal response factor technique for calculating transient heat conduction through walls and roofs. The program also includes a routine which can calculate the "floating" temperature of those rooms with limited heating or air-conditioning and natural ventilation, based on the actual net heat loss or gain to the room.

INPUT REQUIREMENTS: Building parameters such as wall properties, ceiling and floor properties, window area and location, operating schedules. Weather data from U.S. Weather Bureau tapes.

PROGRAM OUTPUTS: Hour-by-hour heating or cooling loads and temperatures for each zone for the entire year. Design-day data may be examined as an option.

PROGRAMMING LANGUAGE: FORTRAN V

DEVELOPER: T. Kusuda - National Bureau of Standards

DOCUMENTATION: NBSLD, the Computer Program for Heating and Cooling Loads in Buildings, NBS Building Science Series 69, (1976), and ASHRAE bulletin entitled, Algorithms for Building Heat Transfer Subroutines, 1975.

AVAILABILITY: Source code and documentation may be obtained for a very nominal fee. For more information contact:

James P. Barnett
Thermal Engineering Section
Center for Building Technology, IAT
National Bureau of Standards
Washington, D.C. 20234

ADDITIONAL COMMENTS: NBSLD, while only a loads program, is generally recognized as the most comprehensive and accurate program for that purpose that exists today. It has been linked with the Meriwether ESAS programs, and with NECAP to produce two hybrids with complete analysis capability.

NASA's ENERGY COST ANALYSIS PROGRAM
(Code Name: NECAP)

DESCRIPTION: The NECAP program follows the procedures outlined in the ASHRAE booklet "Procedures for Determining Heating and Cooling Loads for Energy Calculation" to estimate the energy requirements for buildings. The program is actually a set of six individual computer programs including (1) a Response Factor program, (2) a Data Verification program, (3) a Thermal Loads Analysis Program, (4) a Variable Temperature program (5) a System and Equipment Simulation program, and (6) an Owning and Operating Cost program. Standard wall construction and schedules can be used to simplify program input. The program is an extension of the Energy Utilization Program developed for the U.S. Postal Service, but incorporates extensive modification to improve its usability including completely revised documentation.

INPUT REQUIREMENTS: Building parameters, building coordinate system, azimuth angle, surface description and tilt angle, floor, ceiling, furnishing data, space description, thermostat schedules, type of energy distribution system and pertinent parameters, cost information.

PROGRAM OUTPUTS: Response factors if requested, summary of design day weather, space and building loads, surface shadow pictures and shadow calculations, recommended space heat extraction and addition rates, variable temperature loads if requested, zone air flows, summary of loads not met, equipment capacity summary, monthly and annual energy summary.

PROGRAMMING LANGUAGE: FORTRAN IV

SYSTEM IMPLEMENTATION: CDC 6400, 6600

AVAILABILITY: Program is available through

COSMIC
Suite 112
Barrow Hall
University of Georgia
Athens, Georgia 30602
Ph. (404) 542-3265
Attn. Ron English

For more information contact:

R. N. Jensen
Technical Assistant
Construction Engineering Branch
Mail Stop 227
Langley Research Center
Hampton, Virginia 23665

ADDITIONAL COMMENTS: NECAP, an extension of the early-developed Postal Service energy analysis program, is in turn the basis for a number of subsequent developments, because it is the most comprehensive program in the public domain. One of the subsequent developments is the CERL program, which is presently being developed for the U.S. Air Force and is described in a subsequent entry in this Survey. Another is the proposed development of an extremely comprehensive and accurate public domain program for use by the States and by ERDA for use in creating and implementing energy conservation standards for buildings.

GARD PROGRAM FOR FACILITY/HVAC DESIGN AND ENERGY ANALYSIS (Code Name: SCOUT)

DESCRIPTION: The SCOUT program follows procedures outlined in the ASHRAE booklets "Procedures for Determining Heating and Cooling Loads for Energy Calculations" and "Procedures for Simulating the Performance of Components and Systems for Energy Calculations" to estimate the energy requirements for buildings. SCOUT is an extension of NASA's Energy/Cost Analysis Program (NECAP), but incorporates complete data verification, a full set of self-instructional input forms, 13 distribution system simulations, plus packaged systems and heat recovery devices, and complete internal restructuring resulting in reduced core requirements and reduced running times.

INPUT REQUREMENTS: Building construction and orientation parameters, exterior surface description and orientation data, interior floor, ceiling, wall and furnishing data, optional shadow data (requiring building coordinate system), space description data, thermostat schedules, space heat addition and extraction rates, optional user-specified air flows, type of energy distribution and conversion systems, and pertinent cost information.

PROGRAM OUTPUTS: Space and building peak loads broken out into components, optional surface and space response factors, optional design-day weather summary, optional shadow pictures and calculations, recommended space air flows and heat extraction and addition rates, equipment capacity summary, summary of space and system loads not met, monthly and annual energy summary by system and building, minimum and maximum space temperatures, and payback period comparison of alternatives.

PROGRAMMING LANGUAGE: FORTRAN IV

SYSTEM IMPLEMENTATION: CDC 6400 and 6600, IBM 360 and 370, Univac 1108 and Univac Spectra 70/46

AVAILABILITY: Services involving the use of SCOUT are available from GARD, Inc.

GARD, Inc.
7449 N. Natchez Avenue
Niles, Illinois 60648
Ph. (312) 647-9000
Attn. SCOUT Support Team

SCOUT will be available on nationwide computer networks in 1976.

TRANE AIR CONDITIONING ECONOMICS
(Code Name: TRACE)

DESCRIPTION: The program calculates peak and hourly zone loads based on coincident hourly climatic data for temperature, solar radiation, wind and humidity of typical days in the year representing seasonal variations. The average days are compiled based on the most recent ten years of U.S. Weather Bureau data. The design phase then receives input from the load phase as well as system type information and zone design information, and calculates supply air quantities and temperatures. The system simulation phase utilizes hourly zone loads and calculates return air quantities and temperatures and accounts for system loads (such as reheat loads). The equipment simulation phase takes the hourly output from the system simulation phase and calculates the annual energy consumption based on part-load performance data which is available on tape. The economic phase then does an economic comparison of the various design alternatives based on input consisting of energy consumption data, utility rate structures, and expected installation and maintenance costs.

INPUT REQUIREMENTS: Building, system, and equipment descriptions. Economic factors.

PROGRAM OUTPUTS: Peak building loads, building and equipment yearly and monthly energy consumption, economic comparison of life-cycle cost for up to four alternatives in one computer run.

MAINTAINED BY: Applications Engineering, The Trane Company, LaCrosse, Wisconsin.

AVAILABILITY: Program source code is proprietary but the program may be utilized through local Trane representatives or through timesharing systems. For more information, contact the local Trane representative or:

Applications Engineering
The Trane Company
3600 Pammel Creek Road
LaCrosse, Wisconsin 45601

ADDITIONAL COMMENTS: Load calculation procedures follow ASHRAE recommended procedures (1967). Although weather data is input for the whole year (8760 hours), the program calculates from this data twelve "typical weather days" (each with values for 24 hours), one for each calendar month. All subsequent monthly energy requirements are based on use of the same "typical weather day" profile for a whole month.

All simulations of air-side systems and mechanical equipment performance are of an equilibrium nature. Part load performance curves for equipment (mostly TRANE equipment) can be either input by the user, or by user option, some are available on computer files. A wide range of total system configurations can be simulated by utilizing appropriate individual simulations. Separate scheduling of different types of base and auxiliary loads allows for close reproduction of actual situations. Total energy plant configurations and thermal storage devices are not simulated, but are presently being developed.

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CONVERSION FACTOR TO METRIC (S.I.) UNITS

Physical Quantity	To Convert from	To	Multiply by
Energy	Btu	J	$1 \cdot 056 \times 10^{3}$
	kWh	J	$3 \cdot 60 \times 10^{6}$
	Wh	J	$3 \cdot 60 \times 10^{3}$
Heat flux rate	$Btu/ft^{2} \cdot h$	W/m^{2}	$3 \cdot 15$
Thermal conductivity	$Btu \cdot in/h \cdot ft^{2} \cdot °F$	$W/m \cdot K$	$1 \cdot 442 \times 10^{-1}$
Energy flux	Btu/ft^{2}	J/m^{2}	$1 \cdot 136 \times 10^{4}$
Thermal conductance	$Btu/h \cdot ft^{2} . °F$	$W/m^{2} \cdot K$	$5 \cdot 68$
Heat capacity	$Btu/lbm \cdot °F$	$J/kg \cdot K$	$4 \cdot 19$
Power	Btu/h	W	$2 \cdot 93 \times 10^{-1}$
	horsepower	W	$7 \cdot 46 \times 10^{2}$
Temperature	Fahrenheit	Celsius	$t_c = (t_f - 32)/1 \cdot 8$
Absolute temperature	Rankine	Kelvin	$0 \cdot 556$
Temperature Difference	Fahrenheit	ΔK	$0 \cdot 556$
Thermal Resistance	$°F \cdot h \cdot ft^{2}/Btu$	$K \cdot m^{2}/W$	$1 \cdot 761 \times 10^{-1}$
Distance	ft	m	$3 \cdot 05 \times 10^{-1}$
	in	m	$2 \cdot 54 \times 10^{-2}$
Volumetric flow rate	ft^{3}/min	m^{3}/s	$4 \cdot 72 \times 10^{-4}$
	gal/min	m^{3}/s	$6 \cdot 31 \times 10^{-5}$
Area	ft^{2}	m^{2}	$9 \cdot 29 \times 10^{-2}$
	in^{2}	m^{2}	$6 \cdot 45 \times 10^{-4}$
Velocity	ft/sec	m/s	$3 \cdot 05 \times 10^{-1}$
	mi/h	m/s	$4 \cdot 47 \times 10^{-1}$
Illumination level	footcandle	lm/m^{2}	$1 \cdot 076 \times 10^{1}$
Volume	gal	m^{3}	$3 \cdot 79 \times 10^{-3}$
	ft^{3}	m^{3}	$2 \cdot 83 \times 10^{-2}$
Pressure	in Hg	Pa	$3 \cdot 39 \times 10^{3}$
	in H_2O	Pa	$2 \cdot 49 \times 10^{2}$
	psi	Pa	$6 \cdot 89 \times 10^{3}$
Permeance	Perm	$kg/Pa \cdot s \cdot m^{2}$	$5 \cdot 75 \times 10^{-11}$
Force	lbf	N	$4 \cdot 45$
Mass	lbm	kg	$4 \cdot 54 \times 10^{-1}$
Density	lbm/ft^{3}	kg/m^{3}	$1 \cdot 602 \times 10^{1}$

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15. SUPPLEMENTARY NOTES

16. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)*

This report provides detailed technical material on various energy conservation actions for existing Air Force facilities and utility systems. It is specifically tailored to serve as a working document for Base engineers and technical personnel. The report covers energy conservation for Air Force facilities, including the equipment for providing hot water, space heating and cooling, lighting, and humidification. It also covers central plant systems and underground distribution systems (hot water, steam, and chilled water). It does not cover energy conservation measured for tactical or mission-related equipment such as ground vehicles or fighter aircraft.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*
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