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RESEARCH REPORT

METHODS FOR DISPOSING OF EXCESS SHELTER HEAT

Prepared for

OFFICE OF CIVIL DEFENSE
DEPARTMENT OF ARMY-OSA

Under
Contract No. OCD-OS-62-191
Subtask 1422A

by

John D. Hummell, David E. Bearint,
and Lawrence J. Flanigan

August 1964

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Battelle Memorial Institute
505 King Avenue
Columbus, Ohio 43201

Shelter Research 1400
Auxiliary Systems
Subtask 1422A
Unconventional Heat Sink Study

SUMMARY

OF

RESEARCH REPORT

METHODS FOR DISPOSING OF EXCESS SHELTER HEAT

August, 1964

OCD-OS-62-191
SUBTASK 1422A

Battelle Memorial Institute
Columbus, Ohio

Summary Prepared by
Battelle Memorial Institute
August, 1964

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Battelle Memorial Institute

August 31, 1964

Office of Civil Defense
Department of Defense
The Pentagon
Washington 25, D. C.

Attention Director of Research

Dear Sir:

Attached is our Research Report, "Methods for Disposing of Excess Shelter Heat", prepared under Contract No. OCD-OS-62-191, Subtask 1422A. Copies of this report are being distributed in accordance with the Shelter Research Program Standard Distribution List, including Attachment 1, which was sent to us by the Deputy Assistant Director for Research.

Under this contract the technical and economic aspects of conventional and novel cooling techniques utilizing natural and stored heat sinks have been studied. The conventional techniques considered include simple cooling with ventilation air or water and, where artificial cooling would be necessary, mechanical-vapor-compression and absorption machines. Concepts for novel cooling systems were evolved and evaluated for cooling shelters under circumstances which would preclude the use of conventional systems.

Our recommendations for further research include:

(1) Study of a shelter cooling system using methyl alcohol as the heat sink and as the fuel

(2) Development of an absorption refrigeration device specifically for shelter cooling

(3) Design and cost study of heat exchangers for air-cycle shelter cooling.

We have submitted proposals on these three subjects to the Civil Defense Technical Office at the Stanford Research Institute.

We believe this has been an extremely significant project in the research program of the Office of Civil Defense and we urge that the research recommended as a result of this study be undertaken.

Sincerely,

James A. Eibling

James A. Eibling
Group Director
Thermal Systems Research

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SUMMARY

of

RESEARCH REPORT

on

METHODS FOR DISPOSING OF
EXCESS SHELTER HEAT

SCOPE

The purpose of this study was to conceive and to evaluate cooling methods which could be used to dispose of excess heat in fallout, blast, and CW/BW shelters throughout the United States. A major effort was directed toward novel systems able to function under the unique shelter situations for which blast shelters are designed. Novel and conventional cooling methods were compared on the basis of cost effectiveness for the various types of shelters.

The guidelines for the study as suggested by the Office of Civil Defense were:

1. 14-day period of shelter occupancy
2. 24-hour sealed period for blast shelters
3. Long standby period
4. 85 F effective-temperature limit.

OBJECTIVES

The objectives of the study were to:

1. Define heat sinks feasible for shelter cooling
2. Devise cooling methods to utilize the feasible heat sinks
3. Evaluate cooling methods with respect to the technical, economic, and safety factors applicable to various types of shelters.

APPROACH

The technical and economic factors associated with the use of natural and artificial (stored) heat sinks for cooling fallout, blast, and CW/BW shelters were examined. For those heat sinks that appeared to be feasible, cooling systems were devised that would operate under each set of conditions to which shelters might be subjected. Special emphasis was placed on the study of novel or unconventional systems which could function under conditions not tolerable for conventional cooling methods. Such conditions would prevail in the absence of one or more of the following: atmospheric air, natural water, commercial power.

The cooling systems were evaluated on the basis of estimated cost, availability of equipment, maintenance required during standby, reliability, safety, and amount of development required.

FINDINGS

Appropriate heat sinks and cooling systems were found for every combination of circumstances anticipated for fallout, blast, and CW/BW shelters. Three novel cooling systems were conceived for application when conventional cooling systems could not function. The estimated costs of these novel cooling systems are not considered excessive in view of the necessity of shelter cooling.

Heat Sinks

The heat sinks best suited for shelter cooling are:

1. Atmospheric air
2. Natural water
3. Stored water
4. Methyl alcohol
5. Ice
6. Ammonia.

The earth could rarely be considered an economical heat sink.

Techniques for using atmospheric air and water from natural sources as heat sinks are well developed and, when applicable, these heat sinks are the most appropriate for cooling a shelter. However, they cannot always be relied upon under exposure to blast or fire resulting from a nuclear detonation. Water stored in blast-resistant reservoirs can be used economically in conjunction with a high-temperature evaporative condenser for heat rejection from a refrigeration device.

Methyl alcohol, which is relatively inexpensive, could serve as the heat sink for a sealed shelter providing power was available or could be developed on-site to drive an open-cycle vapor-compression machine. The chemically stable alcohol could also be used as engine fuel and thereby eliminate the necessity for petroleum engine fuels which deteriorate in storage.

Ice and ammonia are the heat sink materials usable when a shelter is sealed from the atmosphere and no power is available. Ice is the more acceptable, and it may be suitable for periods other than the sealed period because of the high reliability and safety of its use. In addition, it has the advantage of providing a source of potable water. During standby, the ice would be maintained in a well-insulated space which would be cooled by a relatively small refrigeration machine driven by commercial power.

Liquid ammonia stored in pressure tanks would be a less costly heat sink than ice for blast-resistant installations. The ammonia could be throttled into direct-expansion coils, the only power required being the power to move the shelter air across the coils. Storage of pressurized ammonia within a shelter with the possibility of leakage of toxic vapors presents a serious potential hazard.

Cooling Systems

The appropriateness of any particular cooling system depends upon the type of shelter being considered, the availability of atmospheric air and natural water, the power supply, and the cost. Three novel systems are described below: (1) a refrigerant-vapor-engine system, (2) an open-cycle sulfuric acid absorption system, and (3) an open-cycle mechanical-vapor-compression system. In addition, the possible application of air-cycle cooling systems is described and conventional cooling systems are discussed briefly.

Refrigerant-Vapor-Engine System

A thermodynamic cycle analysis showed that a refrigerant-vapor engine could develop the power required to drive the mechanical equipment for cooling a shelter using ice as the heat sink. Heat released by the shelter occupants would be absorbed in a refrigerant-cooled vaporizer; the high-pressure vapors would expand through an engine and would condense in a heat exchanger at the sink. A refrigerant-liquid pump would be used to transfer the fluid from the low-pressure condenser to the high-pressure vaporizer. The pump and air-moving blowers would be driven by the expansion engine. Because this system is completely self-driven and uses a stored heat sink, it is applicable to a sealed shelter without access to power or natural water.

Open-Cycle Sulfuric Acid Absorption System

An open-cycle absorption system employing sulfuric acid and water as expendable working fluids can be used economically for cooling shelters. The system would resemble a conventional absorption system without the generator. Little energy would be required to operate the pumps and blowers which could possibly be driven manually during the sealed period. The heat sink could be a natural source of water even though the water was warm. Possibly stored water which would be boiled at atmospheric pressure could be used. Research is needed to define methods for limiting equipment corrosion and purging the system of gases.

Open-Cycle Mechanical-Vapor-Compression System

An open-cycle mechanical-vapor-compression system using methyl alcohol as the refrigerant and as the heat sink would be a satisfactory and economical means of cooling a sealed shelter provided combustion air for an auxiliary power system were available. The cooling system would operate at less than atmospheric pressure and, therefore, the toxic methyl alcohol vapor would not leak into the shelter. The toxic vapor could be vented to the atmosphere, burned in a simple combustor, or used as fuel for an engine. Except for the compressor, the system could be assembled from commercially available components.

Air-Cycle Cooling Systems

If economical air-cycle cooling systems could be developed, they would have substantial advantages in that air is used as the working fluid and standby maintenance costs should be comparatively low. Disadvantages would be the large power requirement and the need for an ample supply of water. At present, equipment is commercially available, but it is expensive because it is not designed specifically for air-cycle cooling applications. Techniques would have to be developed for making inexpensive components designed specifically for such equipment. Centrifugal compressors and expanders appear to hold the most promise. Further study is needed to define the design of the water-cooled heat exchanger for the air leaving the compressor.

Conventional Cooling Systems

In many circumstances shelters can be cooled adequately with ventilating air and/or naturally available cool water in heat exchangers. The reliable equipment is available for such systems but their potential use is limited by the availability and temperature of the air and water and availability of a reliable power source.

Mechanical-vapor-compression and absorption-refrigeration machines have, of course, been proven in service. Their application is limited, however, by the availability of power and of an appropriate heat sink. Standby maintenance requirements for high reliability are not well defined but they could be minor in view of commercial experience with refrigeration devices which are used seasonally.

RECOMMENDATIONS

The development of the following novel cooling system is recommended:

1. Open-cycle sulfuric acid absorption system
2. Open-cycle mechanical-vapor-compression system using methyl alcohol

3. Heat-exchanger design for the air-cycle system.

In addition, for systems relying upon an ice heat sink, the refrigerant-vapor engine should be compared with other sources of power which also can function during the sealed period.

Battelle Memorial Institute · COLUMBUS LABORATORIES

August 31, 1964

Office of Civil Defense
Department of Defense
The Pentagon
Washington 25, D. C.

Attention Director of Research

Dear Sir:

Attached is our Research Report, "Methods for Disposing of Excess Shelter Heat", prepared under Contract No. OCD-OS-62-191, Subtask 1422A. Copies of this report are being distributed in accordance with the Shelter Research Program Standard Distribution List, including Attachment 1, which was sent to us by the Deputy Assistant Director for Research.

Under this contract the technical and economic aspects of conventional and novel cooling techniques utilizing natural and stored heat sinks have been studied. The conventional techniques considered include simple cooling with ventilation air or water and, where artificial cooling would be necessary, mechanical-vapor-compression and absorption machines. Concepts for novel cooling systems were evolved and evaluated for cooling shelters under circumstances which would preclude the use of conventional systems.

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Sincerely,

James A. Eibling

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Group Director
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FOREWORD

The research program described in this report was initiated and supported by the Office of Civil Defense as part of their effort to develop concepts for protective shelters with maximum cost effectiveness. The specific objectives of the program were to devise novel methods to dispose of excess shelter heat and to compare novel and conventional methods on the basis of cost effectiveness. Heat disposal methods were to be considered with respect to their applicability in shelters located within the United States.

The information developed in this study is intended for use in the preparation of manuals on shelter design. Therefore, the report is written primarily for technically trained users who have general knowledge of cooling system operation.

The study was conducted at Battelle Memorial Institute by personnel in the Thermal Systems Group of the Mechanical Engineering Department under the direction of Mr. James A. Eibling. Mr. Frank C. Allen of the Office of Civil Defense Research Directorate staff monitored the work and contributed to the over-all planning of the research. His assistance and guidance is greatly appreciated. The authors also wish to acknowledge the contributions of other Battelle staff members, particularly, Mr. Richard E. Barrett, Mr. Ralph I. Mitchell, and Mr. Walter E. Chapin.

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METHODS FOR DISPOSING OF EXCESS SHELTER HEAT

by

John D. Hummell, David E. Bearint,
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INTRODUCTION

Because protective shelters of interest to the Office of Civil Defense must be designed for high occupant density, the task of removing excess heat becomes a major problem. Heat liberated by occupants and by mechanical, electrical, and air revitalization equipment in many instances will be greater than that flowing outward through the shelter walls or removed by the ventilating air. Consequently, some means of disposing of excess heat will be required to maintain a habitable environment within the shelter.

Techniques appropriate for the removal of heat from most shelters will probably differ somewhat from conventional methods for cooling occupied structures. Careful consideration must be given to such factors as: availability of heat sinks, safety requirements, reliability, first cost of equipment, requirements for maintenance during stand-by periods, power requirements, the relatively short operating period, and comfort conditions within the shelter. Some of these factors are unique in protective shelter design and, therefore, cooling methods not conventionally used for comfort cooling were investigated during this program.

Due to differences throughout the United States in weather, soil conditions, and natural heat sinks, a broad and thorough analysis of heat disposal methods and evaluation of design parameters was required. Architects and engineers, having suitable data concerning various possible cooling methods, can select the cooling equipment best suited for local conditions.

To define the limits of this research program and to assure integration of the results into the over-all OCD program, the following guidelines were established by OCD:

1. Mechanical system concepts are to be developed for fallout shelters that can be readily adapted for protection against other effects, including fire, blast, biological contaminants and chemical agents.

2. A 2-week occupancy period is to be considered for planning purposes

3. At some time during the 2-week occupancy period, shelters might be completely isolated from the outside atmosphere for 24 hours

4. The effective temperature in the occupied portion of the shelters should not exceed 85 F

5. Shelters might have to be maintained on a standby basis for many years

6. Shelters might be located anywhere in the United States; therefore, design data must be broadly applicable

7. All methods of heat disposal investigated should be thoroughly described in the report of the study to evaluate the advantages and disadvantages of both applicable and unusable methods

8. Heat transfer through shelter walls should be excluded from consideration. This is the subject of separate studies by other OCD contractors.

SUMMARY

During Battelle's study for the OCD of methods of disposing of excess shelter heat, technical and economic data were developed in three categories: heat sinks, miscellaneous cooling-system components, and refrigeration and dehumidification devices. Various combinations were considered as assembled into cooling systems and these systems were then evaluated with respect to their cost effectiveness in removing excess shelter heat. Special attention was given to the conception of applicable new cooling systems. In the report which follows, a comparison is also made of the advantages, disadvantages, and estimated costs associated with various conventional and novel systems. Areas of needed research are defined for the development of novel systems.

The main body of this report, entitled "Results of Technical and Economic Studies", consists of five major sections:

1. Nuclear Radiation Effects
2. Standby Maintenance Requirements
3. Heat Sinks
4. Cooling System Components
5. Refrigeration and Dehumidification Devices.

Nuclear radiation effects are considered inconsequential because most cooling systems will be protected and because radiation levels will be below that at which equipment is seriously damaged. With respect to standby maintenance, not much information is available concerning experience with long standby periods. Moreover, it is evident that standby maintenance procedures will depend upon the particular equipment selected; therefore, only general statements can be made.

The section covering heat sinks contains thermodynamic data, physical properties, and estimated costs for a variety of materials which were considered. For heat sink materials which appear applicable, storage facilities are described. Calculated heat-transfer rates for cooling ponds, earth coils, and atmospheric condensers are also included.

The last two sections contain detail covering the operation, reliability, and economics of a wide variety of equipment. Unconventional as well as conventional devices are described and, in addition, performance data are given for conventional equipment which could be used advantageously under off-design conditions.

Several unusual approaches to shelter cooling problems which were conceived during this program are discussed in detail. A novel engine which could drive all components of a cooling system and which would have no oxygen requirement is described. Its motivating energy would be the heat rejected by shelter occupants; the heat sink would be ice. Because the engine requires no oxygen it might be particularly attractive for use during the time when it is necessary to seal the shelter from the outside atmosphere.

An unusual open-cycle absorption-refrigeration device which requires little mechanical power and uses water as a heat sink is also described. Sulfuric acid and water are the working fluids. Another suggestion is a cascade arrangement of mechanical-vapor-compression devices. Such an assembly would make possible heat rejection at boiling water temperature and consequently would greatly reduce the need for water storage. Possibilities for a suitable air-cycle cooling system are also examined.

In the final section of the report, entitled "Cooling Systems", is a table showing applicable cooling systems and their costs as a function of the availability of atmospheric air and natural water. This table should be helpful for selection of optimum components to meet the requirements of particular shelters.

The immediately following section, "Why Shelter Cooling is Unique", contains a discussion of the significant differences between ordinary comfort cooling and the cooling of protective shelters.

WHY SHELTER COOLING IS UNIQUE

Considering that air conditioning has become so commonplace, it may seem strange that the cooling of a protective shelter would pose special problems. However, the fact that such a good job has been done, and done repeatedly, in commercial buildings and residences may actually interfere with the proper design of shelter cooling systems. That this was true became evident during discussions with engineers experienced in everyday comfort cooling. After the requirements of shelter cooling had been fully explained, these engineers would suggest the use of commercial systems or equipment which analysis by the project staff had revealed would not be applicable for shelter cooling. These experiences served to emphasize to the staff that shelter designers should approach their task with ingenuity and open minds and try not to rely entirely on past practices.

This section of the report covers some of the factors which make shelter cooling different from ordinary comfort cooling. Any shelter design manual or any recommendations concerning shelter cooling systems should include such background information to help designers develop an appreciation of the unusual requirements of shelter cooling. In introducing this discussion, a brief review of the basic functions of cooling systems is given. Although this information will seem elementary to many readers, it may suggest a new way to view a familiar yet complex subject. Following this is a discussion of the technical and economic aspects of shelter cooling.

FUNCTIONS OF COOLING SYSTEMS

Removing excess heat from an inhabited area requires that air having the necessary temperature and humidity for absorbing heat and moisture released by people and equipment must be provided. This air could come from a natural source such as the atmosphere, or be air conditioned by a cooling system and recirculated within the shelter space.

In discussing the cooling and dehumidifying of air it is convenient to consider that heat and moisture are tangible substances and that they must be picked up, transported, and rejected from the occupied area by the cooling system. The fact that heat is a form of energy and that moisture can exist in either the vapor or liquid state does not change the cooling system functions.

Cooling systems which perform these functions are governed by the following thermodynamic laws:

1. Heat flows naturally only from a higher temperature to a lower temperature

2. Energy must be expended to remove heat from a lower temperature source and reject it to a higher temperature sink

3. Heat removed from a source must be received by a sink

4. Heat may be transferred by energy flow or by mass flow.

In many cases, a practical heat sink having a temperature lower than the temperature required in the shelter occupied space may not be available. Therefore, technically feasible cooling methods can be conveniently divided into two groups:

1. Those depending on the natural flow of heat to a cool sink

2. Those requiring a refrigeration device to pump heat into a warm sink.

Figure 1 shows the two possible heat-flow paths from people and equipment through various intermediate fluids and heat-transfer processes to the heat sink. Temperature is shown along the ordinate to indicate the temperature level of the various stages of the cooling processes.

Heat released by people and equipment is absorbed by the air through conduction and convection. The warm air moves by mass transfer to a cooler heat exchanger which accepts heat from the air. If a cool sink is available, the heat can be transferred to the sink again by a mass transfer of a heat-exchanger fluid and the natural flow of heat from a higher temperature to a lower one. An example of this is a cooling system utilizing cold well water circulated through a heat exchanger.

If the sink temperature is above the temperature of the occupied area, a refrigeration device must be used to pump the heat to the sink, this requires a power input. The refrigeration device, through the mass transfer of its working fluid, removes heat from the cooler heat exchanger placed in the occupied area and delivers it to the higher temperature heat exchanger located at the heat sink. The sink heat exchanger must, of course, operate at a temperature above the sink to provide the temperature differential required for adequate heat transfer. This mode of operation is basic to all refrigeration cooling systems. The various refrigeration systems differ, however, in the number of intermediate working fluids needed in a particular cooling situation.

TECHNICAL FACTORS

The major technical differences between shelter cooling and conventional comfort cooling are associated with:

1. Availability, temperature level, and thermal properties of heat sinks

2. Temperature and humidity limits

3. Heat sources.

Sink Characteristics

The heat sink is the key factor in selecting the cooling method which is best for a particular shelter. Providing a heat sink for ordinary comfort cooling is no problem because atmospheric air can always be used, and in many instances adequate water sources are available. Both of these are at relatively low temperature. These sinks

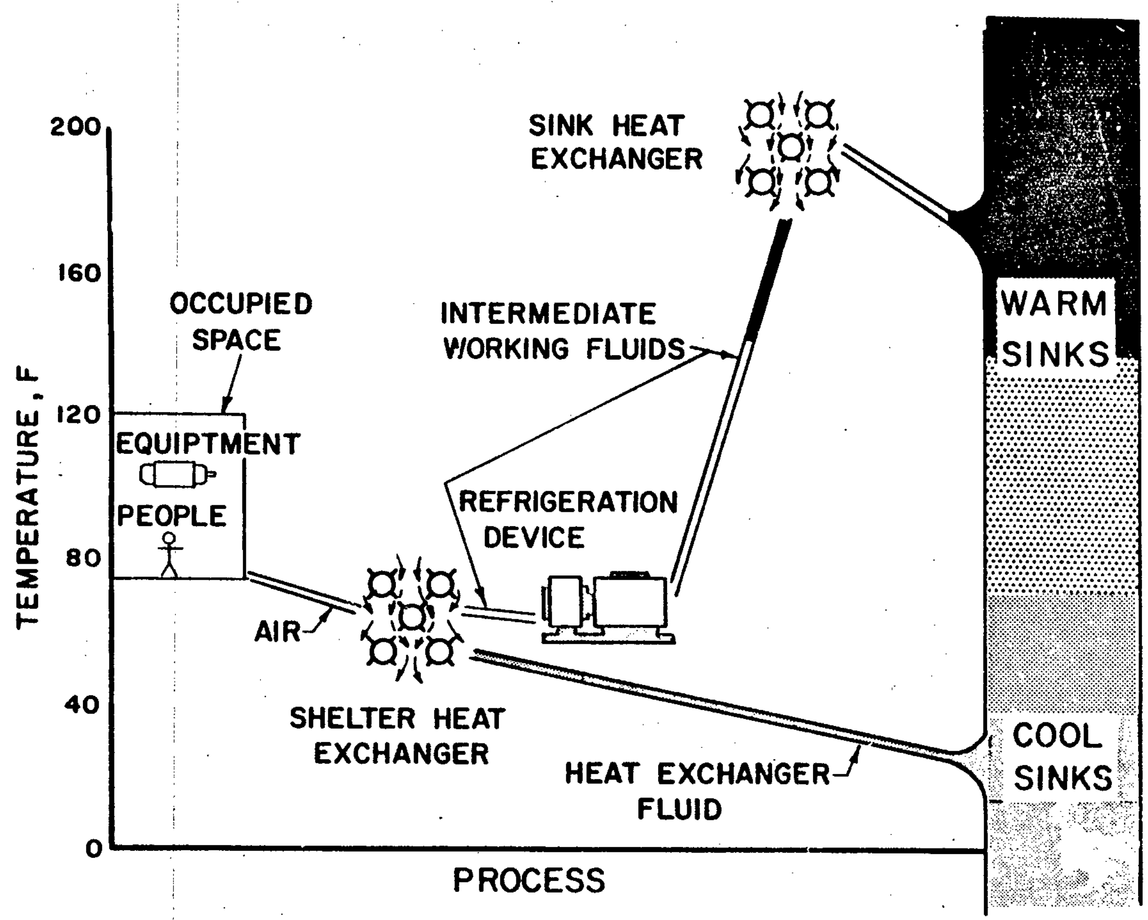

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FIGURE 1. TWO PATHS FOR HEAT FLOW TO THE HEAT SINK

could also be used for shelter systems not required to be blast or fire resistant. However, shelters designed for blast and fire protection must have an invulnerable heat sink and heat-rejection equipment. This makes the use of atmospheric air more difficult and may preclude use of city water systems which are dependent upon vulnerable power.

The sink temperature and the thermal properties of the sink materials define practical and economic methods that can be used for cooling shelters. The sink temperature alone may determine whether a refrigeration machine is required. However, other thermal properties of a sink frequently govern its applicability. For example, although the temperature of the earth is lower than that required in a shelter, the heat conductivity and heat capacity of the earth are so low that large masses are needed for appreciable heat flow.

Air Temperature and Humidity

Many inconveniences which would not be tolerated under normal circumstances will be accepted by shelter occupants. Comfort standards will be appreciably lower than otherwise. Therefore, the thermodynamic processes for shelter cooling can be operated at temperatures higher than those used for conventional comfort and humidity control cooling. Because the entire temperature level of the cooling system can be increased, either the temperature of the sink could be made higher or the temperature differential between the heat sink and the shelter could be made larger to increase the rate of heat transfer. These considerations suggest a search for cooling methods different from those used for conventional comfort cooling.

Heat Sources

Heat loads imposed on shelter cooling systems would be quite different from those imposed on conventional systems. For commercial systems used to cool buildings, the load is largely sensible heat representing heat gained through the structure walls from the outside, heat generated by internal lights and equipment, and heat entering with the ventilating air. The smaller latent load is due primarily to the moisture in the ventilating air. The load contributed by occupants is relatively small except in auditoriums and similar buildings where large numbers of people congregate.

In contrast, the primary source of heat in a shelter would be the shelter occupants, with the ratio of latent heat and sensible heat dependent upon the temperature and humidity in the shelter. Consequently, the ventilating air would leave the shelter with a higher heat content than in conventional situations, the heat-transfer capacity of cooling coils would be increased, and greater quantities of condensate would have to be drained from the coils. This condensate, incidentally, could be a source of potable or domestic water.

Heat released by occupants is of particular concern if manual power is used to drive a cooling system. A human power source, as is well known, has low output. More significant, insofar as this application is concerned, is the low thermal efficiency of the human power source. There is the possibility that the heat released by a man driving cooling system equipment would be more than the cooling effect produced by the system that he was driving.

Figure 2 shows the coefficient of performance (COP)* of a manually operated cooling system needed to remove the heat released by all of the shelter occupants as a function of human power output, human efficiency, and the percentage of occupants working. Man's thermal efficiency ranges between 10 and 20 per cent, with 15 per cent

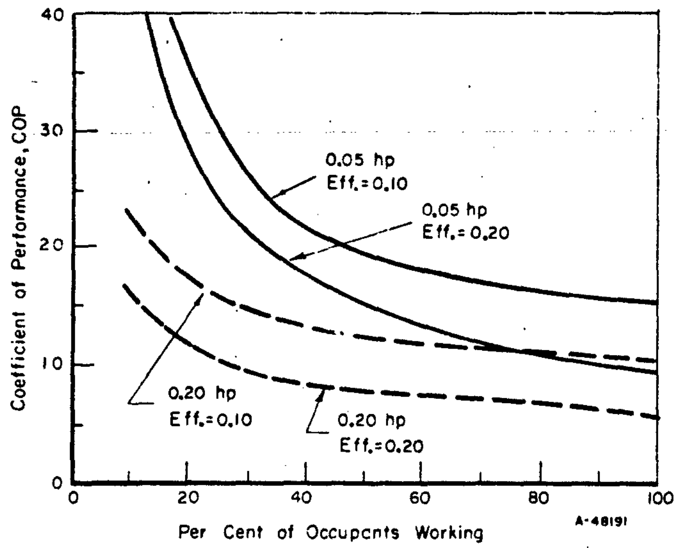


FIGURE 2. REQUIRED COP FOR COOLING WITH MANUAL POWER

being a good average.[1] Of course, the exact power output of any particular shelter occupant is unknown. Moreover, even the power output of a healthy man, working under variable temperature conditions, varies greatly with the length of time the output must be sustained. Also of significance is the working method, i.e., body position and parts of body used. Results of experiments, primarily in the field of manual-powered flight, showed:[2-11]

Average man, 20-minute effort

Arms only, 0.15 hp

Legs only, 0.36 hp

Arms and legs, 0.53 hp

*COP or coefficient of performance is the ratio of the cooling effect to the energy input.

Trained cyclist, 20-minute effort

Legs only, 0.58 hp.

From this it seems reasonable to assume that the power output of the "average shelter occupant" would not exceed 0.2 hp and considering that there may be a preponderance of women, children, aged, etc., occupants in some shelters, the average might be as low as 0.05 hp. This range of output could be developed for an extended period of time only under favorable temperature and humidity conditions. The maximum human power output which can be achieved in still air with no body temperature build up is:(1)

Effective Temperature, F	Power, hp
87	0
76	0.008
71	0.017
65	0.033

Because some build up of body temperature could be tolerated and because there would be some air movement in a shelter, these power outputs are somewhat low. However, they do illustrate the effect of ambient temperature on manual power output.

Not to be overlooked is the oxygen required by a working man. In a shelter, during a sealed period oxygen from a stored supply would have to be used to replenish that required not only to sustain life but also to provide the energy for power. It may well be that the oxygen requirement for an efficient internal combustion engine would be less than that required for manual power. Semi-closed cycle internal combustion engines, using their exhaust gases as diluents, could run on a supply of pure oxygen and fuel. Such an engine would eliminate the problems associated with furnishing a cool area for workers and combining the power outputs of a large number of men.

The location of equipment in a shelter complex will greatly affect its contribution to the load on the cooling system. This is an item which seldom needs consideration in commercial cooling installations. Commercial units are generally placed outside the space to be cooled. The heat given off by the equipment is rejected to atmospheric air which circulated through the separate enclosure of the cooling system. However, in a shelter the cooling equipment will probably be inside the shelter complex and the heat it generates must be included in the total heat load which must be removed by the cooling system, by the ventilating air, and possibly through the shelter walls. Therefore, the net cooling effect in the shelter is the difference between the capacity of the equipment and the amount of heat released by the equipment.

Cooling equipment can operate in higher ambient temperatures than can be tolerated by people. Therefore, the cooling load would be a minimum if all possible equipment were housed in a space which could be ventilated and cooled with the air leaving the occupied space. The shelter designer must consider the aspects of heat rejection for the total shelter system, including the auxiliary power system, and then provide for maximum utilization of resources available for cooling. This will require considerations which are not associated with conventional comfort cooling but which are of primary importance for shelter cooling.

Figure 3 shows the ratio of the total heat to be rejected to a sink, other than prime-mover exhaust heat, divided by the heat removed from the shelter as related to the COP of the cooling system, the type of prime mover, and the efficiency of power transmission to the cooling equipment. These curves are not intended to show that cooling the reciprocating engine is more of a problem than cooling a gas turbine. They are intended to stress the fact that the shelter designer must be alert to the many ramifications of the total heat-removal problem.

ECONOMIC FACTORS

Although it is difficult to analyze the various technical problems involved in shelter cooling, analysis of the economic aspects is even more difficult. The significance and nature of the cost of cooling a shelter are entirely different from those associated with a conventional cooling system. A large part of the cost in conventional cooling is in the operating expense; capital costs can be amortized over a period of many years. For a shelter cooling system, in one sense, there is no operating cost, as such, because all equipment, fuel, power plant, etc., must be in place and ready to operate. After the equipment is used, it seems likely that it would have little, if any, future value. Therefore, it seems logical to assume that the total shelter cooling system cost can be divided into two categories:

1. Capital investment or first cost

2. Standby maintenance expense.

Capital Investment

The capital investment or the first cost must include all the equipment and materials needed to operate the shelter throughout its period of occupancy. The necessary equipment, supplies, and energy to drive the equipment must be available at all times. Therefore, there would be no operating cost. An exception might be in the unusual case of a shelter designed to rely on a commercial power supply. Amortization and salvage of equipment seems to have little meaning in the evaluation of the economic factors.

The relatively short period anticipated for shelter use greatly influences the comparative costs of the various possible cooling systems. Total system cost must be considered as the sum of the cost of mechanical machinery, air-handling equipment, auxiliary power supply, and perhaps an artificial heat sink. The most economical combination would be the one having the lowest first cost. For example, it might be advisable to use a relatively expensive heat sink, such as ice, because it would require only low cost heat exchangers and power requirements would also be very low. If the design is for a large cooling capacity over a short period of time, such a system would be less costly than a large refrigeration machine with its prime mover. However, as the operating time increases, the economics swing in favor of refrigerating machinery.

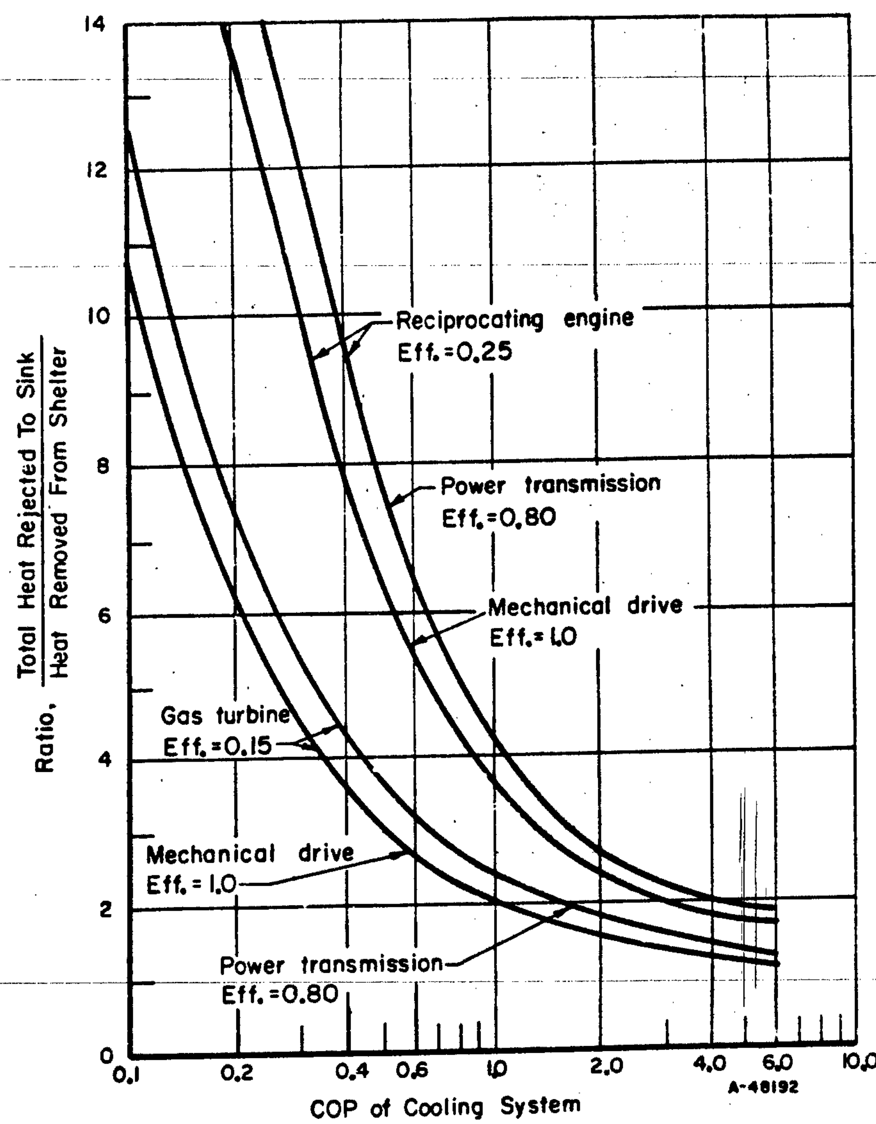


FIGURE 3. RATIO OF HEAT REJECTED TO SINK TO HEAT REMOVED FROM SHELTER FOR VARIOUS DRIVING METHODS

Standby Maintenance Expense

There would be little, if any, resemblance between the maintenance of conventional cooling equipment and shelter cooling equipment. With conventional equipment, maintenance is primarily the repair or the replacement of components because extended operation has caused them to wear, decompose, or become contaminated. Few working components fail because of deterioration with time, and usually these few can be identified and replaced before breakdown. Potential costs can be established on the basis of years of experience with similar types of equipment.

Shelter cooling equipment certainly would not "wear out" in the anticipated two weeks of operation. Rather, the maintenance would be only that required to keep the equipment in reliable condition for immediate operation upon demand. The reliability of cooling systems would be enhanced by frequent test operation and inspection, but these would be expensive during long standby periods. Some compromises must be made; it will be necessary to sacrifice some acceptable degree of reliability for the sake of economy.

Speculation about maintenance costs for standby equipment is dangerous because no specific maintenance tasks can be defined. Many maintenance problems can be identified only after practical experience with specific equipment. It is not possible to give realistic estimates of maintenance costs before the installations of typical cooling systems in shelters and before reliability studies have been made.

CONCLUSIONS

Although numerous cooling methods have been identified as being technically satisfactory for removing heat from shelters, only a few basic ones are truly applicable. Of those methods which are applicable, some are novel, having been conceived during this study, while others incorporate conventional equipment operating under off-design conditions. Still other methods are essentially those in use for commercial comfort cooling. Appropriate cooling systems could be assembled from various types of equipment and power sources. Of primary significance in any particular situation is the availability of air and/or water.

Each combination of circumstances determines the suitability of various possible cooling systems. Therefore, the numerous systems cannot be placed in any logical order with respect to the over-all merits. However, some conclusions can be stated with the reservation that further details must be examined before any final evaluation. The conclusions are discussed here in the same sequence as the details are presented in the report: (1) radiation effects, (2) standby maintenance, (3) heat sinks, (4) cooling system components, and (5) refrigeration and dehumidification devices.

RADIATION EFFECTS

On the basis of information available at this time, it seems that the radiation associated with a nuclear detonation would not exceed 2×10^5 R per hr and this would not damage a well-designed shelter cooling system. The radiation levels which would occur outside the shelter would be orders of magnitude less than the dosage required to significantly change structural materials, electrical insulation, lubricants, refrigerants, and refrigerant seals. Fluids flowing into the shelter from exposed cooling coils would not become radioactive and, therefore, would present no radiation hazard to shelter occupants. While no problems are anticipated with irradiated water, some effects might occur for which there are no helpful data. Perhaps the most significant effect would result if cooling spray water were radiated in the presence of nitrogen. Gaseous nitric acid would form, some of which would be absorbed in the water. However, it is believed that the quantities would be so small that corrosion of any wetted surfaces would be negligible. Irradiated water in a closed system might contain hydrogen and hydroxyl radicals, as well as free hydrogen, oxygen, and hydrogen peroxide. Some of these would promote corrosion, but this should not be serious for a two-week period of operation.

STANDBY MAINTENANCE

Little experience has been reported with respect to cooling equipment reliability after long periods of nonuse. Similarly, data of standby maintenance requirements and on maintenance costs are few. However, some types of air-conditioning equipment in commercial service are used seasonally and remain idle from six months to nearly a year. Cooling equipment for shelters could be expected to be operable after standby periods at least as long. Undoubtedly, reliability of operation would be increased with frequent inspection of equipment and controls, and periodic operation of the more essential components. Unfortunately, no information is available on which to base recommendations for definite maintenance programs. Techniques are well known for controlling corrosion by maintaining a low relative humidity and/or by filling liquid containers with corrosion inhibitors compatible with construction materials. The maintenance of cooling equipment would have to be incorporated into the maintenance program for the entire shelter complex. This would be true especially if the cooling equipment imposed the major portion of the load on the auxiliary power system which may or may not be operated periodically at full power.

HEAT SINKS

The role of the heat sink in cooling the shelter cannot be overemphasized. The sink temperature defines cooling methods which are technically feasible while its cost and thermodynamic properties generally establish the equipment and the power requirements of a cooling system. For these reasons cooling methods can be logically evaluated by considering, first, the factors regarding the heat sink and, second, the mechanical equipment necessary to utilize the sink under conditions which would prevail in a shelter.

Air and water from natural sources and artificially stored water and ice are the only heat sinks which are generally applicable for shelter cooling. Under special circumstances, ammonia, methyl alcohol, and earth should be considered. Techniques for using atmospheric air and natural sources of water as heat sinks are well developed and, when applicable, these are the most appropriate for cooling a shelter. However, these resources cannot always be relied upon under such conditions as blast and fire which are characteristic of a nuclear detonation. Stored water could be used in conjunction with a cooling tower or evaporative condenser for rejection of heat from a refrigeration device. A tower or evaporative condenser operating above design temperature and with an air flow equal to 3 cfm per occupant could dissipate the entire heat load generated by shelter occupants. This suggests a cooling system, completely housed within the shelter, which could reject the excess heat to evaporating water with a mass transfer of the vapor to the atmosphere.

Another potentially interesting use of water, whether stored or natural, is for evaporative cooling of ventilating air. The heat absorption capacity of evaporative cooled air is nearly two times that of atmospheric air. Therefore, with simple equipment and a relatively small quantity of water, ventilation rates necessary for cooling could be reduced by a factor of two. Evaporative cooling may be economically justified for shelters requiring blast valves and high-cost chemical, biological, and radiological filters.

Ice and ammonia are the only artificial heat sinks applicable for use during the period when the shelter is sealed from the atmosphere. Ice is the more acceptable and it may be practical for other than the sealed period because of the high reliability and safety of its use, in addition to its advantage as a source of potable water. During stand-by, storage of ice in a well-insulated space would require refrigeration by a conventional machine driven by commercial electric power. During the sealed period, manual power could be used to operate the pumps and blowers required to reject heat to the ice.

Liquid ammonia stored in pressure tanks would be less costly as a heat sink than ice stored in chambers constructed for high overpressures. The ammonia could be released through valves into direct expansion coils which could be used in conjunction with a blower to cool the shelter air. The power requirements would be even less than that for cooling with ice, but the presence of high-pressure ammonia lines within a shelter and the release of toxic vapors into the atmosphere are potential hazards. The risks could be reduced by use of intermediate cooling fluids and provision for the absorption of the ammonia vapor into a large supply of water which could be too warm for direct cooling of the shelter.

Methyl alcohol, which is relatively low in cost, could be used as a heat sink for a sealed shelter providing power were available by using it in an open-cycle mechanical vapor compression device. For this application, the alcohol would serve not only as the heat sink but also as the working fluid.

Methyl alcohol could also be used as a heat sink by using it in a cooling tower. If alcohol, which vaporizes readily in atmospheric air, were sprayed into a cooling tower on a summer day, the liquid portion leaving the tower would be cool enough to absorb heat from a shelter. Using this method, heat could be rejected to the warm atmosphere without a refrigeration device. The efficiency of alcohol cooling would be significantly affected by the quantity of water vapor in the air. The vapors of methyl alcohol are toxic and the mixture of vapor and air in a tower would be combustible. Because the vapors are combustible, the tower would have to be protected from any ignition source or the vapors would have to be destroyed by controlled burning.

The earth could serve as a heat sink, but the buried piping required would be too costly except where the earth is cool and where it would remain moist throughout the period of heat absorption. The last condition is unlikely except in swamp or marsh areas. Heat from the buried pipe would dry the earth around the pipe; consequently, the thermal conductivity would decrease. The heat flow would be further decreased because the earth would shrink away from the pipe.

COOLING SYSTEMS

Any shelter cooling system will be a combination of many components. Proper selection of compatible and economic units will be the major challenge to be faced by shelter designers. In the final section of this report both conventional and unconventional assemblies of possible cooling system components are tabulated and compared. However, final evaluation and selection must be in terms of the particular requirements of individual shelters.

The five unusual possibilities for cooling system equipment described below were considered during this research program. The first four seem to have considerable promise; the last seems less promising at the moment, but it may prove to have interesting merit after some development work has been done:

1. A refrigerant-vapor engine for driving blowers and pumps

2. An open-cycle sulfuric-absorption device

3. Cascade arrangement of mechanical-vapor-compression devices

4. Open-cycle mechanical vapor compression device using methyl alcohol

5. Air-cycle cooling equipment.

Refrigerant-Vapor Engine

A refrigerant-vapor engine requiring no oxygen and motivated by the heat released by the shelter occupants could be used to drive blowers and pumps if a sink were available with a temperature below 35 F, such as a stored ice sink. Heat released by the occupants would vaporize a refrigerant in a heat exchanger; the vapors at a higher pressure would be expanded through the engine and would condense in a heat exchanger at about 40 F. A refrigerant-liquid pump would transfer the fluid from the low-pressure condenser to the high-pressure vapor generator. Such an engine would be especially attractive during a sealed shelter period. Since appropriate engines of this type are not commercially available, a development program would be required. However, no major problems are foreseen. One type of this engine, the rotary-vane expansion engine, has been developed for special military applications. Suitable refrigerants and lubricants are commercially available. Condensation, which is ordinarily a problem in high-temperature vapor engines, would not exist.

Open-Cycle Sulfuric-Acid Absorption System

An open-cycle sulfuric-acid absorption system would have the advantage of low power requirement; therefore, it could be used during a shelter closed period, provided an adequate water supply was available. The water supply could be at a temperature above that of water used for direct cooling. The only mechanical power required would be for pumping the fluids and driving the blower on a heat exchanger. With this system no refrigeration for a heat sink would be required during standby because the sulfuric acid and water could be stored at earth temperature. The system would be less hazardous than a liquid-ammonia system because the entire system could operate at pressures less than atmospheric.

Cascade Mechanical-Vapor Compression

Mechanical-vapor-compression (MVC) devices would be attractive for use during a sealed shelter period when the air supply is suitable only for combustion and if a stored heat sink is required. Two such devices, which normally reject heat at temperatures slightly above atmospheric, could be coupled to form a cascade arrangement. This would make possible heat rejection to stored water at normal boiling temperature. Consequently, a minimum quantity of water would be required. One MCV unit would absorb heat below the shelter temperature and reject it to the second which would increase the temperature above 212 F.

Open-Cycle Mechanical-Vapor Compression

An open-cycle mechanical-vapor-compression system using methyl alcohol as the refrigerant and as the heat sink would be a satisfactory and economical way to cool a sealed shelter provided combustion air for an auxiliary power system were available.

The cooling system would operate at less than atmospheric pressure and, therefore, no leakage of toxic alcohol vapor would occur in the shelter. For disposal, the vapor could be vented to the atmosphere, burned in a simple combustor, or used as fuel for an engine. Except for the compressor, the system could be assembled from commercially available components. A rotary-vane compressor requiring no lubrication appears to be the most desirable.

Air-Cycle Cooling Systems

If economical air-cycle cooling systems could be developed, they would have substantial advantages in that air is used as the working fluid, standby maintenance should be comparatively inexpensive, and ventilation could be provided with the same mechanical equipment. Disadvantages would be the large power requirement and the need for an ample supply of water. At present equipment is commercially available, but it is expensive and it is not designed specifically for air-cycle cooling applications. Techniques would have to be developed for making inexpensive components designed specifically for shelter cooling. Centrifugal compressors and expanders appear to hold the most promise.

RECOMMENDATIONS FOR FUTURE STUDY

In the study of methods for disposing of excess shelter heat, five cooling systems were identified which warrant further research for particular shelter cooling situations. An open-cycle absorption system, a refrigerant-vapor-engine system, and an open-cycle mechanical-vapor-compression system are novel ones conceived during the course of the project. Mechanical-vapor-compression machines in cascade systems and air-cycle systems are nonconventional systems in the sense that they are not used for comfort cooling; however, such systems have been applied to special cooling tasks.

An open-cycle absorption system is the only system which can reject heat to a warm-water heat sink with an extremely small power input. This absorption system would be most useful during a sealed period, when the power source would have to be the shelter occupants or some kind of closed-cycle engine. The estimated cost of equipment and materials is reasonable when sulfuric acid and water are considered as the working fluids. The configuration of the acid-water unit would resemble the absorber shell of a conventional lithium-bromide-water absorption machine. Research is needed to determine materials of construction, to establish methods for storing concentrated sulfuric acid, to design the purging system for removal of noncondensable gases, and to devise methods for disposing of spent sulfuric acid.

The refrigerant-vapor-engine system appears attractive for the sealed period if an ice heat sink is required. The main advantage of this system is that it uses the heat released by the shelter occupants as the source of energy to drive the system's water pumps and air blowers. No oxygen would be required for either manual power or a closed-cycle engine. Commercial heat exchangers could be used, but a vapor expander and a refrigerant liquid pump would have to be developed. Expanders and pumps have been built for specialized military and waste-heat-recovery applications.

An alcohol-vapor compressor is needed for use with an open-cycle mechanical vapor-compression system which uses methyl alcohol as the refrigerant and heat sink. This system has merit with respect to cost, safety, low maintenance, and high reliability for cooling blast shelters needing a stored heat sink. Other than the compressor, the system could be assembled from commercially available equipment. A low cost compressor requiring no lubrication appears to be the most desirable type. Development of such a compressor should not be too difficult in that it would be similar to carbon-vane-rotary compressors used in vacuum service.

A cascade arrangement of two nearly conventional mechanical-vapor-compression machines could be used to reject heat to boiling water if power were available. This system would be applicable to the situation in which a limited quantity of water was available and when atmospheric air would be suitable only for combustion in an engine. If the water would have to be stored, the cost of its storage space in a blast-resistant facility would be much less than the cost for storing other heat-sink materials. The problems to be resolved are those associated with the operation of mechanical-vapor-compression machines at high temperatures. Refrigerants and lubricants decompose at elevated temperatures and this is aggravated by their contact with some of the materials used in these machines. It is believed that the deterioration would not be significant for the short-term operation in shelter cooling; however, data are needed to verify this and to determine if other problems would arise.

An air-cycle system could be used to cool shelters if a large power source and a large heat sink were available. Such systems are attractive because air is used as the working fluid and because the stand-by maintenance for high reliability would be low. The cost of the system and its power source would probably be high, but perhaps it may be acceptable in view of the merits of the system. Relatively small high-speed centrifugal compressors, expanders, and mechanical drives would have to be developed for high efficiency and low production costs. For the water-cooled heat exchanger and air piping, design details need to be established to incorporate commercially available components into the system.

- **RESULTS OF TECHNICAL AND ECONOMIC STUDIES**
  - NUCLEAR RADIATION EFFECTS
  - STANDBY MAINTENANCE REQUIREMENTS
  - HEAT SINKS
  - COOLING SYSTEM COMPONENTS
  - REFRIGERATION + DEHUMIDIFICATION DEVICES

INTRODUCTION

This section presents a detailed evaluation of methods for disposing of excess shelter heat. Both technical and economic factors are covered in each of five major categories:

1. Nuclear Radiation Effects
2. Stand-By Maintenance
3. Heat Sinks
4. Cooling System Components
5. Refrigeration and Dehumidification Devices

Nuclear radiation effects are discussed with respect to cooling system working fluids, electrical insulation, lubrication, construction materials, and the possibilities of radiation being carried into the shelter area by the cooling system. Data are included which show that radiation effects would be negligible for any of the cooling methods discussed in this report.

Standby maintenance is treated as a separate topic to facilitate coverage of problems common to various cooling systems. Specific maintenance problems related to particular components are covered later in the appropriate sections of the report.

The section on heat sinks includes data on the thermodynamic and physical properties of materials, material costs, and requirements for associated or supporting equipment. The equipment items discussed are those required to obtain and to maintain the heat-sink material for subsequent heat absorption.

In the "Cooling System Components" section, other equipment items which would be needed to transfer heat from the air in a shelter to a heat sink are discussed. These items include air-handling units, direct expansion coils, fans, pumps, air filters, blast valves, and evaporative condensers.

The discussion of refrigeration devices covers various methods and machines that can remove heat from a shelter and pump it to a sink having a temperature higher than that of the shelter. Both conventional machines and unusual devices which are not presently used for ordinary comfort cooling are considered. Performance data, power requirements, temperature limits, costs, and space requirements are covered. Dehumidification is closely related to refrigeration and dehumidification devices are also covered in this section.

NUCLEAR RADIATION EFFECTS

Various studies have been made of the probable radiation levels which would result from hypothetical nuclear attacks.[12-14] One study[12] predicts that the chances are less than 1 in 100 that the peak gamma flux would exceed 10^5 R per hr one hour after a detonation. Although such estimates reflect the qualifying assumptions and the method used to predict the radiation level, they are the best estimates available.

Assuming a peak gamma flux of about 10^5 R per hr, the corresponding integrated gamma radiation exposure would be 2×10^5 R during a 2-week period. At these radiation levels, it appears that equipment would not be significantly damaged from either direct radiation or from that associated with fallout. For the probable radiation duration and level, the characteristics of materials used in cooling systems would be essentially unchanged. Fluids flowing in coils exposed to radiation would not become radioactive and, therefore, they would present no radiation hazard to occupants. Lubricants, refrigerants, electrical insulation, and shaft seals would not be significantly affected.

Structural metals would not be physically changed by gamma flux exposures of less than 10^{12} R. Because the maximum expected integrated exposure is only 2×10^5 R, the physical properties of the metals in structures and equipment would not be altered. Also, no radioisotopes would be produced.

With respect to the functional properties of electrical insulation and lubricants, radiation greater than 10^7 R is required to make any change. Therefore, electric motors, electric wiring, and lubricated bearings could be used outside of the shelter, if expedient, without risk of failure due to radiation effects.

Refrigerants and refrigerant seals in mechanical cooling systems might sustain some damage; however, they would still perform satisfactorily for a 2-week period. A review of some experiments performed with refrigerant R-11 showed that initial breakdown of the refrigerant occurred when exposed to a gamma flux of 1.5×10^4 R per hr.[15] A cooling system was operated to determine the effects of this refrigerant breakdown on system performance. The results showed that a total exposure of 3.6×10^6 R was required before the efficiency of the system decreased and the compressor began to overheat. The overheating was probably caused by decomposition of the oil.

When ammonia is exposed to an equivalent of 10^9 R, only approximately 3 per cent of it will decompose into hydrogen and nitrogen. Thus, it appears that ammonia would not be affected appreciably at 10^5 R.

Refrigerant seals, made of silicone or neoprene rubber, metal, or carbon can withstand a dose of approximately 10^6 R before they lose their effectiveness.

While no problems are anticipated with irradiated water, it is expected that some effects might be evident. Unfortunately, there are no definite data. Perhaps the most significant effect would result if spray water were radiated in the presence of nitrogen. Gaseous nitric acid would form, some of which would be absorbed in the water. However, it is believed that the quantities would be so small that corrosion of the wetted surfaces would be negligible.

In a closed system, irradiated water might contain hydrogen and hydroxyl radicals, as well as free hydrogen, oxygen, and hydrogen peroxide. Some of these would be slightly corrosive; but again, corrosion would not be serious for a 2-week period of operation.

STANDBY MAINTENANCE REQUIREMENTS

There is very little available information on the maintenance of cooling systems that must be idle for long periods of time. The only real experience in this area is that acquired with commercial air-conditioning equipment. Many commercial units remain idle for periods longer than six months; they are then put into operation without difficulty. Whether this idle period could be extended appreciably depends largely upon the particular equipment in question. Discussed below are general topics related to stand-by maintenance and preservation of equipment. Additional information concerning specific equipment is included later in the appropriate technical discussions. However, the information is incomplete. Much additional work is needed to develop reliable data. In the meantime, the treatment of the subject can be only speculative.

Corrosion of metals, the growth of mold and mildew, and moisture pickup by electrical insulation are negligible in air when the relative humidity is below 30 per cent. Suitable humidity could be maintained easily and economically in a water-tight shelter during a standby period. Solid desiccant or mechanical dehumidifiers could be operated with commercial power.

Heat exchangers, pumps, and pipelines handling water could be preserved by filling them with water and adding a corrosion inhibitor. Ethylene glycol or brine solutions could be used to prevent freezing. Saturated solutions of sodium chloride and calcium chloride have been found to be less corrosive than the dilute solutions. Also, corrosion inhibitors are available for use with salt solutions. Pumps and heat exchangers are available for handling fresh water or sea water, and these should be adequate for any shelter application.

For water or brine systems, corrosion inhibitors are available. These include sodium dichromate, sodium nitrite, and mixed inhibitors such as sodium hexametaphosphate with zinc chromate, or sodium benzoate with sodium nitrite. The best selection of corrosion inhibitor in any particular instance depends upon the construction materials used in the system. For example, if there are lead-tin soldered joints in the water-cooling system, sodium nitrite cannot be used alone but must be combined with sodium benzoate. Similarly, if there are organic pump seals in the water system, the level of the sodium dichromate has to be maintained below the point at which the chromate will attack the seal. In shelter design, corrosion engineers, mechanical engineers, and materials specialists should collaborate in choosing the best corrosion inhibitor for particular systems.

Special precautions must be taken to assure proper functioning of electrical contacts. Contacts can be kept free of corrosion by keeping them dry, hermetically sealing them, or using corrosion-resistant materials.

Lubricated bearings are known to function satisfactorily if they are turned over occasionally. In general, lubricated bearings can be statically stored at a relative humidity of 30 per cent with no serious corrosion problems developing. Metal-to-metal galling or fretting corrosion can occur in loaded static bearings subjected to severe vibration. Under such circumstances, which probably would be rare in shelters, the load should be removed from bearings. No time schedule for bearing exercise can be specified. Bearings are known to function satisfactorily after nearly one year of idleness on agricultural and processing machines which are used seasonally. It seems probable that a bearing made of low ductility metals and packed with a moisture-resistant, noncorrosive lubricant could remain idle for years and still function reliably when put into service. However, data are needed to verify this.

In setting up a stand-by maintenance program, the entire cooling system, including the auxiliary power plant, must be considered. For example, heat exchangers could not be filled with a corrosion inhibitor and sealed if the cooling system were to be used as the load for exercising the auxiliary power system. Likewise, if the power system was maintained by encapsulation, any other equipment requiring periodic operation would have to be driven by another power source.

• HEAT SINKS

- EARTH
- ATMOSPHERIC AIR
- WATER - NATURAL SOURCE
- STORED MATERIALS
    - ICE
    - WATER
    - METHYL ALCOHOL
    - REFRIGERANTS
    - FUELS
    - COMPOUNDS WHICH REACT ENDOTHERMALLY
    - OTHER MATERIALS

HEAT SINKS

The term heat sink is used to designate the medium which absorbs rejected heat. Neglecting the insignificant amount of heat lost by radiation to the atmosphere, all of the heat removed from a shelter must be absorbed by the heat sink. This heat is manifested in an increase in the enthalpy of the heat-sink material. The enthalpy increase may result in a temperature rise, a phase change, or a combination of the two.

The term heat sink has taken on various meanings in the field of air conditioning and refrigeration because an explicit definition has not been required. Sometimes the term is used to refer to a heat-transport medium, such as the air passing through a cooling tower, or to a piece of equipment such as an evaporative condenser. For this report, "heat sink" is defined as the medium which absorbs heat with a resulting increase in enthalpy. In some circumstances the heat may be rejected from a shelter in such a manner that the heat sink medium may be confused with a heat-transport medium. For example, heat is carried away from a cooling tower by the atmospheric air, but the heat is absorbed by the water during a phase change from liquid to vapor. Therefore, the water is the heat-sink medium while the air is the transport medium in that it is used to carry away the vapor.

In some instances, multiple heat sinks are utilized. For example, a high-temperature evaporative condenser rejects heat to both the air and the water passing through the condenser. The sensible heat of the air is increased by a temperature rise, the water absorbs heat during a phase change, and the vapor is carried away in the air.

The characteristics of the heat sink determine the methods that can be used to cool shelters and also affect the cost of the cooling system. Accordingly, a study was made of the technical and economic aspects of using various available mediums as heat sinks.

Technical factors of interest pertain to the thermodynamic and physical properties of the materials. Economic factors include the cost of artificial materials for heat sinks and the cost of appropriate storage facilities. Equipment needed to transfer the heat to the heat sink is discussed in the "Cooling System Components" section.

For this study, heat-sink materials were divided into four categories as follows:

1. Earth
2. Atmospheric air
3. Water, natural source
4. Stored materials.

The first three, earth, air, and water, are available naturally, therefore, only their heat-absorption characteristics need to be evaluated.

The fourth type, stored materials, necessitate evaluation on the basis of their useful temperature, quantities required, storage requirements, and costs.

EARTH

On first examination, the earth may seem to be a good heat sink for shelter cooling systems. The earth has been used as both a heat source and a heat sink for residential heat pumps but only to an extremely limited extent. It has not been widely used as a heat sink in commercial applications because of its low thermal capacity, its unpredictable thermal properties, and its unfavorable economic aspects. The cost of installing and maintaining an underground heat exchanger is higher than that of providing an aboveground heat exchanger cooled by air or water. For shelter cooling systems, the blast resistance of earth heat sinks should not be overlooked in over-all evaluations, but it seems doubtful that this advantage can justify the higher costs involved. Costs are discussed in the section "Cooling System Components".

The properties of the earth which most significantly affect its use as a heat sink are:

1. Thermal conductivity

2. Density

3. Specific heat

4. Temperature.

The thermal conductivity is the most important property affecting the rate of heat flow through the earth. The thermal conductivity of moist soil is several times that of dry soil; however, moisture migrates through the soil in the same direction that heat flows. Therefore, the thermal conductivity of the soil decreases with use. In addition, as the soil dries, its volume decreases causing the soil to crack and withdraw from heat-exchanger surfaces. The resulting air spaces further reduce heat transfer between the heat exchanger and the soil. Thus, except for permanently wet or marshy soils or in instances where the heat exchanger could be buried below the water-table depth, underground heat exchangers should be designed for dry soil to insure sufficient heat-sink capacity in continued use.

Table 1 shows the thermal properties of various types of earth and stone. Stone is included to show that its properties are quite similar to those of soils. From the table it can be seen that the thermal conductivity of the dry soils varies considerably with composition and moisture content, the value being between 0.085 and 0.225. The heat-transfer rate through the soil is nearly proportional to the thermal conductivity. This wide range of values attests to the uncertainties which surround the design of a reliable yet minimum-cost earth heat sink.

TABLE 1. THERMAL PROPERTIES OF VARIOUS SOILS(17-19)

Soil	Thermal Conductivity, Btu/(hr sq ft F/ft)	Density, lb/cu ft	Specific Heat, Btu/lb F
Average dry soil	0.175	95	0.20
Low-conductivity dry soil	0.085	86	0.19
High-conductivity dry soil	0.225	126	0.18
High-conductivity wet soil	1.16	142	0.25
Granite	1.0 to 2.3	165	0.195
Dry sand	0.19	90 to 120	0.195
Limestone	0.2 to 0.75	155	0.215
Marble	1.2 to 1.7	170	0.210
Lava	0.49	--	--
Concrete	0.50 to 1.0	144	0.156

Earth temperature is another major factor that affects the use of the earth as a heat sink. The minimum and maximum earth temperatures that can be expected at a 6-foot depth throughout the United States are about 35 and 90 F.[16] In many sections of the country the cooling system would have to utilize a refrigeration machine to increase the temperature differential between the fluid circulated in the coil and earth for heat transfer.

ATMOSPHERIC AIR

Atmospheric air is the only universally available heat sink, although its temperature and humidity levels tend to limit its usefulness in some geographical areas. Contamination of the air by bomb bursts, fire, and chemical and biological warfare agents must also be considered.

Two methods for utilizing atmospheric air are: (1) the direct cooling of the shelter by ventilating air and (2) cooling of the condenser or high-side heat exchanger of a refrigeration machine. A corollary of the first method is evaporative cooling of inlet ventilating air wherein water is also used. Evaporative cooled air can absorb nearly twice as much heat as atmospheric air.

The temperature of atmospheric air is the only physical property which requires discussion here. Air temperature tables are published in the 1963 Edition of the ASHRAE Guide[20] and these data were used for calculations in this study. These temperatures were measured and recorded by various weather stations throughout the United States.

WATER, NATURAL SOURCE

Natural sources of water may be used as heat sinks in one of several ways depending upon the quantity available, the temperature, the chemical composition, and the solid content. Possible methods range from rejecting heat to large quantities of cool water, with a small water temperature rise, to rejecting heat to relatively small quantities of possibly warmer water by taking advantage of the high latent heat of vaporization. In all cases, commercial equipment is available and experience has already established the requirements for designing such cooling systems.

Consideration of water as an available heat sink implies that an ample supply is reasonably accessible and that the supply as well as the equipment for using the water will remain operational after the standby period and after an attack. Determining locations of water resources and assessing the probable effects of blast were not within the scope of this research program. Also, inasmuch as the techniques for using water from lakes or rivers is reasonably well established, the following discussion is limited to the use of water from wells.

Techniques for drilling and maintaining water wells have been developed primarily because of the need for water by small cities and industries located in regions where surface water is not available.

Well capacity can be adequately predicted by drilling test wells. Test wells can also show the structure of the water-bearing earth or rock strata. The earth or rock structure and the chemical content of the water establish whether or not the well characteristics are likely to change with use, or with time and nonuse.

The only concern with the standby maintenance of wells might be the buildup of chemical deposits and corrosion in the well equipment. Basically, water is classified as being "hard" or "soft" from the percentage of mineral matter in solution. The minerals commonly found in solution include: the carbonates of calcium, sodium, potassium, magnesium, iron, and manganese; the chlorides of calcium and sodium; and the sulfides of iron and magnesium.

Other contaminates include various dissolved gases, including natural gas, sulfur dioxide, and hydrogen sulfide. Water containing these gases is not necessarily "hard". The presence of sulfur compounds may also indicate the presence of sulfuric acid. Chlorides in the water indicate the possible presence of hydrochloric acid.

Carbonates, in the presence of air, tend to deposit as scale on metallic parts. There should be no problem with shelter cooling equipment which would be operated for a two-week period. However, deposits would build up in well casings if the water level fluctuated. Each time the level was lowered the water remaining on the walls of the pipe would evaporate and leave its chemical constituents. These chemicals would not be dissolved by the water when it rose again and, therefore, the deposit would grow. For many wells the deposits at any one location would be so small to be of no consequence. However, with high-chemical-content water the deposits could become serious. It seems probable that the rate of chemical deposition would be negligible if all openings to the well were sealed to prevent any circulation of air to the water surface. This would eliminate the evaporation which is necessary for deposit formation.

Beyond the scope of this report, and in need of a great deal of further research, is the subject of the deposition of iron compounds through the action of crenothrex bacteria. In a few wells, the casing walls have become so encrusted with iron sludge that it w almost impossible to remove the pumps. The best preventive is the frequent and heavy introduction of chlorine into the well. The time interval, amount of chemical, and method of introduction seems to be controversial.

The best method of using the water would depend upon the water temperature, the quantity available in relation to the heat load, and the cost of providing the water. As has been stated, it is beyond the scope of this report to list the availability of water re es at various locations. However, the temperatures of surface and ground wa the continental United States have been plotted, and piping and power costs can be adequately estimated. Details on the various methods for using water and the cost of the equipment to build up the cooling systems are presented in the section on "Cooling Systems". Therefore, at this point, it is of interest to examine the temperature range of naturally available water.

Figures 4 and 5 show the average well water temperature and the average surface water temperature for the months of July and August. Of particular interest are the areas where water temperatures are below 70 F. Well water temperature is below 70 F throughout the United States except in the extreme south. Throughout the summer, surface water temperautres remain below 70 F in the northwest and extreme north. Water at temperatures below 70 F could be used in a heat exchanger located in the occupied shelter space to maintain an effective temperature of 85 F.

The cost of wells and their approximate pumping capacities are generally known by the water developers in their respective areas. No adequate account of these factors could be included in this report. However, a good "rule of thumb" value for the cost of drilled wells is $1 per ft of depth per inch of casing diameter. Approximate capacities and costs of representative wells are given in Table 2.

TABLE 2. APPROXIMATE CAPACITIES AND COSTS OF WELLS

Diameter Casing, inches	Maximum Capacity, gpm	Cost of Well, dollars per gpm per 100 ft of well depth
6	125	4.80
8	300	2.66
10	500	2.00
12	1,000	1.20
20	2,000	1.00

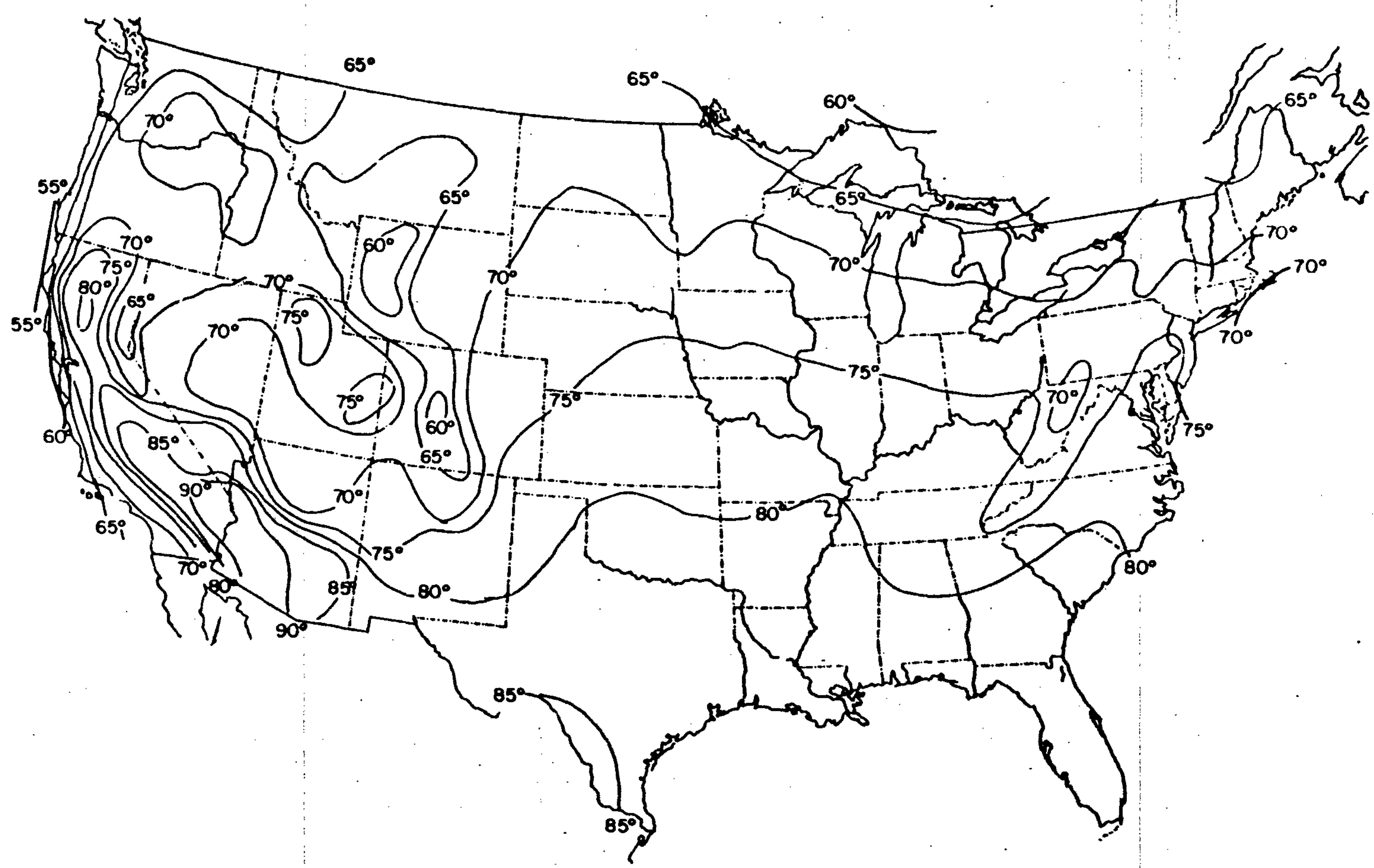


FIGURE 4. SURFACE WATER TEMPERATURES – JULY AND AUGUST

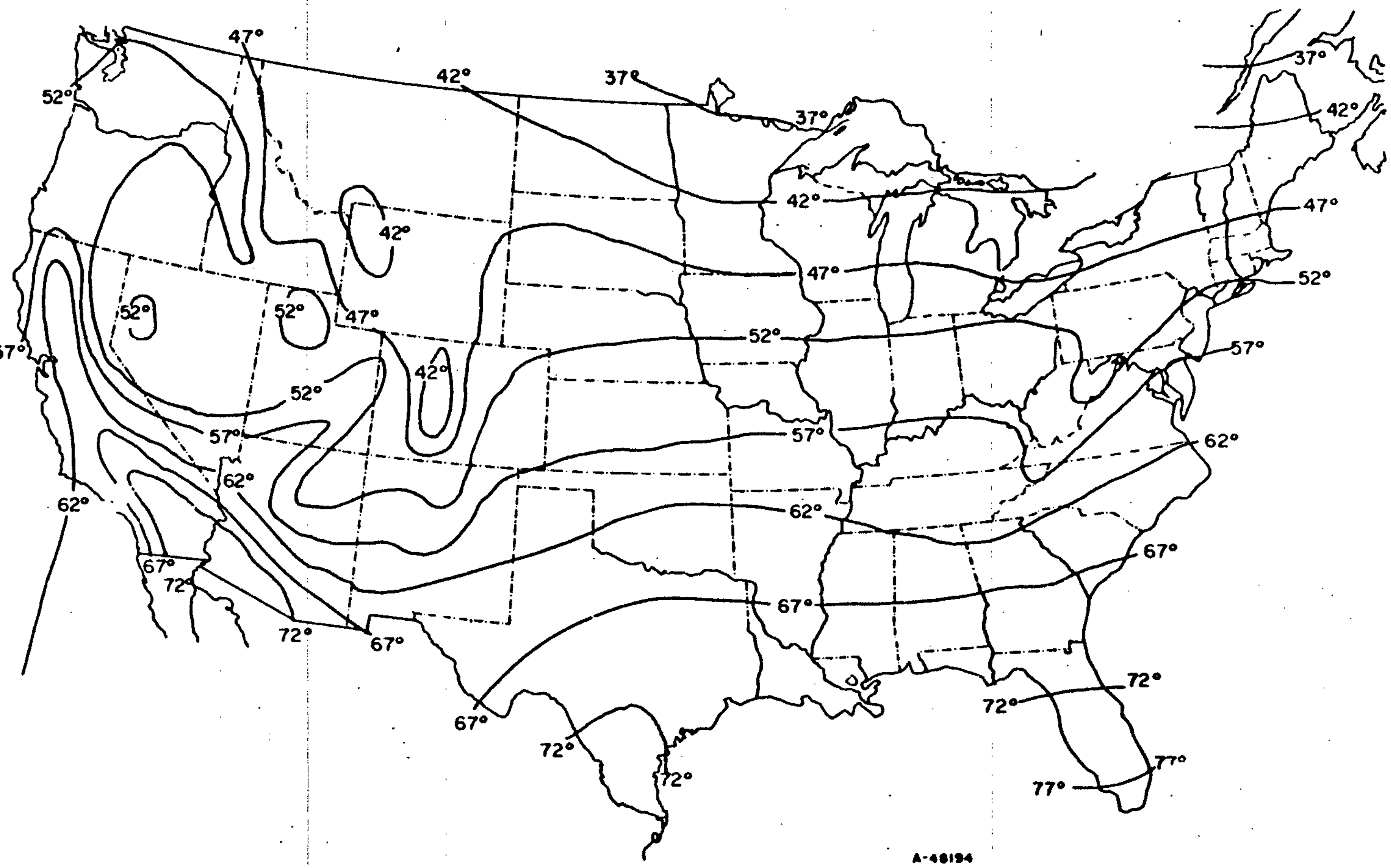


FIGURE 5. AVERAGE TEMPERATURE OF GROUND WATER AT DEPTHS OF 30 TO 60 FT

STORED MATERIALS

With low cost a prime consideration in shelter construction, ice and water are the best stored materials for general use as heat sinks. Under some circumstances, however, methyl alcohol or ammonia might be considered. These conclusions were made following a thorough search for materials which could be used as heat-sink media from both economic and practical standpoints. The search for other materials was stimulated by the unique shelter cooling requirements for a 14-day period of operation, the possibility of the shelter's being sealed from the atmosphere for 24 hours, the long standby period, the need for reliability, and the high capital cost and maintenance requirements of conventional cooling systems with their auxiliary power systems.

Literally hundreds of materials were evaluated using the physical and thermodynamic properties listed in chemistry and physics publications, and cost estimates obtained from manufacturers and suppliers. Factors considered are:

1. Pressure and temperature for phase changes
2. Specific heat
3. Heat of fusion
4. Heat of vaporization
5. Storage requirement
6. Toxicity
7. Flammability
8. Cost.

A discussion of the techniques of employing only ice, water, alcohol, and ammonia as stored heat sinks would leave this subject incomplete. Unanswered would be questions concerning the use of well-known heat-absorbing materials and the possibilities that, with some innovation, a heat-absorbing material having a secondary use could be utilized. An example of the latter would be the storage of liquified petroleum gas which would absorb heat during evaporation and subsequently be used as fuel for an auxiliary power system. Numerous possibilities exist which are technically feasible but which are not applicable for practical reasons.

Cost considerations alone eliminate the use of many materials as stored heat sinks. From strictly practical considerations, ice surpasses any other material. In fact, ice is nearly ideal in that no hazards are involved with its use; it could be used as the source of potable water; very little equipment and power are required to utilize it as a heat sink; and present technology and equipment are adequate for designing and constructing ice-sink systems. Therefore, if any material is to be superior to ice, its total requirements for use must be less costly. These requirements include the cost of the material, its storage facility, and the equipment required to utilize it. These factors are discussed for the following materials:

1. Ice
2. Water
3. Methyl alcohol
4. Refrigerants
5. Fuels
6. Compounds which react endothermically
7. Other materials.

ICE

As was indicated previously, in many respects an ice heat sink would be nearly ideal for cooling shelters. Low-power requirement, safety, supply of potable water, and equipment available for building a cooling system are some of the more obvious advantages. Somewhat less obvious are the facts that ice may well be the only practical heat sink during a sealed period, and that an ice sink for such a period could be utilized by the chilled-water distribution system which was part of a refrigeration cooling system. Details of the application of ice to example shelter-cooling situations are discussed in the section entitled "Cooling Systems". Included here are the heat transfer and cost considerations directly related to the heat-sink facility.

To a limited extent, ice is used for comfort cooling in churches where the cooling load is large but of short duration. In some instances, ice is obtained from an ice company, but in others ice is made continuously on the site during nonuse periods by a small refrigeration system. Therefore, the ice-making refrigeration system can be considerably smaller and less expensive than a comfort-cooling refrigeration system which would be large enough to handle the short-term cooling load. The experience gained from the design and operations of these systems would be applicable for cooling shelters. One point worth mentioning here is that the heat-transfer rate between still water and a piece of ice would be between 30 and 40 Btu per hr per sq ft per F temperature difference. This is not high enough for complete utilization of ice floating in water. Therefore, circulation of the water over or around the ice would be necessary.

Maintaining a supply of ice would require some combination of a refrigeration machine and insulated storage space. During the standby period, the refrigeration machine could be run by commercial power and attended by local service organizations. The size of the refrigeration unit and the power to operate it would depend primarily upon the thermal insulation of the storage space.

Insulation materials, their thermal conductivity, and their approximate installed cost for ice storage are:

Material	Thermal Conductivity, Btu/(hr sq ft F/in.)	Installed Cost, $ per sq ft per in. of thickness
Cork	0.27	0.35 to 0.40
Polystyrene	0.29	0.35 to 0.40
Polyurethane	0.15	0.80

Cork and polystyrene have been used for some time for cold storage insulation. Polyurethane is relatively new on the market and at the present time it costs approximately twice as much as cork or polystyrene. However, its higher cost is offset by its lower thermal conductivity, so the cost of insulating a storage facility is nearly the same for all three. The cost of polyurethane reflects the nature of the material and the manufacturing processes; therefore, its cost relative to the cost of other materials probably will not decrease appreciably.

Because the cost of applicable insulation is directly proportional to its thermal resistance, the heat loss from an insulated storage space would be inversely proportional to the insulation cost. Therefore, the approximate heat loss in terms of temperature difference and insulation cost is:

$$\begin{array}{c}\text{Heat Loss,}\\ \text{Btu per hr per sq ft}\end{array} = \frac{0.11 \times \text{temperature difference across insulation}}{\text{insulation cost, \$ per sq ft}}$$

For example, if the temperature difference between the outer and inner surfaces of a buried ice tank is 25 F (55 F - 30 F), and if the tank is insulated with 6 inches of cork which costs $2.40 per sq ft, the resultant heat loss would be:

$$\text{Heat loss} = \frac{0.11 \times 25}{2.40} = 1.15 \frac{\text{Btu}}{\text{hr sq ft}} .$$

With this heat loss through storage chamber walls, a refrigeration device driven by a 1-hp motor would be required to maintain a volume of ice having a heat storage capacity of 1×10^8 Btu. A heat sink having a capacity of 1×10^8 Btu would be adequate for storing the entire heat load from approximately 60 people for a period of 14 days. The cost of the refrigeration device to maintain the ice would be a negligible part of the total cost of such an ice heat-sink cooling system.

The total storage costs for ice would depend upon the structural requirements of the storage chamber. No requirements or cost data are available for structures which would be satisfactory for storing ice. However, storage chambers would be adequate if they were of the same construction as the shelter. Storage chamber costs were determined from published estimated shelter costs which are as follows:

Type of Shelter	Shelter Cost, $ per cu ft
Fallout	0.70
Blast, 30 psi	4.00
Blast, 100 psi	7.00

Figure 6 is a plot of the estimated cost of storing ice in various types of structures with the equivalent of 4 to 8 inches of cork insulation. About 160 Btu of heat could be absorbed per pound of ice if the final water temperature were 50 F. Ice weighs about 56 lb per cu ft. Near cities, ice can be purchased in large quantities at about $8 per ton delivered. This would make the ice cost about $0.025 per 1,000 Btu.

WATER

Water has the highest specific heat and the highest latent heat of vaporization of any known material. These thermodynamic properties along with practical and economic considerations place water in a class far above most other materials. In fact, the phase change temperature is the only property which could be surpassed.

The method of using water as a heat sink would depend primarily upon the availability of atmospheric air. The latent heat of vaporization could be utilized best with evaporative coolers or with a combination of a refrigeration machine and a cooling tower or a spray pond. All these arrangements require the use of atmospheric air to absorb the water vapor and for combustion air for a power system to drive equipment.

It seems unlikely that substantial quantities of power would be available at a time when atmospheric air could not be used for carrying off water vapor. But, if such a condition did exist, a refrigerant machine could be used to pump heat into a quantity of water to raise its temperature or even to boil the water at a pressure slightly above atmospheric pressure. The possibilities of using refrigeration machines with sink temperatures up to 212 F are discussed in the section on "Refrigeration and Dehumidification Devices".

With numerous methods for utilizing stored water being technically feasible, the practical uses are defined by the economic factors. The cost of water storage in various types of facilities are shown in Figure 7.

The costs of the underground steel tanks are typical of those for fuel storage at service stations. The cost of burying such tanks is about the same as for providing foundations for aboveground installations. The costs of the open tanks were assumed to be the same as for swimming pools having the water surface at ground level.

The underground steel tanks and the open tanks were assumed to have no blast resistance.

METHYL ALCOHOL

Methyl alcohol is one unconventional heat sink that may be applicable for cooling shelters. It can be used in two ways; in a cooling tower at atmospheric pressure, and in an open-cycle mechanical vapor compression refrigeration device at below atmospheric pressure. Both methods utilize the latent heat of vaporization of the alcohol.

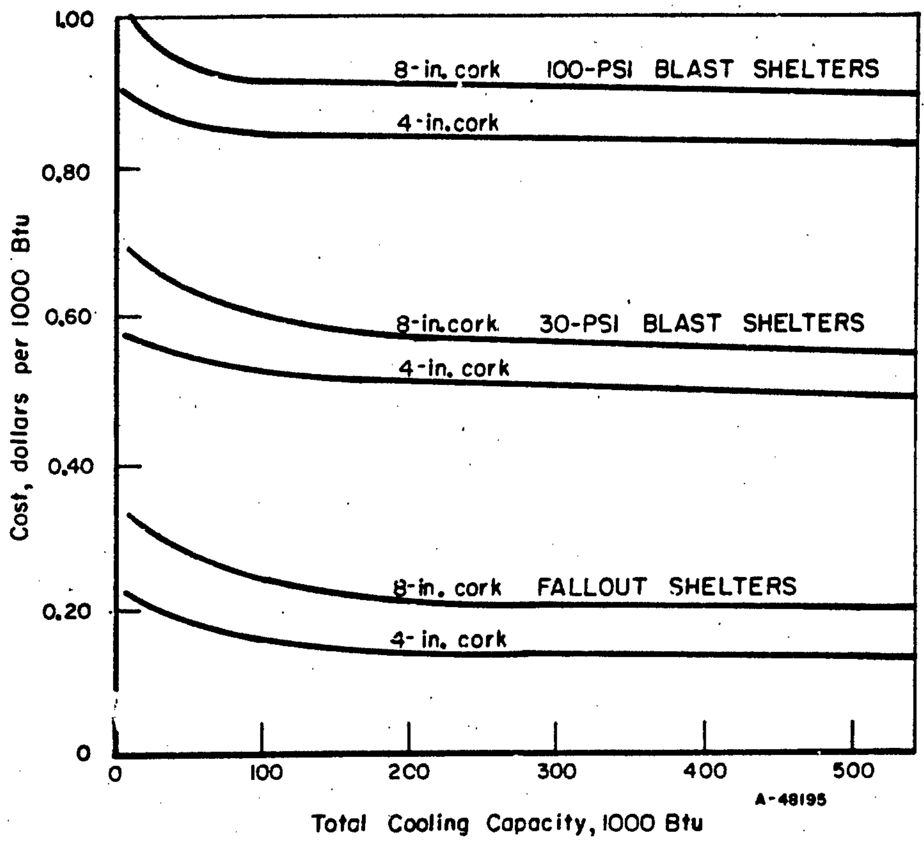


FIGURE 6. COST OF ICE-STORAGE CHAMBERS

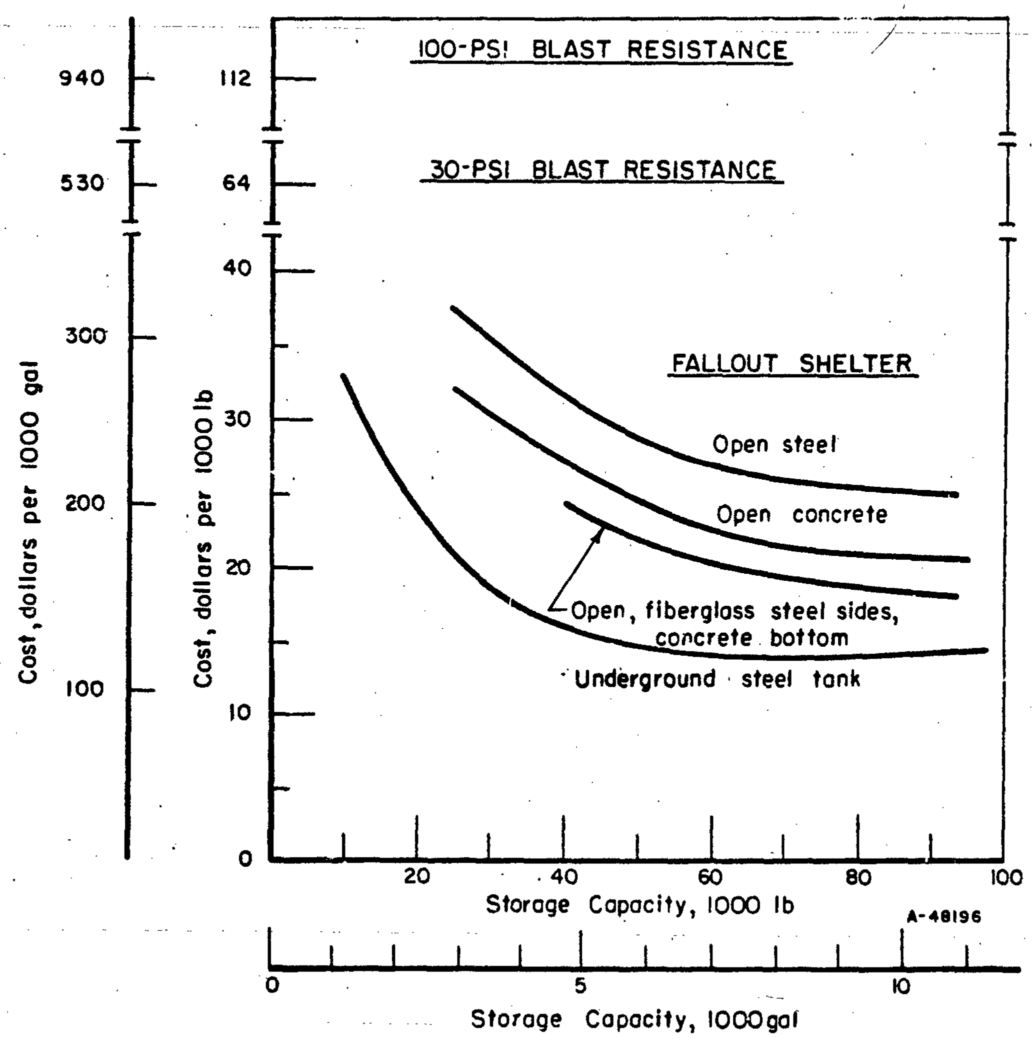


FIGURE 7. COST OF WATER STORAGE

Methyl alcohol sprayed through a cooling tower could leave the tower at a temperature low enough for direct cooling of a shelter. This would offer a way to reject heat from a shelter to the atmosphere, even on a hot summer day, without a refrigeration machine. The effectiveness and the cost of this method would vary with the temperature and humidity of the atmospheric air. Cooling towers and methods for disposing of the toxic air-alcohol vapor mixture would have to be developed. Methyl alcohol properties pertinent to its use in a cooling tower are:

Specific gravity	0.792
Boiling temperature	149 F at 14.7 psia
Heat of vaporization	473 Btu per lb at 75 F
Estimated cost	$0.045 per lb
Vapor pressure	122 mm of Hg at 75 F.

The vapor pressure of methyl alcohol is about 6 times the vapor pressure of water; this suggests its use as a heat sink in conjunction with a cooling tower. An alcohol-air psychrometric chart shows that alcohol would be cooled to about 50 F by vaporization into dry air. However, such a chart is not generally applicable because vaporization of alcohol in atmospheric air would be complicated by the presence of the water vapor. Alcohol has a high affinity for water. When sprayed into the air the alcohol would partially evaporate, but at the same time it would absorb water vapor. The solution circulated through the tower would reach an equilibrium composition of water and alcohol depending upon the moisture content of the air and the temperature of the alcohol-water solution.

Figure 8 is a logarithmic plot of the partial pressures of methyl alcohol-water vapors over methyl alcohol-water solutions. Partial pressure data of the mixture are not published below 200 mm of mercury and the lower pressures are of concern for this application. Therefore, the chart shows the results of calculations made from data for the pure liquid extrapolated into regions where experimental evidence is lacking. The lines on the chart appear logical and indicate reasonable values; however, experimental data are needed to establish the validity of the extrapolation.

Because alcohol absorbs water vapor, the alcohol content of the alcohol-water mixture in the cooling tower would have to be maintained by discarding some of the mixture and by adding pure alcohol. In effect, this lowers the quantity of heat which could be absorbed from the shelter with a given supply of alcohol. The quantity of heat which could be absorbed per pound of alcohol used would depend upon:

1. Temperature of alcohol-water mixture leaving the tower
2. Concentration of alcohol in the alcohol-water mixture
3. Wet-bulb temperature or moisture content of the air
4. Dry-bulb temperature of the air.

Figure 9 shows a plot of the heat which could be absorbed per pound of alcohol used with atmospheric air at 78 F wb and 96 F db. This example uses temperatures

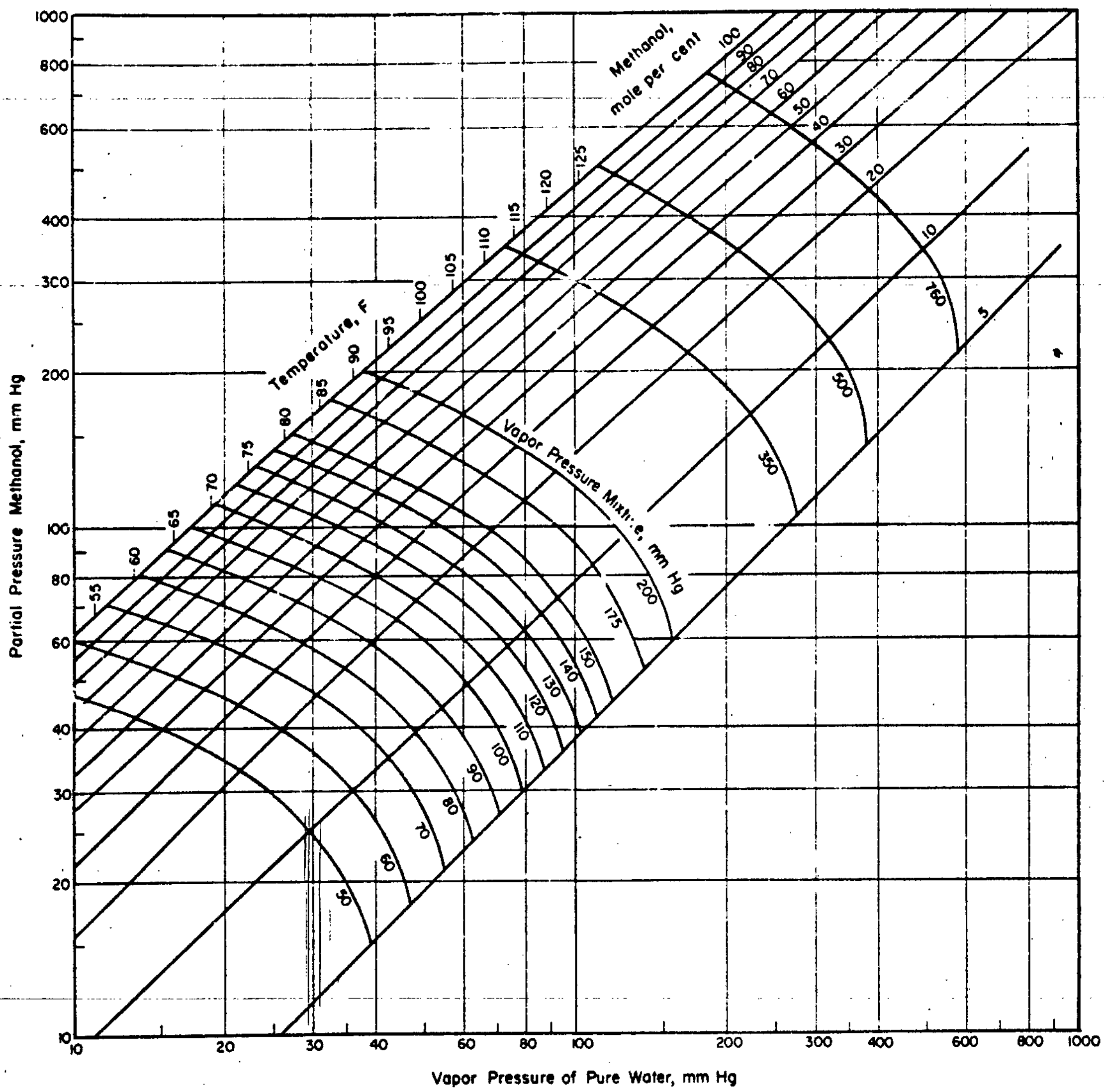


FIGURE 8. LOGARITHMIC PLOT OF PARTIAL PRESSURE OF METHANOL IN VAPOR OVER METHANOL-WATER SOLUTIONS AS A FUNCTION OF VAPOR PRESSURE OF PURE WATER AT THE SAME TEMPERATURE

typical of a hot humid day. For the calculations it was assumed that complete temperature and mass-transfer equilibrium were reached between the alcohol, water, and air. For best utilization of the alcohol, the mole fraction of alcohol in the alcohol-water mixture should be between 0.4 and 0.5.

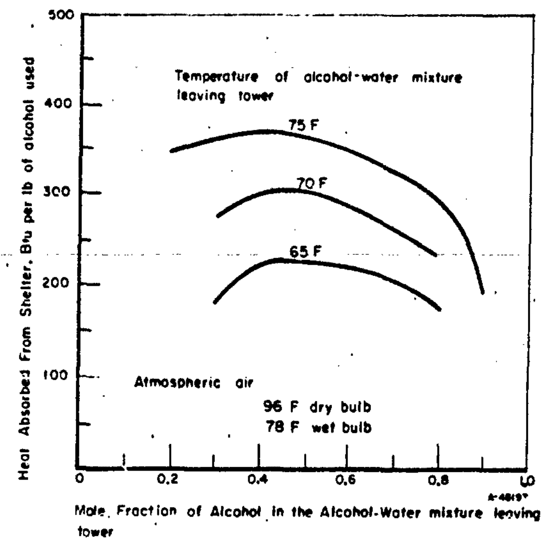


FIGURE 9. OPERATING CHARACTERISTICS OF A METHYL ALCOHOL COOLING TOWER

At a constant atmospheric air temperature, the amount of heat which could be absorbed from the shelter per pound of alcohol used decreases with decreasing temperature of the alcohol-water mixture leaving the tower. This is caused primarily by an increasing amount of water vapor from the atmospheric air being absorbed by the liquid alcohol. Consequently, at lower mixture temperatures more alcohol-water mixture would have to be discarded and more pure alcohol added to maintain the optimum alcohol concentration. Secondarily, a larger portion of the latent heat of the alcohol would be absorbed by the atmospheric air passing through the tower. For effective cooling of the shelter air, the temperature of the alcohol-water solution leaving the tower could not exceed 75 F.

Maximum utilization of alcohol would be obtained with completely dry air entering the cooling tower. This could be approached by dehumidifying the inlet air with possible regeneration of the dehumidifier by heat obtained by burning the alcohol vapors leaving the tower. While this could be done, the cost of the dehumidifier would be greater than the saving in the cost of the alcohol and its storage facility.

The performance characteristics of an alcohol heat-sink system using a cooling tower can best be shown by an example. For this example, the atmospheric air was assumed to have a dry-bulb temperature of 96 F and a wet-bulb temperature of 78 F. It was also assumed that the cool solution leaving the cooling tower would be at a temperature of 75 F.

Figure 10 shows the performance characteristics of the system operating at the assumed conditions. These calculated results were obtained by the use of the chart in Figure 8 to determine heat and mass-transfer balances. The temperature of the cool solution was chosen as 75 F for this example because: (1) higher temperatures could not be used in the shelter heat exchanger (70 F would be more applicable), and (2) lower temperatures would require higher alcohol concentration, thus larger quantities of alcohol would be discarded.

The flow-rate data reveal that the design of an alcohol cooling tower would be considerably different from that of a conventional water cooling tower when compared on the basis of heat transfer. For a heat-transfer rate of 1,000 Btu per hr, the flow rates would be:

Ideal Alcohol Tower	Typical Water Tower
0.66 gpm circulate alcohol-water mixture	0.25 gpm circulated water
4.8 cfm atmospheric air	25 cfm of atmospheric air

The circulation rate of the alcohol-water mixture is higher because, for this example, it was used with a 5 F temperature differential. The air-flow rate for the alcohol tower would be about one-fifth that for the water tower because of the relatively large quantity of alcohol vapor which could be removed by the air. Therefore, water cooling towers would not be applicable for use with alcohol. Rather, new configurations would have to be established to fully utilize the latent heat of alcohol for absorbing heat from a shelter.

Methyl alcohol is toxic to humans. The tolerance limit in air is 100 ppm for continuous exposure. This excludes the use of an alcohol tower under circumstances where the air leaving the tower could pass into the shelter ventilating air system.

Methyl alcohol is flammable over a relatively wide range of fuel-air ratios. Explosive limits are between 6.7 and 36.5 per cent by volume of alcohol in air. The water vapor in atmospheric air would not appreciably affect the explosive limits. The flammability of alcohol presents another danger but it also suggests a possible disposal method. The mixture leaving the cooling tower could be kept within the flammable range and the vapors destroyed by burning in a combustion chamber. This does not preclude the necessity of preventing ignition of the vapors within the tower. Flash arresters installed at both the inlet and outlet of the tower might be adequate, but a thorough study of fire prevention methods is needed.

With a required air-flow rate of only 5 cfm per 1,000 Btu per hr of heat absorbed, it might be advantageous to install the tower in the shelter structure providing adequate provisions were made to protect against blast and fire. Otherwise, extremely costly tower construction would be required to withstand blast pressures and to exclude fire.

The cost of providing alcohol as a heat sink would include the cost of alcohol and the storage costs which would be the same as for water storage. Assuming that in a cooling tower one pound of alcohol would absorb 300 Btu of heat from the shelter, the total costs per 1,000 Btu would be:

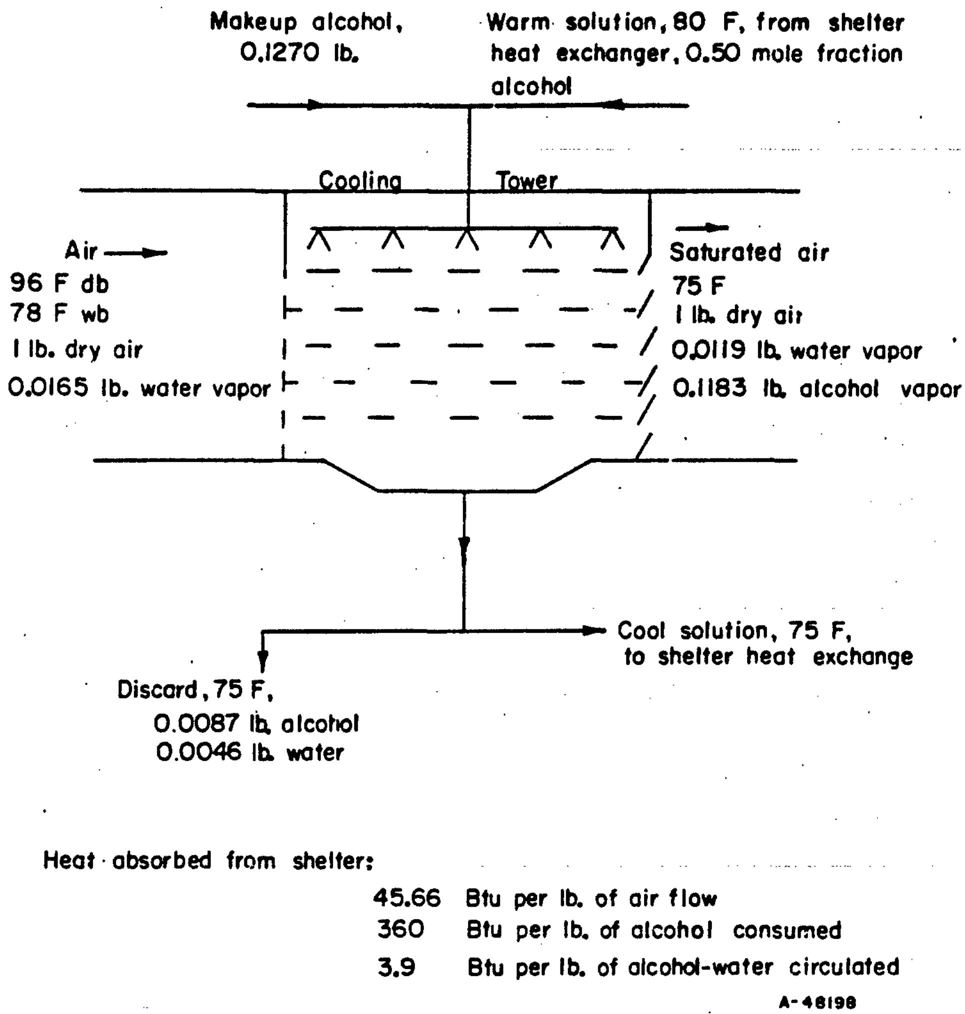


FIGURE 10. CALCULATED EQUILIBRIUM CONDITIONS FOR ALCOHOL COOLING TOWER

	Cost, dollars per 1,000 Btu		
Alcohol	0.15	0.15	0.15
Underground steel tank	0.06		
30-psi blast storage		0.27	
100-psi blast storage			0.47
Total	0.21	0.42	0.62

As was previously pointed out, methyl alcohol can also be used in an open-cycle mechanical-vapor-compression refrigeration device. A system of this type would operate at below atmospheric pressure. In Figure 8 it can be seen that at a pressure of 1.4 psia methyl alcohol boils at a temperature of 60 F. If this boiling is made to occur in a heat exchanger, direct cooling of the shelter ventilating air is possible. At 1.4 psia the latent heat of vaporization of the alcohol is 500 Btu per pound. Therefore, the cost of providing alcohol as a heat sink for use in an open-cycle refrigeration device would be:

	Cost, dollars per 1,000 Btu		
Alcohol	0.09	0.09	0.09
Underground steel tank	0.04		
30-psi blast storage		0.16	
100-psi blast storage			0.28
Total	0.13	0.25	0.37

Venting of alcohol vapors to the atmosphere would present fire and toxicity hazards similar to those associated with using alcohol in cooling towers which was discussed previously.

The equipment required to utilize methyl alcohol at low pressures is discussed in the "Refrigeration and Dehumidification Devices" section.

REFRIGERANTS

Many refrigerants have atmospheric pressure boiling temperatures well below the required shelter temperature. Such liquids, stored in underground pressure tanks, could be throttled through direct expansion coils to absorb heat from a shelter space. The vapors would be vented to the atmosphere. However, the cost of these materials, ammonia being a possible exception, make such an operation prohibitively expensive.

Table 3 shows physical and thermodynamic properties and costs of various refrigerant-type materials. With the exception of propane and ammonia, the costs of the refrigerants alone are much greater than the cost of ice and its storage facilities.* Propane can be eliminated because its cost is nearly eight times that of ice and, in addition, its storage costs would be at least twice that of ice. On a volume basis, propane would absorb about 5,000 Btu per cubic foot compared with 9,000 Btu for ice.

*Cost of ice - $0.025 per 1,000 Btu
Cost of ice storage facility - fallout shelter, $0.19; 30 psi blast shelter, $0.55; 100 psi blast shelter, $0.90.

TABLE 3. PROPERTIES OF REFRIGERANT-TYPE MATERIALS

Refrigerant	Properties of Liquid at 55 F: Pressure, psia	Properties of Liquid at 55 F: Density, lb/cu ft	Heat Absorption From and at 55 F, Btu/lb	Estimated Cost, $ per 1,000 Btu
Ammonia (R-717)	98.1	38.8	543	0.09
Butane (R-600)	23.8	36.6	163	0.93
Carbon dioxide (R-744)	698.9	52.2	113	0.68
Ethane (R-170)	466.1	22.7	294	4.80
Ethylene (R-1150)	840.0	11.4	91	8.30
R-12	66.7	84.5	66	11.50
R-13	390.8	63.0	45	16.60
R-21	17.0	87.1	103	7.35
R-22	108.0	77.4	88	8.65
R-114	20.7	93.2	57	13.20
R-500	78.9	74.9	79	9.60
Methyl chloride (R-40)	56.8	58.4	172	2.33
Nitrous oxide (R-744A)	632.5	50.8	117	6.45
Propane (R-290)	99.3	32.0	166	0.18[a]
Propylene (R-1270)	119.0	32.6	169	3.30
Sulfur dioxide (R-764)	37.1	87.4	159	1.90

(a) Propane costs may vary from $0.01 to $0.04 per pound depending upon location.

Ammonia has a heat-absorption capacity of 21,000 Btu per cubic foot. Buried ammonia storage tanks would cost about $4.50 per cubic foot or the equivalent of $0.22 per 1,000 Btu of cooling capacity. Therefore, the total storage and ammonia cost would be about $0.31 per 1,000 Btu. This is more costly than the total for ice in a fallout type of storage structure, but less than ice in storage structures similar to blast shelters.

The hazard of ammonia storage and subsequent use for shelter cooling could not be fully evaluated during this research program, but a few comments are in order. The threshold limit[21] of maximum allowable concentration for ammonia vapor in air is 100 ppm for an 8-hour daily exposure. For continuous exposure, the limit would be 50 ppm or less. Therefore, absolutely no leakage would be permissible within the shelter and the exhausted ammonia vapor could not be allowed to contaminate the ventilating air supply. Contamination of the ventilating air would not be a problem during a sealed period, and with a properly designed exhaust system it would not be a problem during normal operation. Because ammonia vapors are less dense than air, a rather simple exhaust system could be used.

Ammonia vapors could also be disposed of by burning. However, burning would produce gases even more toxic than the ammonia vapors. Nitrogen dioxide which would be formed has a threshold limit[21] of 5 ppm for an 8-hour exposure daily. Also, ammonia burned in the presence of hydrocarbons forms hydrogen cyanide for which the threshold limit[21] is 10 ppm. Therefore, it seems unlikely that burning the ammonia would be practical, even though the total quantity of toxic products would be less than the quantity of ammonia burned.

If a large quantity of water were available, provisions could be made to absorb the ammonia vapor in the water, forming ammonium hydroxide. The latent heat of the ammonia vapor and the heat of solution would both have to be absorbed by the resulting ammonia hydroxide. For example, if one pound of ammonia vapor were absorbed in six pounds of 70 F water, the resulting temperature would be 140 F.

FUELS

Hydrocarbon fuels, with the exception of the methyl alcohol which was discussed previously, would not be satisfactory heat sinks. They are not suitable either alone as heat sink materials or as a combination heat sink and fuel supply for an auxiliary power system. The reasons are:

1. Fuels are too expensive to be used strictly as heat sinks
2. Fuel quantities required for engines would be too small for significant heat-sink capacity
3. Fractional distillation of some fuels used as heat sinks might hamper engine operation
4. Fuel vapors released into the atmosphere would cause fire hazards.

Table 4 is a list of typical hydrocarbon fuels and their pertinent properties. Estimated costs are included.

TABLE 4. HYDROCARBON FUELS

Fuel	Specific Gravity Liquid	Boiling Temp at 14.7 psia, F	Heat of Vaporization, Btu per lb	Cost as Heat Sink, $ per 1,000 Btu
Butane	0.579	31	156-165	0.20
Ethane	0.546	-127	175	4.80
Ethyl alcohol	0.785	172	367-396	0.25
Fuel oil	0.82-1.00	350-700	80-120	0.25
Gasoline	0.70-0.75	100-400	116-145	0.30
Kerosene	0.78-0.82	200-600	90-110	0.30
Methane	0.424	-258	248	0.30
Methyl alcohol	0.792	149	482-502	0.09
Propane	0.582	-44	147-167	0.18

All of the fuels are considerably more expensive than ice as a heat sink. Therefore, a fuel heat sink could be justified only if the fuel could also serve another purpose. The only possibility seems to be its subsequent use as fuel for auxiliary engines. However, the heat absorption capabilities of fuels are low and, therefore, a much larger quantity of fuel would be needed for use as a heat sink than would be needed for fuel for an auxiliary power system. The shaft energy which would be obtained from a given quantity of fuel is from 12 to 80 times the latent heat of vaporization of the same quantity of fuel. Therefore, no work was done to evaluate any method for using fuel as a heat sink in conjunction with its use as a fuel.

COMPOUNDS WHICH REACT ENDOTHERMICALLY

Compounds which provide endothermic reactions are too costly for consideration as shelter heat sinks. Various compounds could be separately stored at earth temperature and produce a cooling effect when mixed.[22] Typical combinations are: ammonium nitrate and water; sodium nitrate and dilute nitric acid; ammonium nitrate, sodium carbonate, and water; sodium sulfate and dilute sulfuric acid. However, the cooling effect would be so small compared with the cost of the materials that such heat sinks are not applicable.

Ammonium nitrate mixed in equal parts with water produces a typical endothermic reaction with a heat absorption capacity of about 47 Btu per pound of mixture. Ammonium nitrate costs about $0.034 per pound or in terms of cooling capacity, $0.36 per 1,000 Btu. This is more than 13 times the cost of ice.

OTHER MATERIALS

Many chemical compounds absorb heat with a phase change at attractive temperatures. Ones with phase changes between 55 and 70 F could be stored underground at ambient temperatures and could be used to absorb heat by conduction and convection directly from the shelter. Others with phase changes between 70 and 120 F could be utilized with conventional refrigeration machines. Ones with phase change temperatures above 120 F would require custom refrigeration machines although these could be made up of conventional components. Even though all these technical possibilities exist, these heat-sink mediums are too costly for such applications. Because of this, the technical problems of heat transfer and storage requirements associated with them were not studied.

Table 5 is a list of a few of the many compounds studied which undergo phase changes at temperatures between 32 and 212 F. Their pertinent properties as heat sinks are included. Ice, water, and methyl alcohol have been included to show their cost and storage volume advantages over all the other materials.

Sodium sulfate decahydrate, sodium carbonate decahydrate, and calcium chloride hexahydrate are crystalline substances which have been previously considered as heat storage materials.[24] Because of this, and also because these are among the least expensive materials being considered here, a few comments about them follow.

During a phase change with the addition of heat, the hydrates change in crystalline structure. Essentially some of the combined water is driven out of the crystals and a

TABLE 5. STORED HEAT SINKS(23-26)

Material	Chemical Formula	Phase Change Temperature, F	Latent Heat Btu/lb	Latent Heat Btu/cu ft	Cost, dollars per 1,000 Btu heat absorbed
Solid liquid					
Nitrogen pentoxide	N_2O_5	85	139	14,300	
Gallium	Ga	85	34	12,500	
Disodium phosphate	$Na_2HPO_4 \cdot 12H_2O$	97-125	120	11,400	0.28
Hydrazine	NH_2NH_2	36	170	10,530	110.00
Sodium carbonate	$Na_2CO_3 \cdot 10H_2O$	90	115	10,350	0.16
Sodium sulfate (Glaubers salt)	$Na_2SO_4 \cdot 10H_2O$	90	108	9,840	0.11
Sodium thiosulfate	$Na_2S_2O_3 \cdot 5H_2O$	120	90	9,640	0.56
Water	H_2O	32	144	8,350	0.025
Sulfuric acid	$H_2SO_4 \cdot H_2O$	47	72	8,070	0.14
Formic acid	HCOOH	48	106	8,050	1.80
Calcium chloride	$CaCl_2 \cdot 6H_2O$	84	75	7,870	0.13
Glycerol	$C_3H_8O_3$	64	86	6,800	2.30
Acetic acid	$C_2H_4O_2$	62	81	4,450	2.20
p-Xylene (ethylbenzene)	$C_6H_4(CH_3)_2$	56	71	3,830	0.70
Benzene		42	55	3,020	0.75
Liquid vapor or solid vapor					
Water	H_2O	212	970	60,300	Nil
Hydrofluoric acid	HF	67	650	40,300	0.25
Ethylene oxide	$(CH_2)_2O$	49	250	30,800	0.92
Methyl alcohol	CH_3OH	148	472	23,600	0.09
Hydrocyanic acid	HCN	79	378	16,240	0.32

solution of water and the salt is formed. When subsequently cooled, some, but not all, of the salt returns to the hydrate crystal. Therefore, with each cycle of heating and cooling, the quantity of hydrate decreases. This is one drawback to their use as a heat storage material for commercial applications. This would be no problem for the shelter cooling application because no cycling would take place. However, this property would need to be considered in the shipping and handling of the hydrates prio to their being sealed in underground storage chambers.

Only the sodium sulfate decahydrate is commercially available in large quantities, but sodium carbonate decahydrates and calcium chloride hexahydrate could be produced if the need arose. The anhydrous form of these chemicals, which is readily available, could not be mixed with water to form the desired hydrates. No significant technical problems are foreseen regarding efflorescence or deliquescence of the hydrates during shipping and handling if it is done according to well-established methods. Prolonged storage would be possible in sealed containers at temperatures below the phase change temperature.

Solid sulfuric acid is an attractive material in that it has a melting temperature of 47 F. However, its cost is about six times that of ice. The storage volume, insulated storage facility, and refrigeration machine needed would be about the same as for storing ice. Because of its much higher cost it is not a desirable heat sink material.

• COOLING SYSTEM COMPONENTS

- REFRIGERANT-VAPOR ENGINES
- EARTH COILS
- COOLING TOWERS
- VERTICAL-TUBE CONDENSERS
- QUIET SURFACE PONDS
- SPRAY PONDS
- AIR-HANDLING UNITS
- AIR-COOLED CONDENSERS
- EVAPORATIVE CONDENSERS
- EVAPORATIVE COOLERS
- DEHUMIDIFIERS
- AIR FILTERS
- BLAST VALVES
- BLOWERS
- WATER PUMPS
- PACKAGE BOILERS
- MECHANICAL POWER

COOLING SYSTEM COMPONENTS

This section covers the technical and economic aspects of cooling system components other than refrigeration devices. Among these are heat exchangers, pumps, blowers, motors, and air filters which can be combined in a variety of ways to make up cooling systems. These components can be divided into three classes: (1) novel or unconventional, (2) conventional, operated at other than design conditions, and (3) conventional, operated at design conditions.

The novel or unconventional components that are discussed are:

1. Refrigerant-vapor engine

2. Earth coils

3. Vertical-tube atmospheric condenser

The characteristics of conventional components have been developed through years of use and continual modifications, and the state of the art is such that basic improvements that would significantly improve their performance for shelter cooling are unlikely. However, their effectiveness would be enhanced by operating them at "off-design" conditions. These "off-design" conditions and the estimated performance are discussed with respect to:

1. Evaporative coolers

2. Cooling towers

3. Evaporative condensers

4. Quiet surface and spray ponds.

Typical costs and operating data are shown for the conventional components which would be used in a conventional manner. This information, even though well established, is presented here for convenience and to permit future updating of economic data. Condensers and evaporators which are integral parts of refrigeration machines are discussed along with the machines.

REFRIGERANT-VAPOR ENGINES

A cycle analysis indicated that a refrigerant-vapor engine could be used to drive a shelter cooling system if ice were used as the heat sink. Such an engine is especially applicable during a sealed period when oxygen would be limited for either manual or internal-combustion-engine power. With an over-all cycle efficiency of 3.4 per cent, sufficient power could be developed to drive blowers and pumps for removing heat from the shelter air and rejecting it to chilled water. Common refrigerants have thermodynamic and physical properties such that the pressures and flow rates would be moderate, and the shelter equipment requirements seem well within established design practice. The development of an engine operating on a refrigerant vapor at 70 F would not present many of the difficulties which have hampered the use of refrigerant-expansion engines at higher temperatures.

Figure 11 is a schematic diagram of a refrigerant-vapor-engine cooling system and a pressure-enthalpy diagram describing the operation of the engine. The cycle analysis was made using 70 F vapor from the vapor generator and 40 F liquid leaving the condenser. These temperatures are valid for practical heat exchangers and the temperatures are ones which could be expected within a shelter provided with an ice heat sink.

Table 6 shows vapor-generator and condenser pressures, and the associated flow rates of a vapor engine for removing shelter heat at a rate of 100,000 Btu per hr. Of the common refrigerants shown, R-22 is one of the more practical from the standpoint of pressure, volume flow rates, and safety for use in shelter cooling.

Figure 12 shows the power output of an engine operating on R-22 at various overall engine efficiencies with a heat input to the cycle of 100,000 Btu per hr. Also shown is the power required to drive the pumps and blowers for the complete cooling system. Power requirements to operate the system were determined from the flow rates and estimated pressure losses for commercial heat exchangers. It was assumed that the liquid pump would be directly coupled to the engine with the air blower either directly coupled or driven by an electric motor. Power requirements would be:

	Horsepower Required, hp	
Equipment	Direct Mechanical Drive	Electric Drive (Generator Eff., 90 per cent; Motor Eff., 85 per cent)
Liquid pumps	0.465	0.465 (direct-drive only)
Air blowers	0.700	0.915
TOTAL	1.165	1.380

A direct-drive system seems to be the most appropriate for shelter cooling. Such a system would require a minimum of equipment and could be operated with a lower efficiency engine. It could be nearly self-governing with the engine output (or speed) matched to the cooling load (or blower speed) and the power requirement. Any additional control could be provided with simple manually controlled or temperature-controlled throttling valves.

Electric-drive systems would allow more freedom in the location of the engine and the blower. This feature might warrant the use of an electric drive in a large shelter where cooling units would have to be remotely located from a centrally located ice heat sink. A power unit consisting of one engine, one refrigerant condenser, and one electric generator would supply electric power to the blowers on the evaporators. Each evaporator would require refrigerant lines to the power unit. Advantages of such a system would have to be weighed against those of multiple small units each containing its engine, mechanically driven blower, evaporator, and condenser. Each condenser would be cooled with water piped from the heat sink.

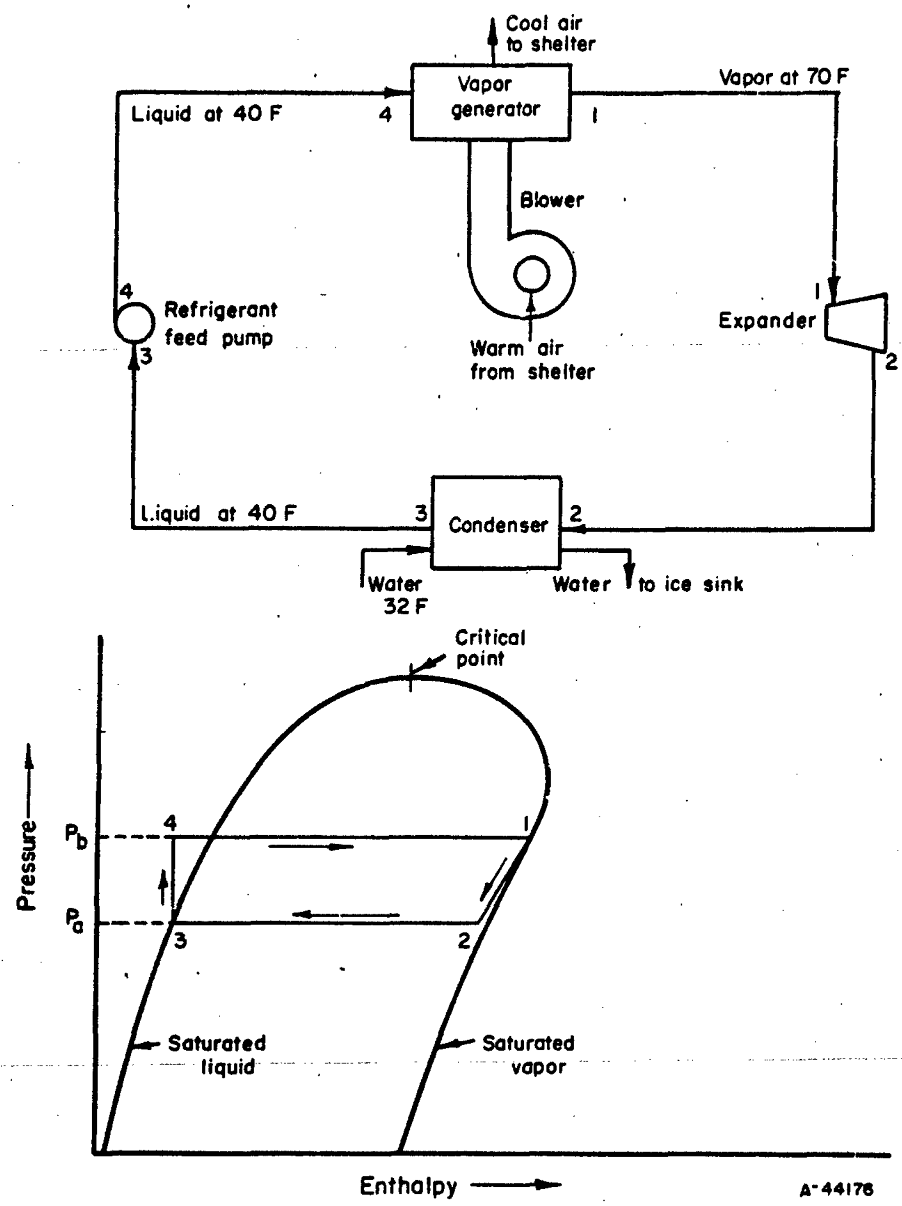


FIGURE 11. SCHEMATIC FLOW DIAGRAM AND PRESSURE-ENTHALPY DIAGRAM FOR THE REFRIGERANT-VAPOR-ENGINE SYSTEM

TABLE 6. REFRIGERANT-VAPOR-ENGINE SYSTEM PRESSURES AND FLOW RATES FOR A COOLING LOAD OF 100,000 BTU PER HR

Refrigerant	Vapor Generator Pressure at 70 F, psia	Condenser Pressure at 40 F, psia	Generator Pressure / Condenser Pressure	Flow Rate to Remove 100,000 Btu/Hr	
				Lb/Hr	Cu Ft per Hour at Engine Inlet
R-11	13.41	7.03	1.91	1,180	3,000
R-12	84.82	51.68	1.64	1,454	717
R-13	473.4	319.6	1.48	2,910	168
R-21	23.08	12.32	1.87	920	2,120
R-22	137.2	83.72	1.64	1,113	445
R-40	73.41	43.25	1.70	570	787
R-113	5.52	2.65	2.08	1,374	7,440
R-114	27.57	15.22	1.81	1,587	1,800
R-290	124.0	78.0	1.59	596	526
R-500	100.51	60.94	1.65	1,215	597
R-600	31.6	17.7	1.79	577	1,720
R-717	128.8	73.32	1.76	184	427
R-744	852.5	567.3	1.50	1,163	93
R-764	49.62	27.10	1.83	627	996

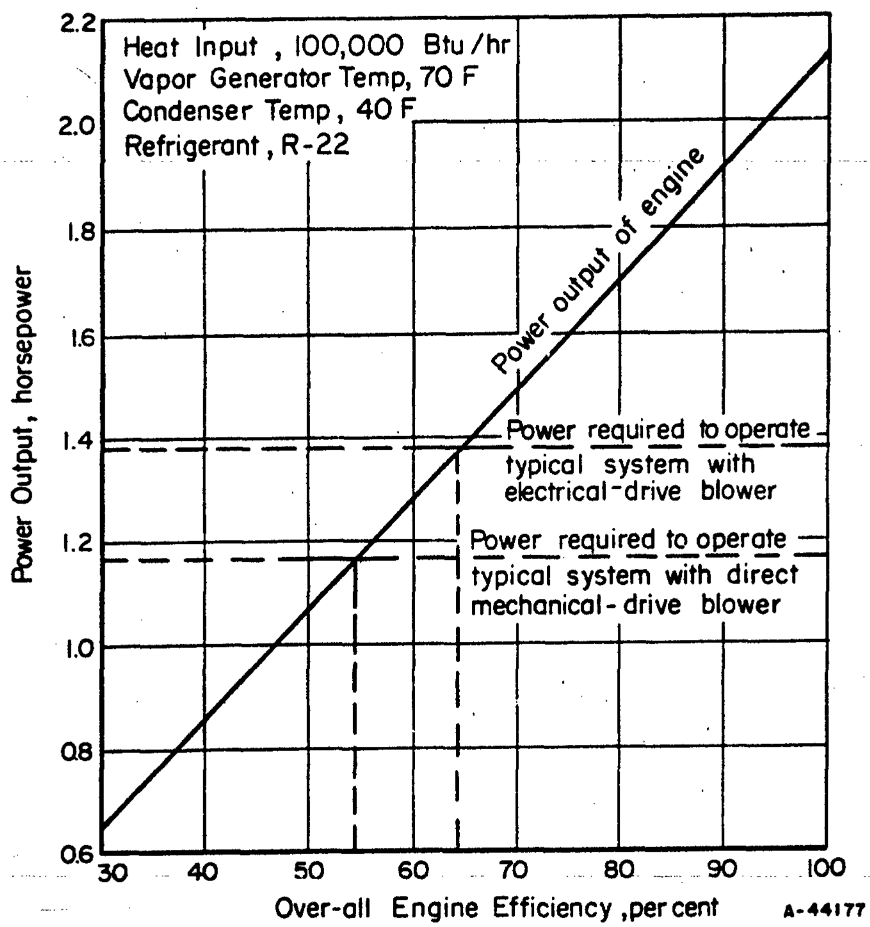


FIGURE 12. OPERATING PARAMETERS FOR A REFRIGERANT-VAPOR-ENGINE SYSTEM

For the relatively low compression ratio required, refrigerant-vapor engines could be designed with an over-all efficiency of about 70 per cent. At an operating temperature of 70 F, no problems with vapor condensation within the engine, with lubrication, or with refrigerant decomposition would be encountered. These have restricted the use of refrigerant-vapor engines at the higher temperatures necessary to obtain thermal efficiencies high enough for commerical applications.

For this report, the cost of the engine and its small liquid-refrigerant pump is estimated to be $30 per cfm of gas at the engine inlet. This estimate was made assuming that the engine would be similar to a rotary vane air compressor and that being a specialized piece of equipment not commercially available, the cost would probably be 5 to 6 times that of a rotary vane air compressor.

EARTH COILS

The practical considerations and the physical properties of earth with respect to its use as a heat sink were discussed in the "Heat Sink" section. Presented here are estimates of the heat-transfer characteristics and cost of buried coils.

Various analytical methods have been proposed for calculating the heat-transfer rate from buried surfaces to the earth surrounding them. No experimental data are available for evaluating the adequacy of any of these methods but they are sufficiently accurate for present purposes. The major factors affecting the heat transfer are:

1. Soil thermal conductivity
2. Heat-transfer surface shape and size
3. Temperature difference between heat-transfer surface and earth
4. Duration of operation.

An isolated straight pipe provides the best configuration for the maximum heat transfer per unit of surface area. Also, the reduction of heat transfer with time of operation is less for isolated pipes than for any other configuration. For most soil types and for a 14-day period of continuous operation, pipes must be spaced 4 to 8 feet apart to be isolated effectively.

The heat-transfer rate is roughly proportional to the soil conductivity and the temperature difference between the pipe and the earth. The other factors influence the heat transfer in a more complex way.

Figure 13 shows the calculated heat transfer from the pipe to average dry soil and to wet soil for various pipe diameters which would be obtained after 14 days of operation. The heat transfer to wet soil would be about 10 times that to dry soil, but wet soil heat transfer would rarely be possible. As discussed in the "Heat Sink" section, heat flowing into the earth drives moisture in the same direction as the heat flow. Therefore, even though the earth might be wet at the start of operation, it would become dry around the pipe as heat continued to flow into the earth. For this reason,

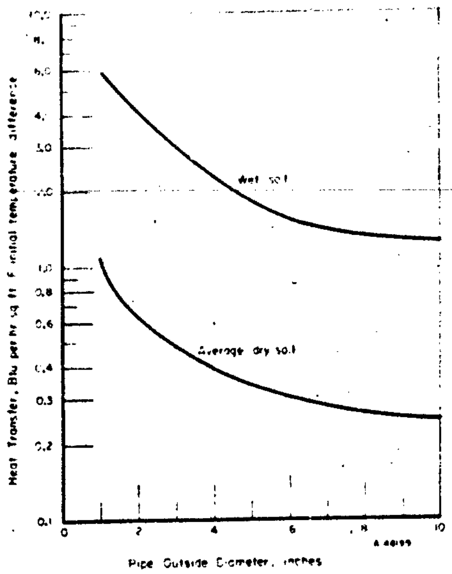


FIGURE 13. HEAT TRANSFER FROM PIPES TO SOIL AFTER A 14-DAY OPERATING PERIOD

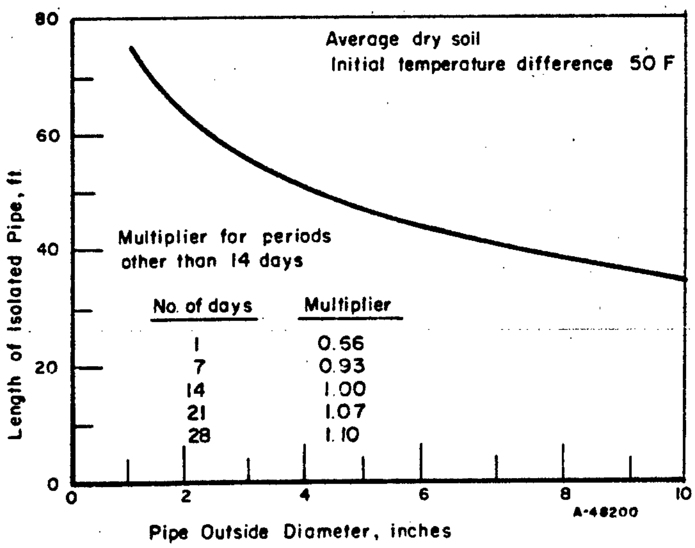


FIGURE 14. LENGTH OF PIPE REQUIRED TO TRANSFER 1000 BTU PER HR TO THE EARTH

the remaining part of this discussion will be concerned only with average dry soil conditions.

The difference between the shelter temperature and the earth temperature would be too small to provide adequate heat transfer directly to the earth. Therefore, a refrigeration device would be required to pump the shelter heat to a higher temperature. Because of the possibility of leaks in the buried pipe, the pipe should not be used directly as a refrigerant condenser but rather as a cooler for an intermediate fluid such as water or brine. With a conventional mechanical-vapor-compression machine operating with a condensing temperature of, say, 125 F, the brine or water could enter the coil at 115 F. The temperature difference between the pipe and the earth would then be about 50 F.

Figure 14 shows the length of pipe required to reject 1,000 Btu per hr to an average dry soil for 14 days with an initial temperature difference of 50 F between the pipe and the earth. The multipliers shown permit calculation of the length of pipe needed for operating periods other than 14 days. These show that for only one day of operation the pipe length could be two-thirds of that required for a full 14-day period. Operation could be considerably extended with little addition of pipe or slightly reduced heat-transfer capacity. Considering the fact that the pipes would have to be at least 4 feet apart, considerable area would be required for an earth coil. For example, if 20 Btu per hr were rejected per foot of pipe, an area of about 200 square feet would be required to reject the heat load of two average people.

The total cost of using earth coils would consist of the cost of excavating and backfilling the trench as well as the procurement and installation of the pipe. Typical costs for digging and backfilling are:

> 35 cents per linear foot for the first 3 feet of depth plus 5 cents per linear foot per foot of depth in excess of 3 feet.

Data are lacking on the cost of installing a network of piping such as would be required for a complete cooling system. The earth coil would probably be less expensive than a building piping system requiring hangers and a relatively large number of direction changes. Therefore, it was assumed that the earth coil would cost about three-fourths that of a building pipe system or 75 cents per foot of length per inch of diameter for copper, steel, or plastic pipe. These estimates used with the pipe lengths shown in Figure 14 give the following costs for earth coils in trench 8 feet deep:

Pipe Outside Diameter, in.	Total Cost,* $ per 1,000 Btu per hr
1	100
2	130
4	230
6	290

*Based on heat-transfer data in Figure 13.

The costs for the earth coil alone would be equal to or greater than the total cost of an ice heat sink. Should a shelter be located where wet soil would be accessible for the sink and where the soil would remain wet during heat addition, the coil costs could be as low as one-tenth of those shown above. Under such circumstances, an earth-sink system with the required refrigeration machine and power source might be economically feasible.

COOLING TOWERS

Cooling towers that are operated above their normal design temperature appear especially attractive for shelter cooling. The exhaust ventilating air flow would be sufficient to carry off all of the water vapor released by a tower having an inlet water temperature of 115 to 130 F. Water at these temperatures could be used to cool the condensers of mechanical vapor-compression refrigeration machines. This application is described in more detail later.

Cooling towers which are commercially available could be used to supply cooling water for refrigeration machines provided sufficient quantities of air and water were available. The primary reason for using cooling towers is that the latent heat of vaporization of water can be utilized as the heat sink at a temperature slightly above the wet-bulb temperature of the air with little draft loss. Therefore, when only a limited supply of water is available for use as a heat sink, a cooling tower will provide the maximum utilization of the water. Cool water could be supplied not only for a refrigeration machine, but also for cooling an auxiliary power system.

Packaged or factory-built cooling towers are available in sizes up to 3,000,000 Btu per hr of heat-dissipating capacity. These units, or multiple units if necessary, could handle the heat rejection from refrigeration machines used to cool shelters of any size. Custom-built towers having very large capacities such as for electric power generation are constructed on the site. In general, in the larger sizes the performance features and the costs of packaged and custom-built towers are similar.

Descriptions and operating characteristics of cooling towers for conventional use are adequately covered in the literature. Therefore, the discussion here will emphasize to those aspects which are particularly pertinent for shelter cooling.

Packaged cooling towers are made with steel exteriors and fills of redwood, treated fir, or some plastics. The towers are constructed and properly painted or coated to withstand many years of use and exposure to the weather. Models are available for exterior installation with a fan to provide air circulation. Also available are towers for inside mounting.

In some circumstances cooling towers for shelters could be exposed to the atmosphere and operated in a conventional manner. For a typical operation, a water circulation rate of 0.25 gpm and an air flow rate of 25 cfm would be required for rejecting 1,000 Btu per hr of heat. This corresponds to a water evaporation rate of about 1 lb per hr. Water leaving the condenser of a refrigeration machine would enter the tower at 16 to 25 F above the air wet-bulb temperature. Leaving the tower, the water temperature would be 6 to 13 F above the wet-bulb temperature. The total power required to drive the pump and fans of atmospheric cooling towers is approximately 0.01 hp per 1,000 Btu per hr of heat rejection.

The heat-rejection rate of a tower with a given air flow can be substantially increased by raising the temperature of the water entering the tower. This feature is especially attractive for the use of towers installed within the protective walls of a shelter. Lower air requirements reduce the size of ducts and fans, and the power required to handle the air.

Figure 15 shows the heat absorbed per lb of air for various inlet water temperatures, and for various water flow rates with a constant volume of air flow. These data were obtained from a manufacturer of cooling towers.(27) The curves show that the exhaust ventilating air could be used to operate a cooling tower which would have sufficient capacity to reject the entire quantity of heat released by the occupants. A ventilating air rate of 3 cfm per person is equal to about 13 lb per hr. Assuming that the heat to be dissipated by the cooling tower is 600 Btu per hr per person, the heat to be handled per lb of air would be about 45 Btu. This could be dissipated using an inlet water temperature between 115 and 130 F. As shown by Figure 15, for a given heat absorption rate, lower water temperatures can be used with higher water flow rates. Lower water temperatures would result in higher efficiency of the refrigeration machine but the higher water rates would require larger pumps, piping, and power for pumping. A detailed analysis of given design conditions would show the most appropriate system to use with respect to the many factors which could be varied.

Figure 16 shows the cost of conventionally designed and operated cooling towers. The ratings are based on a wet-bulb air temperature of 78 F, a 10 degree cooling range, and 0.25 gpm of water and 25 cfm of air flow per 1,000 Btu per hr of heat rejection. These data are shown with relation to the heat dissipated by the evaporation of water and not with relation to the cooling effect in the shelter. The heat to be handled by the tower would be the sum of the cooling load and the power input to the refrigeration machine cooling the shelter. The power input depends upon the type and the efficiency of the refrigeration machine used as discussed in the "Refrigeration Devices" section of this report.

Cooling tower costs for typical installations are as follows:

	Per Cent of Total Cost	
Item	Small Tower	Large Tower
Tower, including casing, fill, and fan	20	35
Pump	15	10
Piping	27	30
Wiring	4	10
Starters	14	5
Miscellaneous	20	10

The piping costs were estimated on the basis of 200 ft of total piping. This would permit the tower to be located up to about 75 ft from the equipment using the tower water. Under some situations the tower might have to be located further away and these piping costs would not be realistic. For estimating installed piping costs, a water piping cost of about $1 per in. of diameter per ft of length should be used.

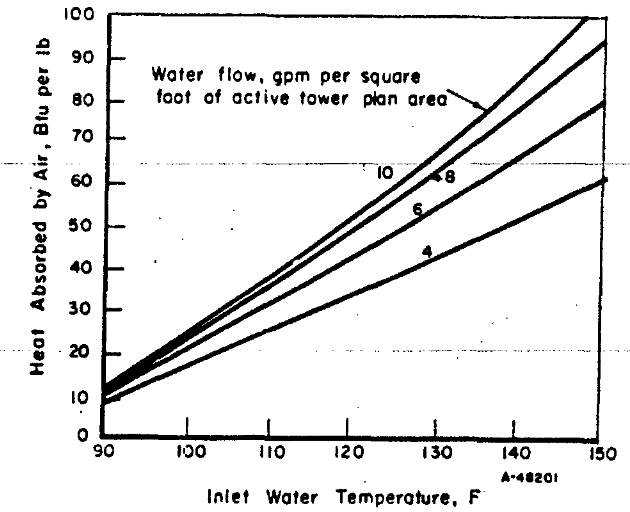


FIGURE 15. OPERATING PARAMETERS FOR PACKAGED MECHANICAL-DRAFT COOLING TOWER

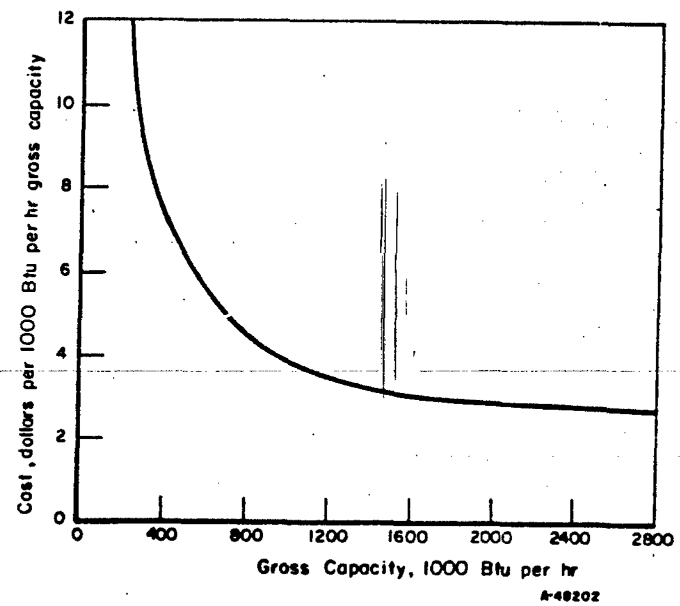


FIGURE 16. COST OF PACKAGE MECHANICAL-DRAFT COOLING TOWERS, EXTERIOR DESIGN

Cooling towers occupy about 0.5 cubic feet per 1,000 Btu per hr of capacity when operating at their design ratings.

VERTICAL-TUBE CONDENSER

A system of tubes or pipes extending from cooling equipment in a shelter into the atmosphere was considered as a heat-transfer device to utilize the air as a heat sink. It is technically feasible to use such tube surfaces as a condenser for a refrigeration machine, but the cost would be considerably greater than that for other applicable cooling methods. Discussed here are the heat-transfer characteristics and the costs of vertical-tube condensers.

Vertical tubes would receive nearly a minimum amount of solar radiation and, therefore, would provide the lowest condensing temperature for a refrigeration device. Exposed surfaces on a sunny day reach an equilibrium temperature above atmospheric temperature such that the heat loss to the atmosphere is equal to the heat absorbed from solar radiation. Condensing temperatures would have to be many degrees higher than the equilibrium temperatures to obtain condensing heat-transfer rates worthy of consideration.

The heat-transfer rate (convection and low temperature radiation) to the atmosphere from surfaces 25 to 75 F above ambient temperature with little or no wind is about 2 Btu per hr sq ft F.

The intensity of solar radiation at any location on the earth's surface varies widely from day to day depending upon atmospheric conditions. On a clear day, vertical surfaces in the sun are exposed for many hours to direct and diffuse radiation of intensities greater than 150 Btu per hr sq ft. Surfaces shaded from direct sunlight but open to the sky would be exposed to diffuse solar radiation at intensities of 20 to 30 Btu per hr sq ft. The quantity of radiation absorbed by a tube would be dependent upon the shading and the absorptivity of the tube surface. Table 7 shows the approximate differences between the atmospheric and the equilibrium temperatures which could be expected for tubes in the sun on a day with no wind.

Figure 17 shows the effect of condensing temperature on heat-transfer rates and on the length of 3-inch-OD tubes required to dissipate 1,000 Btu per hr at an ambient temperature of 95 F. In preparing the figure, it was assumed that each tube was standing free in the atmosphere and that its heat-transfer characteristics were unaffected by adjacent tubes.

Blast resistance is an attractive feature of a heavy-wall tube condenser cooled by the atmosphere. Therefore, any shading techniques or surface treatments to reflect solar radiation would have to be effective after exposure to blast effects. Surface treatments might easily be made ineffective by fire or by deposits of dirt and smoke.

A rule of thumb figure for cost of erected pipe is $1 per foot per inch of pipe diameter. At these prices the condenser cost would be more than $75 per 1,000 Btu per hr of heat rejected at a temperature of 130 F. This is as much as the cost of ice and its storage facility. Condensing temperatures higher than 130 F would require less pipe but more expensive refrigeration machinery.

TABLE 7. EQUILIBRIUM TEMPERATURES FOR VERTICAL TUBES IN THE ATMOSPHERE

Description of Tube and Exposure	Absorptivity to Solar Radiation	Equilibrium Temperature, deg F above ambient temperature
Iron tube, scaled surface, exposed to sunlight on one side	0.7 to 0.8	25 to 35
Iron tubes, scaled surface, shaded from direct sunlight, exposed to diffuse radiation	0.7 to 0.8	10 to 15
White or aluminum tube, exposed to sunlight on one side	0.3 to 0.4	10 to 15
White or aluminum tube, shaded from direct sunlight, exposed to diffuse radiation	0.3 to 0.4	5 to 10

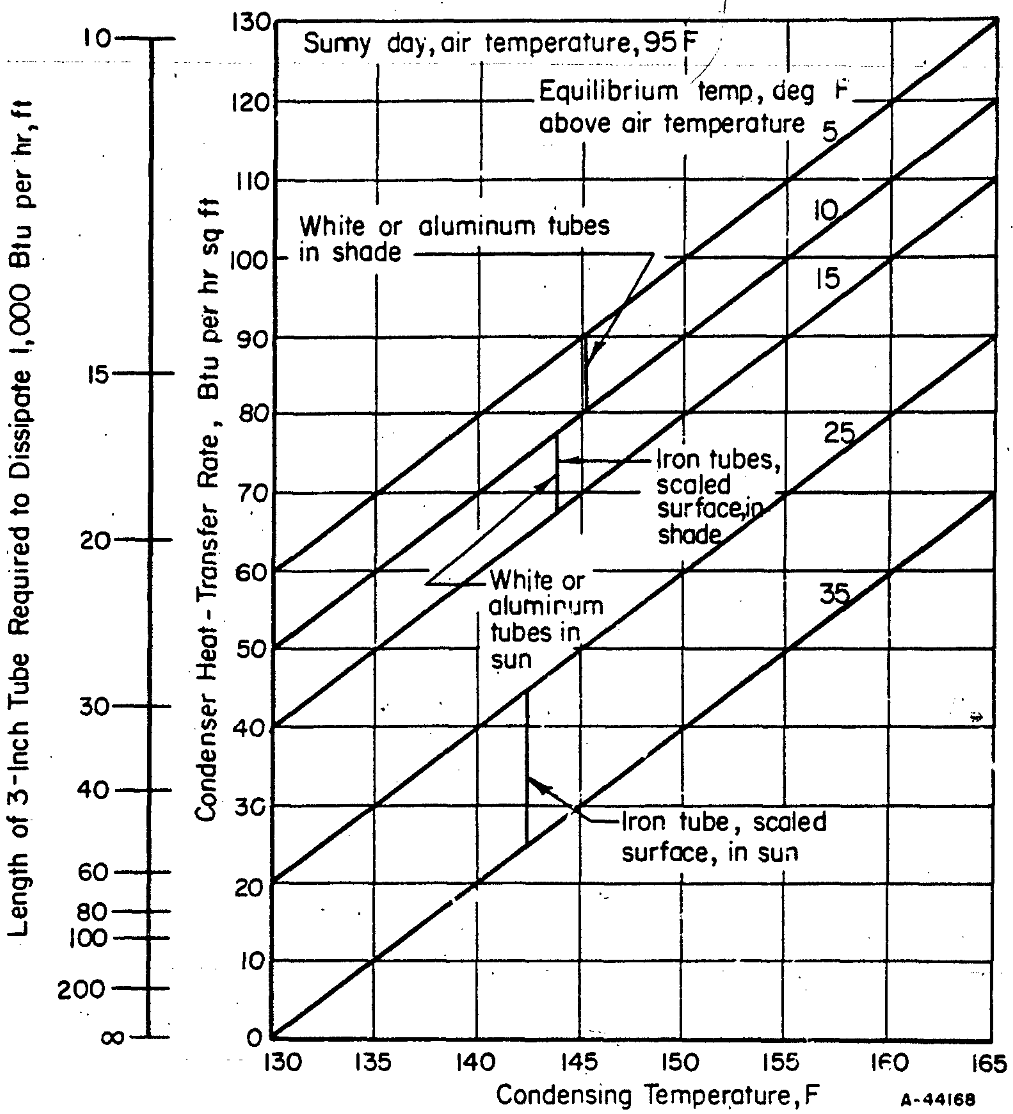


FIGURE 17. HEAT-TRANSFER RATES AND TUBE LENGTH FOR VERTICAL-TUBE CONDENSERS

QUIET SURFACE PONDS

A warm-water surface, exposed to the atmosphere, would be a practical means of dissipating heat from a refrigeration machine under some circumstances. This conclusion is based on the calculated heat transfer from water surfaces to the atmosphere and with consideration of the physical features of ponds, whether natural or artifical such as swimming pools. Of course, there would be no problem with an extremely large body of natural water such as a lake or river. In such cases, the water surfaces are so large that the heat added to the water would increase the water temperature only a few degrees. Also, the equilibrium water temperature would generally be below the atmospheric temperature. The use of ponds with water surfaces so small that the water temperature would be appreciably above the atmospheric temperature poses special problems. Discussed are the heat-transfer characteristics of exposed water surfaces, some of the practical aspects of using small ponds, and the cost of small ponds.

Heat from an exposed water surface at a temperature above the atmospheric temperature flows to the atmosphere by conduction, convection, radiation, and mass transfer of water vapor. Heat-transfer rates by these mechanisms are governed by the water temperature and the atmospheric conditions of wind, wet-bulb temperature, and dry-bulb temperature. The influencing factors are included in equations, which have been developed to calculate the rates of heat transfer. Experimental and operational data have verified the validity of the equations for use with water temperatures within a few degrees of the atmospheric temperature. Although the calculated results for water temperatures more than a few degrees above atmospheric temperature appear to be reasonable, no experimental data have been found to substantiate them. Figures 18 through 24 show the calculated heat-transfer characteristics for practical values of the governing variables.

Figure 18 shows the heat loss by evaporation in still air for various water temperatures, relative humidities, and dry-bulb air temperatures. As can be seen from the figure, the heat loss rate increases rapidly with increasing water temperature. Dry-bulb air temperature has very little effect especially at low relative humidities.

Figure 19 shows the effect of air velocity on evaporative heat loss expressed as a multiplier to be used in conjunction with Figure 18. From the curve it can be seen that even slight winds greatly increase the evaporation rate. For example, the evaporation rate with a 3-mile per hour wind is two times the evaporation rate with still air.

Figure 20 shows the convection heat loss in still air from an open water surface for various water and dry-bulb air temperatures. The heat-loss increases as the difference between the water temperature and air temperature increases. The increase is more than proportional to the temperature difference because the convection coefficient also increases with increasing temperature differences. The heat-transfer coefficients used to obtain the curves are the coefficients for a horizontal flat plate facing upward.

In Figure 21, the effect of wind velocity on the convective heat loss is shown. At higher temperature differences between the water and the air, the effect of wind is not as pronounced as it is at low temperature differences. This is because the still air convection heat losses shown in Figure 20 include air movement by natural convection;

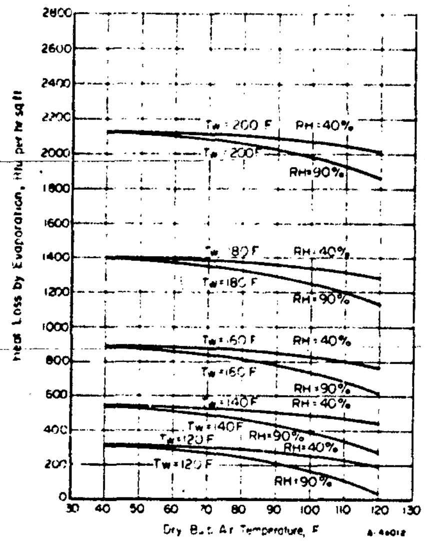


FIGURE 18. HEAT LOSS BY EVAPORATION IN STILL AIR FROM AN OPEN WATER TANK FOR VARIOUS WATER TEMPERATURES (Tw), RELATIVE HUMIDITIES (RH), AND DRY-BULB AIR TEMPERATURES

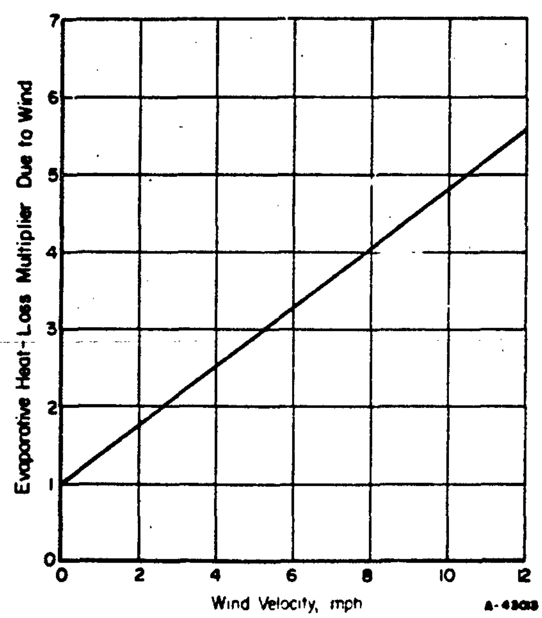


FIGURE 19. EVAPORATIVE HEAT-LOSS MULTIPLIER DUE TO WIND

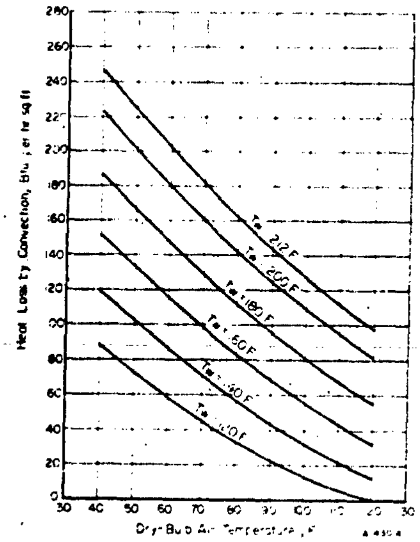


FIGURE 20. HEAT LOSS BY CONVECTION IN STILL AIR FROM AN OPEN WATER TANK FOR VARIOUS WATER TEMPERATURES (Tw) AND DRY-BULB AIR TEMPERATURES

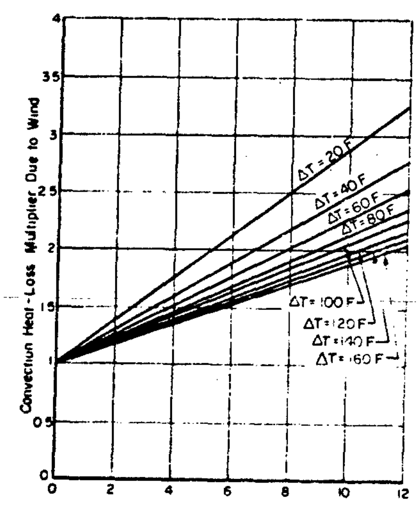


FIGURE 21. CONVECTION HEAT-LOSS MULTIPLIER DUE TO WIND FOR VARIOUS VALUES OF WATER-AIR TEMPERATURE DIFFERENCE (Δ T)

so, at any wind speed, the percentage increase in air movement is less at higher temperature differences. The values obtained in Figure 21 are multipliers to be used with the heat loss rates shown in Figure 20.

The heat gain by solar radiation varies with the time of year, the latitude, and atmospheric conditions. Although solar radiation is instantaneous, its effect on temperature depends upon the average radiation over a period of time and the thermal capacity of the object receiving the radiation.

Figure 22 shows the heat gain from solar radiation for an exposed pond as a function of time of year and latitude for the northern hemisphere. The heat gain is expressed as a 24-hour average and is based on a water absorptivity of 0.96. The values shown are maximums which would occur on clear days. Monthly average values would be lower, being 50 to 60 per cent of the given values for the eastern United States and 80 to 90 per cent for the arid southwest.

Figure 23 shows the heat loss by low temperature radiation from a water surface to the air for various air and water temperatures. The calculations were made with a water emissivity of 0.96 and assuming the sky to be a black body at atmospheric air temperature. This sky condition would be approached only on a high humidity, overcast day. On a clear day, the low-temperature radiation from the water might be as much as 85 Btu per hr sq ft. more than that shown in Figure 23.

The heat transfer to dry earth around an artificial or natural body of water would be so small that it can be neglected as a means of heat rejection. Heat-transfer rates would not exceed 10 Btu per hr sq ft. at any realistic temperature difference between the water and the dry earth.

Heat rejection from a water surface by all the various methods would be about 600 Btu per hr sq ft on summer days with water at a temperature of 120 F. Water at 120 F could be used to cool conventional mechanical vapor-compression refrigeration machines which are designed to be air cooled. With water at 150 F the heat rejection rate would be about 1500 Btu per hr sq ft. Refrigeration machines could be cooled with 150 F water but their design would differ somewhat from that used in conventional air-conditioning practice. With a 30-degree increase in water temperature, the water surface area required would be reduced by a factor of 2-1/2. However, this advantage would be offset by the requirement of a modified refrigeration machine which would be less efficient because of the larger temperature span over which it would operate. A detailed analysis of the specific situation would be required to determine the optimum system.

There are many ways in which an open-surface water pond could be utilized for cooling a refrigeration machine. However, in view of the many other possibilities, such as spray ponds, cooling towers, evaporative condensers, and air-cooled heat exchangers, it would seem that a pond would be used only because it was naturally available, or because it could be made blast resistant. A discussion of the effects of blast is beyond the scope of this report, but a few points are mentioned to aid in the evaluation of open-surface ponds.

Artificial ponds could be made with a deep basin like a swimming pool. The water would be stored in the pond during the standby period and then evaporated from this same reservoir when necessary. Points to consider for this type of construction are:

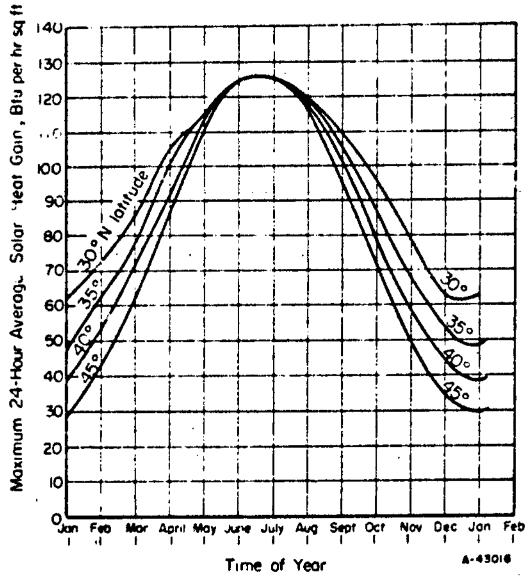


FIGURE 22. MAXIMUM 24-HOUR AVERAGE SOLAR HEAT GAIN FOR WATER WITH AN ABSORPTIVITY OF 0.96 AS A FUNCTION OF LATITUDE AND TIME OF YEAR

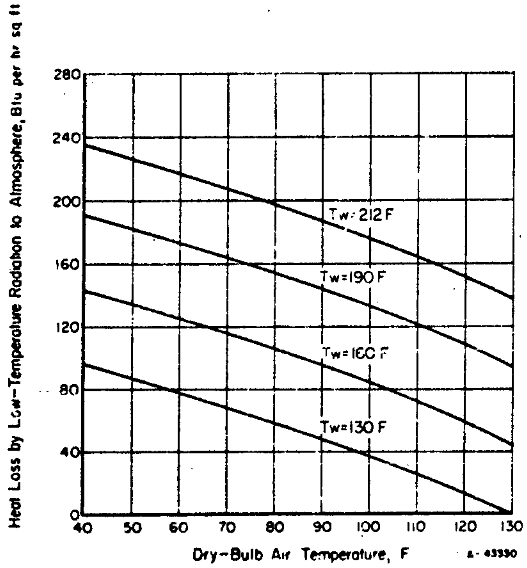


FIGURE 23. HEAT LOSS BY LOW-TEMPERATURE RADIATION TO THE ATMOSPHERE FROM AN OPEN WATER SURFACE FOR VARIOUS WATER TEMPERATURES (Tw), AND AIR TEMPERATURES

1. Water blown out of reservoir by blast and wind following detonation

2. Reduction in heat-transfer rates because of dust and debris on water surface

3. Reduction in heat-transfer rate with lowering of water level

4. Maintenance of reservoir during winter months in cold climates

5. Stand-by hazards associated with open ponds in populated areas.

An exposed water surface could be provided with a shallow-basin pond and an independent source of water. The basin could be a shallow concrete or steel structure to be used only for providing sufficient water surface and not for the storage of water. During operation, water from a separate source, either natural or artificial, would be pumped into the shallow basin to maintain the appropriate water level. As for any open-surface pond, debris on and above the water surface would reduce the heat transfer from the water. In other aspects the shallow pond would have the following advantages compared with the deep basin type:

1. Only a small portion of the total water supply would be lost due to blast

2. Controlled water level would assure the anticipated heat-transfer rate

3. Maintenance and safety problems would be less during the standby period.

It might appear that a cover over the pond would be the solution to many of the practical problems listed above. Certainly a cover would provide adequate protection during stand-by, but its effect on the reliability of the operation after a blast is questionable. In fact, it might be that the shelter cooling system and, therefore, the pond would be in use at the time of the blast. Under such circumstances even ponds equipped with covers would be open and could receive debris and have the water blown out.

The cost of artificial ponds would vary considerably with size, materials of construction, and techniques used in construction. Swimming pools are constructed of various combinations of concrete, steel, vinyl sheets, and steel-reinforced fiberglass. Any of these construction materials would be satisfactory for a water reservoir. The major difference between them being the maintenance required during the standby and the resistance to blast damage. Concrete pools subjected to freezing and thawing crack, and steel ponds require occasional painting to prevent rusting. Structurally, the concrete and steel ponds are stronger than the other types and might be more desirable from the standpoint of blast resistance.

Figure 24 shows the approximate installed costs for deep-basin water ponds with construction similar to that of swimming pools. These costs are typical for ponds installed with their surface at grade level and with water depths of about 4 feet.

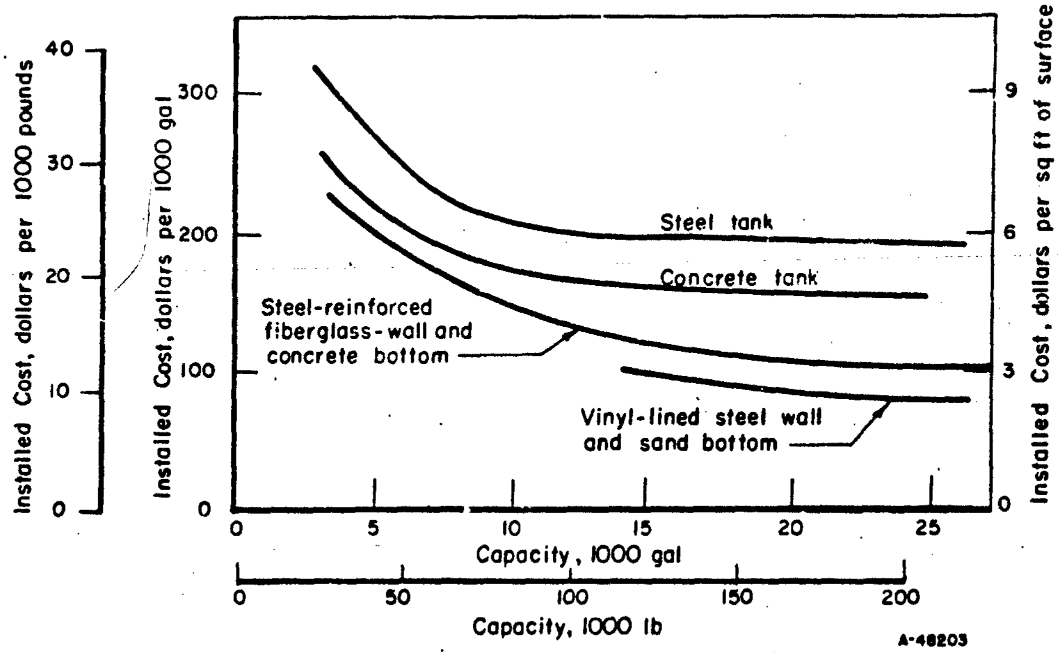


FIGURE 24. INSTALLED COST OF OPEN WATER-STORAGE TANKS (TANKS FOUR FEET DEEP WITH WATER SURFACE AT GROUND LEVEL)

Shallow basin ponds could be constructed with the bottom sloping up at the edges to provide the proper depth. The bottom itself might also be sloped to provide the desired water flow between the inlets and outlets. With relatively small slopes at the edges and over the bottom, the construction would be essentially similar to a water-tight slab. Costs for such ponds would vary from a few cents per square foot for sand-covered vinyl to the following for concrete over a gravel fill:

Thickness of Concrete Slab, inches	Cost, \$ per sq ft
4	0.45
6	0.57
8	0.69
10	0.81
12	0.93

The cost of shallow-basin ponds, on the basis of surface area, would be much lower than the cost of a deep-basin pond excluding the cost of a separate water storage facility. However, to be a comparable facility, water storage cost would have to be added to the cost of the shallow-basin pond. Conventional underground blast-resistant steel or concrete water storage tanks cost as much as \$1,000 per 1,000 gallons. With the water storage cost added to that of the shallow-basin pond, there is no conclusive cost advantage for either the shallow pond or the deep pond.

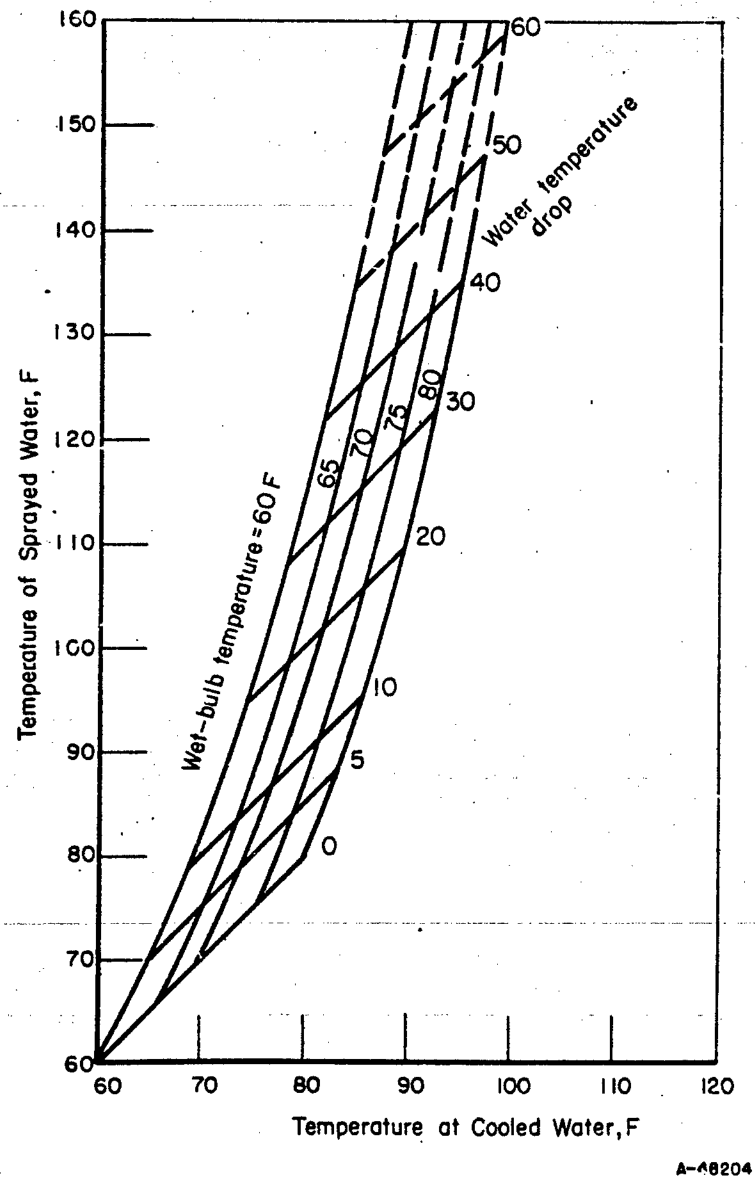


FIGURE 25. TYPICAL SPRAY-POND TEMPERATURE CHARACTERISTICS

SPRAY PONDS

For heat-dissipation capacity equal to that of a quiet surface pond, spray ponds would require only one-third to one-sixth the area. They would be considerably more blast resistant than cooling towers or evaporative condensers located above ground, which also utilize the latent heat of vaporization of water.

The cooling capacity of spray ponds is dependent upon the volume of the atmosphere swept by the water spray. Sprays in typical ponds now in use rise 12 feet above the pond surface from nozzles located 2 to 6 feet above the pond surface. The nozzles operate at water pressures of 4 to 10 psig with 5 to 7 psig being most common. Nozzles usually have a water capacity of 35 to 50 gpm and are located in groups of 4 to 6 nozzles per 12 feet of pipe length. The distance beween rows of nozzles is from 13 to 38 feet with 25 feet being most common. The water spray from each nozzle will strike the surface of the pond in a circular pattern having a radius of about 10 feet.

Figure 25 shows the relation between inlet and outlet water temperatures and air wet-bulb temperature for a typical spray pond installation. Other operating characteristics of conventional installations are:

Inlet water temperature - 20 to 25 F above wet-bulb temperature
Pond water temperature - 10 F above wet-bulb temperature
Heat dissipation rate - 1800 to 3600 Btu per hr sq ft of surface
Spray loading - 120 to 240 lb per hr sq ft of surface
Evaporation rate - roughly 1 per cent of water sprayed
Drift loss - 1 to 3 per cent of water sprayed.

Evaporation and drift losses would be especially significant if the natural source of water was limited or if stored water were used. The drift losses listed are typical for nozzles 25 to 35 feet from the edge of ponds without windbreaks, and for nozzles 12 to 20 feet from the edge of ponds protected by louvered fences 12 feet high.

The power required to spray the water would be less than 3 per cent of the total power required to operate a refrigeration cooling system. The cost of spray ponds would be similar to that shown in Figure 24 for open surface water ponds. The additional cost for piping and nozzles for the spray system would be about $1 per sq ft of pond surface.

AIR-HANDLING UNITS

An air-handling unit is an air-liquid heat exchanger consisting of a heat-transfer coil, casing, blower, and controls. Air-handling units are the most practical and the economical solution for cooling areas remote from the supply of chilled liquid. In general, the power requirements would be less and the pipe diameters would be smaller for distributing chilled liquid to remote areas of a shelter than for distributing conditioned air.

The installation and the use of air-handling units in shelters would be the same as for industrial and residential purposes. Therefore, no special skills would be required of the mechanical contractors who would design and erect this portion of a cooling system.

Years of design and practical experience have established optimum configurations of air-handling units from the standpoints of both cost and operation. The finned-tubed banks generally are made up of 1/2-in. or 5/8-in. tubes with 8 to 14 fins per inch of tube length. The tubes may be connected with various header arrangements to provide the required flow paths. Face air velocities between 400 and 600 feet per minute are used for most installations. These velocities are limited by the tendency to blow condensate through the unit and by noise and pressure drop.

The heat-transfer capacities of air-handling units are shown in manufacturers' catalogues for given temperatures and relative humidities. A common set of rating conditions are: air entering at 80 F with relative humidity of 50 per cent, air leaving at 55 F and nearly saturated, and 45 F liquid in the coil. These conditions are typical in comfort air-conditioning practice, but shelter cooling conditions could be quite different. The heat-transfer capacity of a given coil varies significantly with the temperature difference between the air and the liquid in the coil, and also with the relative humidity of the air. Therefore, the air-handling unit should be selected on the basis of the conditions under which it is to operate.

Figure 26 shows the approximate heat-transfer characteristics of air-handling units for various temperature differences and relative humidities of the inlet air. At a given temperature difference, the heat-transfer rate with 90 per cent relative humidity air may be 1.5 times the rate with 50 per cent relative humidity air. The higher rate is the result of higher heat-transfer coefficient for the water vapor condensing upon the surfaces than for the convective cooling of dry air. When shelters require cooling, the relative humidity will probably be high and, therefore, the air-handling units would probably be smaller than those used for conventional comfort air conditioning.

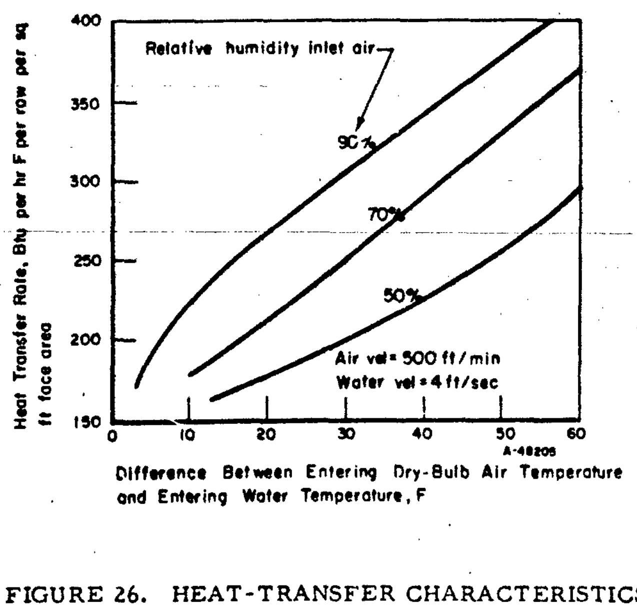


FIGURE 26. HEAT-TRANSFER CHARACTERISTICS OF AIR-HANDLING UNITS

Air-handling units with inlet water at temperatures up to 70 F could maintain an effective temperature of 85 F in a shelter. The air-handling unit design would depend upon the temperature of the water supply and the desired temperature change of the water and air passing through the heat exchanger. Numerous combinations of flows and temperature changes could be used satisfactorily. The most appropriate combination would depend primarily upon water temperature and pumping power requirements.

Table 8 shows typical operating characteristics of various units based on finned-tube heat-exchanger data taken from performance tables published by manufacturers. The published data were extrapolated to include temperatures and humidities corresponding to an effective temperature of 85 F to estimate the effect of condensation on the heat-transfer rates. The results may be in error somewhat, but more accurate values would require test data on heat-exchanger performance under the desired operating conditions.

The coefficients of performance shown in the table were calculated as the net cooling capacity (100,000 Btu per hr minus power to drive system) divided by the power required to drive the system. For these examples the following assumptions were made: blower efficiency, 60 per cent; pump efficiency, 70 per cent; shelter effective temperature, 85 F; and saturated air leaving the heat exchanger. Power is shown for pumping water through the heat exchanger. Also shown is the pumping power with an additional pressure loss of 100 feet of water to show the effect of well depth and pipe-friction loss on power requirements.

All equipment required for disposing of excess shelter heat with naturally available cool water could be manually operated. The pumps and blowers would not require sensitive speed control and their power requirements would be low enough for them to be driven manually.

The cost of air-handling units varies considerably with the supplied accessories. The basic unit consists of a casing, blower, coil, motor, and drive. In addition to these, filter box, temperature control, and face and by-pass dampers are available. The accessories which could be advantageously used would depend upon other features of the cooling system. If air circulation within the shelter were desirable, even though artificial cooling were not, this circulation could be provided with the blower of a unit equipped with by-pass dampers. It seems unlikely that the filter box would be needed. However, filters do keep the coils clean and free from plugging. Temperature control is another option which may or may not be required on each air-handling unit. This would depend entirely upon the control technique used for the cooling system.

Table 9 shows typical features of air-handling units and the breakdown of their costs. The costs include all expenses which would be incurred in putting the unit into operation. They include contractor's cost, shipping and handling, miscellaneous supplies, and contractor's profit.

Figure 27 shows a plot of the cost of the basic air-handling unit which includes the casing, blower, motor and drive, wiring, starter, and coil. Also shown is the cost of a unit with dampers, filters, and temperature controls. These costs were used to estimate the total costs of cooling systems which are described in the "Cooling Systems" section of this report.

TABLE 8. TYPICAL WATER-COOLED HEAT-EXCHANGER PERFORMANCE FOR ABSORBING 100,000 BTU PER HR

Heat Exchanger		Shorter Flow			Power, hp			Coefficient of Performance	
No. of Rows	Face Area, sq ft	Temperature, F Inlet	Temperature, F Outlet	Quantity, gal/min	Pump Water Through Heat Exchanger	Pump Water Through Heat Exchanger Plus 100-Ft Head of Water	Drive Air Blower	Water Through Heat Exchanger	Water Through Heat Exchanger Plus 100-Ft Head of Water
				Air Flow, 4300 Cfm; Temperature Change, 10 F					
1	7.6	40	50	18.7	0.009	0.685	0.119	307	48
1	8.8	40	60	9.4	0.005	0.343	0.131	289	82
1	10.3	40	70	6.2	0.004	0.228	0.151	253	103
1	11.9	50	60	18.7	0.014	0.690	0.160	225	45
2	7.2	50	70	9.4	0.009	0.346	0.187	200	73
2	10.4	60	70	18.7	0.025	0.700	0.243	146	41
3	9.5	60	80	9.4	0.017	0.345	0.324	114	57
6	8.9	70	80	18.7	0.063	0.740	0.546	64	30
				Air Flow, 2100 Cfm; Temperature Change, 20 F					
2	4.3	40	50	18.7	0.021	0.697	0.116	287	47
2	4.9	40	60	9.4	0.012	0.350	0.130	277	81
3	3.9	40	70	6.2	0.009	0.233	0.152	242	101
3	4.8	50	60	18.7	0.035	0.710	0.172	189	43
4	4.5	50	70	9.4	0.022	0.360	0.211	168	68
6	4.5	60	70	18.7	0.065	0.741	0.293	109	37
9	4.5	60	80	9.4	0.048	0.386	0.423	83	47

TABLE 9. DESCRIPTIONS AND COSTS OF TYPICAL AIR-HANDLING UNITS

Unit cooling capacity,* Btu per hr	60,000	120,000	240,000	360,000
Air flow, cfm	2,000	4,000	8,000	12,000
Coil face area, sq ft	3	7.5	13.7	20.6
Fanpower, hp	1	1.5	3	5
Space requirement with filters and dampers	0.5 cu ft per 1,000 Btu per hr			
Cost Breakdown, Total Installed Cost, Dollars				
Casing and blower	600	780	1,110	1,600
Motor and drive	80	100	160	230
Wiring and starter	150	150	230	300
Coil, six row	490	820	1,250	1,760
Face and bypass dampers	260	340	460	570
Temperature control	360	350	360	360
Filter box	100	180	320	380

*Capacity with air entering at 60 F with relative humidity of 50 per cent, air leaving at 55 F nearly saturated, liquid in coil 45 F.

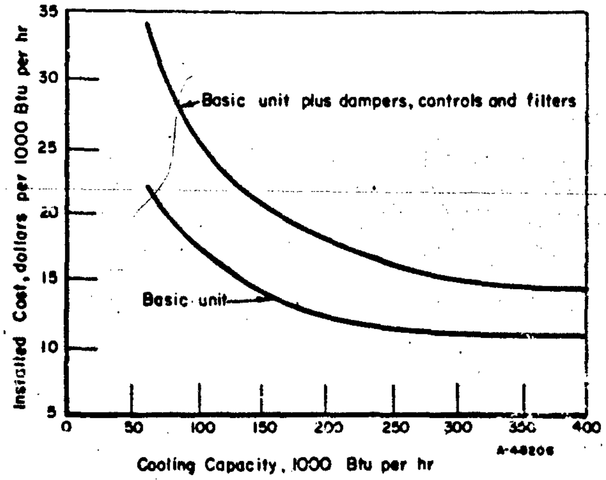


FIGURE 27. INSTALLED COST OF AIR-HANDLING UNITS, SIX-ROW COIL

AIR-COOLED CONDENSERS

Air-cooled condensers must be used for refrigeration devices when no water is available. Conventional units would be suitable for use with shelter cooling systems. Presented here are typical costs and operating features of conventional forced-air condensers.

A few natural-draft air-cooled condensers have been installed to reduce operating costs at the expense of higher capital cost. These would not be applicable for the shelter situation which favors the short-term, higher operating cost with a lower initial cost.

Typical features and nominal rating temperatures for air-cooled condensers used with mechanical-vapor-compression machines are:

Condensing temperature - 125 F
Entering air temperature - 95 F
Leaving air temperature - 110 to 115 F
Air flow - 4.0 to 4.5 lb per 1,000 Btu
Space requirement - 1/2 cu ft per 1,000 Btu
Power requirement - 1 hp per 150,000 to 200,000 Btu per hr.

Figure 28 shows the installed costs of conventional air-cooled condensers with air dampers, head-pressure valve, 100 ft of refrigerant piping on each line, fan, and motor.

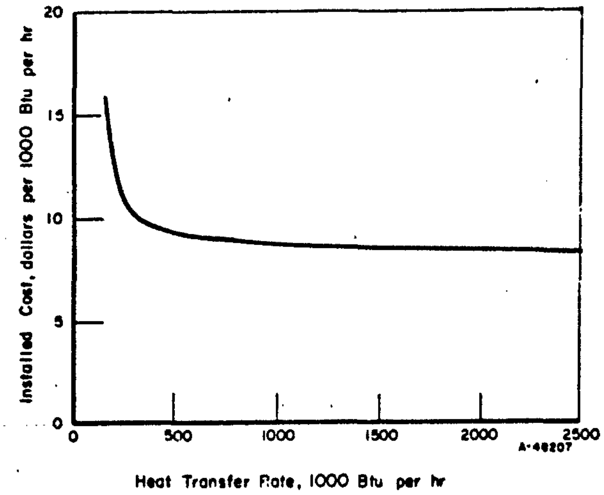


FIGURE 28. COST OF CONVENTIONAL AIR-COOLED REFRIGERANT CONDENSERS

EVAPORATIVE CONDENSERS

All the heat released by shelter occupants could be rejected with the exhaust ventilating air by using a cooling system consisting of an evaporative condenser and a conventional mechanical-vapor-compression refrigeration machine. Such a condenser would have to operate at a temperature 10 to 20 F above the temperatures normally used for evaporative condensers with an unlimited source of air. Conventional mechanical refrigeration machines could operate at this higher condenser temperature. By using an evaporative condenser, the entire cooling system could be conveniently housed within the protective walls of the shelter.

The effectiveness of evaporative condensers is directly related to the heat content of the air passing through it.

Figure 29 shows the total heat content of mixtures of air and water-vapor. The heat rejected to the exhaust-ventilating air by an evaporative condenser would be the difference between the heat content of the air leaving condenser and the heat content of air at an effective temperature of 85 F. Two of many possible operating conditions are:

Air flow, cfm per person	3	5
Heat to be absorbed, Btu per hr per person	600	600
Heat to be absorbed per lb of dry air, Btu	44.5	36.7
Temperature air entering condenser, F	92	92
Temperature air leaving condenser, F		
at 80 per cent relative humidity	115	106
at 60 per cent relative humidity	122	115

Condensing temperature – 10 to 15 F above the temperature of air leaving the condenser.

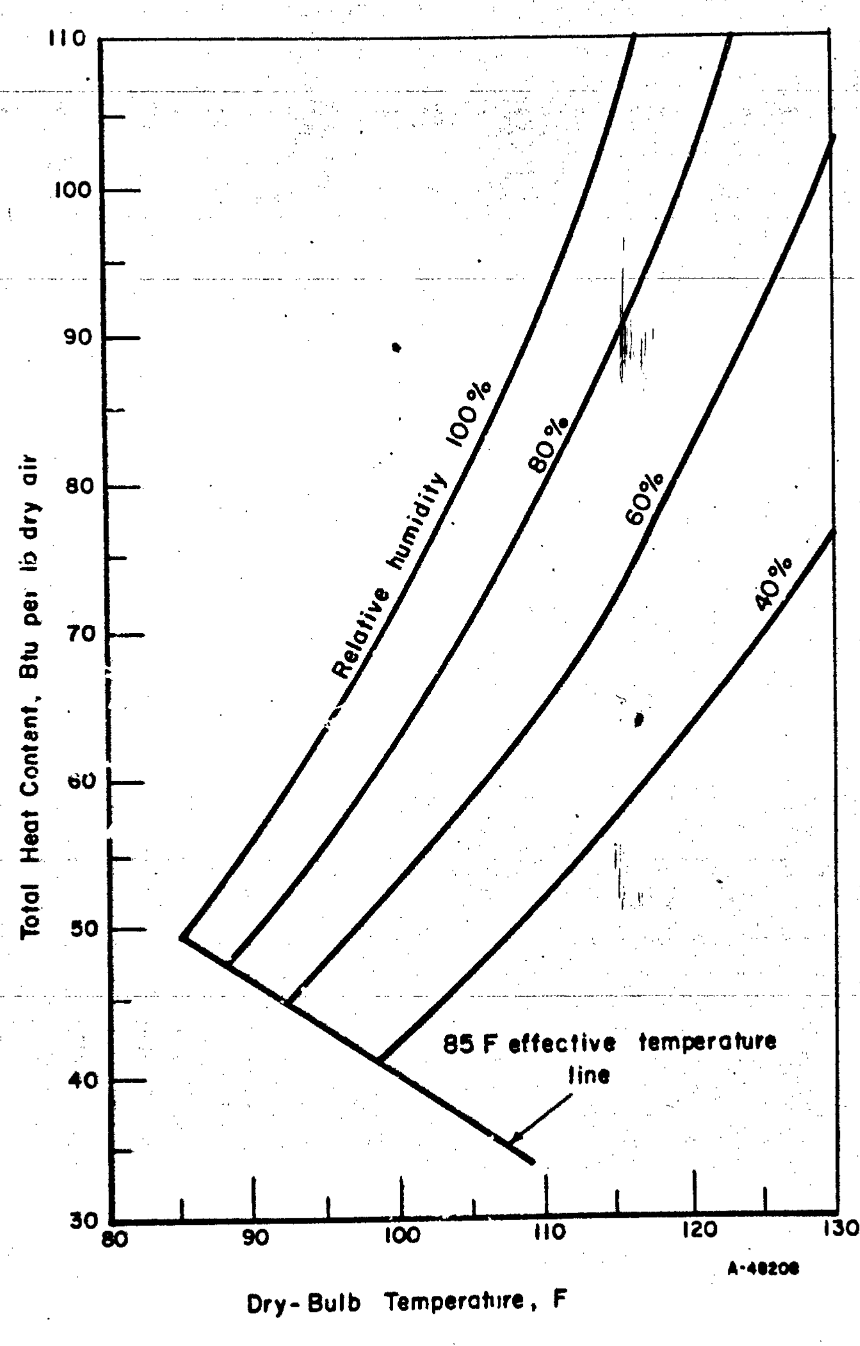


FIGURE 29. TOTAL HEAT CONTENT OF AIR AND WATER-VAPOR MIXTURES

No experimental or operating data are available to show the degree of saturation which could be obtained with various designs of evaporative condensers operating at these higher temperatures. Schutte and Koerting Company have done some work in this area, but they had no data for release at the time this report was written.[28] Conventional evaporative coolers operate with 3/8 to 1/2 inch of water pressure drop over a 3 to 4 row coil and achieve 80 to 90 per cent saturation of the air. These units have a heat-rejection rate of about 10 Btu per pound of air. It is estimated that at the higher temperatures a conventional 3 to 4 row coil would provide 60 per cent saturation and a 6 to 10 row coil with a pressure drop of about 1 inch of water would achieve 80 per cent saturation. Data are needed to verify the performance of the 6 to 10 row coil.

Figure 30 shows the installed cost of evaporative cooled condensers with fans, water pumps, motors, air dampers, 200 feet of refrigerant piping, and minor accessories required for a complete installation. Other requirements to operate the units are:

<u>Conventional Units or High-Temperature Units for 60 Per Cent Saturation</u>

1 hp per 150,000 to 225,000 Btu per hr
1 to 0.6 cubic feet of space per 1,000 Btu per hr of capacity

<u>High-Temperature Units for 80 Per Cent Saturation</u>

2 hp per 150,000 to 225,000 Btu per hr
1.2 to 0.8 cubic feet of space per 1,000 Btu per hr of capacity.

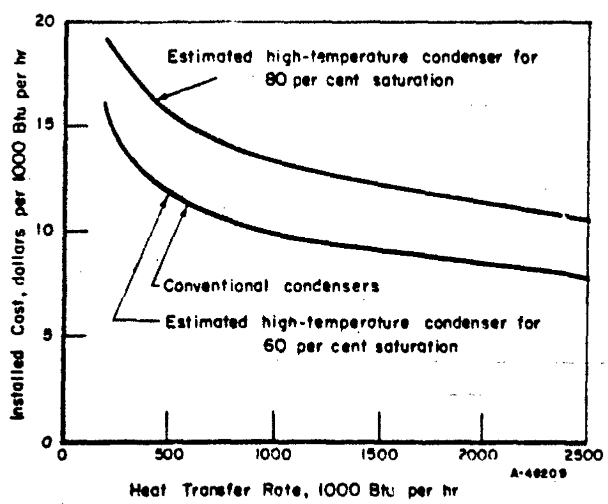


FIGURE 30. COST OF CONVENTIONAL AND HIGH-TEMPERATURE EVAPORATIVE CONDENSERS

Evaporative condensers would be most effectively used by installing them in the shelter exhaust air duct. It seems unlikely that a conventional evaporative condenser would be installed in the atmosphere outside of a shelter. Long refrigerant lines and the increased possibility of damage with resultant loss of refrigerant would be significant disadvantages. Rather than using an evaporative condenser, a more practical solution would be the use of a cooling tower in the atmosphere and a water-cooled condenser in the shelter with the refrigeration machine.

EVAPORATIVE COOLERS

Evaporative cooling of shelter ventilating air can more than double the heat-absorption capacity of the air. Evaporative cooling is used commercially in hot, dry areas, but its use is diminishing because of its inadequacies in controlling humidity. However, with an effective temperature limit of 85 F, evaporative cooling could be used for shelter cooling in any area of the United States.

The function of evaporative coolers is to adiabatically saturate the air with vapor from evaporating water. The heat required to evaporate water to provide the vapor decreases the sensible heat of the air with an equivalent increase in the latent heat. This process requires intimate contact between the water and the air. Two types of evaporative coolers are available, both of which expose large water surface areas to the air. These are:

1. Spray type
2. Wetted mat type.

The degree of saturation of the air depends upon the nature of its contact with the water in the liquid phase. With sufficient time and flow turbulence, complete saturation can be approached. However, the equipment requirements to do this are impractical; consequently a compromise involving partial saturation and less complex equipment is ordinarily made.

The operation of evaporative coolers can most easily be shown with the aid of a psychrometric chart.

Figure 31 is a psychrometric chart with 80, 85, and 90 effective temperature lines superimposed. Evaporative cooling of the air and heat absorption from the shelter would follow a process ad indicated by Points 1, 2, and 3. Point 1 corresponds to the temperature and the humidity of the air entering the evaporative cooler. As water evaporates into the air the process continues along the Line 1-2 to Point 2 with essentially no change in total heat of the air-vapor mixture. The sensible heat of the air is reduced with a decrease in dry bulb temperature while the latent heat and moisture content of the air increases. The evaporative cooling process shown is for the case where the inlet water temperature is equal to the wet bulb temperature of the inlet air. Corrections must be included in calculations if the inlet water temperature does not equal the inlet air wet bulb temperature.

Point 2 shows the properties of the air leaving the evaporative cooler. The degree of saturation depends upon the evaporative cooler design and its operation. The relative humidity of the air increases in the evaporative cooler as they dry bulb air temperature

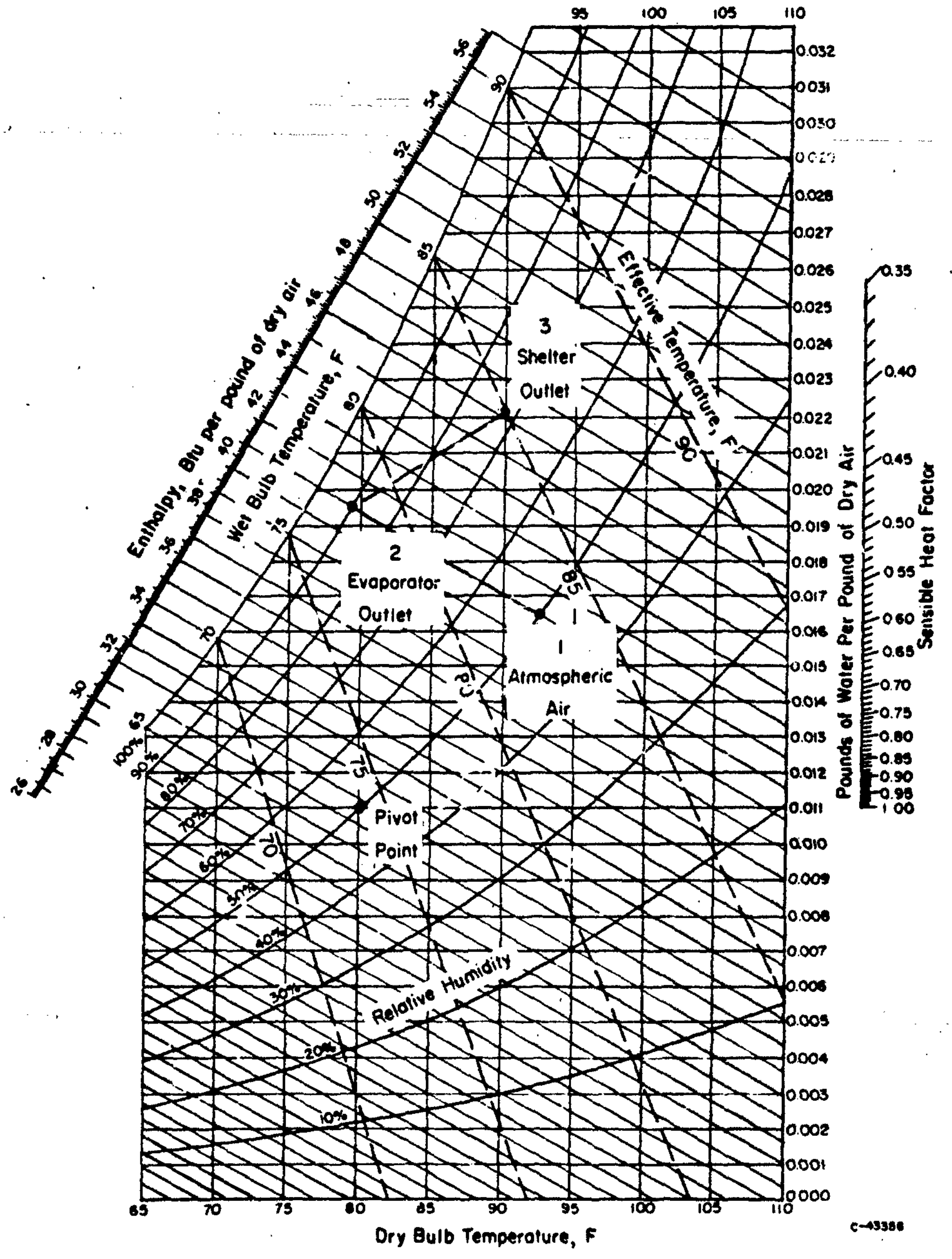


FIGURE 31. PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE LINES AND PROCESS FOR EVAPORATIVE COOLING

decreases. Relative humidities above 90 per cent are easily obtained. High relative humidities are desired to produce more cooling per pound of air but are not desirable from the standpoint of condensation which may occur on shelter walls or other surfaces which are below the dewpoint temperature.

From Point 2 to Point 3 the air absorbs heat from the shelter. The slope of this line depends upon the ratio of sensible to latent heat which must be absorbed from the shelter. For humans the sensible heat release is determined by the dry bulb temperature. For this example, sensible heat release equal to 47 per cent of the total was used to draw the line shown on the chart. The end point, Point 3, is established by the effective temperature to be maintained in the shelter.

The amount of heat absorbed from the shelter by a pound of air depends upon the enthalpy increase between Point 2 and Point 3. From the chart it is apparent that maximum heat absorption would occur with air leaving the shelter at 100 per cent relative humidity. However, with a sensible heat factor of 0.47, the relative humidity of the cooling air decreases with increasing temperature and therefore, would not approach 100 per cent humidity. Saturation at the shelter outlet can be approached by successive stages of evaporative cooling of the air within the shelter or by removal of some sensible heat by other means.

The heat absorbed by shelter walls would influence the humidity in the shelter and the effectiveness of evaporative cooling. The heat absorbed would be sensible heat (when relative humidities are below 100 per cent) thereby reducing the amount of sensible heat to be removed by the evaporatively cooling. Essentially, this would be the same as decreasing the sensible heat factor. As the sensible heat factor decreases, the relative humidity and enthalpy at Point 3 increase, which results in more heat being removed from the shelter per pound of air. With an effective sensible heat factor of about 0.25 heat absorption proceeds at constant relative humidity. While heat flow through shelter walls is not within the scope of this project it should not be neglected in the analysis of evaporative cooling systems.

The efficiency of evaporative coolers is defined as:

$$\frac{\text{Inlet dry-bulb temperature - Outlet dry-bulb temperature}}{\text{Inlet dry-bulb temperature - Inlet wet-bulb temperature}}.$$

Figure 32 shows the air flow rates required to remove 600 Btu per hr of heat from a shelter employing an evaporative cooler as a function of geographical areas in the continental United States. For comparison, the air flow rates required for direct air cooling are shown in Figure 33.

By using evaporative cooling the ventilation rate could be reduced to about one-half that required with direct air cooling in the west and north; to about one-third in the south and southeast.

Following are brief descriptions of the two most common types of evaporative coolers.

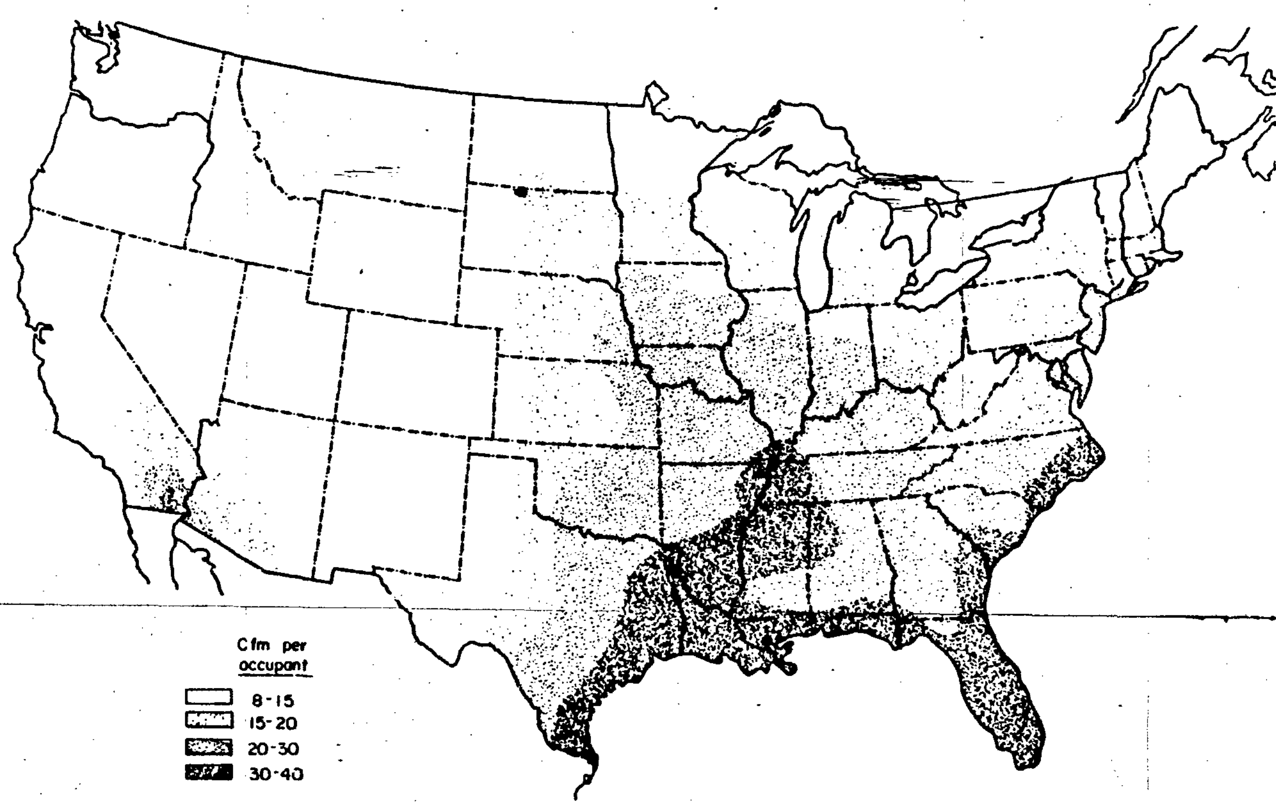


FIGURE 32. SHELTER VENTILATION RATES REQUIRED WITH EVAPORATIVE COOLING

Based on:

600 Btu per hr per occupant.

Design wet bulb temperature exceeded less than 5 per cent of the year.

Design dry bulb temperature exceeded less than 5 per cent of the year, June through September, air leaving evaporative cooler at 90 per cent relative humidity.

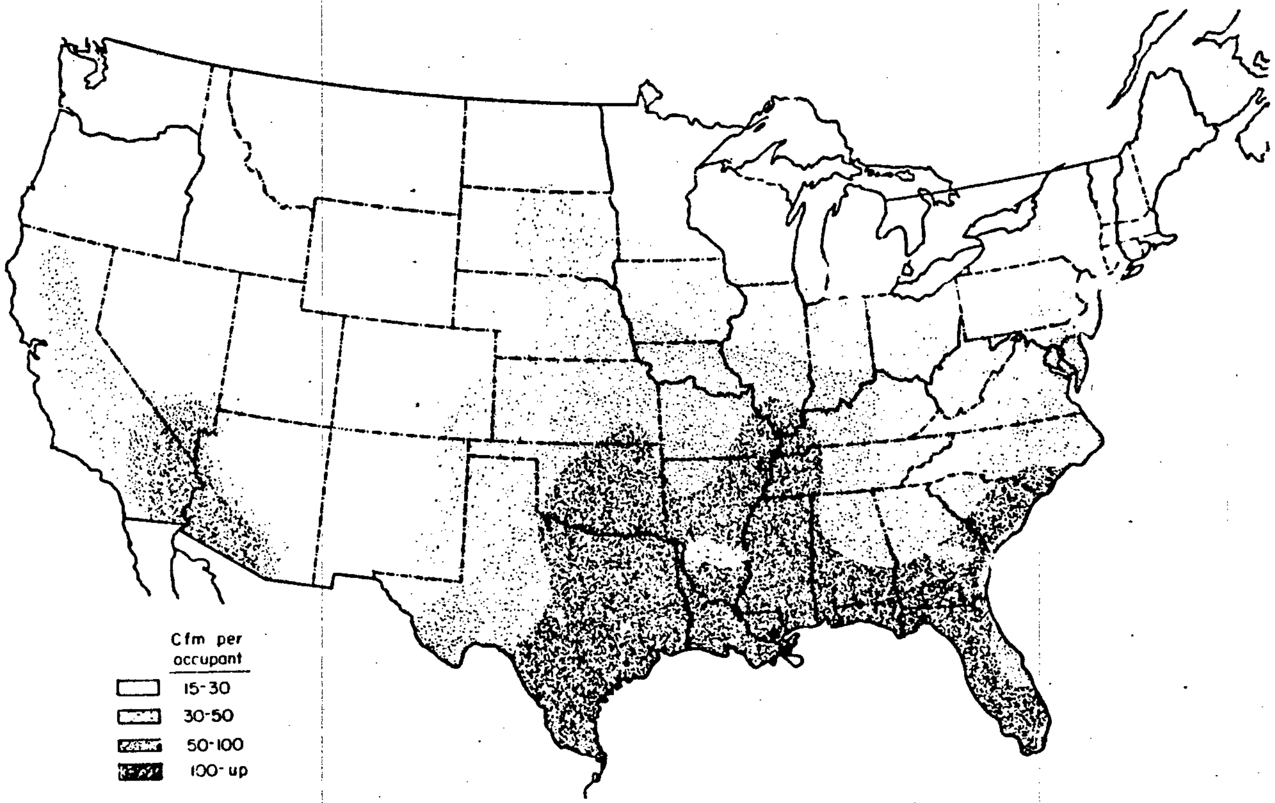


FIGURE 33. SHELTER VENTILATION RATES REQUIRED WITH DIRECT AIR COOLING

Based on:
600 Btu per hr per occupant.
Design wet bulb temperatures exceeded less than 5 per cent of year.
Design dry bulb temperatures exceeded less than 5 per cent of time, June through September.

Spray Type

Spray type evaporative coolers consist of a casing, inlet baffles for air-flow distribution, one concurrent spray system, one countercurrent spray system, and water-flooded mist eliminators. The casings in a typical design are about nearly / feet long in the direction of air flow with face areas to correspond to the desired capacity. The main features and operating characteristics of this typical design are:

Spray water - 10 gpm per 1,000 cfm of air at 20 psi

Mist eliminator water - 4 gpm per foot of width at 10 psi

Air flow - 500 feet per minute face velocity, 0.30 inch of water pressure drop

Space requirement - 13 to 14 cubic feet per 1,000 cfm of air

Evaporative efficiency - 90 per cent.

The costs of the spray-type coolers of various capacities are shown in Figure 34.

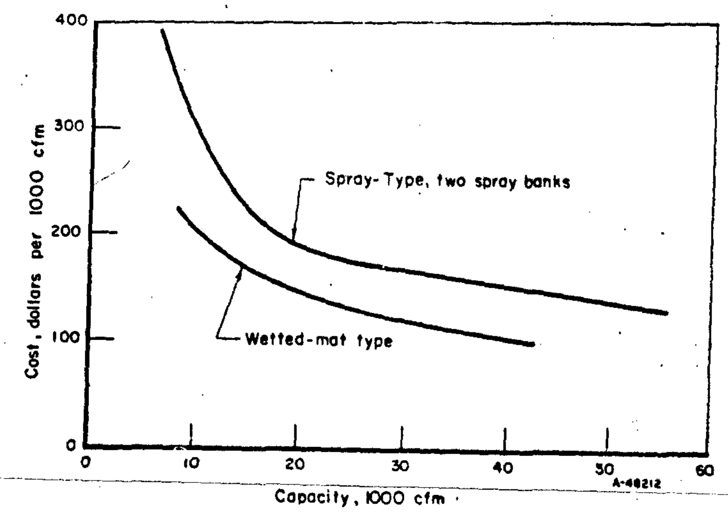


FIGURE 34. INSTALLED COST OF EVAPORATIVE COOLERS INCLUDING ELECTRIC PUMPS; BLOWERS NOT INCLUDED

Wetted Mat Type

Wetted-mat-type evaporative coolers consist of a casing, inlet baffles for air-flow distribution, a countercurrent spray system, and a fine fiber mat. Because the mat is an effective mist eliminator a finer spray can be used in this type than can be used in spray type. The finer spray increases the water surface area in contact with the air and therefore, only one spray system is used. In operation part of the water spray evaporates into the air and the remainder is carried into and trapped by the mat, which provides additional wetted surface for evaporation.

A typical unit is 3-1/2 feet long in the direction of air flow with face areas to correspond to the desired capacity. Listed below are other features of this typical design.

Spray water - 0.20 gpm per 1,000 cfm air at 40 psi

Air flow - 400 feet per minute face velocity, 0.30 inch of water pressure drop

Space requirement - 8.7 cubic feet per 1,000 cfm of air

Evaporative efficiency - 80 per cent.

The wetted mat would act as a filter for removing particles from the air. No work was done to determine if it could be substituted for any of the filters used for cleaning shelter ventilating air. However, the physical features of the mats indicate it would be an excellent prefilter.

Figure 34 also shows the cost of wetted-mat surface type evaporative coolers.

DEHUMIDIFIERS

While water-cooled chemical dehumidifiers could be used to dehumidify the shelter ventilating air, a more attractive use is in conjunction with an evaporative cooler to remove both sensible and latent heat. Chemical dehumidifiers are used commercially for humidity control in industrial plants and in hospitals. The lithium chloride, which is the material most commonly used in chemical dehumidifiers, also removes airborne bacteria and mold spores. This is advantageous in hospital operating rooms, food-processing plants, and may be desirable for shelters. Other desirable features of either a dehumidifier or a dehumidifier-evaporative-cooler combination are:

1. Heat can be rejected to water at temperatures up to at least 90 F
2. Equipment is not pressurized, except for the boiler which supplies steam for regeneration
3. Lithium chloride is neither toxic nor corrosive and has a low vapor pressure
4. Equipment has few moving parts which tends to decrease maintenance requirements and increase reliability.

Chemical dehumidifiers are relatively large, must be assembled on the site, require a source of steam and atmospheric air for regeneration, and need water for cooling. The amount of moisture a given size dehumidifier can remove depends upon the temperature and flow rate of the cooling water, the quantity of moisture in the air, and the surface area of the absorbing lithium chloride. Consequently, equipment performance cannot be generalized making it necessary to carry out a design and cost analysis for each specific installation. This has been done in the following example. For this example, inlet air at 95 F dry bulb and 79 F wet bulb (85 F effectiv temperature) was selected.

Figure 35 is a schematic drawing of a typical lithium bromide dehumidifier showing the operating characteristics of the system for the inlet conditions selected.

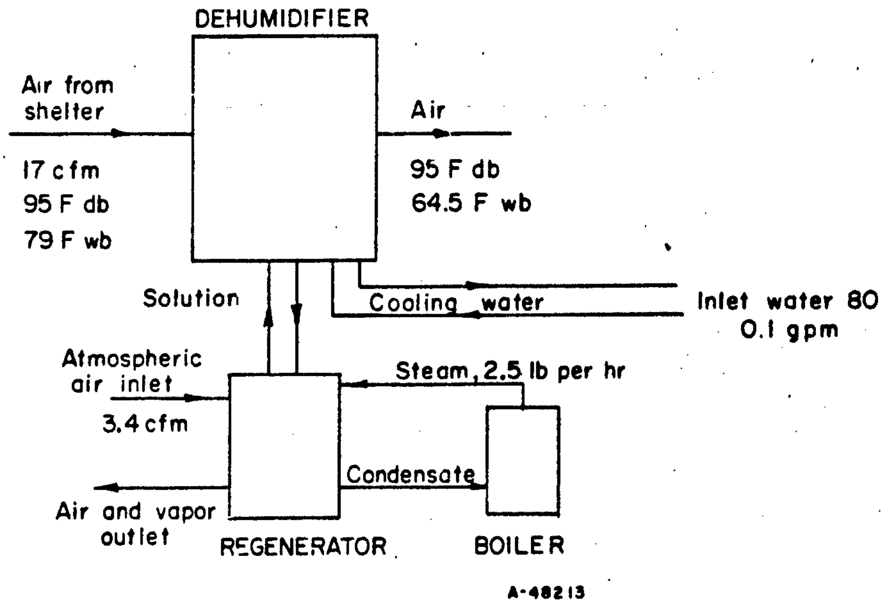


FIGURE 35. DEHUMIDIFICATION SYSTEM FOR REMOVING WATER VAPOR AT THE RATE OF 1 LB PER HR

The air leaving the dehumidifier could be used for direct cooling of the shelter, in which case, the quantity of air to be handled would be quite large. For this example, each pound of air would remove 5 Btu. In addition, because of the sensible heat ratio of the cooling load, the air leaving the shelter would have a dry-bulb temperature of 106 F (85 F effective temperature) which would probably not be tolerable.

A more effective and practical use for a dehumidifier would be in conjunction with an evaporative cooler. This system has two advantages. One, the amount of heat removed per pound of air would be more than double that removed by the dehumidifier alone. For this example, 13 Btu per pound of air. Two, exhaust air leaving the shelter would have a dry-bulb temperature of 95 F (85 F effective temperature) which would definitely be tolerable.

The dehumidifier-evaporative-cooler system would also be advantageous from a cost standpoint because evaporative coolers are relatively inexpensive compared with the dehumidifying equipment. The estimated installed cost for the equipment shown in Figure 35 which, when used with an evaporative cooler, would produce approximately 1,000 Btu per hr of cooling is:

	Cost, dollars
Dehumidifier	27
Regenerator	13
Boiler	20
	60

The dehumidifier and regenerator would occupy about 1 cubic foot of space and require approximately 1/30 hp per 1,000 Btu per hr of cooling.

AIR FILTERS

Air filters are not cooling system components directly associated with the removal of excess shelter heat. However, in some instances their cost and performance characteristics must be considered in determining the total cost of cooling systems.

For special shelters designed to provide BW/CW protection, two types of filters may be required; particulate and gas. Particulate filters which will remove solid particles and liquid droplets consist of a medium efficiency prefilter and a high or extreme efficiency filter. The prefilter is used to remove the larger size particles, thereby reducing the loading on the higher efficiency filter. Gas filters which will remove only gases and vapors consist of a bed of activated charcoal on which the material to be filtered is adsorbed. Such filters are not required for failout shelters.

Table 10 shows the performance characteristics of shelter ventilating-air filters.

TABLE 10. PERFORMANCE CHARACTERISTICS OF SHELTER VENTILATING AIR FILTERS

				Rated Performance	
Filter Type	Function	Process	Approximate Space Requirement, cu ft per 1,000 cfm	Flow, cfm per sq ft face area	Pressure Drop, inches of water
Medium efficiency	Removes coarse particles, protects subsequent filters	Inertial impaction	0.5	300	0.1
High efficiency	Stops smallest particles	Filtration	4	150	1.0
Gas, activated charcoal	Removes poisonous gases and vapors	Adsorption	15	100	0.8

Figure 36 shows the installed cost of the three types of filters as a function of air flow rate.

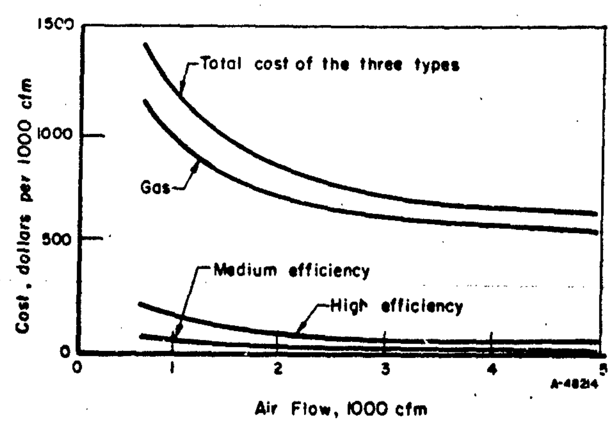


FIGURE 36. INSTALLED COSTS OF FILTERS

BLAST VALVES

Blast valves are of interest in this program in that their cost and operating characteristics affect the cost of some cooling systems presented later. The flow characteristics and cost of commercially available tubular and poppet valves were obtained from manufacturers. A brief description of the valves for which cost data were obtained in included.

Tubular valves consist of a set of spring-mounted stainless-steel tubes and a matching set of fixed solid rods. Under the initial pressure of a blast wave the spring-mounted tubes move to mesh with the solid rods, thereby sealing the opening. Upon restoration of normal atmospheric pressure, the valve opens automatically. The tubular valves close in 5 milliseconds or less. The basic module has an air-flow rating of 1,000 cfm at 1 inch of water pressure drop. Multiple modules can be arranged for air-flow capacities up to 15,000 cfm.

Poppet valves are available with closing mechanisms actuated by blast pressure, triggered by a sensor and actuated by a dual arrangement such that either blast pressure or a sensor system would function. The blast-actuated poppet valves close in 9 to 24 milliseconds, depending upon the blast pressure and the size of the valve. The sensor-triggered poppet valves actuated by compressed air, and poppet valves equipped with a dual closing arrangement, close in 30 to 100 milliseconds depending upon the valve size. The poppet valves are built in nominal sizes with diameters up to 60 inches and with flow capacities as high as 35,000 cfm at 1 inch of water pressure loss.

Figure 37 shows the installed cost of the various blast valves versus their rated capacity. These data are used in the "Cooling System" section of this report. The cost of an infrared sensor would be between $1,500 and $8,000. This has not been included in the cost of the valves.

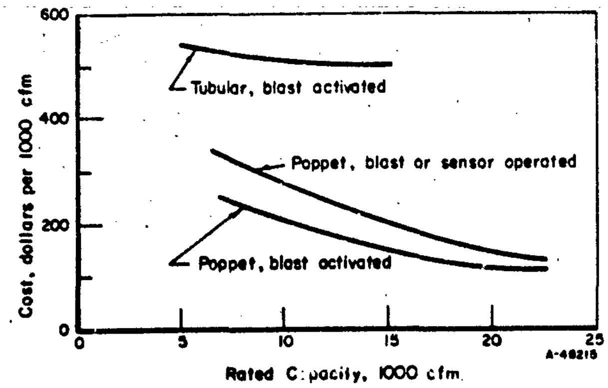


FIGURE 37. COST OF BLAST VALVES

The blast valve cost per cfm could be reduced by increasing the air flow through a given valve at the expense of increased air-handling costs resulting from the higher pressure loss. In addition, the valve-closing mechanism would have to be modified to compensate for the increased pressure drop across the valve.

BLOWERS

Blower considerations of prime importance are the power requirements and the blower costs. Power requirements are important not only because of the demands upon the power system, but also because the power input to the blower would appear as heat in the air leaving the blower. Because shelters are expected to be slightly pressurized the blower must be at the air intake; therefore, the temperature of the air entering the shelter would be somewhat higher than the atmospheric temperature.

The horsepower input to a ventilating blower shaft is approximately;

$$\text{hp} = \frac{1.57 \times 10^{-4} \times \text{pressure rise, inches of water} \times \text{flow, cfm}}{\text{blower efficiency, decimal}} .$$

The following tabulation shows typical values of power input and air temperature rise for blowers having an efficiency of 60 per cent.

Pressure Rise, inches of water	Power Input per 1,000 cfm, hp	Temperature Rise of Air, F
2	0.52	1.2
4	1.04	2.4
6	1.56	3.6

If the blowers were driven by an electric motor in the air stream, the temperature rise would be 10 to 20 per cent greater than that shown above. If the blowers were driven by a reciprocating combustion engine, which would be cooled by the blower air, the temperature rise would be two to three times that shown in the tabulation.

Figure 38 shows the cost of installed blowers driven by electric motors or driven directly with engines. The costs of the engine-generator set at $150 per hp is included to emphasize the fact that the cost of air at higher pressures is greatly influenced by the type of driver on the blower.

WATER PUMPS

The centrifugal pump is widely used in industrial applications and is available in a variety of sizes and discharge pressures. The centrifugal pump is also well suited for use in shelter cooling because of its low initial cost, small space requirements, low maintenance requirements, and quiet operation. Seals that would not be adversely affected by long periods of nonuse are available.

For pumping water containing large quantities of solids or corrosive compounds, pumps specifically designed for this type of service are available.

The factors of particular interest here are pump costs and power requirements.

The power input in horsepower required for a pump is:

$$hp = \frac{8.3 \times \text{flow rate, gpm} \times \text{head, ft}}{\text{pump efficiency, decimal}} .$$

Figure 39 shows the installed cost of centrifugal pumps equipped with close-coupled electric motors for a range of capacities.

PACKAGED BOILERS

Figure 40 shows the cost of low pressure oil-fired boilers which would be suitable for use in shelter cooling systems. These boilers leave the factory fully assembled and ready for connection to steam, water, and electric services.

Boilers of this class have a thermal efficiency of approximately 75 per cent and, therefore, their fuel consumption would be about 0.08 pound per hour per 1,000 Btu per hr of heat output. The boiler space requirement would be about 0.12 cubic feet per 1,000 Btu per hr of heat output.

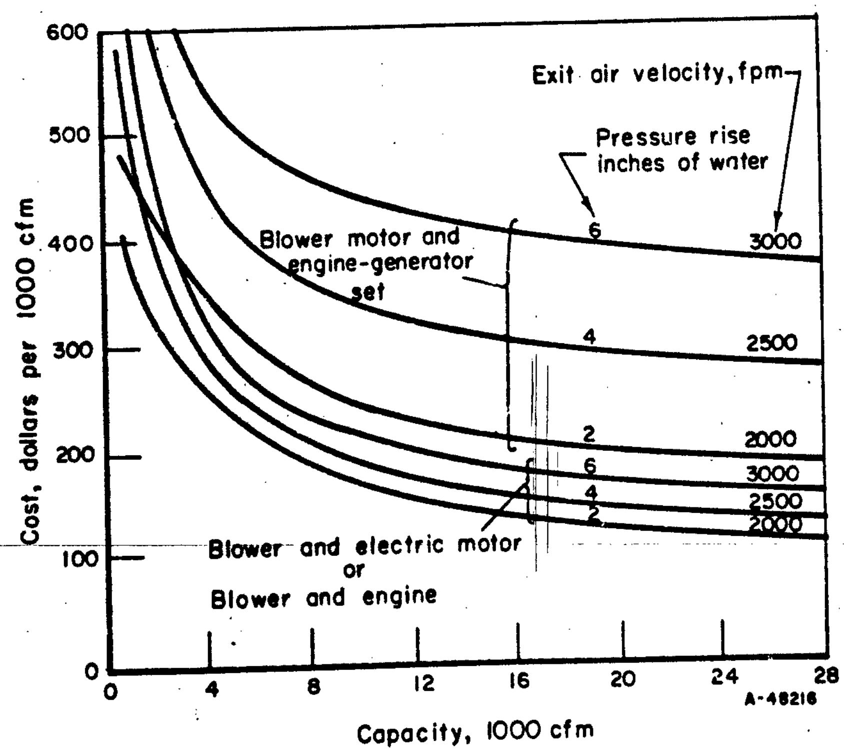


FIGURE 38. COST OF BLOWERS AND DRIVERS

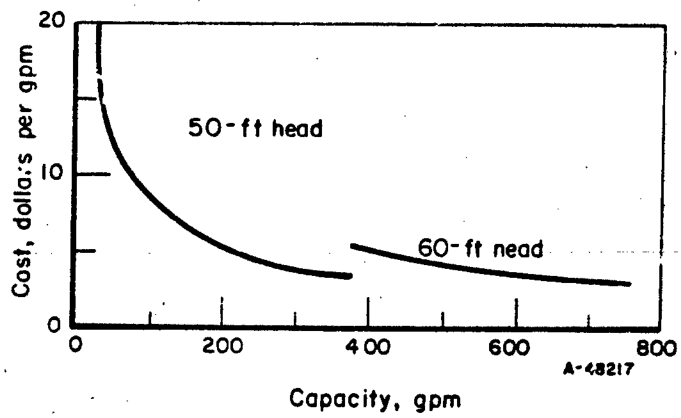


FIGURE 39. INSTALLED COST OF CENTRIFUGAL PUMPS WITH CLOSE-COUPLED ELECTRIC MOTORS

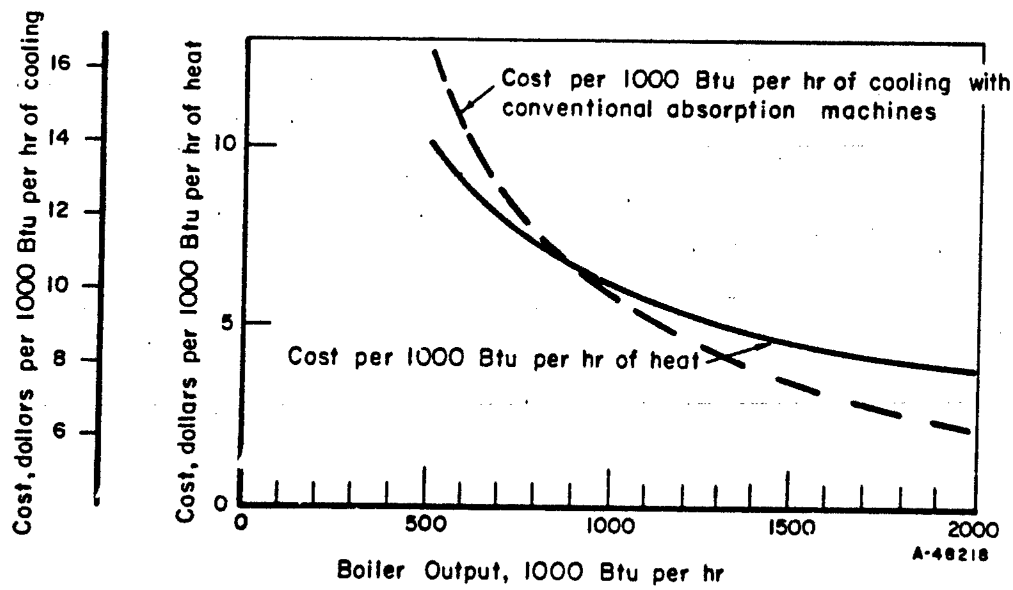


FIGURE 40. INSTALLED COST OF LOW-PRESSURE PACKAGED OIL-FIRED BOILERS

MECHANICAL POWER

The cost of providing mechanical power for operating a cooling system may be a significant factor in the total cooling system cost. The requirements and costs of auxiliary power systems were the subject of a separate study conducted by Battelle for the Office of Civil Defense under Contract No. OCD-OS-62-190, Subtask 1411C. The final report is entitled "A Study of the Minimum Requirements for Auxiliary Power Systems for Community Shelters".

As might be imagined, it is difficult to generalize the cost of auxiliary power systems. However, for the purposes of this study it was deemed necessary. The costs which were selected for use in determining the total cost of cooling systems are:

Power Output, hp	Cost, dollars per hp	
	Engine	Engine-Generator
Up to 10	100	200
10 to 40	70	150
40 to 100	50	100
100 and up	40	80

The above costs are typical for standard industrial installations and do not include the provision of unusual requirements for the engine such as blast-resistant air intakes and exhausts, shelter space, and custom cooling equipment or heat sinks. These requirements need to be considered in relationship with the total shelter facility.

• REFRIGERATION + DEHUMIDIFICATION DEVICES

- PHASE CHANGE OF WORKING FLUIDS
- NO PHASE CHANGE OF WORKING FLUIDS
- SOLID-STATE DEVICES

REFRIGERATION AND DEHUMIDIFICATION DEVICES

Any shelter cooling system that utilizes a heat sink at a temperature above the shelter temperature will require a refrigeration device. Of the many techniques of cooling, only two have major commercial significance: mechanical vapor compression (MVC) and absorption. Commercial machines are applicable for shelter cooling, but because of the unique requirements of this application, for example, lack of on-site power, long standby period, short operating period, and unavailability of a natural heat sink, other cooling techniques using unconventional equipment might be advantageously applied to shelter cooling. Therefore, these were investigated and novel devices were devised.

The novel or unconventional devices of particular interest are: (1) cascade arrangement of MVC machines, (2) MVC machine without a condenser using methyl alcohol as the working fluid, (3) open-cycle absorption machine using sulfuric acid and water as the working fluids, and (4) semi-open air-cycle machine.

For discussion purposes, refrigeration and dehumidification systems may be divided into three primary groups: (1) those that have phase change of the working fluid, (2) those that do not have phase change of the working fluid, and (3) those that use solid-state devices.

In the first group, "phase change of the working fluid", are the systems that perform their refrigeration or dehumidification functions by means of a sequence or cycle of thermodynamic processes, one or more of which involves a phase change of the working fluid or fluids. As will become evident, only the liquid-vapor phase change is applicable. In the second group, "no phase change of the working fluid", are those devices that are normally considered to operate by a sequence or cycle of processes none of which involves a phase change of the working fluid. Here the gaseous phase is most applicable. In the last group, "solid-state devices", are the semi-conductor devices that operate by means of various combinations of coupling phenomena between temperature, electrical, and magnetic potentials. The Peltier thermoelectric device is perhaps the best known of this group.

All of the devices considered, including those that were found to be not applicable to shelter cooling, are discussed in the following sections.

PHASE CHANGE OF WORKING FLUIDS

Refrigeration and dehumidification devices in which the working fluid undergoes a phase change can be grouped according to utilization of the working fluid and according to the form in which the energy required for operation must be supplied.

Two methods of using the working fluids are available. One, they can be used once in an open-cycle device (expendable fluid) and discarded, or, two, they can be regenerated and used repeatedly in a closed-cycle device (nonexpendable fluid).

The form of the energy required to operate the device may be: (1) shaft power (mechanically motivated), (2) heat (thermally motivated), or (3) stored in the working fluids (self-motivated).

NONEXPENDABLE FLUIDS – MECHANICALLY MOTIVATED DEVICES

The following discussion deals with mechanically motivated refrigeration devices that operate on a closed cycle and with a phase change of the working fluid. The liquid-vapor phase change is the only one applicable and all devices covered operate on the mechanical-vapor-compression (MVC) cycle. The discussion is divided as follows: single-stage, multistage, and nonisothermal.

The air-conditioning and refrigeration industry today is based, to a great degree, on MVC refrigeration devices. These operate on a thermodynamic cycle that is well understood and with equipment that is, for the most part, highly developed. Much information is readily available, particularly in the American Society of Heating, Refrigeration, and Air Conditioning Engineers Guide and Data Books and it will not be repeated here. Rather, problems arising because of the unique requirements of shelters will be considered and information will be presented for use in estimating the cost of complete shelter cooling systems.

MVC – Single Stage

Single-stage MVC equipment can operate with a heat sink at normal (less than 120 F) or abnormal (above 120 F) temperature.

Disregarding standby requirements and assuming normal temperature conditions, there is little reason to question the technical applicability of present MVC equipment. The commercial trend is to increase market attractiveness by cost reduction. This is being accomplished, in part, by the wide manufacture and use of factory-assembled equipment. Known under the general title of "unitary air conditioners", this equipment is presently available from a large number of manufacturers in unit capacities from 24,000 to 900,000 Btu per hr (2 to 75 tons). Most units employ hermetic or semihermetic reciprocating compressors and are designed to operate with Refrigerant 12, 22, or 500. Condensers are air, water, or evaporative cooled. Both direct-expansion and chiller models are available. The units are divided into four classes, A, B, C, and D, depending on the relative location of the various components. Class A includes all single-package direct-expansion assemblies. Classes B and C include direct-expansion units with the components divided into two packages. Class D comprises chiller-type units.

Complementing the unitary types is a wide range of built-up equipment for those installations that require custom-designed systems.

The performance of MVC equipment varies with the particular installation and the operating conditions. Detailed information is best obtained from the manufacturer. However, Table 11 gives a representative range of performance values.

TABLE 11. REPRESENTATIVE PERFORMANCE OF MVC EQUIPMENT UNDER NORMAL CONDITIONS

Type	Cooled	Coefficient of Performance (COP)	Heat-Rejection Factor (HRF)
Unitary or built-up	Air	1.8 – 3.2	1.6 – 1.3
	Water or evaporative	2.6 – 4.4	1.4 – 1.2

The unitary types are most simply and economically installed; they are available for inside and outside placement and for all arrangements of mounting. For example, floor, wall, and ceiling mounts are available. The plumbing connections required to and between components depends on how the unit is cooled and if it is a direct-expansion or a chiller unit. The installation of built-up systems is somewhat more involved. Individual components must be placed and then interconnected. Since they are not precharged with refrigerant, it is necessary that they be evacuated of noncondensible gases and moisture and then charged with refrigerant. All types, however, require electrical wiring. The cost of providing the electrical power connections to a unit is a significant economic consideration. A good working figure is $40 per horsepower.

The requirements for the standby period must be further investigated. Conventional air-conditioning equipment is installed for use either during only the summer months or on a continual year-round basis. During this research, instances were found in which equipment was kept on a standby basis for extended periods. At present it appears that the best approach to insuring reliable operation is periodic exercising of the equipment along with replacement of critical seals and control components as necessary.

The safety aspects of MVC equipment under normal operating conditions are generally considered excellent. However, for shelter applications consideration must be given to the high refrigerant pressures that can exist in this equipment and the effects of possible refrigerant leaks into the shelter. Separating the equipment from the occupied area may be advisable.

From the standpoint of comfort, attention must be given to keeping noise and vibration to an acceptable level. As most of this equipment employs reciprocating compressors with self-actuated valves, there are vibration and noise problems which can best be solved by separating the equipment from the occupied portion of the shelter.

The configurations of MVC equipment are varied. In general, however, the space required is as shown in Table 12. The larger capacity units make better use of available space.

TABLE 12. SPACE REQUIREMENTS FOR MVC EQUIPMENT

Type	Capacity, Btu per hr	Sq Ft per 1000 Btu per Hr	Cu Ft per 1000 Btu per Hr
Unitary	24,000 - 900,000	0.46 – 0.11	0.42 – 0.11

The comparative costs of different types of MVC equipment are shown in Figure 41. The dashed lines represent built-up equipment; as can be seen the installed cost decreases as unit capacity increases. For moderate-production quantities this is a normal cost curve. Unitary equipment is represented by the solid lines. These curves deviate from the normal pattern because of mass-production quantities. This is particularly true of the popular single-assembly units of lower capacities. In general, water-cooled unitary models are less expensive than air cooled, and evaporative-cooled units are about equal in cost to air-cooled units. The cost of chiller units runs 5 to 10 per cent above the cost of direct-expansion units.

Because a survival shelter might not have access to a normal temperature sink, the refrigeration device might have to be designed to make use of a sink at a higher than normal temperature level. This sink could conceivably be water boiling at 212 F. While MVC equipment using higher than normal condensing temperatures is known, rarely has such equipment been designed for sink temperatures above about 160 F. Some examples of high-condensing temperature MVC equipment are: heat pumps, ground-based military equipment, and specialized aircraft equipment.

Several disadvantages and problems are associated with high-temperature condensing. A disadvantage can be thought of as something that lends itself to a rather obvious, direct, and, therefore, generally successful solution. A problem, on the other hand, is of a more complex nature and it remains a problem. For example, some of the disadvantages of high-temperature condensing are: lower COP, higher condensing pressures, reduced compressor capacities, and increased mechanical loadings. These disadvantages can be overcome by such direct methods as, respectively, increasing the capacity of the motivating energy source, using heat exchangers, tubing, and fittings of higher pressure rating, and employing compressors of larger capacity and of more rugged design. The problems, however, center around the degradation of the refrigerant and lubricant and are not as easily solved.

Figures 42 through 48 show calculated operating characteristics for a simple MVC cycle operating with a wide range of refrigerants at condensing temperatures up to 212 F. The refrigerants used are listed in Table 13. The basic assumption for the calculations were: an internal evaporator temperature of 50 F, an internal compression efficiency of 75 per cent, and a mechanical efficiency of 90 per cent. Figures 42, 43, and 44 show, respectively, the effects of various refrigerants and a variable condensing temperature on the coefficient of performance, heat-rejection factor, and required compressor power. Figures 45, 46, and 47 are, respectively, the pressure ratio across the compressor, refrigerant mass flow rate, and refrigerant volumetric flow rate at the compressor inlet. Figure 48 shows vapor pressure-temperature plots for the refrigerants studied.

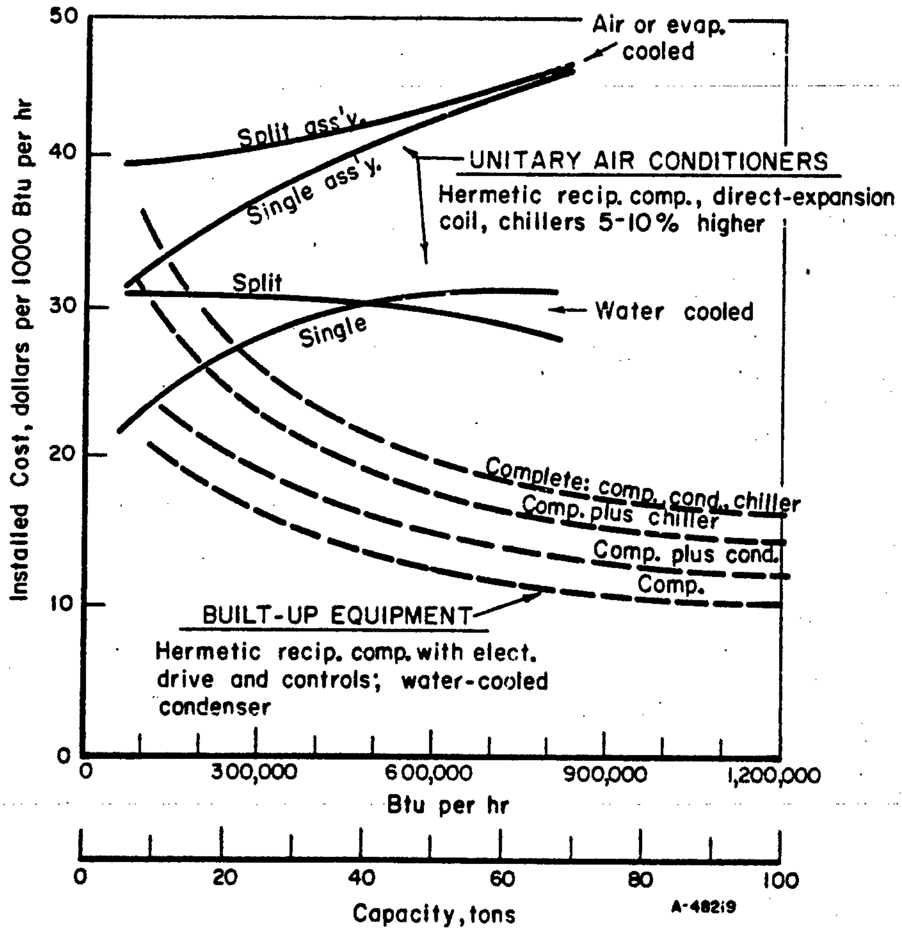


FIGURE 41. COMPARATIVE COSTS OF UNITARY AND BUILT-UP MVC AIR-CONDITIONING EQUIPMENT

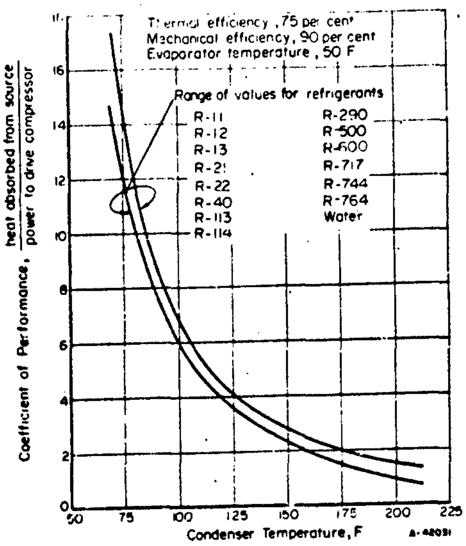


FIGURE 42. RANGE OF COEFFICIENT OF PERFORMANCE FOR VARIOUS REFRIGERANTS

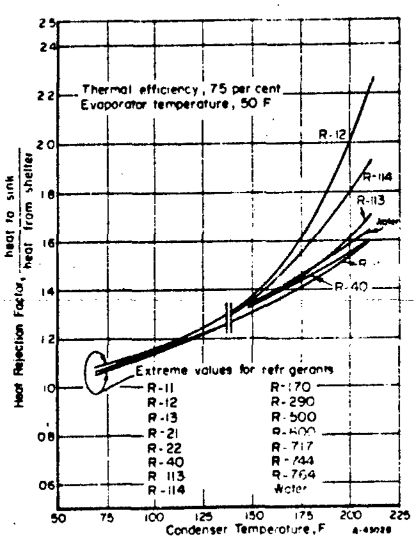


FIGURE 43. RATIO OF HEAT REJECTION TO HEAT REMOVAL FOR SIMPLE SATURATION CYCLE WITH COMPRESSOR THERMAL EFFICIENCY OF 75 PER CENT

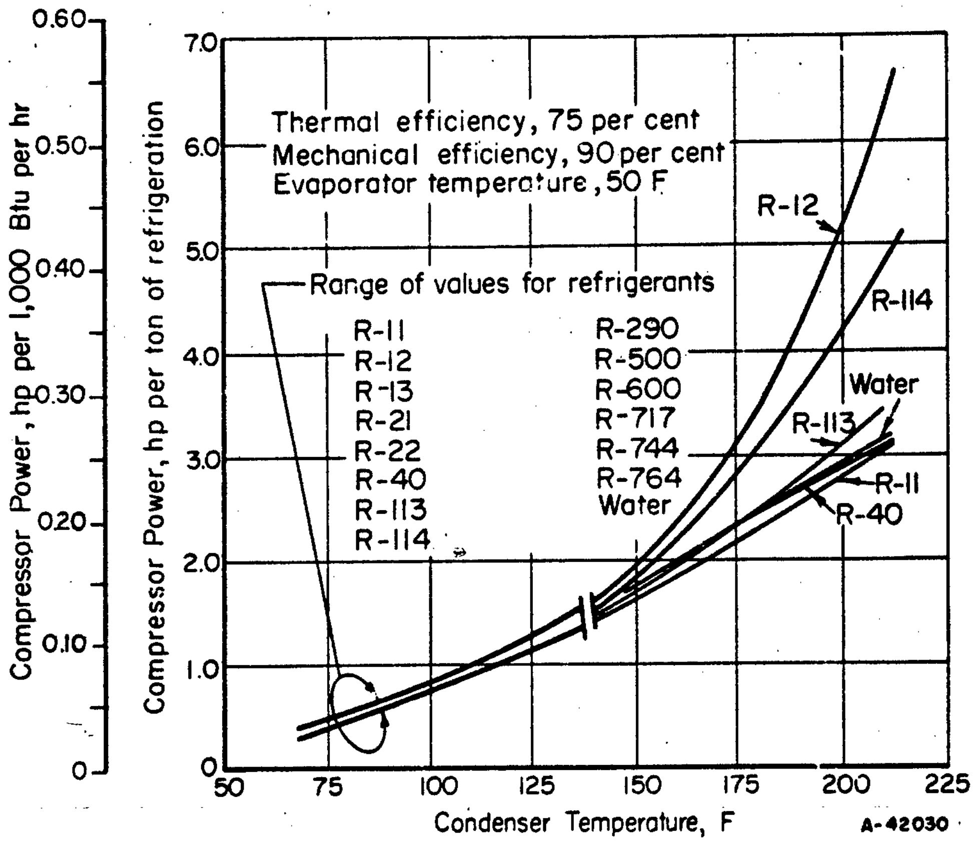


FIGURE 44. COMPRESSOR POWER FOR SIMPLE SATURATION CYCLES WITH THERMAL EFFICIENCY OF 75 PER CENT AND MECHANICAL EFFICIENCY OF 90 PER CENT

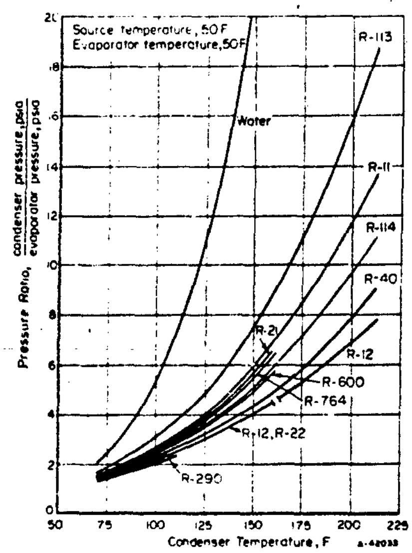


FIGURE 45. CONDENSER-TO-EVAPORATOR PRESSURE RATIO FOR VARIOUS REFRIGERANTS

Evaporator temperature 50 F.

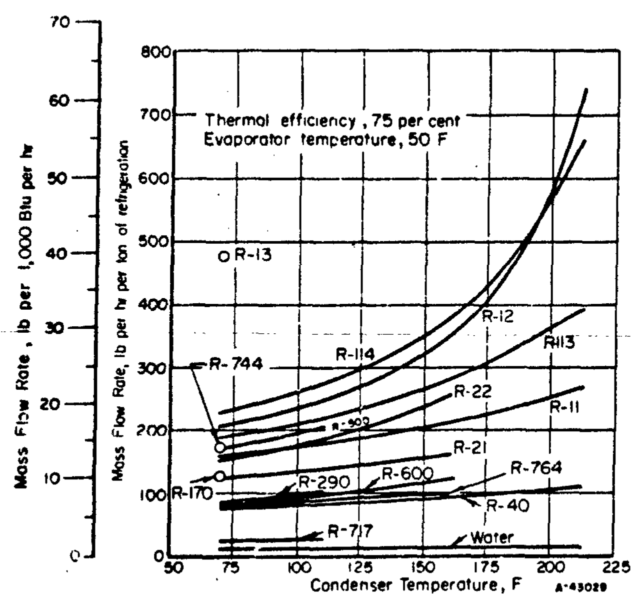


FIGURE 46. REFRIGERANT MASS FLOW RATE FOR SIMPLE SATURATION CYCLE WITH COMPRESSOR THERMAL EFFICIENCY OF 75 PER CENT

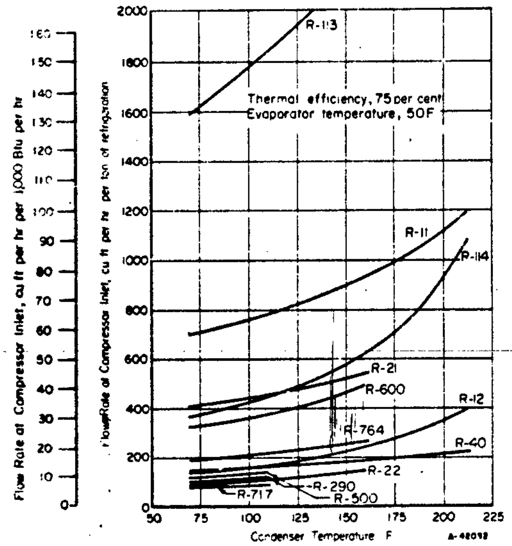


FIGURE 47. VOLUME FLOW RATE AT THE COMPRESSOR INLET FOR SIMPLE SATURATION CYCLE

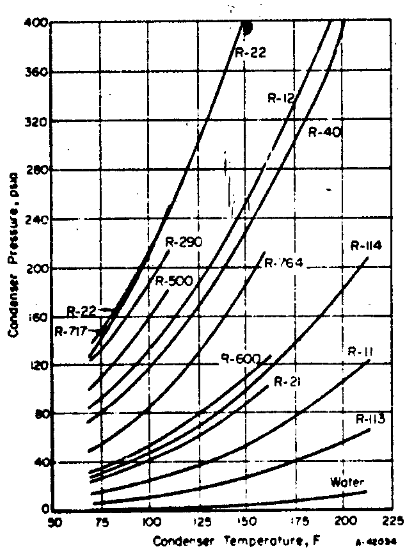


FIGURE 48. CONDENSER PRESSURES FOR VARIOUS TEMPERATURES OF REFRIGERANTS

TABLE 13. REFRIGERANTS USED FOR CYCLE ANALYSES

ASHRAE Number	Chemical Name	Formula	Mol. Wt.	Normal Boiling Point, Deg F
		HALOCARBON COMPOUNDS		
11	Trichloromonofluoromethane	CCl_3F	137.4	74.8
12	Dichlorodifluoromethane	CCl_2F_2	120.9	-21.6
13	Monochlorotrifluoromethane	$CClF_3$	104.5	-114.6
21	Dichloromonofluoromethane	$CHCl_2F$	102.9	48.1
22	Monochlorodifluoromethane	$CHClF_2$	86.5	-41.4
30	Methylene Chloride	CH_2Cl_2	84.9	105.2
40	Methyl Chloride	CH_3Cl	50.5	-10.8
113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.4	117.6
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$	170.9	38.4
152a	Difluoroethane	CH_3CHF_2	66	-12.4
		MISCELLANEOUS ORGANIC COMPOUNDS		
		Azeotropes		
500	Refrigerants 22/152a, 73.8/26.2 wt%	CCl_2F_2/CH_3CHF_2	99.29	-28.0
		Hydrocarbons		
170	Ethane	CH_3CH_3	30	-127.5
290	Propane	$CH_3CH_2CH_3$	44	-44.2
600	Butane	$CH_3CH_2CH_2CH_3$	58.1	31.3
		INORGANIC COMPOUNDS		
717	Ammonia	NH_3	17	-28.0
718	Water	H_2O	18	212
729	Air		29	-318
744	Carbon dioxide	CO_2	44	-109 (subl)
764	Sulfur dioxide	SO_2	64	14.0

As previously stated the problems of high-temperature condensing are degradation of the refrigerant and lubricant. This degradation is a thermal decomposition of the refrigerant and lubricant which is complicated by chemical interaction between the refrigerant and lubricant, the materials of construction, and any contaminants that may be originally present in the system or formed during its operation. The products of this degradation appear in all phases, that is, gaseous, liquid, and solid; and they are the ultimate source of problems. For example, a gaseous product will collect in the heat exchangers, increasing the absolute pressure level and reducing capacity by interferring with the mass transfer of the condensing or evaporating refrigerant. The most harmful liquid products are those that are acidic. These attack critical mechanical components such as compressor valves, expansion valves, and bearing surfaces within the compressors. The solid or sludge products hinder operation by fouling surfaces and blocking vital refrigerant and lubricant passages.

With presently available information on decomposition it is impossible to predict the amount of difficulty there might be in the development of an MVC device capable of rejecting heat to a boiling water sink. While some data on decomposition rates are available, generally they are for temperatures well below 212 F. Moreover, there is no reliable method for relating these rates to a reduction in performance or failure of a particular system. On the other hand, one consideration that favors a successful development for shelter applications is the short operating period required. Two weeks is quite short when compared with normal operating requirements for refrigeration and air-conditioning equipment.

MVC – Multistage

In the MVC cycle the pressure ratio across the compression device is a direct function of the temperature difference between the evaporator and the condenser. Any increase in this temperature difference results in an increase in the pressure ratio. Depending on the particular application, at some point the pressure ratio eventually reaches a value where, for any one of several reasons, it becomes impractical to accomplish the compression process in a single stage. The solution is to employ two or more stages of compression. This can be done through either direct staging or cascade staging.

Direct staging is also known as compound staging. In this arrangement the compressors are connected in series as shown in Figure 49A. Refrigerant vapor leaving the evaporator is compressed to an intermediate pressure by the first compressor and then discharged to an intercooler. In the intercooler the refrigerant vapor is desuperheated to some degree before it is introduced to the intake of the second compressor. If only two compressors are used, the refrigerant vapor is discharged from the second compressor to the condenser. The intercooler desuperheats the refrigerant vapor by throttling liquid refrigerant from the condenser to the interstage pressure at the intercooler. Figure 49A shows a shell-and-coil type intercooler; two others, a flash type and a dry-expansion type are also available. In addition to desuperheating the vapor, the intercooler reduces the temperature of the liquefied refrigerant below condenser conditions, thereby increasing its capacity to absorb heat in the evaporator.

Cascade staging of two compressors in somewhat less involved. As shown in Figure 49B this arrangement is simply two separate vapor compression cycles joined thermally at an intermediate heat exchanger. The intermediate heat exchanger, or cascade condenser, combines the condenser for the lower cycle and the evaporator for the upper cycle.

The widest application of staged compression is in the field of low-temperature refrigeration. Single-stage refrigeration devices are acceptable down to between 0 and -40 F, depending on the capacity of the unit. At lower evaporator temperatures the drop-off in capacity and operating efficiency becomes significant. The increased pressure ratio also results in an undesirable increase in the vapor discharge temperatures. For evaporator temperatures down to about -125 F, compound compression can be used. The limitation here is associated mainly with the refrigerant, that is, one is required with a boiling temperature below the desired operating temperature and a reasonable condensing pressure. For temperatures below -125 F, cascade staging is usually required.

The application of staged compression to the shelter situation is obviously directed toward high-temperature condensing and, in particular, toward the ultimate employment of boiling water as the heat sink. The advantages of compound over single-stage compression at high-temperature condensing conditions would be increased efficiency and

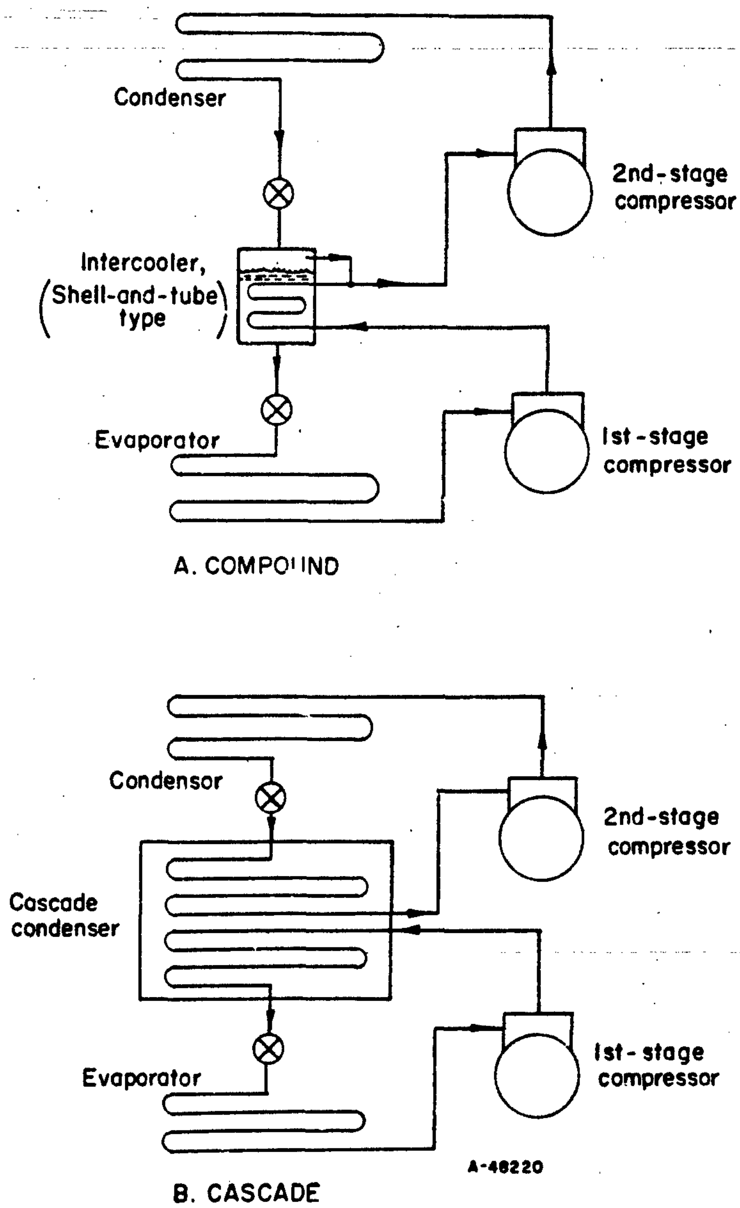


FIGURE 49. ARRANGEMENT OF COMPONENTS FOR TWO-STAGE COMPOUND AND CASCADE COMPRESSION

reduced vapor discharge temperatures. The reduction in vapor discharge temperatures is of major significance. As pointed out in the preceding discussion on single-stage devices, the high-temperature degradation of the refrigerant and lubricant appears to be the factor that would limit the successful development of a high condensing temperature MVC device.

The advantages and disadvantages of compound and cascade compression should be compared. Since the cascade system is made of separate refrigerant circuits, it is possible to employ different refrigerants. The degradation problem must be overcome in either system and it appears that the cascade system would facilitate the matching of a refrigerant and a lubricant for high-temperature duty. Also, with different refrigerants, component design would be simplified since pressure levels, specific volumes etc., could be better selected. The disadvantage with the cascade system would be a somewhat reduced operating efficiency due to the temperature difference that must be overcome at the cascade condenser.

Selecting the two-stage cascade system for high-temperature condensing, some example calculations were made for a number of refrigerant combinations. For these calculations the following assumptions were made: evaporator temperature, 50 F; cascade condenser temperature, 110 F; final condenser temperature, 212 F; internal compressor efficiency, 75 per cent; and compressor mechanical efficiency, 90 per cent. The results of the calculations are listed in Table 14. Also included are the results of a single-stage MVC cycle operating between 50 and 212 F with several refrigerants. This information is of value in the final cooling system analysis where heat sink capacities and the size of the motivating energy source are required.

On the basis that a two-stage cascade MVC system is the equivalent of two separate devices, the cost must be considered as double that of a single-stage MVC system.

MVC – Nonisothermal

Nonisothermal refrigeration is a concept that makes use of an often overlooked fact that the Carnot cycle is not invariably the most efficient cycle possible. For example, Figure 50 shows a series of schematic temperature-entropy diagrams. Part "a" shows two lines on a T-S plot which represent heat source and heat sink fluids that have a finite specific heat. Upon passing through the heat exchangers of a refrigeration device, the heat source fluid will, therefore, undergo a temperature decrease and the heat sink fluid a temperature increase. In Figure 50b a Carnot-cycle refrigerator that is capable of operating with the same source and sink has been superimposed on the diagram. Note that in order for the Carnot cycle working fluid to accept and reject heat isothermally, its lower temperature must be somewhat below the minimum temperature of the heat source. As can be seen, the heat exchanging processes between the refrigerant working fluid and the source and sink materials will be quite irreversible. A cycle which has a much closer approach to external irreversibility is shown in Figure 50c. Here the temperature profiles of the refrigerant working fluid closely match those of the heat source and heat sink materials. This cycle obviously requires less work input for the same refrigeration effect.

TABLE 14. CYCLE PARAMETERS WITH CONDENSER TEMPERATURE OF 212 F, EVAPORATOR TEMPERATURE, 50 F; THERMAL EFFICIENCY, 75 PER CENT COMPRESSOR MECHANICAL EFFICIENCY, 90 PER CENT

	Refrigerant	Evaporator Pressure, psia	Condenser Pressure, psia	Condenser-to-Evaporator Pressure Ratio	Flow Rate per 1000 Btu per Hr: Cu Ft per Hr	Flow Rate per 1000 Btu per Hr: Lb per Hr	Heat to Sink / Heat From Source	Compressor Power per 1000 Btu per hr, hp	COP
Simple-Vapor-Compression Cycles									
	R-11	8.80	120	13.61	99.4	22.4	1.62	0.27	1.46
	R-12	61.4	475	7.74	41.5	61.5	2.28	0.56	0.71
	R-40	51.99	465	8.94	17.6	9.2	1.61	0.27	1.47
	R-11	3.43	64	18.72	274	32.6	1.72	0.31	1.26
	R-114	18.73	207	11.1	89.5	55.0	1.94	0.44	0.90
Cascade Cycle, 110 F Intermediate Temperature									
Low temp	R-114	18.73	54.4	2.90	37.4	22.9			
High temp	R-114	54.4	207	3.81	26.6	45.0			
Total							1.61	0.29	1.36
Low temp	R-12	61.4	151	2.46	13.8	20.5			
High temp	R-114	54.4	207	3.81	26.7	45.0			
Total							1.61	0.29	1.38
Low temp	R-114	18.73	54.4	2.90	37.4	22.9			
High temp	R-12	151	475	3.14	15.1	54.4			
Total							1.80	0.35	1.12
Low temp	R-113	3.43	12.8	3.72	154	18.2			
High temp	R-113	12.8	64.1	5.03	75.6	29.9			
Total							1.56	0.25	1.61
Low temp	R-21	15.33	47.4	3.09	38.1	11.3			
High temp	R-11	28.09	120	4.27	33.8	22.5			
Total							1.54	0.24	1.66
Low temp	R-21	15.33	47.4	3.09	38.1	11.3			
High temp	R-113	12.76	64.11	5.03	75.0	29.6			
Total							1.55	0.24	1.64
Low temp	R-11	8.80	28.09	3.19	65.3	14.8			
High temp	R-113	12.76	64.11	5.03	75.5	29.8			
Total							1.55	0.24	1.62

TABLE 14. (Continued)

	Refrigerant	Evaporator Pressure, psia	Condenser Pressure, psia	Condenser-to-Evaporator Pressure Ratio	Flow Rate per 1000 Btu per Hr: Cu Ft per Hr	Flow Rate per 1000 Btu per Hr: Lb per Hr	Heat to Sink / Heat From Source	Compressor Power per 1000 Btu per hr, hp	COP
Low temp	R-717	89.19	247.0	2.77	86.2	26.2			
High temp	R-40	134.5	465	3.46	94.7	123.3			
Total							1.56	2.96	1.59
Low temp	R-40	51.99	135.4	2.60	157.6	82.0			
High temp	R-40	134.5	465	3.46	94.0	122			
Total							1.55	2.92	1.62
Low temp	R-764	33.45	99.76	2.98	208	88.7			
High temp	R-11	28.09	120	4.27	407	272			
Total							1.55	2.87	1.64

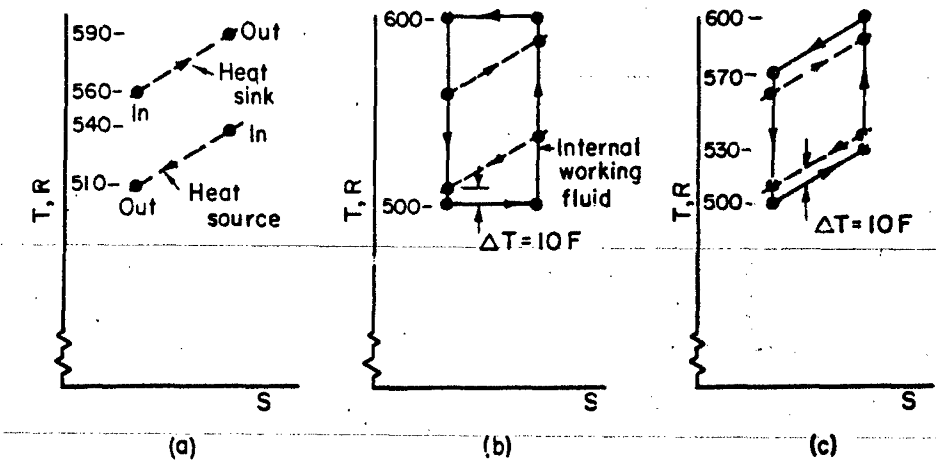


FIGURE 50. TEMPERATURE-ENTROPY DIAGRAMS FOR A HEAT SINK, A CARNOT CYCLE, AND A NONISOTHERMAL REFRIGERATION DEVICE

In brief then, the most efficient refrigeration cycle is one that consists of processes that are thermodynamically reversible both internally and externally. The non-isothermal approach attempts to reduce externally induced irreversibilities by reducing log-mean temperature differences. This, in turn, is accomplished by matching the temperature profiles of the internal working fluid more closely to those of the heat source and heat sink materials.

A numerical example of the improvement possible can be obtained by considering the temperatures shown in Figure 50. One assumption made is that a minimum temperature difference of 10 F always exists. The numerical calculations are as follows:

Isothermal Case

$$COP = \frac{T_L}{(T_S - T_L)} = \frac{500}{(600-500)} = \frac{500}{100} = 5$$

Nonisothermal Case

$$COP = \frac{T_{L_{mean}}}{T_{S_{mean}} - T_{L_{mean}}} = \frac{515}{(585-515)}$$

$$= \frac{515}{70} = 7.35$$

The 47 per cent increase in COP for the nonisothermal case is due to the smaller mean temperature span over which the heat is pumped and the higher absolute mean temperature of the heat source.

Attempts to adapt this concept to MVC refrigeration have been made, but with limited success. The approach is to use a binary mixture of refrigerants with one component being somewhat more volatile than the other. In the evaporator, then, a larger percentage of the more volatile component is vaporized first at the lower temperature. As the remaining solution becomes richer in the less volatile component, the boiling temperature increases. The reverse occurs in the condenser where the less volatile component tends to condense first at the higher temperature, followed by an increasing percentage of the more volatile component as the condensing temperature decreases.

Limited success in the past has apparently been due to some lack of appreciation for the thermodynamic irreversibilities that can be easily induced as a mixture is moving between the liquid and vapor phases. If equilibrium conditions are not approached, internal irreversibilities develop which help to negate any improvements that may be made externally. In the few published cases in which the investigators were aware of these problems, some limited improvement in the operation of actual equipment was demonstrated.[30] However, it appears that there is still a great deal of work to be done and much to be learned before this concept will become practical.

The advantages of using nonisothermal MVC systems in the shelter situation are obviously along the lines of less installed horsepower required to drive the device and smaller heat sink needs. However, in view of the state of the art, these savings would probably not justify the research and development costs.

NONEXPENDABLE FLUIDS - THERMALLY MOTIVATED DEVICES

Frequently, there is a lack of appreciation for the fundamental fact that thermally motivated refrigeration devices are somewhat subtle combinations of heat engines driving heat pumps. A simple thermally motivated refrigerator operates at three temperature levels: below sink temperature, at sink temperature, and above sink temperature. Thus, the energy required so that the refrigerator section can pump heat from below sink temperature to sink temperature is derived from the engine section which allows heat to drop from above sink temperature to sink temperature. A familiar example is the simple absorption-cycle refrigerator in which the generator operates above, the condenser and absorber at, and the evaporator below sink temperature. Here, the condenser and evaporator can be thought of as the heat-pump section that is driven by the heat-engine section made up of the generator and absorber. Figure 51 shows a schematic arrangement of components for the simple absorption cycle.

An understanding of the above is important for several reasons. First, it makes obvious the fact that the use of direct heat motivated refrigerators for shelter cooling offers a potential for significantly reducing the amount of installed horsepower required in a shelter in the form of internal combustion engines. For some shelters and shelter conditions this could be a significant consideration. Second, it explains why these devices have heat-rejection factors that are larger than those usually reported for mechanically motivated refrigerators for which the heat rejected by the source of motivating energy is ordinarily not included. Third, it is indispensable to an understanding of how direct thermally motivated refrigeration devices can be made more efficient. An example here is a complex absorption cycle that accepts its motivating heat at a temperature level higher than that possible with a simple cycle. This allows the motivating heat to drop over an increased temperature span for potentially more efficient engine operation.

The possibility of using new combinations of working fluids for closed-cycle absorption devices was given some study during this program. This was found to be an old problem that has received a great deal of both theoretical and practical

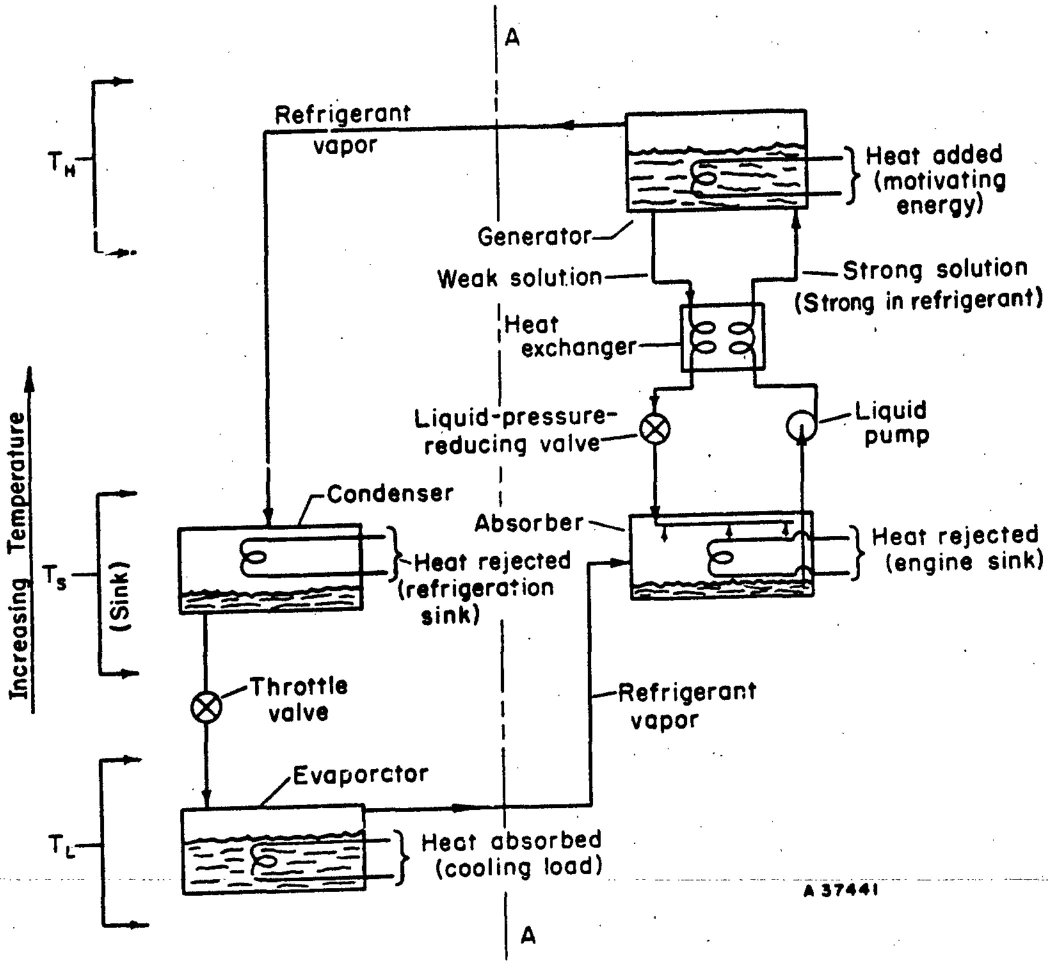


FIGURE 51. ARRANGEMENT OF COMPONENTS FOR THE BASIC ABSORPTION-REFRIGERATION CYCLE

investigation.[30, 31] The many requirements for a successful working fluid combination have eliminated all but ammonia-water and lithium bromide-water as being practical. While it is true that shelter application presents some unique conditions, in particular, short-term operation, there were no new combinations of refrigerant and absorbent found that appeared to offer any significant advantages.

Absorption - Lithium Bromide and Water

Conventional absorption-cycle refrigeration equipment, operating on the simple cycle and employing water as the refrigerant and lithium bromide as the absorbent, is available in unit capacities ranging from approximately 3 to 1,000 tons. Two basic types are available: one designed for residential and small commercial applications and the other for large commercial air-conditioning applications and industrial process cooling. The residential type covers the unit capacity range between 33,600 and 600,000 Btu per hr (2.8 and 50 tons). Unit capacities for the commercial type range from 624,000 to 12,000,000 Btu per hr (52 to 1,000 tons).

The basic difference between residential- and commercial-type units is the method used to circulate the working fluids within and between the various components. In the commercial-type units, electrically driven pumps are used to transfer and recirculate the absorbing solution and liquid refrigerant. The residential units accomplish working fluid transfer by thermal means. Heat is applied in the generator to cause the working fluids to rise to higher elevations. Return of the working fluids to the generator, by way of the other components, is by gravity heads and pressure differences. The small pressure difference, amounting to about 60 mm Hg, between the high and low pressure sides of the lithium bromide-water cycle makes this method feasible. Of course, the electrically pumped commercial units make more efficient use of area and volume for purposes of heat and mass transfer, and are therefore more compact machines.

An additional difference between residential- and commercial-type units is the method of applying the motivating heat to the device. A large percentage of the residential types are designed for direct firing with a gaseous fuel such as manufactured gas, liquid petroleum gas (LPG), or natural gas. Although oil-fired units are not available, they have been constructed, are feasible, and if required could be developed for shelter use with little difficulty. Units with capacities under about 6 tons are exclusively direct fired. Residential units of over 72,000 Btu per hr capacity are available either direct or indirect fired. Indirect fired units use low-pressure steam, usually between 2 and 15 psig, with 10 to 12 psig being the most common. All commercial-type equipment is designed for indirect firing with low-pressure steam or high-temperature liquids.

At present, residential-type units are manufactured by only one company which is closely associated with the gas industry.

The construction of the residential types is of welded steel to produce a completely sealed or hermetic unit. They are basically water chillers; however, direct-expansion units are available in capacities below 72,000 Btu per hr. Heat rejection from the units is either to well water, remote cooling tower, or the atmosphere by way of an integral cooling tower. Air-cooled lithium bromide-water equipment is not available. The reason for this is that direct cooling cannot reliably maintain the absorber at a temperature below that where lithium bromide hydrates can form in the absorbing solution. This crystallization is highly undesirable as it clogs liquid lines and heat exchangers.

The commercial-type equipment is presently available from three separate manufacturers of air-conditioning equipment. Absorption refrigeration is ideally suited to those air-conditioning applications where low-pressure steam is available, such as from a boiler used to carry the winter heating load. The construction of the commercial units is basically welded steel to form a semihermetic unit. Certain joints can be disassembled to permit repair or replacement of pumps and certain control components. These units are exclusively chillers, either of water or a process fluid. Heat rejection is normally to well water or a remote cooling tower.

One universal problem associated with lithium bromide-water absorption refrigerators is the leakage of air into the equipment. With water as the refrigerant working fluid, the absolute operating pressures are quite low being, respectively, about 10 mm Hg and 70 mm Hg on the low and high side components. The inward leakage of air is highly undesirable for two reasons: it accelerates corrosion, and it interfers with mass transfer. Of these, the mass-transfer problem is the least troublesome and from one aspect quite useful. If it were not for the fact that the lithium bromide solution is quite corrosive when in contact with the oxygen in the air, the leakage problem would be easily overcome with a suitable purge pump. The semihermetic commercial units employ purge pumps of one design or another, but with limited capacities. Purge pump capacity is limited as a safety measure to prevent extensive long-term corrosion damage in the event of a sizable air leak. If a sizable leak should develop, air would interfere with the refrigerant vapor mass-transfer processes and the machine would automatically lose refrigeration capacity. With a limited-capacity purge pump, the leak would have to be repaired before the machine could be restored to service. For shelter purposes, the operating period would be quite short. A much higher corrosion rate could be tolerated. Absorption equipment specified for shelter use should, therefore, be equipped with high-capacity purge pumps to insure that cooling can be maintained. This is also possible with hermetically sealed residential types which are equipped with service taps for the periodic purging of noncondensibles.

The thermodynamic performance of both the residential and the commercial types benefits from the advantages inherent in the use of a nonvolatile absorbent. Nominally, both types develop a COP of about 0.67 and a HRF of about 2.6. These values are based on quantity of heat added at the generator, the net cooling capacity developed at the evaporator, and the quantity of heat rejected at the absorber and condenser. In the case of direct-fired residential units, the COP is usually based on the heating value of the fuel. This method takes into account a heat-exchanger efficiency which does not appear in the ratings of indirect-fired units. Table 15 lists the significant information.

TABLE 15. NOMINAL PERFORMANCE OF SIMPLE-CYCLE LITHIUM BROMIDE-WATER ABSORPTION EQUIPMENT

Type	Fired	Capacity, 1,000 Btu per hr	COP	HRF
Residential	Direct	34 to 300	0.5*	2.6
Residential	Indirect	180 to 600	0.67	2.6
Commercial	Indirect	624 to 12,000	0.67	2.6

*Based on heating value of fuel.

Deviations from the nominal figures shown in Table 15 do exist and are functions of the type and make of unit, sink and evaporator temperatures, and steam pressures where the unit is indirect fired. Detailed information of this type is best obtained from individual manufacturers.

One consideration that may limit the application of lithium bromide absorption equipment to shelters is the rather narrow range of sink temperatures that can be tolerated. As already mentioned, direct air-cooled units are not available since the required upper temperature limit at the absorber cannot be maintained reliably. The problem stems from the fact that as the sink temperature rises while maintaining the same evaporator temperature, the absorbing solution in the absorber must be of a higher absorbent concentration. Unfortunately, this increase in solution concentration can only be carried so far due to the existence of a phase boundary where a precipitate of a lithium bromide hydrate begins to form. If allowed to form, the hydrate will clog the solution return line in the solution heat exchanger and perhaps portions of the absorber. In any case, solution flow is disrupted and the unit is incapable of functioning properly. Under normal air-conditioning conditions when chilled water at 40 to 50 F is being supplied, the inlet sink water temperature is limited to about 90 F. This provides a cushion of about 15 F before hydrate crystallization can occur. For shelter applications where higher chilled water temperatures of 60 to 70 F can be used, the allowable inlet sink temperature may be increased to about 100 F or perhaps as high as 110 F. A similar crystallization problem exists if the inlet sink temperature is too low. This, however, is rather easily overcome and should not pose a serious problem.

The auxiliary power requirements of lithium bromide-water absorption cycle equipment are very low. For example, a 60, 000 Btu per hr residential type direct-fired unit supplying cool air is equipped with a 3/4-hp electric motor to power the air blower. A similar unit functioning as a water chiller is equipped with a 1/6-hp motor to power the chilled-water circulating pump. The requirements for commercial-type equipment are equally low. One manufacturer lists three electric motors totaling 5 hp for a unit rated at 624, 000 Btu per hr, and only 19 hp for a unit rated at 12, 000, 000 Btu per hr. On the commercial-type units the motors are used to power refrigerant and solution pumps and in some designs a purge pump.

Load modulation on the residential type units is accomplished by an on-off control of the fuel burner. In some models two burners are used and a 50 per cent reduction in capacity is possible by operating only one burner. The indirect-fired residential-type units have steam valves to provide the same type of load modulation. The commercial units, on the other hand, have virtually infinite variability over their entire load range. Basically this is made possible by pumping to the generator only that quantity of solution that is required to carry the cooling load. Overloads do not harm either type of unit and only result in a rise in the temperature of the outlet chilled water or air. The time and skill required to bring these units to full operation from their stand-by condition depends, of course, on the degree of "mothballing". Basically, however, these devices are not complicated and operator skill required should be at a minimum. Once charged with the working fluids, supplied with a sink fluid, and provided with motivating heat, these units rapidly take up load.

Installation of both types of units is basically a plumbing job. Other than for leveling the units, the foundation need not be of special design. Once placed and leveled the units are piped or ducted to their source of cooling load and heat sink fluid. A fuel supply line and combustion air supply and flue are required for the direct-fired units. The indirect-fired units using steam require a supply line and a condensate return line. Low-power electrical lines are required to operate the pump, fan motors, and control system.

One installation requirement is storing the lithium bromide absorbing solution outside these units during the stand-by period. This solution is corrosive and for long periods of standby it would be safer to delay charging the unit with the solution. The unit itself can be kept under a deep vacuum continually. Compared to pressurizing the device with an inert gas, such as nitrogen, the vacuum arrangement will allow the rapid detection of leaks that may develop and will also reduce the required capacity of the purge pump.

Standby maintenance on these units, providing they remain sealed air tight, would be primarily a job of seeing that external valves, pumps, and controls are in operable condition. Internal pumps on the commercial-type units not charged with solution could be exercised by temporarily charging the machine with water to test circulation.

Controls are simple and of low power levels because no large electrical motors are needed. Electrical controls are used for the residential types and electrical or pneumatic are available on the commercial types.

Safety and comfort aspects of both are good. These devices operate at below atmospheric pressure, and the lithium bromide solution is corrosive, but not toxic or otherwise highly dangerous. Noise and vibration levels are both low.

Representative size data are shown in Table 16. These figures reveal that the pumped commercial-type units make more efficient use of available volume and floor space. In general, the units can be described as long and high.

TABLE 16. SIZE DATA FOR SIMPLE-CYCLE LITHIUM BROMIDE-WATER ABSORPTION EQUIPMENT

Type	Fired	Configuration	Sq Ft Per 1,000 Btu per hr	Cu Ft Per 1,000 Btu per hr
Residential	Direct	Direct expansion without cooling tower	0.2 to 0.3	1 to 1.5
		Water chiller without cooling tower	0.1 to 0.3	0.7 to 2
		Water chiller integral cooling tower	0.4 to 0.5	2 to 3
	Indirect	Water chiller without tower or boiler	>0.1	0.5
Commercial	Indirect	Water chiller without tower or boiler	>0.1	0.2 to 0.5

The installed cost of lithium bromide-water absorption equipment is shown in Figure 52.

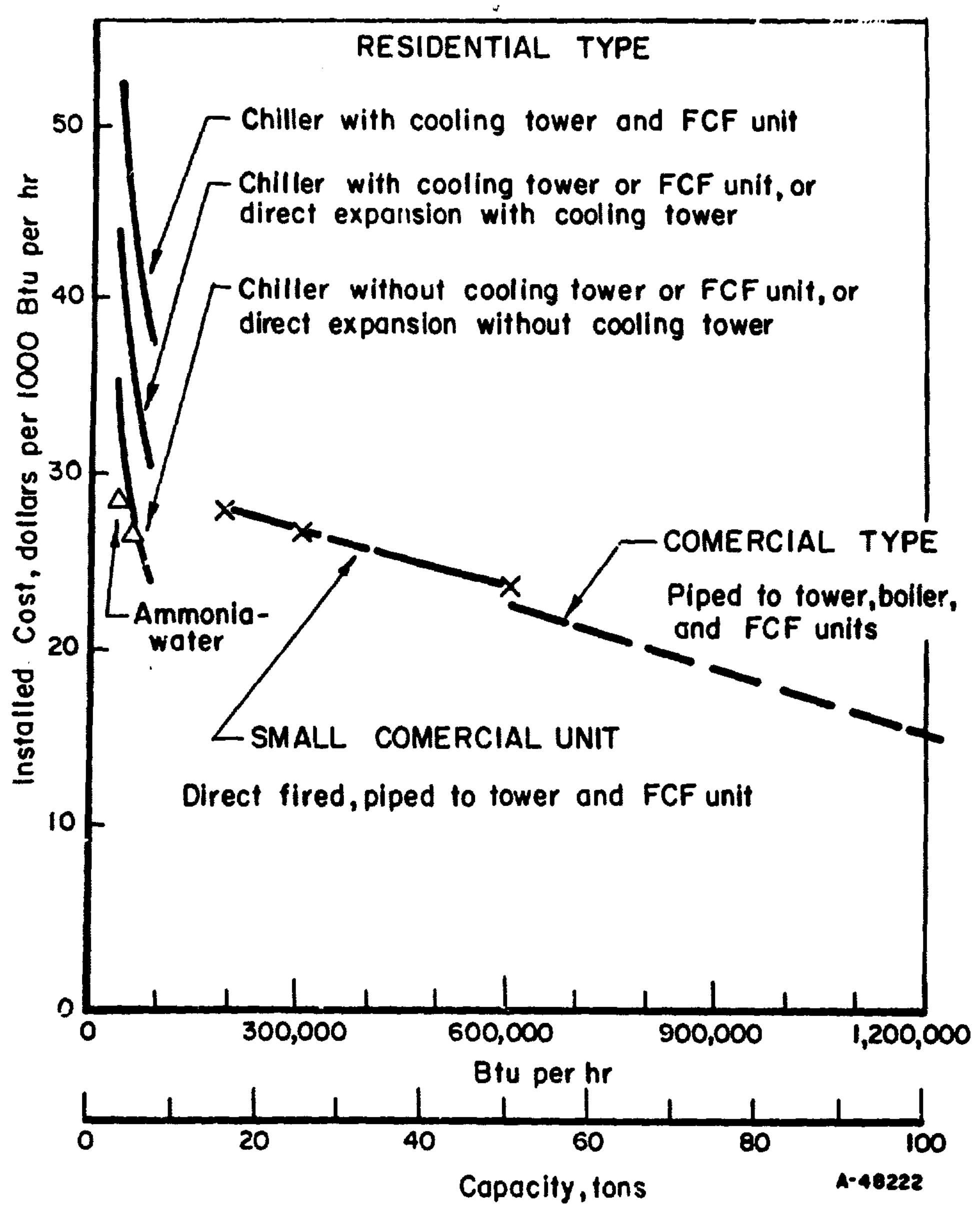


FIGURE 52. INSTALLED COST OF LITHIUM BROMIDE-WATER ABSORPTION CYCLE AIR-CONDITIONING EQUIPMENT

Absorption — Ammonia and Water

The availability of ammonia-water absorption cycle air-conditioning equipment is somewhat limited. At the present time, there is only one manufacturer on the market with relatively small capacity residential-type units.[33] Another organization has a residential-type unit well along in the field test stage but has not entered production. Larger capacity ammonia-water absorption refrigerators are built; however, these are generally very large capacity custom-designed systems for industrial process cooling applications at temperatures below 32 F. For process cooling above 32 F and general air-conditioning applications, where low-pressure steam is available, the mechanically simpler, safer, and more efficient lithium bromide-water equipment is preferred.

In the ammonia-water absorption cycle, ammonia serves as the refrigerant and water as the absorbent. The advantages of this combination over lithium bromide-water are: it can operate at evaporator temperatures lower than the freezing point of water, and it is not sink-temperature limited because of the formation of a hydrate. Some of the comparative disadvantages are: ammonia is toxic, internal pressure levels are well above atmospheric, and the absorbent is volatile. Only the second advantage, not being sink-temperature limited because of hydrate formation, justifies consideration of the use of the ammonia-water combination of air conditioning. This feature allows the residential units to be directly air cooled in contrast to the lithium bromide-water units which must be water or evaporative cooled.

The two sizes of units presently available on the market have capacities of 36,000 and 54,000 Btu per hr, are air cooled, and are designed to be gas fired exclusively. To avoid toxicity problems, they are designed as water chillers and to be installed outdoors. Construction is of the completely welded hermetically sealed type with solution transfer from absorber to generator by means of a trap system that is thermally motivated. On the basis of the heating value of the fuel, the units develop a COP of about 0.3 for a total HRF of about 4.3. Nominally they are rated with an inlet air-sink temperature of 95 F, but they have been run at up to 115 F. This is about the upper limit due mainly to pressure-vessel safety considerations. At 115 F inlet air sink temperature, the maximum working fluid pressure is about 300 psia and it rises rapidly with increases in sink temperature.

Auxiliary power requirements are low. Each unit is equipped with a 1/8-hp motor to power the chilled water pump and a 1/2- or 3/4-hp motor to power the fans for the 36,000- and 54,000-Btu per hr units, respectively. All motors are 115-v, 60-cycle, single-phase. The fans deliver 125 cfm per 1,000 Btu per hr of refrigeration.

Capacity modulation is by on-off control of the fuel supply. Time required to reach full capacity is not a problem and, as with other absorption machines, overloads do not damage.

Installation of these units is simple. Special foundations are not required. Being package units, they require only access to a supply of cooling and combustion air, a vent, connections to the chilled water circuit, electrical service, and a fuel supply. As for the direct-fired residential lithium bromide units, there is no technical reason that prevents the ammonia-water units from being directly fired with oil. Standby maintenance should be moderate on a hermetically sealed unit of this type. The corrosion problem is not nearly as severe as it is with lithium bromide-water units, which would allow the ammonia-water units to remain charged with working fluid during the standby period. The quantity of charge would require periodic checking as would the general operation of the unit to insure that pumps, fans, fuel system, and control system were fully operable.

Some safety measures must be observed. Because ammonia is used as the refrigerant, these units must be isolated from the habitable shelter area and its air intakes. High pressures are another safety consideration. Under normal operating conditions, pressures of 300 psia exist in the generator and condenser. Comfort aspects are good. Since there are no large rotating parts, these units are free of severe vibration problems. Isolation of the unit from the shelter should eliminate transmission of the sink-air fan noise.

Performance and size data are given in Table 17. The units could be described as cube shaped.

TABLE 17. PERFORMANCE AND SIZE DATA FOR AVAILABLE AMMONIA-WATER ABSORPTION EQUIPMENT

Type	Capacity	Fired	COP	HRF	Sq Ft Per 1,000 Btu per Hr	Cu Ft Per 1,000 Btu per Hr
Residential	3 to 4.5 ton	Direct	0.3[a]	4.3[b] (3.4)	0.3 to 0.4	1.7

(a) Based on heating value of fuel.
(b) 4.3 overall, 3.4 based on heat rejected at condenser and absorber.

The approximate installed costs for the 36,000 and 54,000 Btu per hr models are, respectively, $29 and $27 per 1,000 Btu per hr of cooling. These values are also shown in Figure 52.

A prototype ammonia-water unit which is under development differs in several ways from the one just described. The most important difference is that it uses an electrically driven diaphragm pump for solution transfer. One of the most difficult problems in the design of residential size ammonia-water absorption air conditioners is the transfer of the solution from the absorber to the generator. This is a high-head, low-volume situation with the additional requirements of no leakage, high reliability, and inexpensive construction. The unit presently on the market meets these requirements with a trap system. While the trap system has proven reliable, it is a thermally motivated device and quite inefficient. An electrically pumped unit is more desirable as it not only improves the COP, but allows reductions in the reservoir and heat exchanger sizes because of the more uniform pumping action. The COP of the electrically pumped unit is reported to be in the range of 0.4 to 0.45, based on the heating value of the fuel.

It is possible at the present time to build ammonia-water absorption equipment for shelter applications. Units could be of most any capacity, direct or indirect fired, and water or air cooled. Again, however, the only advantage they would have over lithium bromide-water units would be an ability to operate at somewhat higher sink temperatures. This too, however, has its limiting considerations. First, the ammonia-water absorption machine is the more structurally complicated of the two. This is partially due to the higher pressure levels that must be contained since ammonia is the refrigerant. The higher pressures dictate a construction quite different from that of the rather simple and straightforward design of lithium bromide-water units. Operation at sink temperatures above those normally available would mean an even stronger and no doubt more expensive design. A second consideration is that the ammonia-water absorption cycle is inherently less efficient thermodynamically due to the volatility of the absorbent. To improve the thermodynamic performance, analyzers and rectifiers are employed. This also requires greater structural complication. More directly, higher sink temperatures can mean the need for higher generator temperatures which increase the percentage of undesirable water vapor coming off the generator. This problem, however, can be reduced somewhat because of the higher evaporator temperatures allowable in shelter applications.

Considering the above, special ammonia-water absorption-cycle equipment would offer a direct thermally motivated refrigeration device capable of operating at sink temperatures above those possible with lithium bromide-water equipment, but with the disadvantages of higher cost, lower efficiency, and more safety hazards.

Absorption – Ammonia-Water-Hydrogen

The three-fluid absorption cycle refrigerator is a device specially designed and suited for low-capacity applications where high reliability is desired. No advantages were found that would serve as a basis for recommending further development of this device for shelter cooling.

In the preceding discussion on ammonia-water absorption devices, mention is made of the fact that the transfer of solution from the absorber to the generator is a difficult design problem. Part of the problem is brought about by the pressure difference between these two components. For example, under normal air-conditioning conditions the absorber pressure will be about 55 psia while the generator pressure may be close to 300 psia. One way to overcome this problem is to put a third noncondensible gas such as hydrogen into the low-pressure components, that is, the evaporator and absorber. Enough gas is added to bring the total pressure in these components up to that in the generator and the condenser. This eliminates the pressure difference and solution transfer can be accomplished by very simple means.

Application of the three-fluid absorption refrigeration cycle has been in the field of home and portable-type refrigerators; it has not been used in other applications because of its awkward size and shape. This concept permits a completely hermetic design with all fluid transfer motivated thermally. The cooling capacity per cubic foot of machine is quite low, partially because of the hydrogen in the evaporator and absorber. To elaborate, the transfer of refrigerant between the liquid and vapor phases in the evaporator and absorber becomes a mass diffusion process through a second gas. Mass diffusion rates under these conditions are significantly lower than those possible when only the refrigerant vapor is present. For very small capacity applications, such as household refrigerators, the size of the unit is not unreasonable. However, for air-conditioning capacities the size of the components becomes far too large to be practical. Another consideration is relative placement of components. Since much of the internal working fluid circulation is by gravity due to density differences, there are limitations on the vertical placement of components.

Absorption – Double Effect

The objective of using the double-effect absorption principle is to improve the efficiency or COP of thermally motivated absorption refrigerators. The fundamental approach is to increase the potential thermodynamic efficiency of the engine section of the cycle by adding the motivating heat at a higher temperature level than is possible in the simple absorption cycle. To accomplish this, an additional component, termed a first-effect generator, is added to the components normally associated with the simple absorption cycle.

Figure 53 shows schematically the arrangement of the components. During operation, the first-effect generator accepts a solution rich in refrigerant from the absorber by way of two solution heat exchangers. Externally supplied motivating heat is added to the solution and approximately half the refrigerant to be removed from the solution is vaporized at the first-effect generator. The somewhat depleted solution is then conducted to a second-effect generator. In a simple absorption cycle this would be the only generator.

The second-effect generator is not heated by an external source but by the condensing refrigerant vapor produced in the first-effect generator. In other words, the second-effect generator serves as the condenser for the first-effect generator. The vapor from the second-effect generator is then liquefied in the usual condenser through rejection of its latent heat to the sink. The refrigerant that is condensed after heating the second-effect generator is also conducted to the condenser where a small portion flashes off into vapor. The flash vapor is subsequently liquefied by additional heat rejection to the sink. The two liquefied refrigerant streams at sink temperature in the condenser are then available for throttling to evaporator conditions to absorb the cooling load. Next, the refrigerant vapor produced in the evaporator is taken into solution at the absorber with heat rejection to the sink to complete the cycle.

An idealized analysis of the simple absorption cycle would show that it has a maximum possible COP of 1. Such an analysis would be based on ideal conditions. It would be assumed that the latent heat of the refrigerant is constant and is the only significant quantity to be considered. A similar analysis of the double-effect cycle would show a maximum COP of 2 possible. Of course, neither cycle permits this maximum to be reached because of inevitable thermodynamic irreversibilities.

Equipment operating on the double-effect principle is just becoming available. At present there is only one manufacturer marketing such a unit.[34] This is a direct-fire type of 180,000 Btu per hr capacity using lithium bromide and water as the working-fluid combination. Larger unit capacities are planned. The construction is primarily of steel, welded to form a hermetic unit similar to the simple-cycle residential unit already described. A necessary addition, however, is a solution pump to transfer the solution from the absorber to the first-effect generator against an increased pressure difference.

The available units are designed as water chillers and use water supplied from an external source or a cooling tower as a heat sink. While the units are fitted for direct firing with gaseous fuels, modification for oil firing should not be a major development problem. Indirect-fired units are possible; however, steam pressures higher than 15 psig will be required due to the higher temperatures required in the first-effect generator.

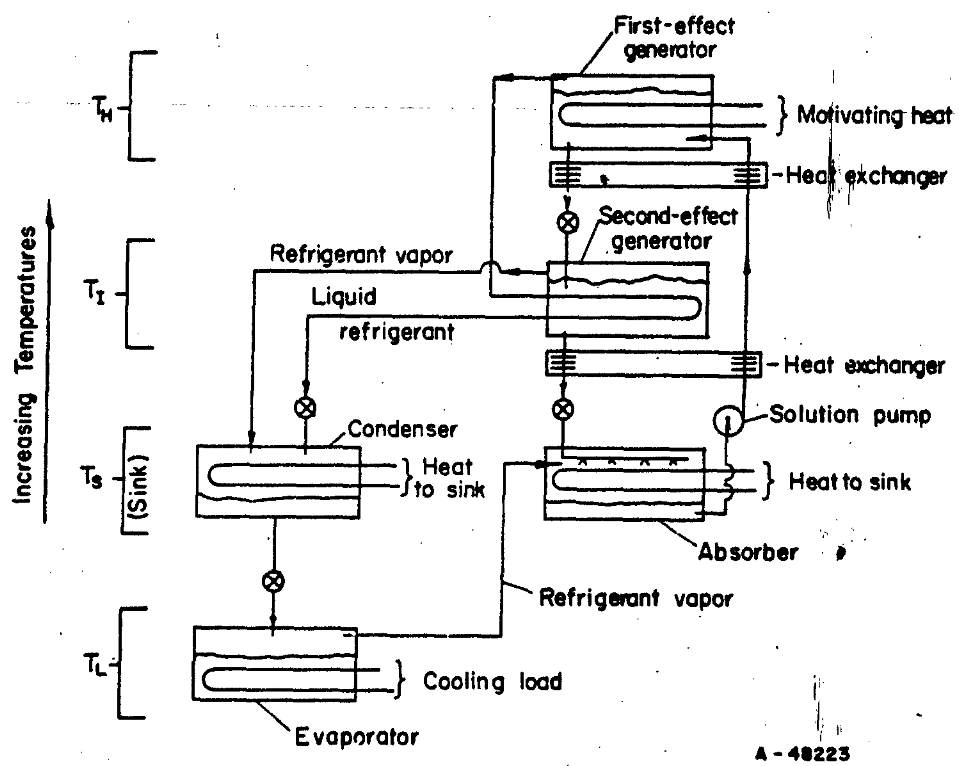


FIGURE 53. ARRANGEMENT OF COMPONENTS FOR THE DOUBLE-EFFECT ABSORPTION CYCLE

From a performance standpoint, these units achieve a COP of 0.75 based on the heating value of the fuel supplied. The HRF is approximately 2 based on the heat rejected at the condenser and absorber. These are significant improvements, the previously described simple-cycle equipment having about 0.5 COP and 2.6 HRF.

In general, these double-effect units have the same advantages and disadvantages as the simple-cycle units of similar construction. One exception may be in the area of corrosion. The double-effect units reach higher solution temperatures than do the simple-cycle units. This aggravates the corrosion problem that is always associated with lithium bromide solutions. Experience will reveal whether this potential source of trouble has been adequately controlled. For shelter applications, corrosion should not be a problem because of the short operating period.

Table 18 lists some pertinent performance and physical data.

TABLE 18. PERFORMANCE AND PHYSICAL DATA FOR DOUBLE-EFFECT ABSORPTION EQUIPMENT

Type	Unit Capacity	How Fired	COP[a]	HRF[b]	Sq Ft Per 1,000 Btu per Hr	Cu Ft Per 1,000 Btu per Hr
Small commercial	180,000 Btu per hr	Direct	0.75	2.0	0.1	1

(a) Based on heating value of fuel and net cooling effect.
(b) Based on heat rejected at condenser and absorber and net cooling effect.

Absorption – Absorption-Resorption

Absorption cycle refrigeration devices that operate on the absorption-resorption principle[35] are complex and equipment of this type is not available. The objective and fundamental approach in using the absorption-resorption cycle is the same as with the double-effect cycle. The intention is to improve the COP by adding the motivating heat at a higher temperature level than is possible in a simple absorption cycle. To accomplish this, two additional components, a resorber and a second evaporator, are added as shown in Figure 54.

Starting at the condenser with liquid refrigerant available at sink temperature, the first process is throttling to the first evaporator where half the total cooling load is absorbed. The refrigerant vapor produced in the first evaporator is taken into solution at the resorber with rejection of heat to the sink. From this point the simple absorption cycle and the absorber-resorber cycle differ in operation. In a simple cycle, the solution rich in refrigerant would be transferred to the generator by way of a solution heat exchanger. In this cycle the solution is sent to a second evaporator by way of a solution heat exchanger. The heat exchanger partially cools the solution which is then throttled to the lower pressure and temperature level existing in the second evaporator. In the second evaporator, refrigerant is driven from solution into the vapor phase as it absorbs the remainder of the cooling load. The solution weak in refrigerant is returned to the resorber by means of a pump and via the solution heat exchanger. The refrigerant vapor produced in the second evaporator is taken into solution at the absorber with rejection of heat to the sink. To do this, the solution in the absorber must be quite weak in refrigerant. Producing this weak solution and still maintaining the vapor pressure needed to condense refrigerant vapor at sink temperature in the condenser requires that the generator operate at a higher temperature level than in a simple cycle.

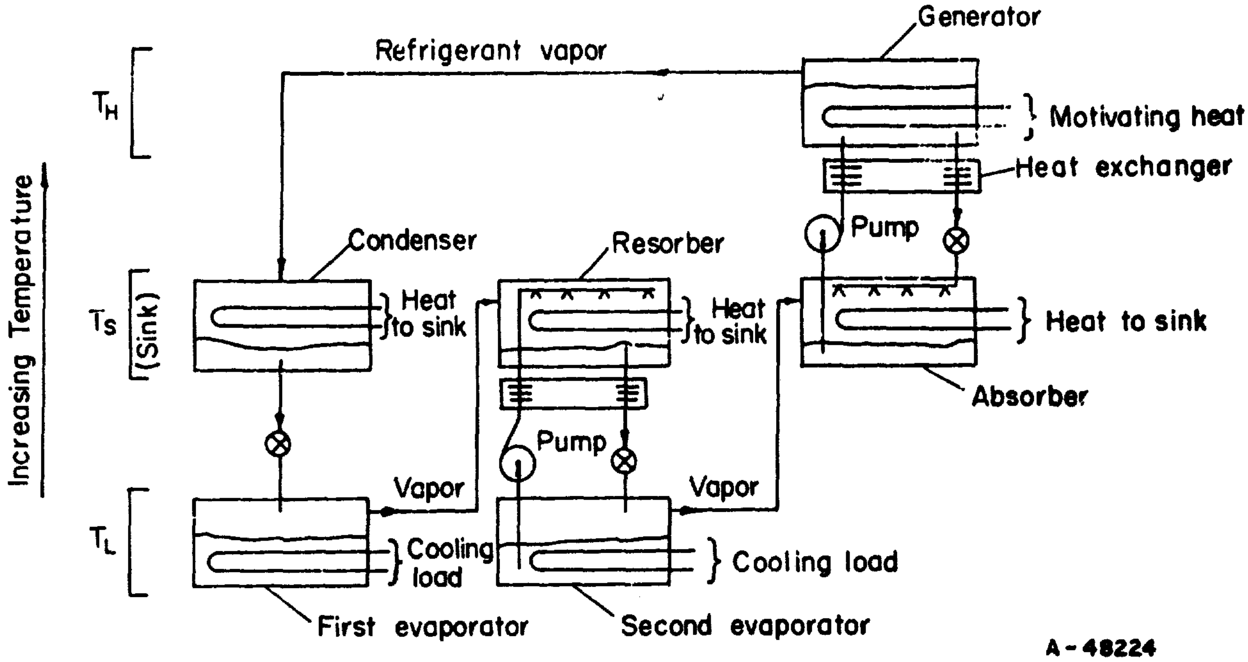


FIGURE 54. ARRANGEMENT OF COMPONENTS FOR THE ABSORPTION-RESORPTION CYCLE

Equipment operating on this cycle is not available because of limitations imposed by practical working fluid combinations. For example, the lithium bromide-water combination is unsuitable for two reasons. First, the lithium bromide would crystallize and, second, the extremely low pressures and correspondingly high specific volumes of the water vapor in the second evaporator and absorber would make the size of these components unreasonably large. While the ammonia-water combination is not subject to crystallization or size problems, it is unsuitable because of the excessive mechanical complication of the equipment. The high generator temperatures mean increased water vapor content in the vapors coming from the generator. This, in turn, requires more effective analyzers and rectifiers. For the combination of ammonia and water, the gain in thermodynamic efficiency offered by this cycle does not justify the tremendous added mechanical complication.

Adsorption

Another possibility with direct thermally motivated closed-cycle refrigeration devices is to use a solid adsorbent in place of the usual liquid absorbent. This has been attempted in a number of cases with very limited success. Most of the suggested designs have employed intermittent operation in order to circumvent the problems associated with transferring a solid between the high- and low-pressure sides of the cycle. Other inherent problems are: (1) poor heat transfer to and from the adsorbent material during the adsorption and regeneration processes and (2) physical deterioration of the adsorbent material.

Adsorption equipment of this type is not presently available. No basis for recommending its development for possible use in the shelters could be found.

Ejector

Closed-cycle vapor-compression refrigeration devices can employ ejectors to accomplish the compression process. A schematic arrangement of the components forming the ejector cycle is shown in Figure 55. As with other direct thermally motivated refrigeration cycles, there are three temperature levels. The condenser operates at the sink temperature, the evaporator below it and the vapor generator above it. The ejector raises the vapor pressure of the refrigerant vapor from evaporator conditions to a pressure that permits the vapor to condense in the condenser at the sink temperature.

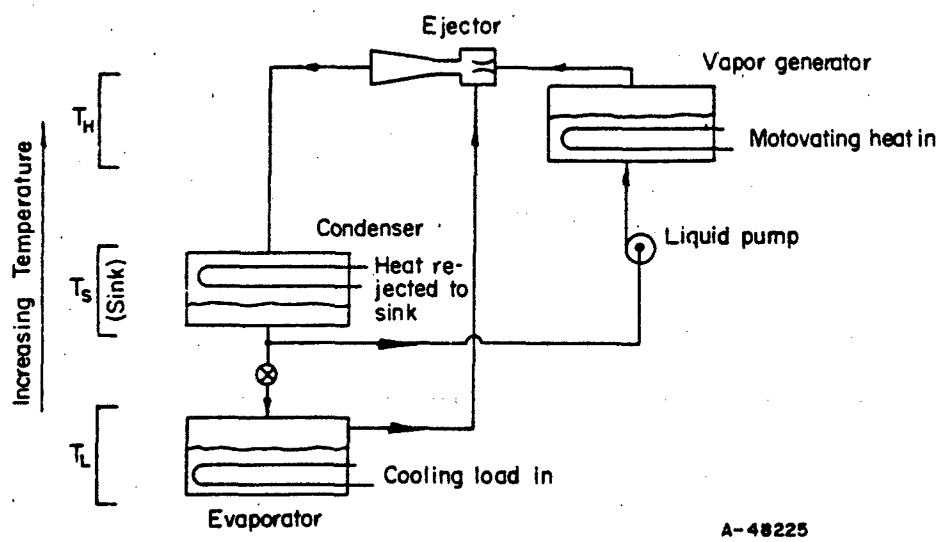


FIGURE 55. ARRANGEMENT OF COMPONENTS FOR THE EJECTOR CYCLE

For shelter application the ejector device offers two basic advantages over other refrigeration devices. First, the device is static; with the exception of small auxiliary liquid pumps it has no major moving parts. This makes it extremely reliable, low on maintenance requirements, and easy to operate. Second, water is used both as the refrigerant-working fluid and as the engine-section or motive-working fluid. Water costs are reasonable and water is often easily available at a site; therefore, providing for possible loss of working fluid during operation does not present critical technical or economic problems.

The disadvantages of the ejector cycle are: it is usually somewhat inefficient thermodynamically and it is very sensitive to the temperature span over which heat is to be pumped.

The ejector-cycle refrigerator has had fairly wide application in the past; is employed today, but mainly for specialized purposes. A few examples of past applications are: residential air conditioning, commercial air conditioning, and railroad freight-car refrigeration. Railroad use stopped when the steam locomotive was replaced by the diesel type. Among other things, the lower thermal efficiency of the ejector cycle made it uneconomical for continuous air-conditioning applications, and it is no longer used by the major air-conditioning equipment manufacturers. However, water-chiller units from about 300,000- to 6,000,000-Btu per hr capacity are still available on a custom-built basis.(36) Units have also been built with capacities in the range of 70,000 Btu per hr and some employed two different working fluids. None were outstandingly successful.

The deficiencies of ejector-cycle equipment are the result of the thermodynamic performance and operating characteristics of the ejector as a vapor-compression device. Thermodynamically it is an inefficient device. Losses due to turbulence are significant. The mixing process in which the high-velocity motive-working fluid entrains the low-velocity refrigerant-working fluid involves the degradation of highly available kinetic energy into less available forms. Since supersonic velocities are attained, some of the pressure recovery in the diffuser sections is across a shock boundary in an irreversible process. The remainder of the pressure recovery occurs in a normal divergent subsonic diffuser which is limited, by practical considerations of length, to something less than a optimum configuration. These inefficiencies result in lower coefficients of performance and, therefore, higher rejection factors as compared with other direct thermally motivated refrigerators.

Ejectors are designed to pump a given quantity of refrigerant over a given vapor-pressure difference with a corresponding given motivating-fluid pressure. Deviations from these design conditions produce different results. For example, if the cooling load increases there is a corresponding increase in evaporator temperature and vapor pressure. The ejector makes good use of this since it is able to pump a larger quantity of vapor over the reduced vapor-pressure difference. On the other hand, a decrease in cooling load can result in a freeze-up in the evaporator since the ejector must pump a reduced amount of vapor over an increased pressure difference. Load can be modulated by throttling the motivating fluid, but this is somewhat inefficient. The most serious operating problem arises when an ejector is running at design load and the condensing temperature rises. If this exceeds a certain limit, the ejector can no longer pump vapor over the increased pressure difference. When this occurs, the hot motivating fluid is diverted to the evaporator and the device becomes a heater rather than a refrigerator. Controls can be devised to prevent this from occurring.

The ejector equipment available for air-conditioning applications uses water as both the refrigerant and the motivating working fluid. Evaporators are of the open or flash-tank type which make possible reducing the temperature and vapor pressure lift to a minimum. Condensers are either of the surface type, cooled by water from a well or cooling tower, or of the direct evaporative-cooled type. The direct evaporative-cooled condenser is another device employed to reduce the condensing temperature and, therefore, the vapor-pressure lift against which the ejector must operate. The ejectors are normally designed to operate with 100-psig steam. They will operate quite inefficiently at 2 psig. Steam pressures over 100 psig produce very slight gains in performance.

Some typical performance curves for steam ejector refrigeration equipment are shown in Figure 56. For a condensing temperature of 105 F and a leaving chilled water temperature of 60 F, the steam rate is approximately 1.55 lb per hr per 1,000 Btu per hr for a COP of 0.64 and a HRF of 2.7. This condition could be obtained with a surface-type condenser cooled by 85 F water from a cooling tower if it is assumed there is a 10 F rise through the condenser and the internal condensing temperature is 10 F above that, or 105 F. Direct evaporative cooling can provide lower internal condensing temperatures with corresponding increases in operating efficiency and reduced heat-rejection factors. Chilled-water temperatures below 60 F have the opposite effect as shown in Figure 56.

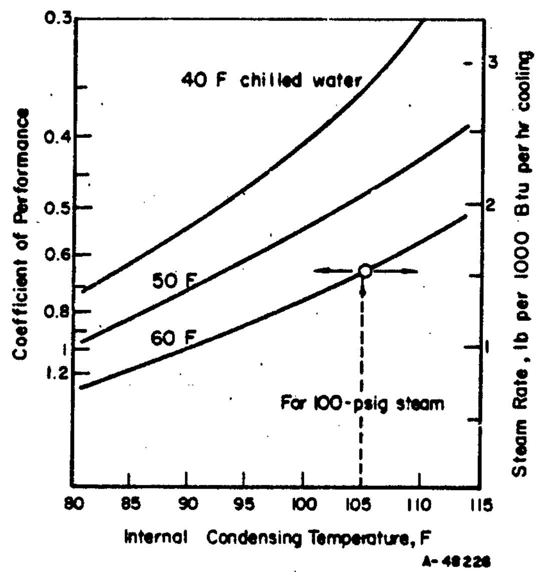


FIGURE 56. TYPICAL PERFORMANCE CURVES FOR STEAM-EJECTOR REFRIGERATION DEVICES

Presently available equipment is normally purged of noncondensibles with two-stage ejector systems. The capacities of these systems are several times these required. This is advantageous for shelter applications, because during stand-by this unit could be maintained at atmospheric pressure and during operation some air leakage would be tolerable.

Approximate purchase costs of presently available ejector equipment are shown in Table 19. Quantity production would reduce these prices. The assumptions are for units supplying 60 F chilled water; in the case of the units employing surface-type condensers, 85 F cooling water is to be supplied.

TABLE 19. APPROXIMATE PURCHASE PRICE FOR EJECTOR REFRIGERATION EQUIPMENT

Tons	1,000 Btu per Hr	Cost, dollars per 1000 Btu per hr	
		With Surface Condenser	With Evaporative Condenser
25	300	36.7	62.5
50	600	20	33.4
100	1,200	11.7	20

Negative Heat of Solution

Some fluid combinations exhibit negative heats of solution which suggest the application of endothermal mixing processes to a direct heat-motivated refrigeration cycle.[37] Considering a cycle with two fluids, A and B, Fluid A would be volatile and Fluid B relatively nonvolatile. Starting with the separated fluids at sink temperature, the first process is the endothermic mixing process in a heat exchanger. The resulting solution undergoes a temperature decrease, which allows it to absorb heat from a heat source below sink temperature. After the heat source has been appropriately cooled, the solution is transferred to a distillation chamber for separation. Externally supplied heat is added to vaporize Fluid A which then condenses, rejecting heat to the sink. Fluid B, not mixed with Fluid A, flows from the distillation chamber to be cooled to sink temperature, thereby completing the cycle.

The cooling capacity of such a cycle is directly related to the magnitude of the negative heats of solution. Also, the quantity of motivating heat required at the distillation chamber to effect the separation of A from B is directly related to the latent heat of vaporization of Fluid A. The ratio of the heat of solution to the latent heat of vaporization is a good approximation of the COP possible. Unfortunately, negative heats of solution are smaller than latent heats of vaporization, and in most combinations significantly smaller. Possible thermodynamic efficiency is obviously low.

As with most direct thermally motivated refrigerators, the practical requirements for an acceptable combination of working fluids are many, and few designs are satisfactory. Many of the combinations possible must be eliminated for such reasons as: thermal instability, chemical instability, extreme corrosiveness, toxicity, and formation of precipitates.

In general, this concept does not appear to offer any significant advantages over other direct thermally motivated devices and certainly none for the shelter situation.

EXPENDABLE FLUIDS – MECHANICALLY OR THERMALLY MOTIVATED DEVICES

Discussed here are mechanically or thermally motivated refrigeration devices that operate with loss of the refrigerant to the atmosphere. In essence, these open-cycle devices are the equivalent of previously described closed-cycle devices without their condensers and condenser heat sinks. The refrigerant vapors are pumped from the evaporator to the atmosphere, rather than to a condenser, by mechanical compressors, thermally motivated fluid ejectors, or thermally motivated absorption processes. Thus, the quantity of refrigerant expended is that required to absorb the cooling load. Data on heat sinks show that water and methyl alcohol are the only two fluids which are not too costly for use as expendable refrigerants. Table 20 gives data pertinent to the evaluation of devices using these fluids.

TABLE 20. METHYL ALCOHOL AND WATER PROPERTIES PERTINENT TO OPEN-CYCLE REFRIGERATION DEVICES

	Temp, F	Pressure, mm Hg	Specific Vol, cu ft per lb	Latent Heat, Btu per lb	Ratio, atm pressure to evaporator pressure
Methyl alcohol	60	79	123	500	10.4
Water	60	13.3	1207	1060	57

Mechanical Vapor Compression – No Condenser

An open-cycle MVC device utilizing methyl alcohol as the working fluid can be used economically to cool a shelter if:

1. No aboveground equipment can be used
2. Power is available
3. Heat sink has to be stored
4. Occupied portion of the shelter is sealed.

Water is not applicable as a working fluid because of its high specific volume; the large ratio between evaporator pressure and atmospheric pressure would demand unreasonably large compressors and power supply.

As indicated by Figure 57, cooling with a MVC device without a condenser requires provision of a compressor, an evaporator, and sufficient liquid refrigerant to absorb the entire heat load.

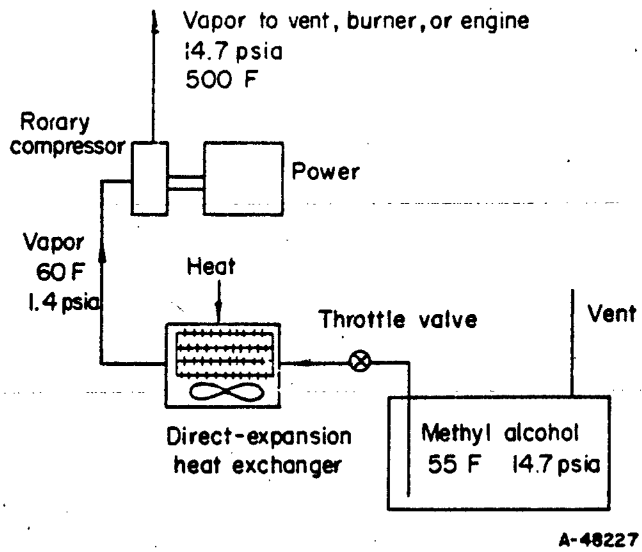


FIGURE 57. OPEN-CYCLE MVC USING METHYL ALCOHOL

Assuming use of methyl alcohol and adiabatic compression at a thermal efficiency of 0.75, the calculated operating charactertistics of an open-cycle MVC device are:

Cooling rate, Btu per hr	1,000
Alcohol flow rate,	
lb per hr	2
cfm vapor at evaporator	4
Compression ratio	11
Compressor power input, hp	0.1
Vapor temperature, F	
Compressor inlet	60
Compressor outlet	500

Two problems which must be solved concern the most suitable design for a compressor to pump the vapor and disposal of the alcohol vapors. It seems possible that a rotary-vane compressor with some type of carbon vanes would be satisfactory as a compressor. Such machines are used for vacuum pumps having pressure differences similar to those associated with the compression of the alcohol vapor. The cost and the applicability of these machines to pumping alcohol vapor must be established. The cost of an alcohol vapor compressor should be about equal to that of a commercial rotary air compressor, or $5 per cfm. Reliability, maintenance requirements, and safety of operation would probably be better than with a closed-cycle refrigeration device.

The toxic alcohol vapors leaving the compressor could be discharged into the atmosphere or burned in a relatively simple naturally aspirated burner. It might be worth while to use the vapors as fuel for an engine. Methyl alcohol is chemically stable and can be stored for long periods of time. During standby, the compressor, evaporator, and piping could be filled with dry air at atmospheric pressure. At startup the compressor would prime the system and put it into operation. During operation the pressure in the system would be below shelter pressure. This would eliminate the possibility of alcohol leakage into the occupied area. Air leaking into the system would be pumped out with the vapor.

The cost of cooling with methyl alcohol in an open-cycle MVC device is attractive as compared with the cost of other methods which could be used under the same circumstances. This method is less costly than using ice because of the lower cost of storing the alcohol with its higher heat absorption at the phase change. The costs of using such equipment for cooling are summarized in Table 21.

TABLE 21. ESTIMATED COST FOR COOLING WITH OPEN-CYCLE MVC AND METHYL ALCOHOL

(Dollars per 1,000 Btu per hr for 300,000 Btu per hr for 14 days)

	Type of Alcohol Storage		
	0 psi	30 psi	100 psi
Alcohol	30	30	30
Alcohol storage	12	54	94
Air-handling unit (direct expansion)	11	11	11
Engine-generator set	15	15	15
Electric motor and wiring	4	4	4
Compressor at $5 per cfm	20	20	20
	92	134	174

Absorption – No Condenser

A thermally motivated open-cycle device would resemble a conventional lithium bromide-water absorption device without the condenser. In such a device the lithium bromide-water solution would be pumped from the absorber to an atmospheric pressure generator where heat would be supplied to drive the water vapor out of solution and into the atmosphere. The lithium bromide would be returned to the absorber to complete its cycle. The absorber would have to be cooled by a heat sink to remove the latent heat of the water and the heat of solution of the water in lithium bromide.

Of the heat rejected from the open-cycle device, approximately one-half would be exhausted with the water vapor leaving the generator and the other half would have to be handled by the absorber cooling water. However, this would not allow the open-cycle device to operate with less cooling water than a closed-cycle device, because in both devices the condensers are cooled by water previously used to cool the absorber. If the absorber outlet water were not used to cool the condenser, it would be dumped at a lower temperature. Therefore, since there is no reduction in the cooling water requirement, the only advantage of the open-cycle device is that no condenser is required. It seems doubtful that this reduction in a conventional component would warrant the provision of higher pressure solution pumps and the higher temperature heat source would be required to operate an atmospheric-pressure generator.

Ejector – No Condenser

Ejectors are, by far, too inefficient to be considered as vapor compressors for an open-cycle MVC device. As for the closed-cycle ejector device, only methyl alcohol and water are practical working fluids. It is doubtful that a single-stage steam ejector working at a pressure of 250 psi would compress any water vapor from a pressure of 0.25 psi (13 mm Hg), which would exist in the evaporator, to atmospheric pressure.[38] One lb of steam at 280 psi might pump 0.3 lb of methyl alcohol vapor over the pressure ratio of 11 required to operate a methyl alcohol evaporator. Higher steam pressures would not significantly increase the amount of alcohol pumped in relation to the amount of steam used. The amount of methyl alcohol pumped per lb of steam would be about 0.17 lb (0.3 divided by 1.8) because the molecular weight of alcohol is about 1.8 times that of water and momentum is conserved in the ejector. Therefore, a steam ejector compressing methyl alcohol vapors from evaporator pressure to atmospheric pressure would require the following approximate fluid flows for 1,000 Btu of cooling:

	Pounds
Methyl alcohol	2
Water for steam	11
Fuel oil to boiler	1
	14

The space required to store the above fluids would be approximately 0.23 cu ft which would be equivalent to a cooling capacity of 4,350 Btu per cu ft of storage space. This is about one-half the 9,000 Btu per cu ft of cooling that can be obtained with ice and, therefore, ejector compressors would not be applicable for open-cycle operation with stored working fluids.

EXPENDABLE FLUID – SELF-MOTIVATED DEVICE

The open-cycle absorption concept involves storing or having available sufficient quantities of liquid absorbent and refrigerant. During the operation of the device, the refrigerant is throttled to an evaporator where it vaporizes as it takes up the cooling load. The absorbent solution maintains evaporator conditions by continuously removing refrigerant vapor from the evaporator and taking it into solution in an absorber. The heat of absorption that is generated in the absorber is rejected to a heat sink. Absorbing solution that becomes rich in refrigerant is removed from the absorber and piped to a spent-solution reservoir. The spent absorbing solution and refrigerant are replaced from storage.

Important to the shelter situation is the extremely low requirement of these devices for external motivating energy. Since the ability to function as a heat pump is entirely in the working fluids as chemical potential, external energy requirements are limited to such items as circulating and purge pumps. Under certain sealed shelter conditions these low-level power demands could be met by storage sources such as batteries or even manually. However, the absorption process must be cooled and access to a heat sink is a definite necessity.

The search for suitable combinations of refrigerant and absorbent was conducted with definite technical and economic requirements in mind. First, the refrigerant should have an atmospheric-pressure boiling point above 70 F. (Materials with lower atmospheric-pressure boiling points can be used directly as expendable heat sinks that exhaust directly to the atmosphere without an absorption process.) Second, the absorbent should have a high chemical affinity for the refrigerant and the refrigerant should have a reasonably high latent heat of vaporization so that storage facility sizes and costs can be kept within reason. Third, both refrigerant and absorbent should have reasonable raw material costs. Fourth, only liquids should be considered acceptable because of the many difficulties associated with the application of a solid to a device of this type.

In the course of the search such obvious combinations as lithium bromide-water and ammonia-water were eliminated, the former because of the high cost of the absorbent and solids problems, the latter because of the low atmospheric pressure boiling temperature of the ammonia refrigerant. Water as a refrigerant and several glycols as the absorbent were also investigated. These were determined to be unsatisfactory on the basis of limited affinity and the relatively high cost of the glycols. Of the many combinations possible, only sulfuric acid and water seemed applicable from practical and economic standpoints.

Absorption – Sulfuric Acid and Water

The combination of absorbent and refrigerant most suitable is the combination of sulfuric acid and water. Little new can be said about the advantages of water. It has a high latent heat of vaporization and as one of the naturally available refrigerants it often need not be stored. Sulfuric acid has the following advantages: a very high affinity for water, low cost because it is one of the world's basic commodities, and ready availability. The high affinity of sulfuric acid for water gives this combination a high ratio of cooling capacity to storage volume plus the potential for high sink temperature operation. The main disadvantages of sulfuric acid are, of course, connected with its corrosiveness.

Ultimately, the feasibility of using the sulfuric acid and water combination is directly dependent on how well the corrosion problem can be resolved. In other words, if it is to be applicable to shelters, means must be devised to store, use, and dispose of sulfuric acid in such a manner that it is entirely safe and still relatively inexpensive to use. Figure 58 is a schematic layout of a proposed system to satisfy these requirements. As shown, the system is composed of three separate facilities to perform the following three functions: pre-operative storage, actual operation, and post-operative storage. Careful consideration of the varied requirements imposed by these three different functions indicated that they could best be met by separate specially designed components. For example, the length of the pre-operative storage period, while not actually known, might be 10 years. On the other hand, the operative period will be relatively short, about 2 weeks.

For the pre-operative period, it is proposed that mild steel tanks be used for storage of both sulfuric acid and water. The acid to be stored is known to the industry as commercial-grade 66 degree Baumé (93.2 per cent H_2SO_4). Fortunately, concentrated sulfuric acid is much less corrosive to mild steel than dilute solutions. This permits the large quantities of this commodity that are manufactured each day to be economically stored, shipped, and handled in relatively inexpensive mild steel containers. Another favorable factor is that a great deal of sulfuric acid handling equipment and experience is already available.[39]

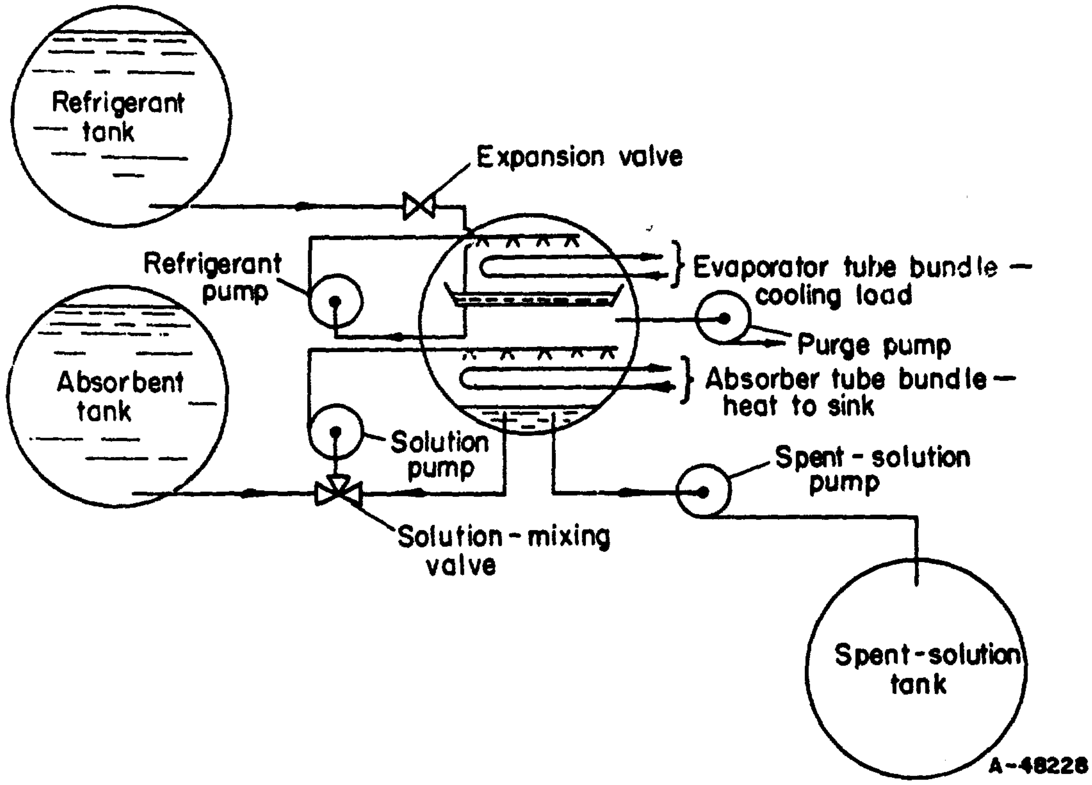


FIGURE 58. OPEN-CYCLE ABSORPTION REFRIGERATION DEVICE

Acid storage tanks would obviously have to be external to the habitable shelter area and so located that a ruptured tank would not pose a danger. It is recommended that the tanks not be buried but provided with ample clearance on all sides to permit inspection and necessary repairs. Since the pre-operative storage tanks would be separate and not built in as an integral part of the over-all system, they could be of almost any size, shape, or number.

The layout of the proposed refrigeration device is identical with that of the absorber-evaporator drums used on some types of commercial lithium bromide-water absorption refrigerators. The evaporator section is located in the upper part of the drum and the absorber in the lower. Circulation pumps for both the liquid refrigerant and absorbent keep the respective heat-exchanger tube bundles covered to provide the needed surface area for mass transfer. The solution mixing valve can be controlled to admit only that amount of concentrated acid required to maintain the cooling load. During the stand-by period the drum can be kept under vacuum continually. This greatly reduces the size of the purge pump while reducing the time required to reach operating conditions. Since the operating drum is separate from the acid storage, air leaks can be detected and easily repaired.

The basic materials of construction for the operating drum can be mild steel with, perhaps, some critical parts, such as the absorber tube bank and spray nozzles, made of more corrosion-resistant materials. The design should take into account the short operating period by allowing for higher rates of metal loss than are presently permitted for commercial absorption units.

The post-operative storage tanks can also be of mild steel. The solution handled here will be more dilute and, therefore, more corrosive. The thought, however, is that these tanks should be designed for a relatively short storage term. If post-operative conditions are at all favorable, this solution can be removed to be used for other purposes or neutralized with readily available materials such as limestone.

Several possibilities for reducing the sulfuric acid corrosion problem were investigated including: use of alternative materials of construction, linings, and inhibitors and use of electrical methods.

Alternative materials of construction that are more resistant to sulfuric acid corrosion than mild steel are available. These are primarily the stainless steels. While it is technically feasible to employ stainless steels, first costs would be 3 to 5 times that of mild steel because of raw material cost. Linings such as glass and some plastics could be applied on mild steel for protection. At present cost, these protectives would increase tank costs again by a factor of 3 to 5 times that of plain mild steel. Production quantities could improve the situation.

The use of chemical corrosion inhibitors looks promising but needs further investigation. Use would be similar to the use of inhibitors in steel-pickling operations. Such inhibitors are primarily amines which allow dilute sulfuric acid solutions to attack the scale but afford some protection to the base metal. Unfortunately, effectiveness of these organic compounds in concentrated solutions is unknown and their chemical stability is questioned. Arsenic is another inhibitor often mentioned, but again no information was found about its effectiveness in concentrated solutions.

Cathodic and anodic electrical protection methods were reviewed. The cathodic method appears undesirable. The required current densities are high and on a continuous basis would be costly. Also, it is difficult to insure freedom from preferential attack due to variations in local current densities. The anodic method, on the other hand, uses intermittent electrical power to build up an oxide film on the metal to be protected. A reference electrode immersed in the corrosive solution is electrically connected to a controller which monitors cell potential between the reference electrode and the tank wall. When the cell potential indicates that the protective film is nearing depletion, the anodic potential is automatically applied to re-establish the film. This method has been applied to the storage of concentrated sulfuric acid in mild steel tanks and 10-fold decreases in iron pickup have been reported.(40)

For complete cooling system cost estimates, calculations were made to determine quantities of working fluids required and heat rejection factors. The results are shown in Figure 59. The assumptions are: an internal refrigerant temperature of 55 F, working fluids stored at 80 F, the solution rejected at the temperature shown is in vapor pressure equilibrium with the refrigerant, and the use of 66 degree Baumé sulfuric acid.

The current price for the sulfuric acid specified is about $24 per ton or 18 cents per gallon. This is very reasonable and is a major factor in the economic feasibility of using an open-cycle absorption refrigerator. For example, assume 120,000 Btu per hr of cooling are to be provided for a period of 24 hours with sink conditions that dictate a spent-solution temperature of 140 F. The total cooling capacity required is about 3,000,000 Btu. From Figure 59 approximately 0.17 gallon of acid is required per 1,000 Btu of cooling. Therefore, approximately 500 gallons of acid are needed. This would cost somewhat less than $100.

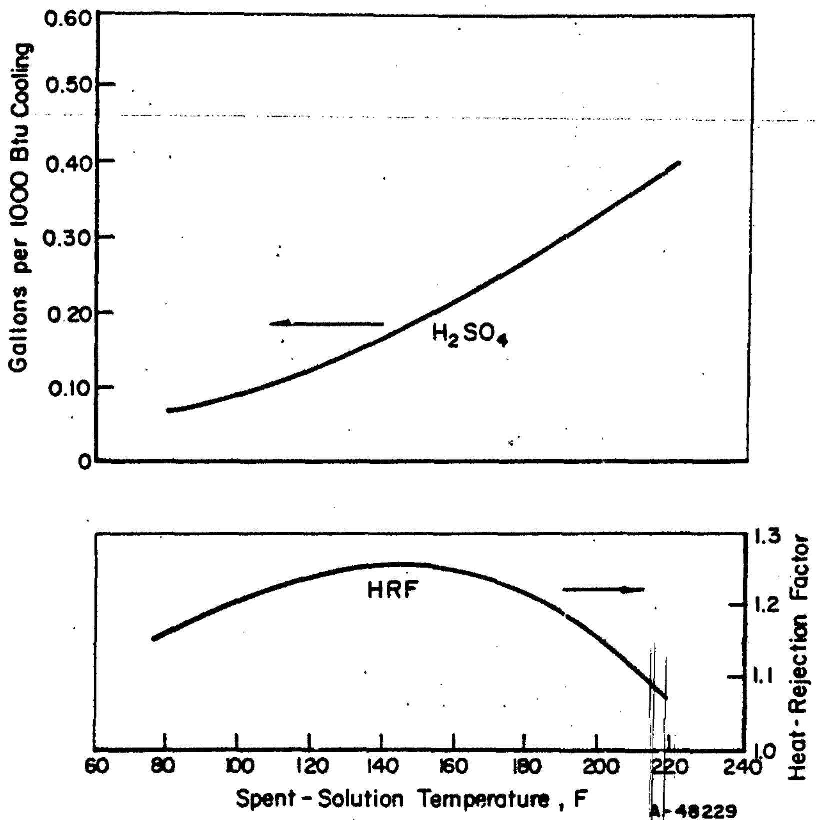


FIGURE 59. CONCENTRATED SULFURIC ACID REQUIREMENTS AND HEAT-REJECTION FACTOR FOR OPEN-CYCLE ABSORPTION REFRIGERATOR

The cost of the absorber-evaporator with pumps, valves, etc., is estimated at one-half that of a commercial two-drum-type bromide-water absorption machine of equal capacity. This is based on the close similarity of the two units.

A potential capability not to be overlooked with this device is that of being able to reject heat to a boiling water sink. Theoretically this is entirely possible; however, the practical feasibility of accomplishing this is again basically a question of corrosion. Solution temperatures would be high, in the range of 220-240 F. Elevated temperatures definitely increase corrosion rates, but this would be offset to some degree by the higher solution concentrations that must be maintained. Corrosion resistant materials plus the short operating period could combine to make this type of operation feasible. A proper evaluation will require further investigations, preferably under actual operating conditions.

The unique technical advantage of such a device is that it can make use of such a limited type water supply as a heat sink when a shelter is sealed off from the atmosphere. Economically, the ability to reject to boiling water offers a reduction in required storage volume compared to simply increasing the stored water temperature by 40 or 50 F. While storage volume for concentrated acid and spent solution are increased by a factor of 2.5 each, water storage is reduced by a factor of 15.

NO PHASE CHANGE OF WORKING FLUIDS

Cooling can be obtained in devices using working fluids which do not undergo a phase change in the thermodynamic cycle. Only gaseous working fluids are applicable and for shelter applications air is the only practical gas.

The four applicable thermodynamic cycles are:

1. Air Cycle
2. Stirling Cycle
3. Joule-Thompson Expansion Cycle
4. Vortex Expansion Cycle.

The first two cycles are ideally thermodynamically reversible and the last two are thermodynamically irreversible.

Of the four listed only the air cycle appears applicable to shelter cooling.

AIR CYCLE

Air-cycle refrigeration machines, in addition to their advantage of using air as the working fluid, would have advantages in shelter applications of low maintenance requirements during standby and as air movers for ventilation. Today, air-cycle machines are used only for special applications because MVC and absorption systems are far superior for normal commercial cooling. Present commercial systems are made up of well-developed and sophisticated components which are relatively high in first cost but very efficient. Thus, their low operating cost, in the long run, overshadows their high initial cost. The economics of shelter cooling are radically different because of the short operati_ period. Operating costs are of secondary importance, initial cost being much more significant. In fact, except for the cost of standby maintenance, the initial cost is essentially the total cost.

For shelter cooling, reliability, safety, and cost requirements are all tied together, and they cannot be meaningfully compared with commercial cooling requirements. Probably the use of air-cycle cooling for shelters will depend on whether or not the merits of an air-cycle system justify the higher first cost.

After the advantages and disadvantages of air-cycle cooling for shelters had been established, technical and economic data were developed for further evaluation of this cooling method. Performance and cost data were obtained from manufacturers of conventional equipment which could be used with some modification for air-cycle systems. Gas turbine specialists of an automotive company were consulted concerning low-cost high-speed centrifugal compressors and expanders. Calculations were made to estimate the cycle efficiency with warm and humid shelter air. Noise problems and air contamination by oil vapor were considered.

The results of this study showed that:

1. Equipment would have to be developed for the shelter air-cycle systems. This can be done within the present state of the art.

2. Small high-speed centrifugal compressors and expanders are best suited for the shelter cooling application.

3. Cost would be high but may be within reason depending largely upon the cost of high-pressure heat exchangers.

Following is a discussion of types of air cycles, cycle and component performance, and component cost.

Figure 60 illustrates the four air-cycle systems considered for shelter cooling. All of the systems consist of a compressor, an expander, and heat exchangers. Each has different advantages and disadvantages. The low-side-open cycle is most suitable for shelter cooling.

The open-cycle system would require shelter pressure to be less than atmospheric pressure. Maintaining such a pressure would probably not be feasible because unfiltered air might leak into the shelter. Also, the occupants could not tolerate a pressure several psi below atmospheric pressure and this would be needed for adequate cooling.

The closed-cycle system would be the least efficient because the temperature differentials required for efficient heat-exchanger operation would increase the temperature span of operation. A closed-cycle machine could be pressurized for increased mass flow for a fixed volume displacement. At higher pressures the equipment would be smaller but would require higher pressure housings, piping, and heat exchangers. Air leakage would seriously reduce the capacity of the machine. The closed-cycle assembly would be considerably more expensive than a partially open one.

The high-side-open cycle would require the use of an air-to-air heat exchanger in the shelter. Pressure on the cold side would have to be considerably below atmospheric pressure to obtain even moderate temperature differences for air-to-air heat transfer. This would require not only a large heat exchanger but also a large volume expander and compressor which would make equipment costs high. However, the high-open-side cycle would have two distinct advantages: (1) the noise problem would be minimized because all openings would be outside of the shelter and (2) the atmosphere would serve directly as the heat sink for both the cooling load and the power input to drive the compressor.

The low-side-open cycle seems most appropriate for shelter-cooling. The heat exchanger at the sink could be water cooled and, therefore, the temperature differential for heat transfer could be relatively high. The size of the heat exchanger would be smaller because of the higher pressures, but its construction and piping would be heavier. The efficiency of the cycle would be higher than that of the closed system, and the compressor and expander size would be as small as possible for any open cycle. Ventilating air could be introduced at the inlet of the compressor. Noise and air contamination by oil would be serious with some types of compressors and expanders. However, only high-speed centrifugal machines seem applicable for shelter cooling. These require no lubrication of parts exposed to the air and their high-frequency noises can be damped with small low-cost mufflers.

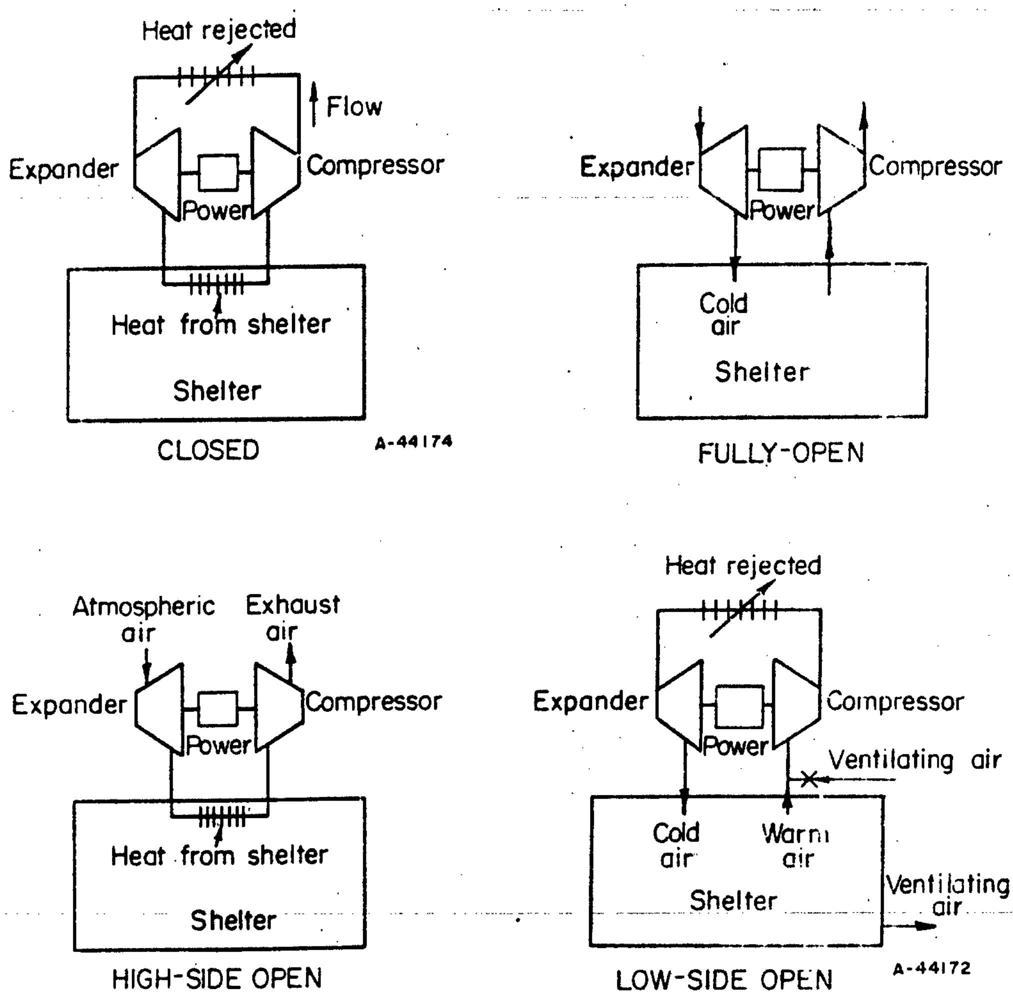


FIGURE 60. POSSIBLE CONFIGURATIONS OF AIR-CYCLE SYSTEMS

The major factors affecting the performance of air-cycle machines are:

1. Compression ratio
2. Expansion ratio
3. Thermal efficiency
4. Mechanical efficiency
5. Temperature at compressor inlet
6. Temperature at expander inlet.

Assuming adiabatic compression and expansion, with no pressure loss between the compressor and the expander, the performance of an air-cycle machine working with dry air can be expressed in the form of an equation:

$$COP = \frac{T_3 - T_1 + \eta_{te}T_1 - \eta_{te}T_1 \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}}{\frac{T_3}{\eta_{mc}\eta_{tc}}\left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}} - \frac{T_3}{\eta_{mc}\eta_{tc}} - \eta_{me}\eta_{te}T_1 + \eta_{me}\eta_{te}T_1\left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}},$$

where

COP = ratio of cooling load to power to drive the compressor

T_3 = temperature of compressor inlet, F absolute

T_1 = temperature at expander inlet, F absolute

P_1 = pressure at expander inlet and compressor outlet

P_2 = pressure at compressor inlet and expander outlet

η_{mc} = mechanical efficiency of compressor

η_{me} = mechanical efficiency of expander

η_{tc} = thermal efficiency of compressor

η_{te} = thermal efficiency of expander

k = exponent of the compression curve.

This equation shows the interrelation of the factors which govern the performance of air-cycle systems. Because of the simplifying assumptions, it is not adequate for accurate calculations using air containing water vapor. Considering the cycle open on the low-side, warm moist air from the shelter would be heated during adiabatic compression. In the heat exchanger, the high-pressure air would lose some of its moisture and this moisture would have to be removed periodically. On expansion, the air would become supersaturated with possible formation of water droplets or even ice crystals at the expander outlet. The prediction of performance with calculations based on the assumption that the outlet gases are air and water vapor at the same temperature is not significantly affected by the state of moisture at the outlet. On the basis of this assumption, performance was calculated as shown in Figure 61, the following being considered representative of shelter cooling conditions:

1. Heat load – 47 per cent latent, 53 per cent sensible
2. Compressor inlet temperature – 85 F effective, actual dry bulb and wet bulb temperatures related to expander outlet temperature
3. Compressor inlet pressure – 14.7 psia
4. Expander inlet temperature – 120 F
5. Saturated air at expander inlet.

Figure 61 shows the calculated COP of an air-cycle machine as related to compression ratio, thermal efficiency, and mechanical efficiency of the compressor and expander. These were calculated assuming no pressure loss in the piping or heat exchanger. While these results are sufficiently accurate for this discussion, they would have to be refined for an accurate estimate of performance after pressure losses were defined. The COP would be about 3 per cent lower for a 1-psi pressure drop between compressor and expander on the heat-exchanger side.

High-quality compressors and expanders having mechanical efficiencies of 0.95 or above and thermal efficiencies of 0.70 to 0.75 are obtainable in the capacities required for shelter cooling. With these efficiencies the COP would be 0.4 to 0.5 at compression and expansion ratios between 2 to 1 and 4 to 1.

Figure 62 shows the calculated heat-rejection factor, the temperature of air to shelter, and the air-flow rates for compression ratios between 1.5 to 1 and 4.0 to 1. The heat-rejection factor is equal to 1 + 1/COP. The curves show that at thermal efficiencies above about 0.75 the heat-rejection factor is nearly the same for compression ratios between 2 to 1 and 4 to 1. However, the air-flow requirement decreases markedly with increase in compression ratio with flow at a compression ratio of 4 to 1 being about two-thirds the flow at a compression ratio of 2 to 1. Therefore, it is desirable to use high compression ratios to minimize the size of the compressor and expander.

With higher compression ratios, the heat exchanger could be smaller because of the lower flow rates and also because of the increased temperature difference for heat transfer to the sink. Assuming sink water flowing in at 90 F and flowing out at 110 F, thermal efficiency of 0.75, and air flowing out at 120 F, the log mean temperature difference for counter-flow heat exchangers would be:

Compression and Expansion Ratio	Log Mean Temperature Difference, F
2 to 1	72
3 to 1	106
4 to 1	130

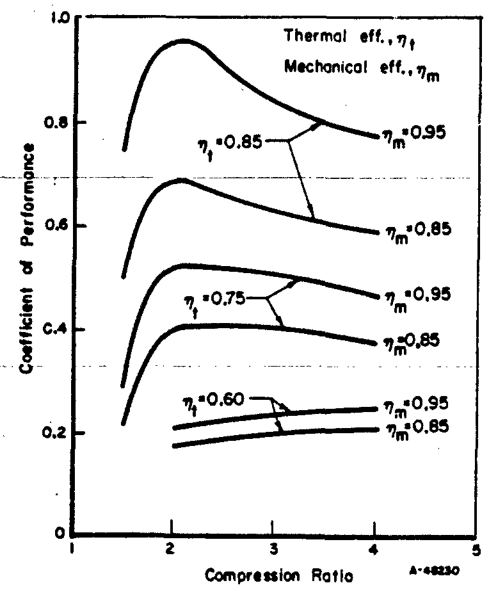


FIGURE 61. COEFFICIENT OF PERFORMANCE FOR AIR-CYCLE REFRIGERATION MACHINES

(Compression ratio equal to expansion ratio)

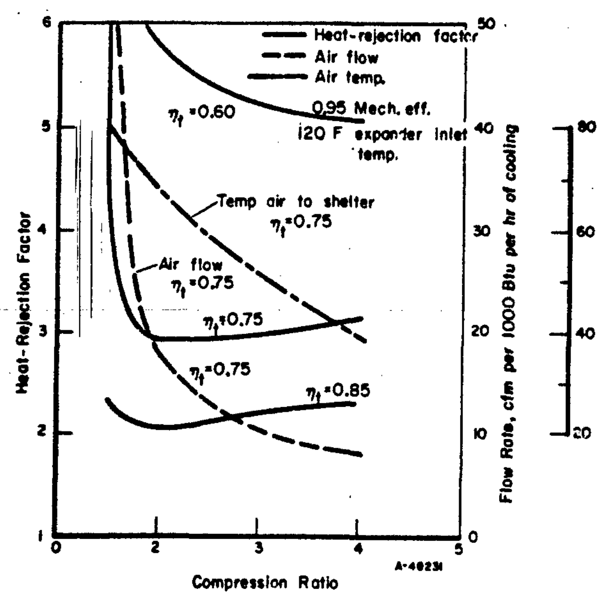


FIGURE 62. CHARACTERISTICS OF AIR-CYCLE COOLING SYSTEM

At the outlet of the expander or the inlet to the shelter, the equilibrium temperature of the air and moisture would be 39 F with an expansion ratio of 4 at 0.75 thermal efficiency. A duct system would be required to distribute the cold air throughout the shelter.

Compressors and expanders can be either dynamic or positive-displacement machines. Dynamic machines include mainly centrifugal and axial-flow compressors and turbines. Positive-displacement machines include reciprocating and rotary types. The most common rotary machines are the sliding-vane, lobe, and screw-type designs. Any of these can be used as compressors or expanders in air-cycle systems.

Table 22 is a summary of the efficiency and the estimated cost of compressors which with the exception of the Roots blower could also be used as expanders.

The estimated costs of compressors and expanders modeled after automotive gas turbine components are necessarily speculative at this time. The estimated cost of $1 per cfm for a complete unit consisting of a compressor, an expander, step-up gearing, and a lubrication system was made with the assistance of gas turbine specialists in the automotive industry. Their advice was valuable because of their experience in designing low-cost systems. Accurate cost estimates were difficult to obtain because of the uncertainty of the number of units required, the manufacturing techniques which would be required for a specific wheel configuration, the configuration of the housing for piping connections, and the bearing design. It was assumed that: (1) a few hundred units would be made, (2) the compressor and expander wheels would resemble the compressor wheel of an automotive turbine, (3) wheels would be mounted back-to-back on a common shaft, and (4) the shaft would be gear driven from a prime mover running at 2400 to 3600 rpm.

The gear-driven high-speed centrifugal units seem to hold more promise than other designs for the following reasons:

1. Low cost
2. Direct coupling to engine
3. Gearing not required when using a gas turbine prime mover
4. No contamination of air by oil
5. Equipment noise easily suppressed.

In the search for low-cost air-cycle equipment, consideration was given to the modification of automotive engines for use as compressors and expanders. Reciprocating internal-combustion engines, with modified cam shafts and valve timing, would make high efficiency air-cycle machines. The power-driven valves, required for a reciprocating expander, would be advantageous in the compressor. The valve action, travel, and timing could be so controlled that the piston speeds could be higher than the speeds of commercial compressors. This would allow higher capacities for a given displacement, thus reducing costs. The speed of commercial compressors is limited by the pressure loss through the small passages in automatic valves and by the erratic action of these valves at high flow velocities.

TABLE 22. ESTIMATED COST AND EFFICIENCY OF COMPRESSORS

Compressor	Compression Ratio	Thermal Efficiency Per Cent	Cost to Contractor, dollars per cfm
High-speed centrifugal, modeled after auto gas turbine	4	0.70 - 0.75 est.	1.00 (include expander and gearing)
Reciprocating, modified auto engine	4	0.85 est.	0.80 - 2.40
Rotary screw, Atlas Copco	2.4	0.80	5.00 - 7.50
Rotary screw, Ingersoll Rand	2	0.60 - 0.70	1.75 - 3.00
Spencer turbine	2	0.60 - 0.65	3.50 - 7.50
Roots blower (not designed for expansion)	2	0.65	1.00 - 2.00
Reciprocating, industrial type	5	0.85	8.00 - 10.00
Centrifugal, Allis-Chalmers	2-3	--	5.00 - 40.00
Rotary vane	3	0.75	4.50 - 5.50

To use a mass-produced automobile engine as a compressor or expander would require only modification of the cam shaft. Such a modified engine would cost approximately $3 per cubic inch of displacement. The flow capacity and efficiency of such a machine would vary with piston speed. Commercial compressors with automatic valves operate with piston speeds up to 1000 feet per minute. Modified automobile engines with mechanically driven valves could operate with piston speeds of at least 2000 feet per minute with little loss in efficiency. Piston speeds of up to 3000 feet per minute appear feasible.

On the basis of a cost of $3 per cubic inch of displacement and a volumetric efficiency of 85 per cent, compressors would cost about $0.80 per cfm if a piston speed of 3000 feet per minute were used with the cost being inversely proportional to speed. At the higher piston speeds the modified automotive engine would be the least expensive compressor. This is the result in part of lower costs inherent in the mass production of competitive automotive engines as contrasted with limited production of rugged long-life commercial compressors. Because of limitations on the maximum size of engines available, one compressor-expander combination would produce a maximum of about 84,000 Btu per hr. (7 tons' of cooling. Therefore, multiple machines would be required to handle large cooling loads. Space and power-connection requirements between multiple units might limit applicability.

Air contamination by oil would further limit the use of reciprocating compressors which require cylinder lubrication. According to the U. S. Public Health Service, information on the effects of oil mist in breathing air is not complete enough to establish an accurate upper tolerance limit for humans.' Experimental studies have been made with animals exposed to white mineral-oil mists at concentrations as high as 100 milligrams per cubic meter. Rabbits exposed 6 hours daily for a period of 1 year experienced no histological changes and no change in the respiratory function. A study of conditions in industry revealed a striking lack of reported illness due to oil mists. The average exposure concentrations were below 15 milligrams per cubic meter, but some exposures exceeded the average by a considerable factor.

A number of operating limits are now used in industry and by control agencies. The Michigan and Detroit Bureaus of Industrial Hygiene use 5 milligrams per cubic meter; some industrial plants find that 10 is adequate to reduce complaints; and other observations indicate that complaints arise when concentrations go above 15.

On the basis of somewhat incomplete information, a tentative limit of 5 milligrams per cubic meter of mineral-oil mist is recommended by the U. S. Public Health Service. The role of additives and fumes resulting from partial decomposition at high temperatures has not been evaluated. However, industrial experience includes human exposure to all types of oil mist.

No evidence is available to establish the maximum allowable concentrations of oil mist in air breathed continuously for days. It seems reasonable to assume that exposure to 5 milligrams of oil mist per cubic meter of air for a single 2-week period would not have detrimental effects. Perhaps some lubricants could be used which would not be noxious.

Oil consumption of good-quality air compressors is normally taken to be 1 quart per 500,000 cubic feet of air. This is equivalent to about 65 milligrams per cubic meter, which is about 13 times the upper limit recommended by the U. S. Public Health Service. This consumption is typical also of an automotive engine with good rings. Oil consumption may be higher for a new compressor or engine because some running is required to set the rings and the seals.

Undoubtedly, some of the oil entrained in the air would not enter the shelter. The air from the compressor would pass through a heat exchanger to be cooled before entering the expander. Oil mist would be deposited on the heat exchanger by inertial impaction and thermal precipitation. Therefore, the oil carried into the shelter might well be principally that added by the expander.

No information was found on the application of a lubricated reciprocating expander, and therefore no data are available on the oil consumption of such a machine. However, in view of the factors which influence the carryover of lubricant, it would be safe to assume that the consumption would be about the same as that for the compressor.

The industrial compressors of the various types are not applicable for air-cycle cooling because of high cost or low efficiency. If any were to be used for special circumstances, the rotary-screw machines would be likely candidates. These machines are designed as industrial compressors in the range of flow rates and pressure ratios which are applicable for shelter cooling. The profiles of the rotors provide internal compression or expansion. Therefore, indentical machines, directly coupled, could be used for the compressor and the expander. Gears are used to drive the mating rotors which do not touch. Because the rotors require no lubrication, air can be handled with no contamination by oil mists or vapors.

Rotary-screw compressors are built with capacities above 10,000 cfm and with single-stage compression ratios of between 2 and 4. The adiabatic efficiency of these machines depends on the configuration of the rotors and the running tolerance between them. The adiabatic efficiency governs the cost of these machines with the higher efficiency machines costing substantially more.

The Atlas Copco rotary-screw compressor is a high efficiency machine. Six-groove gate rotors are mated with four-lobe main rotors. A somewhat circular profile is used for the grooves and lobes of the rotor. The smaller machines have internal gearing to provide rotor speeds higher than that of the input shaft. Single stages can have compression ratios between 2.0 and 3.8. The design of the profile of the screws and the close tolerance between parts result in high efficiencies with a corresponding high cost. Adiabatic efficiencies of 80 per cent can be obtained from these rotary-screw compressors. The Atlas-Copco compressors cost from \$5 to \$7.50 per cfm, depending upon the size of the compressor.

The Ingersoll Rand compressors are made with four-groove rotors and two- or three-lobe main rotors. Rotor speeds can be obtained by direct driving with standard electric motors or internal-combustion engines. These compressors are less expensive than the Atlas Copco compressors, but they also have lower adiabatic efficiencies.

With a maximum compression ratio of 2, the Ingersoll Rand compressors have an adiabatic efficiency between 60 and 70 per cent. These compressors cost between \$1.75 and \$3.00 per cfm. These compressor costs are attractive on a cfm basis, but because of a lower efficiency, the costs of other components of the air-cycle system would be higher. The heat exchanger would be larger, more power would be required to drive the system, and more heat would have to be rejected to the sink.

With low-side-open air-cycle machines noise suppression would be required because the compressor inlet and the expander outlet would be open to the occupied area of the shelter. The type of machine used would determine the amount of suppression required. The high-speed centrifugal machines would produce predominantly high-frequency sound and the other types would produce low-frequency sound. Low-frequency noises are more difficult to suppress than those at higher frequencies. Even so, the low-frequency noise generated by reciprocating and rotary types of compressors could be sufficiently suppressed with mufflers costing less than \$0.25 per cfm of air flow. The cost of mufflers for the high-speed centrifugal machines would be negligible. Muffler design parameters are well known and only limited development should be required to provide the proper muffler for a specific application.

It is understandable that higher thermal efficiencies can be obtained with more sophisticated and costly compressors and expanders. However, the effects of the refinements on the total cost of air-cycle cooling is not so obvious. The two major costs other than the compressor and expander are power and heat exchanger or heat exchangers at the sink.

Table 23 shows, for various compression ratios and thermal efficiencies of compressors and expanders, performance data and estimated costs of the heat exchanger, a cooling tower, and power. Arbitrary costs of compressor-expander combinations are included. These will be used later to show that a considerable increase in the compressor and expander cost can be justified to obtain higher efficiency and consequently lower total cost.

TABLE 23. AIR-CYCLE SYSTEM PERFORMANCE AND ESTIMATED COSTS PER 1,000 BTU PER HR OF SHELTER COOLING

	Performance Data					
Compression Ratio	2:1	2:1	2:1	4:1	4:1	4:1
Thermal Efficiency, Turbine and Compressor	0.60	0.75	0.85	0.60	0.75	0.85
COP of System[(a)]	0.21	0.52	0.96	0.25	0.47	0.77
Air Flow, cfm	28.2	18.0	14.5	10.0	7.80	6.50
Heat to Sink[(a)], Btu per hr	5,800	2,900	2,000	5,000	3,100	2,300
	Estimated Costs, dollars					
Heat Exchanger	96.50	48.50	33.50	83.50	51.50	38.50
Cooling Tower	50.00	25.00	17.50	43.00	27.00	20.50
Power, Diesel Engine	75.00	30.00	16.50	63.00	33.50	20.00
Power, Electric Motor	225.00	90.00	49.00	190.00	100.00	61.00
Compressor and Expander						
$1 per cfm	28.20	18.00	14.50	10.00	7.80	6.50
$3 per cfm	84.60	54.00	43.50	30.00	23.40	19.50
$7 per cfm	197.40	126.00	101.50	70.00	54.60	45.50

(a) Engine cooling not included as part of cooling load.

For this study, the cost of power by conventional diesel engines and electrical apparatus is used as:

$40 per hp for an engine or for an electric motor, motor starter, and motor wiring.

$80 per hp for engine, generator, and wiring to the switch box.

The cost of the heat exchanger and its piping are assumed to be $16.70 per 1,000 Btu per hr of heat transferred. This cost is strictly speculative. The heat exchanger would cool air with water from a large natural source or from a smaller source in connection with a cooling tower. The only off-the-shelf heat exchangers available for such duty are those commonly used as air compressor intercoolers and aftercoolers. These are built for years of heavy-duty service with first cost being secondary to trouble-free operation; therefore, they are not of the class required for the air-cycle system. Consequently, it was necessary to estimate the cost of an adequate heat exchanger. This was done by assuming that, on a Btu per hr basis, the high-pressure exchanger could be made at the same cost as an atmospheric-pressure air cooler used for conventional comfort cooling. The temperature difference for heat transfer in the high-pressure heat exchanger would be at least five times that of the temperature difference in the atmospheric-pressure heat exchanger. Therefore, the higher pressure exchanger could be considerably smaller for the same heat-transfer capacity. It was assumed that the smaller size would offset the cost of the higher pressure casing and piping.

Needed are more accurate costs for heat exchangers and the information on influence of heat-exchanger design on the over-all performance of an air-cycle system. Adequate heat exchangers have been built with standard finned-tube banks housed in standard or custom-made pipe of the desired size. Such designs should be considered for the air-cycle system. The piping from the heat exchanger to the compressor and expander needs to be designed with proper consideration for pressure loss, utilizing where possible standard pipe sizes and fittings for economy.

As shown by the heat-rejection values in Table 23, a relatively large heat sink would be required. For this reason a cooling tower has been included. If a large supply of natural water were available, the tower cost should be replaced by that for obtaining and pumping the water through the heat exchanger.

Figure 63 shows a summation of the costs itemized in Table 23 for a system mechanically connected to a diesel engine. This clearly shows that a premium could be paid for the highly efficient compressor and expander which would reduce the cost of power and heat exchangers. The effect of compression ratio is not adequately displayed because the cost of the heat exchangers was estimated on the basis of only the quantity of heat transferred. Not only is the quantity of heat to the sink less at the higher compression ratios, but also the air flow is lower and the temperature differential for heat transfer is higher. All of these tend to reduce the required size of the higher pressure heat exchanger, but at the same time its construction would have to be heavier to withstand the higher pressures. A detailed design and cost determination is needed to clarify the economics of air-cycle systems.

STIRLING CYCLE

The Stirling-cycle refrigerator is used in the cryogenic field and technically it could be used to cool a shelter. However, in spite of recent improvements, Stirling refrigerators have found little use in ordinary air-conditioning applications, primarily because MVC units for these applications are presently more compact, less expensive, and more durable.

One of several possible configurations of Stirling-cycle coolers is shown in Figure 64. The machine is a closed system consisting of two cylinders connected through heat exchangers and a regenerator. One cylinder is the expansion cylinder and the other the compression cylinder. The two pistons are driven by the same crank mechanism such that the expansion piston is approximately one-quarter revolution ahead of the compression piston.

The heat exchangers transfer heat between the surroundings and the gaseous working fluid inside the unit. Hydrogen or helium is generally used because the fluid

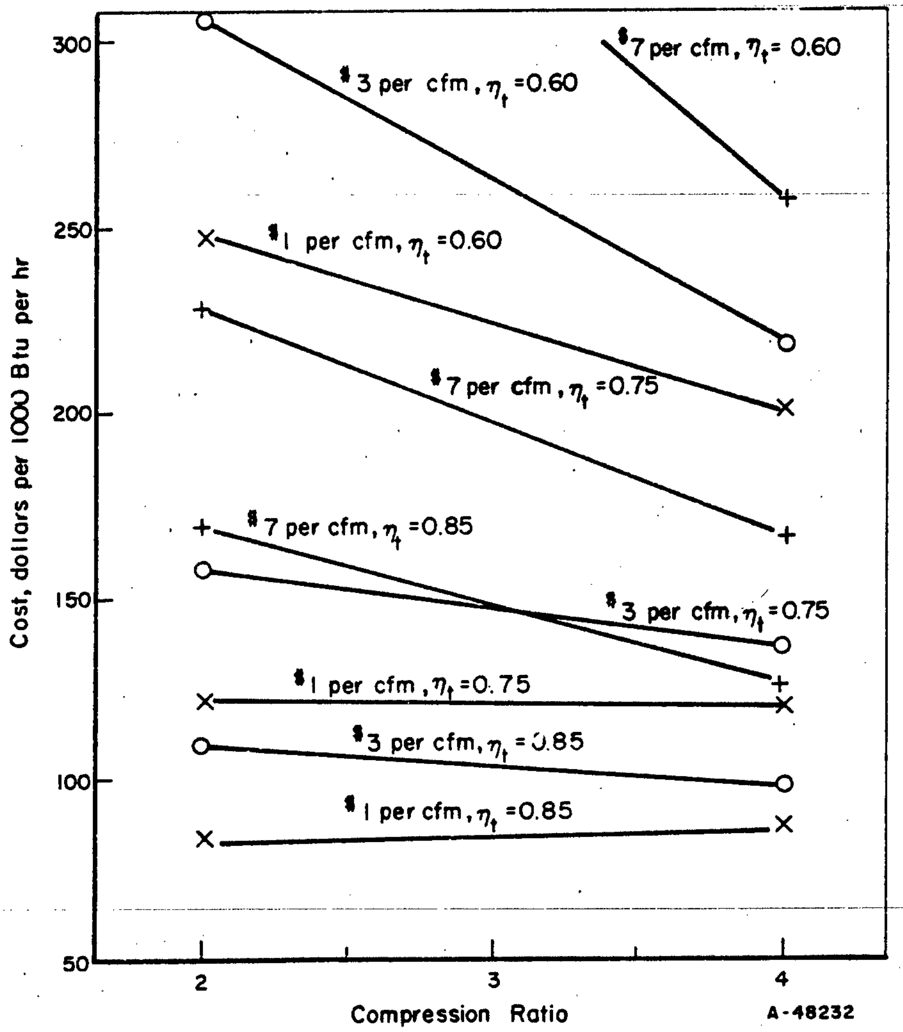


FIGURE 63. COST OF AIR-CYCLE COOLING SYSTEMS

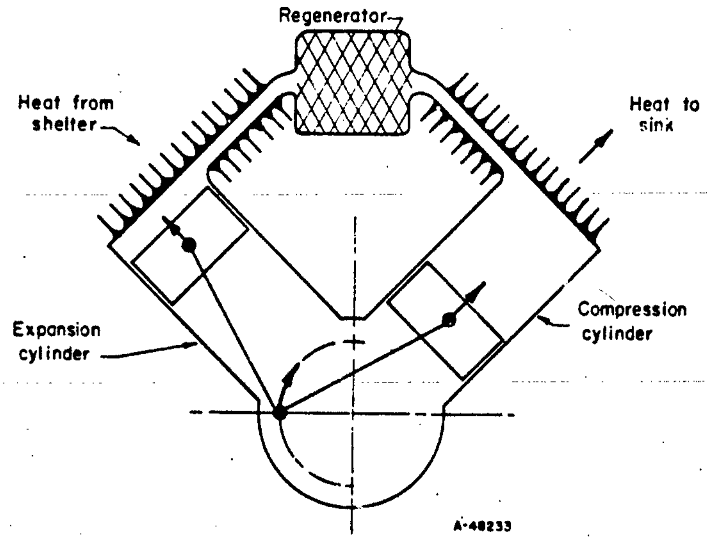


FIGURE 64. STIRLING-CYCLE REFRIGERATOR, RIDER CONFIGURATION

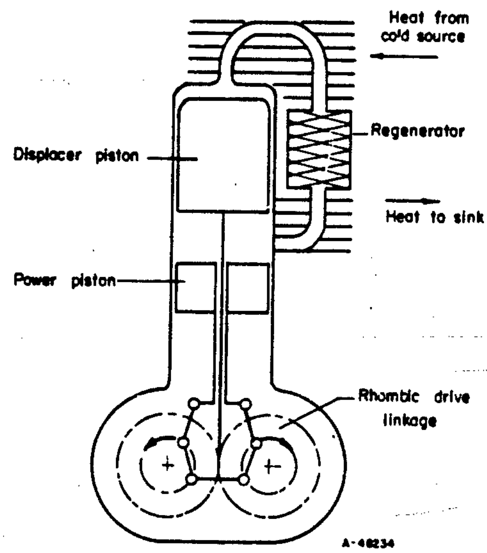


FIGURE 65. STIRLING-CYCLE REFRIGERATOR, PHILIPS DISPLACER CONFIGURATION

pressure losses are less for the quantity of heat transferred. However, air could be used for shelter cooling applications. The expansion cylinder side is in contact with the heat source and the compression cylinder side is in contact with the heat sink. Relatively large heat-exchange surfaces are required to minimize the differences between the internal gas temperatures and the heat source and sink temperatures.

The regenerator usually is constructed of a fine wire matrix. Its function is to alternately store and release heat as the working fluid is forced back and forth between the compression and expansion side by the motion of the pistons.

The heat-transfer surfaces in contact with the heat source and sink tend to maintain the working fluid at a constant temperature. Therefore, during compression heat must be rejected since the compression process raises the temperature of the working fluid. Because of the phasing of the pistons, as illustrated in Figure 64, most of the working fluid is on the compression side during the compression process; consequently, most of the heat is rejected on this side. During the expansion process, the opposite is true. Heat is taken in to maintain a constant temperature as the fluid tends to cool by being expanded. Because most of the fluid is now on the expansion side, most of the heat is taken in on that side. Thus, there is a net heat flow through the unit from the expansion side to the compression side. If the direction of rotation is reversed, the direction of the heat flow will also be reversed. Thus, such a unit could be used for both heating and cooling of a space without changing any of the heat paths. Although the Rider configuration shown in Figure 64 is simpler to describe, the Philips configuration shown in Figure 65 is actually more common at present. This configuration can be shown to be thermodynamically equivalent to the Rider configuration. Such units are currently in use in small capacities as air liquefiers[41] or cryogenic refrigerators which can operate at temperatures as low as 10 K. There is little present commercial use of the heat-engine configuration of the Stirling-cycle unit; however, experimental units are under development for military and space applications.[42] The military potential stems from the fact that it is an external-combustion engine and, consequently, it has the potential for operating quietly. In space, this external-combustion engine is of interest because it can be used for the conversion of solar energy and, in addition, its high efficiency minimizes the size requirements of heat-rejection radiators. Although Stirling-cycle engines are currently regarded somewhat as curosities and although they do not enjoy widespread commercial use, the concept is actually older than most of our current prime movers. The engine was invented in 1816 and was widely used as a source of power in the 19th century. It was particularly attractive at that time because it was not subject to the violent boiler explosions that were being experienced with steam-engine power plants. However, these 19th century units were quite inefficient and bulky and they became obsolete with the development of the modern gasoline and diesel engines.

The modern version of the Stirling-cycle engine is largely the result of the development efforts of the Philips Company of the Netherlands, and it is much like the original version with the exception of the addition of the regenerator and the use of high internal working pressures. As a result, modern Stirling-cycle engines operate at as high an efficiency as any present heat engine and they are far less bulky than their predecessors.

The refrigerating capacity of a Stirling-cycle refrigerator is directly proportional to the mass of working fluid circulated. Thus, the capacity of a given sized unit can be increased by using higher internal pressures; however, even at around 20 atm average internal pressure, the refrigerating capacity per cubic inch of displacement is at the very best only about one-tenth that of a MVC system with a compressor of equal displacement. There are some minor compensating size factors, but generally speaking, a Stirling-cycle refrigeration device would be at least ten times the size and the weight of an equal capacity MVC device.

In Figure 66, the capacity characteristics of Stirling-cycle refrigeration units are shown compared with those of an R22 MVC system. Capacity is shown as a function of the heat-rejection temperature expressed as the ratio of the capacity of the unit to the capacity with a 95 F rejection temperature. As is shown, the capacity of the Stirling-cycle unit is relatively insensitive to the heat-rejection temperature, while the capacity of the MVC system decreases rapidly with increasing temperature. This drop is due in part to the reduced volumetric efficiency of the reciprocating compressor at higher discharge temperatures. If desired, a Stirling-cycle refrigerator could be used with boiling water as a heat sink. Of course, this could not be done without increased power input.

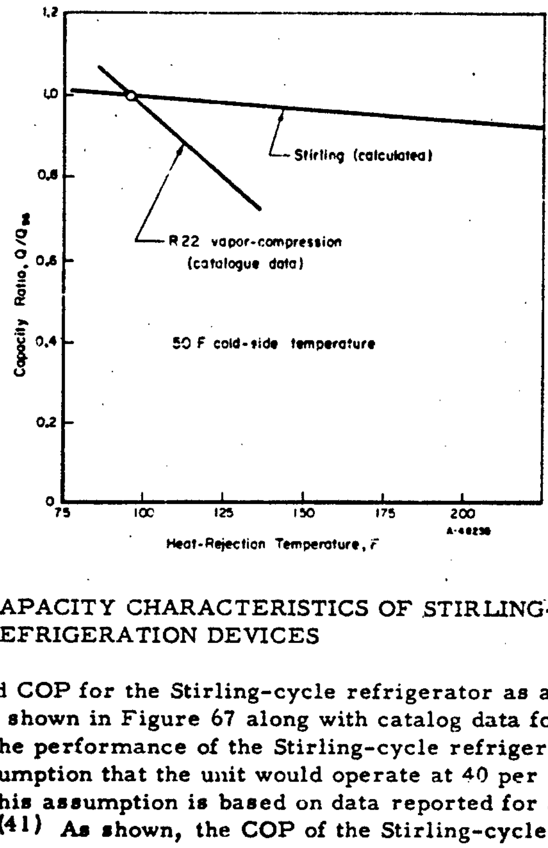


FIGURE 66. CAPACITY CHARACTERISTICS OF STIRLING-CYCLE AND MVC REFRIGERATION DEVICES

The estimated COP for the Stirling-cycle refrigerator as a function of heat rejection temperature is shown in Figure 67 along with catalog data for an R22 MVC system for comparison. The performance of the Stirling-cycle refrigerator was estimated on the basis of the assumption that the unit would operate at 40 per cent of its theoretical ideal efficiency. This assumption is based on data reported for a Philips air-liquefier Stirling-cycle unit.(41) As shown, the COP of the Stirling-cycle unit is roughly comparable to that of the MVC system. In a well-designed Stirling-cycle refrigerator the COP could actually be greater than that for the MVC system. However, this again is a function of the specific configuration of the unit and there are not sufficient operating data in the literature for this temperature range to establish typical performance characteristics.

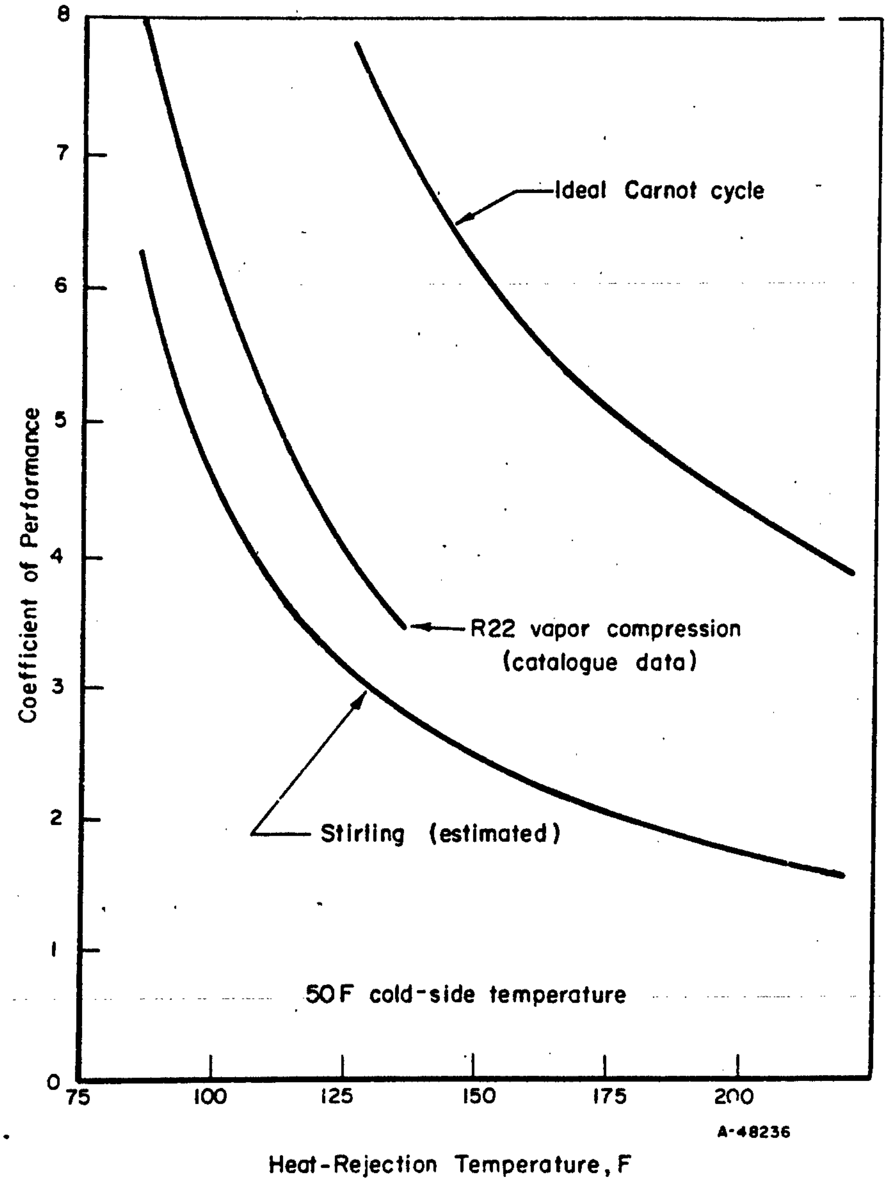


FIGURE 67. ESTIMATED POWER REQUIREMENT OF STIRLING-CYCLE REFRIGERATION DEVICE

The capacity of Stirling-cycle refrigerators can be modulated or controlled by several means. Simplest of these is the typical on-off type of capacity modulation most commonly used in small-capacity systems. Alternatively, since the capacity of the Stirling-cycle unit is directly proportional to the mean pressure in the unit, the refrigerating output can be varied by varying the internal pressure as desired. A third method, not practical in this case because of the mechanical complexities involved, is varying the phase angle between the two pistons to alter the capacity of the unit.

A common disadvantage from the standpoint of durability shared by all Stirling-cycle machinery is that of the sensitivity of this closed-cycle system to contamination of the working fluid, particularly by a liquid lubricant. The presence of the lubricant tends to foul the heat-transfer surfaces, especially if temperatures sufficiently high to decompose the lubricant prevail. There are two alternatives to circumvent this problem. One is the use of self-lubricating surfaces for bearings, rings, etc., such as Teflon-impregnated bearing materials. The other alternative is to use conventional lubricants and minimize the amount of lubricant leakage into the working fluid by the use of highly effective seals.

If the former approach is taken, durability is limited by the naturally higher wear rates experienced by the Teflon-type surfaces compared with the wear rates of normal well-lubricated metallic surfaces in contact. If the second approach is taken, frequent disassembly is necessary to clean the contaminated surfaces because the low-leakage seals that must be used are particularly susceptible to loss of effectiveness as a result of wear.

Despite these difficulties, durabilities of several thousand hours are predicted for Stirling-cycle engines under development and equal or better life should be possible for Stirling-cycle refrigerators.

The noise level generated by Stirling-cycle machinery should be somewhat less than that of MVC machinery. Although mechanical noise from the drive mechanism would be expected to be about the same, because of the absence of discharge or suction valves and the absence of the flow pulsations there will be less over-all noise output.

Since there is no Stirling-cycle refrigeration equipment available at present that is suited for the requirements of shelter cooling, no meaningful cost data are available. Stirling-cycle refrigerators on the market today are special-purpose, low-production, high-cost units which do not serve as a suitable basis for making cost estimates.

As a general class of machinery, Stirling-cycle units are roughly comparable to MVC equipment and some rough cost estimates could be made on this basis. If the cost of the compressor alone in a MVC unit is increased by a factor of about 15, and the normal cost for the rest of the MVC system is added to this figure, a realistic cost estimate for a Stirling-cycle system in high-volume production should result. However, there is the additional complication that production costs are intimately tied to production volume and severe cost penalties would be incurred with low-production volume.

Stirling-cycle units have several features which give them unique advantages for shelter cooling. Perhaps foremost advantage is the possibility of extended shelf-life or inoperative periods without deterioration of the equipment. Although most Stirling-cycle units are charged with hydrogen or helium under high pressure as a working fluid and they are therefore susceptible to leakage problems, air can also be used as the working fluid at the expense of an increase in internal aero-dynamic and friction losses. However, these losses would not be severe and units could be equipped with an integral air compressor so that the units would be self-charging.

In addition, if durabilities of several hundred hours would be acceptable, Stirling-cycle machinery could be made self-lubricating by the use of Teflon bearing and sealing surfaces and, thus, charging or loss of lubricant would not be a problem. Also, since the Stirling-cycle unit is a sealed system, it would be naturally resistant to contamination or corrosion.

The Stirling-cycle unit's tolerance to the heat-sink temperature is also a significant potential advantage for shelter applications. Provided there were sufficient power available to drive the unit, it could operate with only slightly reduced capacity with heat-sink temperatures far in excess of those normally anticipated. Another advantage is the previously-mentioned capability to provide both heating and cooling. Finally, the high efficiency obtained by Stirling-cycle refrigeration units is a significant though not unique advantage.

The primary disadvantages of Stirling-cycle refrigeration equipment are their large size and probable high cost.

JOULE-THOMPSON DEVICES

The Joule-Thompson effect, which is the change in temperature that can accompany the isenthalpic throttling of a gas from a higher to a lower pressure, could be utilized for cooling shelters. The most important application of the Joule-Thompson effect is in the field of low-temperature liquefaction of gases.

For shelter cooling application either an open-cycle or a closed-cycle device could be used. An open-cycle device would consist of a high pressure storage facility and a throttling valve. A closed-cycle device would require a compressor and heat exchangers in addition to the components needed for an open-cycle device. Because of high operating pressures and relatively low efficiency, Joule-Thompson devices are not particularly well suited for shelter cooling application.

Two of the more attractive systems were investigated: an open-cycle using air as the working fluid, and a closed-cycle using ethylene.

In an open-cycle device the minimum amount of equipment is required if the working fluid can be discharged directly into the shelter. Therefore, air was selected as the working fluid. Assuming the air is stored at 200 atmospheres (3000 psi) and at a temperature of 80 F, throttling the air through a suitable valve would reduce its temperature by 63 F due to the Joule-Thompson effect. This would produce 15 Btu of cooling per pound of air used. In addition to the reduction in temperature of the throttled air produced by the Joule-Thompson effect, the flow work that is expended as the gas is expelled from the storage facility would further reduce the air temperature. As the storage tanks neared depletion, the stored-air temperature would approach -76 F; assuming no heat transfer to the stored air during throttling. This final expelled air would have a cooling capacity of 38 Btu per lb. The approximate over-all average cooling capacity would be 27 Btu per lb. With an initial storage density of 15 lb per cu ft, the sensible cooling capacity on a storage-volume basis would be 405 Btu per cu ft which is equivalent to 2.5 cu ft per 1000 Btu. Assuming a sensible heat ratio of 0.47, the storage cooling capacity would be 860 Btu per cu ft or 1.17 cu ft per 1000 Btu.

The major advantages of the open-cycle Joule-Thompson device are its mechanical simplicity, requiring basically just storage tanks and an adjustable valve, its use of air as the working fluid and its suitability for use during the sealed period. Its disadvantages include high working pressures, ineffective utilization of storage space, and high cost of the storage facility.

The closed-cycle Joule-Thompson device investigated uses ethylene (refrigerant 1150) as the working fluid. Ethylene was selected on the basis of its critical temperature of 48.8 F which is just below the temperature level required for shelter use. This allows the throttling process to occur just above the critical point where the isenthalps are nearly parallel to the lines of constant entropy. Schematic temperature-entropy and pressure-enthalpy diagrams of the cycle are shown in Figure 68. Process 1-2 on the diagrams is the throttling of the gas from the higher to the lower pressure with a resultant temperature drop. From 2 to 3 the gas absorbs heat by way of a heat exchanger at constant pressure and variable temperature. Process 3-4 is an isentropic compression of the gas from the low-to the high-side conditions, and 4-1 is a constant-pressure, variable-temperature heat rejection to a sink, again through a heat exchanger.

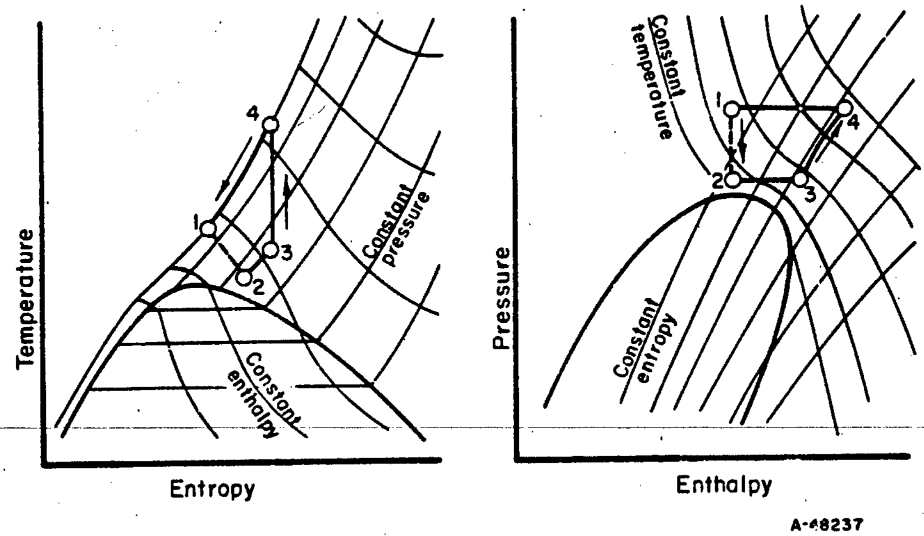


FIGURE 68. JOULE-THOMPSON CYCLE: SCHEMATIC T-S AND P-H DIAGRAMS

For operating conditions, it was assumed that the lowest sink temperature available was 120 F and the high-pressure gas would be available at that temperature. The temperatures in the low-temperature heat exchanger were 60 to 80 F. Isentropic compression from the lower to the higher pressure was assumed. Using the pressure-enthalpy diagram for ethylene available in the ASHRAE Guide and Data Books, the system characteristics and performance shown in Table 24 were obtained.

TABLE 24. CLOSED-CYCLE JOULE-THOMPSON COOLING-SYSTEM CHARACTERISTICS AND PERFORMANCE

State Point	Temperature, F	Pressure, atm	Enthalpy, Btu per lb
1	120	200	178
2	60	55	178
3	80	55	226
4	235	200	262

$$\text{Ideal COP} = \frac{Q_c}{Q_{wk}} = \frac{(h_3-h_2)}{(h_4-h_3)} = \left(\frac{226-178}{262-226}\right) = \frac{48}{36} = 1.3.$$

Consideration was also given to the regenerative heat exchange that is possible between the cool gas leaving the low-temperature heat exchanger and the hot gas leaving the high-temperature heat exchanger. This increased the cooling capacity of the working fluid to 60 Btu per lb with only a slight increase in compression work for a gain in the COP to 1.6.

The possible advantage of a closed-cycle J-T refrigeration device is the elimination of the need for a dynamic expansion machine such as is required for the air-cycle device which is normally considered to operate in the perfect gas regime. The disadvantages would appear to be with the high pressures involved. A search for a more suitable working fluid to overcome this was not successful.

VORTEX-TUBE DEVICES

A vortex tube is a static device that divides a compressed stream into two portions with a temperature increase in one portion and a decrease in the other. Being a static device, the vortex tube accomplishes these temperature changes without the benefit of a transfer of external work. It is also known as a Hilsch tube, Ranque tube, Ranque-Hilsch tube, vortex refrigerator T-tube, and separator tube. It has not been employed extensively as a refrigeration device because of its low efficiency. In general, its practical applications have been few and it remains primarily a laboratory curiosity. Because of its low efficiency, it is not suitable for use in shelter cooling systems.

The most common vortex tube design is shown in Figure 69.[43] This consists of a tube with a control valve at one end, an orifice plate near the other, and a nozzle near the orifice plate which directs the gas into the tube in a tangential direction. The setting of the control valve determines the amount of gas leaving either end and the gas temperatures. An example of the temperatures that can be attained are shown in Figure 70. These plots are taken from data published by Hilsch in 1947[44] and are representative of the temperature developed by a vortex tube using air at various supply pressures.

With all the air leaving through a wide-open control valve, there is no change in the temperature of the gas stream. As the valve is closed and some air begins to leave through the orifice end, the stagnation air temperature at the orifice begins to decrease. At the same time, the air temperature at the valve end begins to increase. Further closure of the valve forces a larger percentage of the air to leave through the orifice and at lower temperatures until at some mass ratio value (μ) a minimum temperature is reached. Continued control valve closure forces more cold air out the orifice end, but at higher temperatures. The air temperature at the valve end continues to increase as the valve closes until at eminent closure there is virtually no temperature change in the air leaving the orifice end, but a large temperature increase in the small amount leaving the valve end.

The mechanism by which the vortex tube operates is still not completely understood; however, one of the more logical explanations is based on the conservation of momentum and the exchange of kinetic energy by viscous shear.[45] Briefly, the gas entering the tube tangentially establishes a vortex where momentum is conserved; that is, the product of tangential velocity and the path radius of each particle is a constant. This type of vortex has slower moving particles near the walls of the tube and faster moving ones near the center. As the vortex moves from the orifice plate region and along the tube toward the valve end, viscous forces tend to establish a vortex with a constant angular velocity, speeding up the slower particles at the expense of slowing down the faster ones. The energy transfer during this process is thought to cause an increase in the stagnation temperature of the gas near the walls of the tube and a decrease in the temperature near the axis. The resulting temperature difference and heat transfer is then thought to cause the cooler gas to expand forcing it out along the axis at the orifice end.

It is conceivable that a vortex tube could be employed in either an open-cycle or in a closed-cycle device in much the same manner as a Joule-Thompson device. Both cases were investigated for the vortex tube. While neither appear to be applicable to shelter cooling, example calculations are included to provide some basis for this evaluation.

For the open-cycle vortex tube device, the cold gas performance curve shown in Figure 70 for a supply pressure of 6 atmospheres might be considered. This plot is shown again in Figure 71 along with a plot of the product of the mass ratio and the temperature depression. The product μ (T_o -T) indicates the cooling capacity per pound of gas supplied to the tube. As shown, this is a maximum at μ = 0.6 where the temperature depression is 70 F. Assuming an air storage temperature of 80 F, direct injection of the cooler air to the shelter with reheating to 80 F, no change in performance of the vortex tube, and a slight change in supply-air temperature, the initial effectiveness of the device would be:

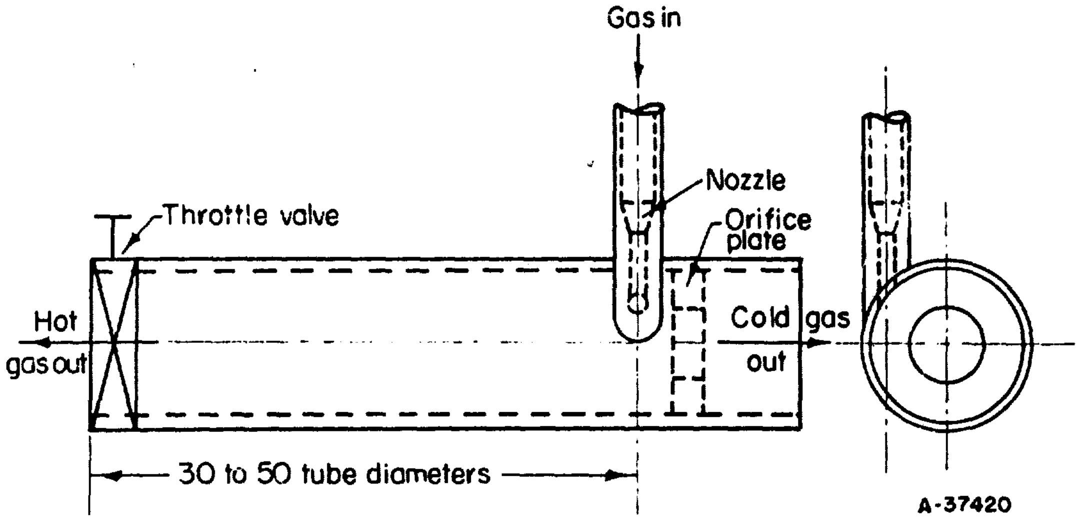
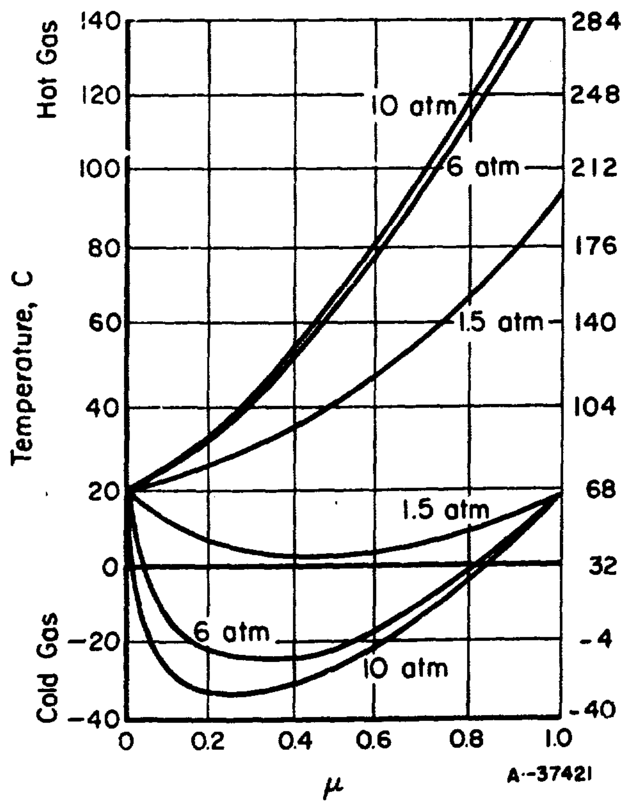


FIGURE 69. COUNTERFLOW-TYPE VORTEX TUBE



Hilsch	Tube No. 1
Main tube ID	4.6 mm
Inlet nozzle ID	1.1 mm
Orifice plate ID	2.2 mm
Inlet air temperature	20 C (68 F)
Exhaust pressure	1 atm

$$\mu = \frac{\text{Cold air mass flow}}{\text{Total air mass flow}}$$

FIGURE 70. TEMPERATURE CURVES FOR A VORTEX TUBE WITH AIR

$$\Delta h = \mu (C_p) (\Delta T)$$

$$= 0.6 (0.24) (70)$$

$$= 10 \text{ Btu per lb air stored.}$$

With a storage-air density of 0.45 lb per cu ft at 6 atmospheres and 80 F, the initial sensible cooling capacity on a storage-volume basis would be 4.5 Btu per cu ft. This is an extremely low space utilization, and higher storage pressures would not significantly improve the situation.

During this program, the vortex tube was analyzed in terms of a semiopen mechanically-motivated cycle as shown in Figure 72. The assumed temperatures are similar to those associated with the closed-cycle Joule-Thompson device. Atmospheric air at 120 F was assumed to be available as a working fluid and as a sink. In Figure 72, the air used as a working fluid is compressed to 1.5 atmospheres, cooled to 120 F by the sink, and then to 96 F by the cooler air leaving the shelter. The compressed air is then directed to the vortex tube where 60 per cent leaves the cold end at 66 F (see Figure 71) and the remainder is rejected to the atmosphere. The 66 F air is then directed to the shelter where it is heated to 80 F and then to the heat exchanger where it cools the air about to enter the vortex tube. Assuming an isentropic compression process, the temperature of the air leaving the compressor would be 190 F and the compression work would be:

$$\text{Ideal } W_k = (\Delta h) = C_p (\Delta T)$$

$$= 0.24 (70)$$

$$= 16.8 \text{ Btu per lb.}$$

The sensible cooling effect per pound of air compressed would be:

$$Q_c = \mu (C_p) (\Delta T)$$

$$= 0.6 (0.24) (14)$$

$$= 2 \text{ Btu per lb.}$$

The resulting COP would then be:

$$\text{Ideal COP} = \frac{Q_c}{W_k} = \frac{2}{16.8} = 0.12.$$

This is a very low efficiency and is some indication as to why this device has not been used extensively.

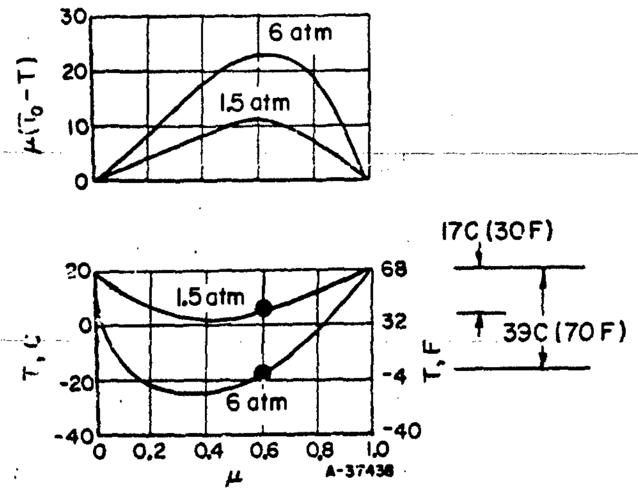


FIGURE 71. TEMPERATURE AND COOLING-CAPACITY CURVES FOR A VORTEX TUBE

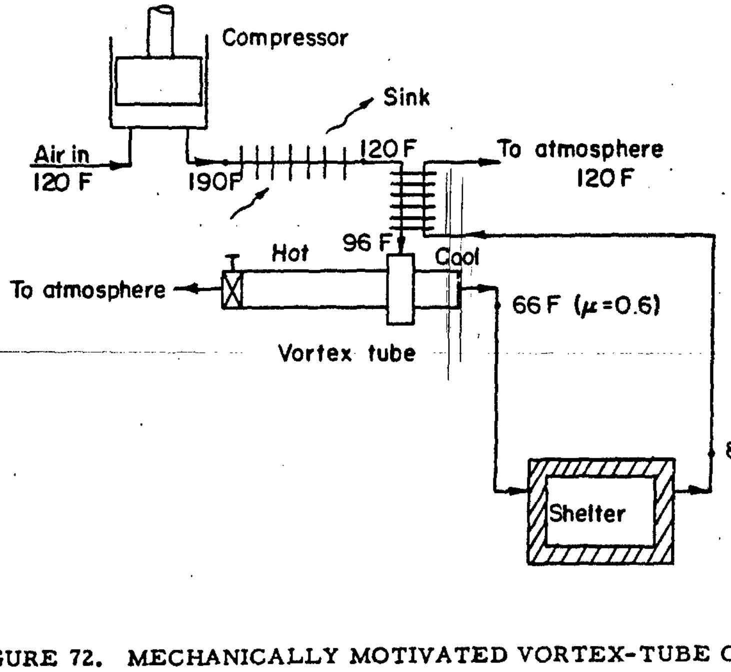


FIGURE 72. MECHANICALLY MOTIVATED VORTEX-TUBE CYCLE

SOLID STATE DEVICES

A cooling effect can be produced by electrically or magnetically stimulating various materials or material combinations. Devices operating on this principle can be thought of as solid-state devices in that the cooling is produced directly in solid materials without the need for a gaseous or liquid working fluid. At this time, the use of solid-state devices for shelter cooling is not practical because of the high first cost.

The cooling effects of interest are:

1. Thermoelectric (Peltier)
2. Galvanomagnetic (Ettingshausen)
3. Thermomagnetic (Righi-Leduc).

Of the three, only thermoelectric cooling has potential for use in shelters. In the past several years a number of commercial applications have been made. However, with few exceptions, all applications have been low capacity devices with ratings below about 1000 Btu per hr. Larger units of up to 120,000 Btu per hr capacity have been and are being developed. These projects are Government sponsored and directed toward determining feasibility for such applications as shipboard air conditioning and refrigeration. Up to the present time there have been no serious attempts to market thermoelectric air conditioning.

Following is a brief description of each of the cooling effects.

THERMOELECTRIC

The thermoelectric effect is the reversible transformation between electrical and thermal energy that occurs at the junction of dissimilar conductors. Figure 73 is a sketch of a thermoelectric device. A direct current passed through the device will produce cooling at one junction and heating at the other depending on the direction of current flow. With presently available materials the COP of a thermoelectric device would be about 1.5 at a temperature difference of 70 F between the hot and the cold junction as compared with a COP of 5 at the same temperature difference for a MVC device.

Thermoelectric refrigeration is not more widely used for several reasons. First, with presently available materials the performance efficiency is inferior to that of conventional refrigeration equipment. Second, thermoelectric material costs are relatively high. Lastly, and perhaps most important, are fabrication costs. While at first glance a thermoelectric device appears to be an extremely simple assembly, this is apparently not the case. Among other items, difficulties have been experienced in obtaining and maintaining satisfactory junctions. The quality control and construction techniques necessary to overcome these difficulties have resulted in high fabrication costs.

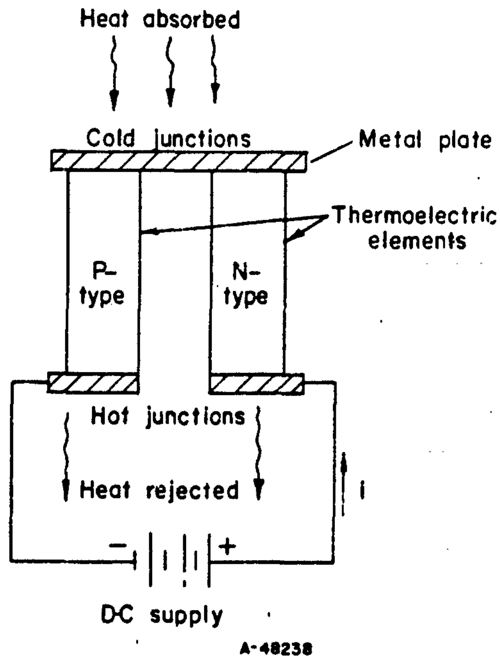


FIGURE 73. THERMOELECTRIC COOLING EFFECT

For shelter applications, thermoelectric devices for air conditioning purposes are technically feasible, but economically unfeasible. Since all large size units are still in the research and development phase, it is difficult to establish a meaningful cost. However, a general consensus of the people contacted is that it is much too expensive for shelter application. Hopes of significantly improving the situation in the near future were not encouraging.

GALVANOMAGNETIC

An electrical conductor lying in a magnetic field and carrying a longitudinal electric current at right angles to the field produces a thermal potential across the conductor that is mutually perpendicular to both field and current. The thermal potential is known as the Ettingshausen Effect. Figure 74 is a sketch showing the galvanomagnetic coupling effects.(46)

At the present time, the Ettingshausen Effect is just beginning to find some applications in the low temperature field, that is, below about 160 K (-170 F). Operation at normal air-conditioning temperatures is hampered by unfavorable mobility of the charge carriers and would require magnetic field densities far beyond present capabilities and therefore, is not technically feasible for shelter cooling.

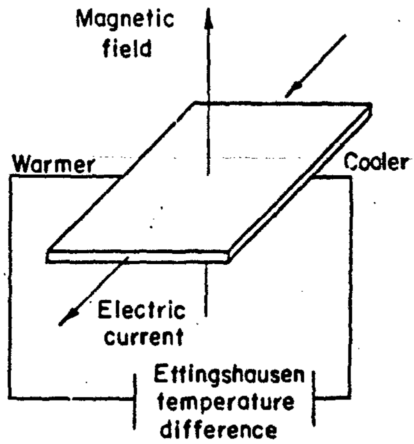


FIGURE 74. GALVANOMAGNETIC COOLING EFFECT

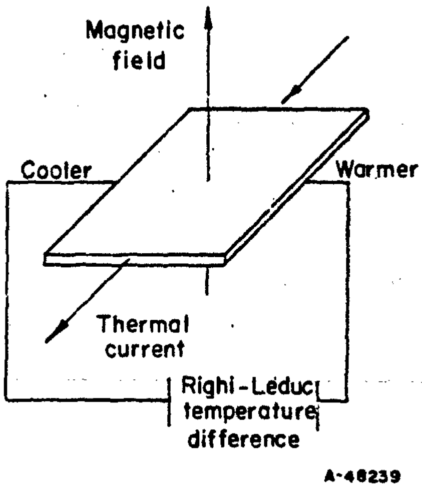


FIGURE 75. THERMOMAGNETIC COOLING EFFECT

THERMOMAGNETIC

Replacing the longitudinal electric current of the Ettingshausen Effect with a thermal current also produces a thermal potential across the conductor. Figure 75 is a sketch showing the thermomagnetic coupling effect known as the Righi-Leduc Effect. This effect is also just beginning to find some applications, mostly in the instrumentation and measurement. At present, it does not appear that the Righi-Leduc Effect is suitable for shelter air-conditioning applications.

- COOLING SYSTEMS

GENERAL CONSIDERATIONS

A cooling system is construed to be the total assembly of equipment and materials required to remove and dispose of excess heat. For example, in a shelter a cooling system utilizing an artificial heat sink might consist of: (1) a supply of ice, (2) an ice-storage facility, (3) a circulating water pump, (4) a heat exchanger, and (5) a power supply. Many other combinations of components can also be used. However, depending upon the degree of availability of air and natural water, certain systems will have greater cost effectiveness than others. Because selection of the best type of cooling system for a particular shelter is so greatly influenced by the availability of useable natural air and water, it is convenient to make a breakdown of possible cooling systems based on this availability. This has been done in Table 25, "Shelter Situations and Possible Cooling Systems".

Table 25 shows cooling systems applicable for use in situations where varying amounts of atmospheric air, natural water, and both atmospheric air and natural water are available. Included in the table for each system considered are: the total cost, the heat sink and power supply to be used, and the main features of the mechanical equipment required. Following the table are more detailed descriptions of the various systems, including schematic diagrams and breakdowns of the cost of the major components.

Not all conceivable shelter cooling systems are included in the table. The systems presented represent a wide range of practical arrangements. The table permits comparison of the costs of a number of systems and illustrates the technique of combining components into useful systems for particular situations.

In general, the systems considered are presented in order of increasing dependence on atmospheric air and natural water. The availability of air is presented in terms of use because air is always available in ample quantities, but it may not always be useable because of high temperature or contamination. The air-use categories considered are: (1) not useable, (2) useable for combustion, (3) useable for shelter ventilation, (4) useable for cooling spray ponds, and (5) useable for operation of other aboveground equipment.

The five categories listed above are cumulative. For example, if atmospheric air is suitable for cooling spray ponds, it is also suitable for shelter ventilation and combustion. In the selection of these use categories, consideration was given to blast

TABLE 25. SHELTER SITUATIONS AND

Situation Number	Air Supply Suitable for: Combustion	Ventilation	Spray Pond[(a)]	Exposed Equipment[(b)]	Natural Water Supply: Limited	Ample 70-90 F	Ample <70 F	Cost[(c)], $ per 1,000 Btu per hr, for Indicated Type of Shelter: Fallout	30 psi	100 psi
1								97(19)	243(29)	381(39)
								105(47)	227(56)	333(64)
								111(33)	257(43)	395(53)
								[illegible]	115(18)	115(18)
2					X			97(17)	182(53)	254(58)
3					X	X		56(37)	90(39)	119(41)
4					X		X	← 18 to 34 →		
5	X							101(23)	247(33)	385(43)
	X							93	120	145
	X							92	134	174
6	X	X						72	94	115
	X	X						NA	15-42	15-42
	X	X						NA	41-55	58-72
	X	X (Except for sealed period)						78	121	164
7	X	X	X					56	78	NA
	X	X	X					78	122	NA
	X	X	X					127	180	NA
8	X	X	X	X				8-35	NA	NA
	X	X	X	X				19-33	NA	NA
	X	X	X	X				56	NA	NA
	X	X	X	X				59	NA	NA
	X	X	X	X				89	NA	NA
	X	X	X	X				120	NA	NA
9	X				X			86	86	86
10	X				X	X		43	43	43
	X				X	X		59	59	59
	X				X	X		106	106	106
11	X				X		X	← 19 to 36 →		
12	X	X			X			66	66	66
13	X	X			X	X		71	71	71
	X	X			X	X		43	43	43
	X	X			X	X		59	59	59
14	X	X			X		X	← 19 to 36 →		
15	X	X	X		X			50	50	NA
	X	X	X		X			66	66	NA
16	X	X	X		X	X		43	43	43
17	X	X	X		X		X	← 19 to 36 →		
18	X	X	X	X	X			8-35	NA	NA
	X	X	X	X	X			14-28	NA	NA
19	X	X	X	X	X	X		43	43	43
20	X	X	X	X	X		X	← 19 to 36 →		

(a) 10-in. concrete slab and piping assumed to be suitable for 30-psi blast.
(b) Exposed equipment refers to commercially available components assumed to have no blast resistance.
(c) Costs for 14 days except those in parentheses, which are for 24 hours.

Abbreviations: Eng. Gen. - Engine-generator set, St. - stored, MVC - Mechanical vapor compression, Atmos. - atmosphere, Ref. Eng. - refrigerant-vapor expansion engine, NA - not applicable.

POSSIBLE COOLING SYSTEMS

Sink	Power	Description: Main Mechanical Features	Cooling System No.
Ice	Manual	Circulate water over ice, O_2 for man power	1
St. Water	Manual	Open-cycle absorption, H_2SO_4-H_2O, O_2 for man power	1A
Ice	Ref. Eng.	Circulate water over ice	1B
Ammonia	Manual	Direct expansion of ammonia, O_2 for man power	1C
Water	Manual	Same as system 1A less stored water	2
Water	Manual	Open-cycle absorption, H_2SO_4-H_2O, O_2 for man power	3
Water	Manual	Circulate water through heat exchanger, O_2 for man power	4
Ice	Eng. Gen.	Same as System 1 plus power	5
St. Water	Eng. Gen.	MVC, cascade cycle, boiling water condenser	5A
Alcohol	Eng. Gen.	Mechanical compression of alcohol vapor, open cycle	5B
St. Water	Eng. Gen.	MVC, evaporative condenser in exhaust air duct	6
Air	Eng. Gen.	Ventilating air, various types of filters, 20 cfm per person	6A
St. Water	Eng. Gen.	Evaporative cool ventilating air, 10 cfm per person	6B
St. Water	Eng. Gen.	System 6 plus water for sensible temp. rise for 24 hrs	6C
St. Water	Eng. Gen.	MVC, spray pond	7
St. Water	Boiler	Absorption refrigeration, spray pond	7A
St. Water	Eng. Gen.	Air-cycle cooling, spray pond	7B
Air	Eng. Gen.	Ventilating air, various filters, 20 cfm per person	8
St. Water	Eng. Gen.	Evaporative cool ventilating air, 10 cfm per person	8A
Atmos.	Eng. Gen.	MVC, air-cooled condenser	8B
St. Water	Eng. Gen.	MVC, cooling tower	8C
St. Water	Boiler	Absorption refrigeration, cooling tower	8D
Alcohol	Eng. Gen.	Alcohol evaporation in a cooling tower	8E
Water	Eng. Gen.	Same as System 5A less stored water	9
Water	Eng. Gen.	MVC, water cooled	10
Water	Boiler	Absorption refrigeration, water cooled	10A
Water	Engine	Air-cycle cooling, water cooled	10B
Water	Eng. Gen.	Circulate cold water	11
Water	Eng. Gen.	Same as System 6 less stored water	12
Water	Eng. Gen.	Dehumidification and evaporative cooling	13
Water	Eng. Gen.	Same as System 10	13A
Water	Boiler	Same as System 10A	13B
Water	Eng. Gen.	Same as System 11	14
Water	Eng. Gen.	Same as System 7 less stored water	15
Water	Boiler	Same as System 7A less stored water	15A
Water	Eng. Gen.	Same as System 10	16
Water	Eng. Gen.	Same as System 11	17
Air	Eng. Gen.	Same as System 8	18
Water	Eng. Gen.	Same as System 8A less stored water	18A
Water	Eng. Gen.	Same as System 10	19
Water	Eng. Gen.	Same as System 11	20

resistance and its effect on cooling-system costs. Cost data are shown in the table for fallout shelters and for shelters affording protection against 30-psi and 100-psi overpressures.

For the first three air categories all cooling equipment would probably be located underground and any desired degree of blast protection could be selected for design. For the fourth and fifth categories, aboveground equipment could be used. Shallow-basin spray ponds are inherently more blast resistant than most other types of aboveground equipment. Therefore, spray ponds are listed in a separate category. For purposes of this report it was assumed that, essentially normal construction practices having been used, spray ponds would be capable of withstanding a 30-psi overpressure. To simplify the economic study, the other aboveground equipment was assumed to have no blast resistance. It was beyond the scope of this program to determine the cost of blast-resistant aboveground equipment.

The availability of natural water is divided into the following categories: (1) no supply, (2) limited supply, (3) ample supply between 70 and 90 F, and (4) ample supply below 70 F.

As was the case in selecting categories of air use, the above categories were arbitrarily chosen. However, they are logical choices. The limited-supply category implies that the water would have to be evaporated to absorb the cooling load. The amply-supply categories were divided at 70 F because water below 70 F could be used directly in a heat exchanger to cool a shelter, whereas if the temperature is above 70 F, a refrigeration device would be required. An upper temperature limit of 90 F was selected as being the maximum temperature which would be expected for natural water.

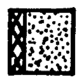
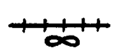
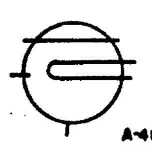
The estimated costs are shown in dollars per 1,000 Btu per hr for cooling systems having a capacity of 300,000 Btu per hr with a heat sink of sufficient capacity for 14 days of operation. In general, the costs on a Btu-per-hr basis for systems having larger capacities would be somewhat less than those shown, and the costs for smaller capacity systems would increase appreciably.

When a stored heat sink is specified, the cost of underground storage space is estimated to be \$0.70 per cu ft for fallout shelters where no blast resistance is required, \$4 per cu ft for 30-psi blast shelters, and \$7 per cu ft for 100-psi blast shelters. The cost for the 0.5 to 2 cu ft per 1,000 Btu per hr of shelter space required to house the equipment installed in the shelter is not included. The cost of mechanical power is included in the cooling system cost. However, the cost of the equipment required to utilize manual power is not included.

In determining the size of the heat sink and, therefore, its cost, the quantity of heat to be rejected to the sink was considered to include the heat to be removed from the occupied space, the energy input required to drive the equipment, and, for the open-cycle absorption system, the heat of mixing of the working fluids. Not included is any waste heat from the power system, whether an engine or manual.

SYSTEM SCHEMATICS AND COST BREAKDOWN

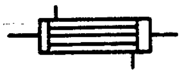
SYMBOLS FOR COOLING SYSTEM SKETCHES

	**Blast valve**
	**Filters**
	**Ventilation blower**
←CH——	**Chilled water flow**
←CHR——	**Chilled water return**
←W——	**Water flow**
←R——	**Refrigerant flow**
←A——	**Alcohol flow**
←S——	**Steam flow**
←©——	**Circulating pump**
	**Air-handling unit**
	**Compressor**
⊗	**Expansion valve**
	**Open-cycle absorption refrigeration device**

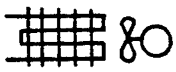
A-48240

Closed-cycle absorption refrigeration device

Boiler



Liquid-cooled condenser



Air-cooled condenser

Source of ample water (well, lake, pond, river, or utility)

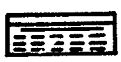
Spray pond

Cooling tower

Evaporative condenser

Water sprayed over ice

Open tank or pond

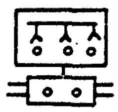


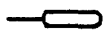
Closed tank or pond

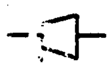
Open tank or pond containing pipe coil

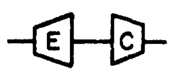
A-48241


Evaporative cooler


Dehumidifier, water cooled, and regenerator


Storage tank, pressurized

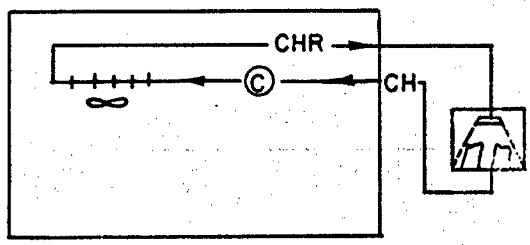

Refrigerant expander



Air-cycle, expander-compressor

A-48242

SYSTEM 1



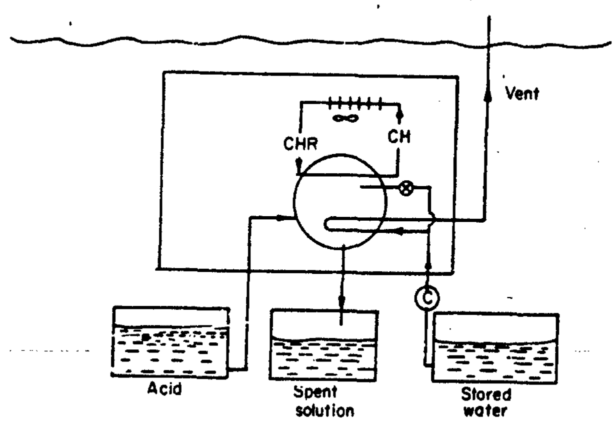
Description: Circulate water over ice and through heat exchanger
Heat sink: Stored ice
Power: Manual
Source of potable water
Extra oxygen required for manual power

Cost:* $ per 1,000 Btu per hr

	Type of Storage For Ice		
	Fallout	30 psi	100 psi
Air-handling unit	11 (11)	11 (11)	11 (11)
Storage	74 (5.3)	220 (15.7)	358 (25.6)
Ice	10 (0.7)	10 (0.7)	10 (0.7)
Pump, 60 gpm	2 (2)	2 (2)	2 (2)
	97 (19.0)	243 (29.4)	381 (39.3)

*Costs in brackets are for 24-hr period.
Cost of standby refrigeration equipment negligible; operating cost proportional to duration of standby.

SYSTEM 1A



A-48244

Description: Open-cycle absorption; absorbent, sulfuric acid; refrigerant, water
Heat sink: boiling water at 212 F
Power: stored or manual
Extra oxygen required for manual power; unconventional equipment; corrosion at high solution temperatures.

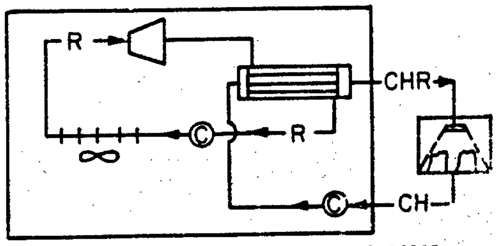
Cost:* $ per 1,000 Btu per hr

	Type of Storage for Fluids		
	Fallout	30 psi	100 psi
Air-handling unit	17(17)	17(17)	17(17)
Absorber unit**	25(25)	25(25)	25(25)
Acid tank	13(0.9)	53(3.8)	83(5.9)
Spent-solution tanks	23(1.6)	68(4.8)	110(7.9)
Acid	18(1.3)	18(1.3)	18(1.3)
Water tank	8(0.6)	45(3.2)	79(5.6)
Pump	1(1)	1(1)	1(1)
	105(47.4)	227(56.1)	333(63.7)

*Costs in brackets for 24-hr period.
**50% more expensive than low-temperature absorber unit.

C

SYSTEM 1B



A-48245

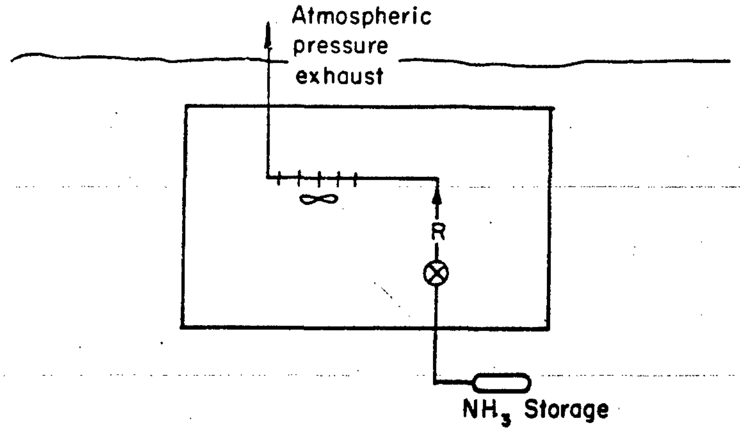
Description: Circulate water over ice and through condenser
Heat sink: Stored ice
Power: Refrigerant-vapor engine
Source of potable water
Oxygen not required for power
Refrigerant-vapor engine not commercially available

Cost:* $ per 1,000 Btu per hr

	Type of Storage for Ice		
	Fallout	30 psi	100 psi
Vapor engine			
Expander	3(3)	3(3)	3(3)
Vapor generator	12(12)	12(12)	12(12)
Vapor condenser	10(10)	10(10)	10(10)
Pump, water	2(2)	2(2)	2(2)
Storage	74(5.3)	220(15.7)	358(25.6)
Ice	10(0.7)	10(0.7)	10(0.7)
	111(33.0)	257(43.4)	395(53.3)

*Costs in brackets for 24-hr period. Cost of standby refrigeration equipment negligible, operating cost proportional to duration of standby.

SYSTEM 1C



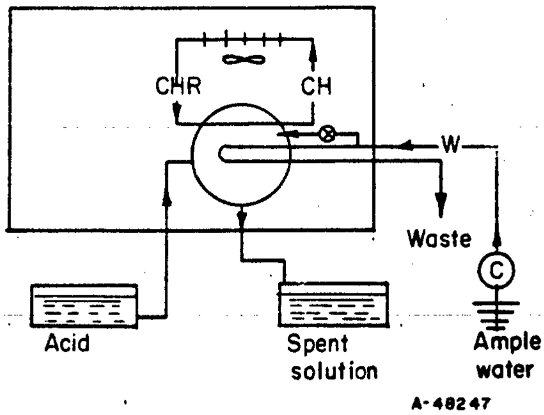
Description: Direct-expansion coils
Heat sink: Stored ammonia
Power: Manual-powered blower
Ammonia toxic
Extra oxygen required for manual power

Cost:* $ per 1,000 Btu per hr

Air-handling unit, direct expansion	11(11)
Storage	74(5.3)
Ammonia	30(2.1)
	115(18.4)

*Costs in brackets are for 24-hr period.

SYSTEM 3



Description: Open-cycle absorption; absorbent, sulfuric acid; refrigerant, water
Heat sink: Natural source water
Power: Manual
Extra oxygen required for manual power
Unconventional equipment

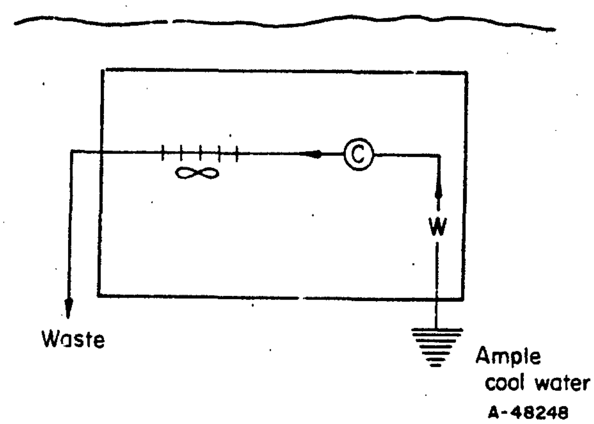
Cost:* $ per 1,000 Btu per hr

	Type of Storage For Fluids		
	Fallout	30 psi	100 psi
Air-handling unit	17(17)	17(17)	17(17)
Absorber unit**	17(17)	17(17)	17(17)
Acid tank	5(0.4)	21(1.5)	33(2.4)
Spent solution tank***	9(0.6)	27(1.9)	44(3.1)
Acid	7(0.5)	7(0.5)	7(0.5)
Pump	1(1.0)	1(1)	1(1)
	56(36.6)	90(38.9)	119(41.0)

*Costs in brackets are for 24-hr period.
**Absorber unit includes pumps.
***Spent solution tank not needed if spent solution can be dumped.

F

SYSTEM 4



Description: Cool water through heat exchanger
Heat sink: Natural source cool water
Power: Manual
Extra oxygen required for manual power

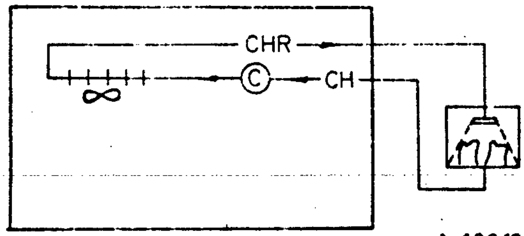
Cost: $ per 1,000 Btu per hr

	Water temperature, F	
	50	70
	Air flow, cfm	
	0.10	0.19
	COP*	
	68	30
Air-handling unit**	15	30
Well, 100 ft	1	1
Pump	2	3
	18	34

*COP calculated for 100-ft head for well.

**Cost adjusted for temperature difference and air flow.

SYSTEM 5



A-48249

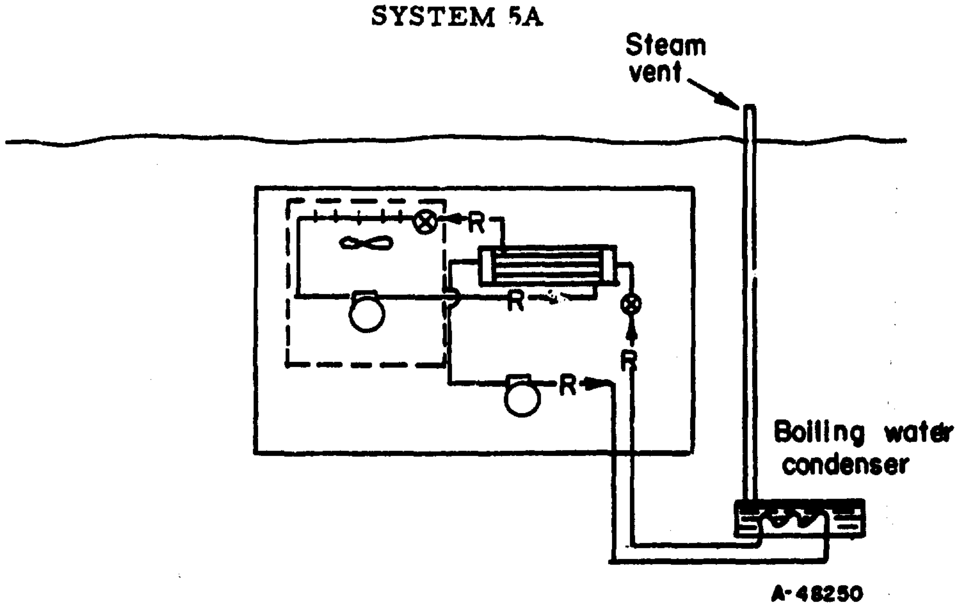
Description: Circulate water over ice and through heat exchanger
Heat sink: Stored ice
Power: Engine-generator set

Cost:* $ per 1,000 Btu per hr

	Type of Storage For Ice		
	Fallout	30 psi	100 psi
Air-handling unit	11(11)	11(11)	11(11)
Storage	74(5.3)	220(15.7)	358(25.6)
Ice	10(0.7)	10(0.7)	10(0.7)
Pump, 60 gpm	2(2)	2(2)	2(2)
Power	4(4)	4(4)	4(4)
	101(23.0)	247(33.4)	385(43.3)

*Costs in brackets are for 24-hr period. Cost of standby refrigeration equipment negligible, operating cost proportional to duration of standby.

SYSTEM 5A

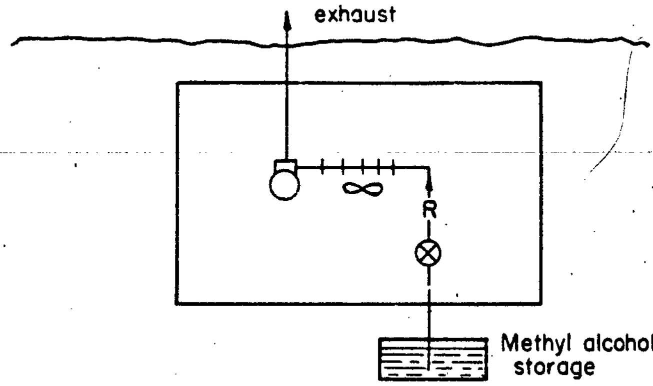


Description: Mechanical vapor compression, cascade cycle, 212 F condenser
Heat sink: Water, phase change
Power: Engine-generator set
Modification of commercial equipment required

Cost: $ per 1,000 Btu per hr	Type of Storage For Water		
	Fallout	30 psi	100 psi
MVC, direct expansion*	60	60	60
Water storage	8	35	60
Power	25	25	25
	93	120	145

*Cost assumed to be twice that of a single-stage MVC unit.

SYSTEM 5B



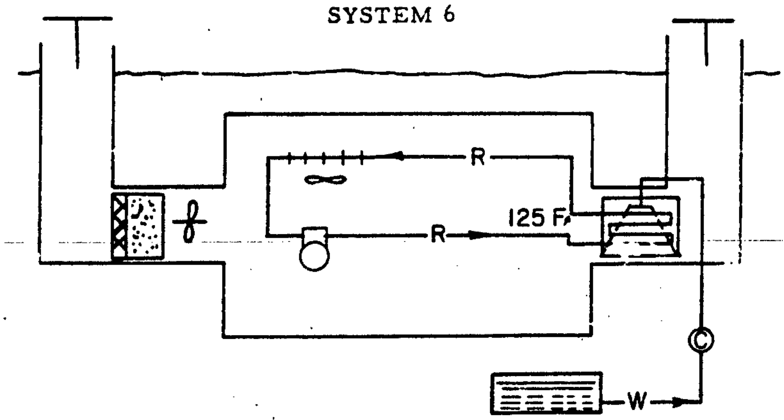
Description: Open-cycle alcohol vapor compression
Heat sink: Stored methyl alcohol
Power: Engine-generator set
Toxic fumes to be exhausted or burned
Compressor may need development

Cost: $ per 1,000 Btu per hr

	Type of Alcohol Storage		
	Fallout	30 psi	100 psi
Alcohol	30	30	30
Storage	12	54	94
Compressor*	20	20	20
Air-handling unit, direct expansion	11	11	11
Electric motor and wiring	4	4	4
Power	15	15	15
	92	134	174

*Cost assumed to be equal to that of commercial rotary-vane air compressors.

SYSTEM 6

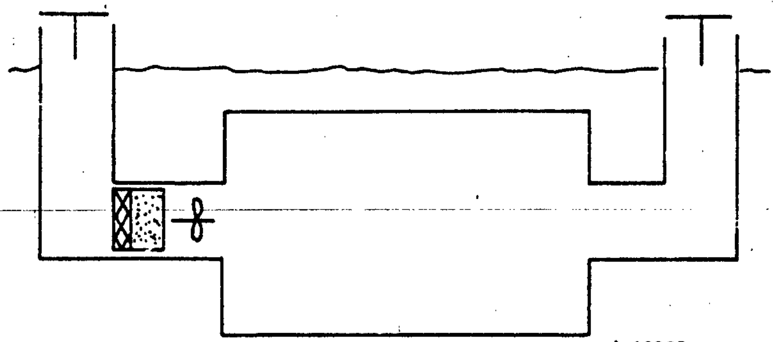


Description: Mechanical vapor compression with evaporative condenser
Heat sink: Stored water
Power: Engine-generator set
Heat can be rejected with ventilating air

Cost: $ per 1,000 Btu per hr

	Type of Storage For Water		
	Fallout	30 psi	100 psi
MVC, direct expansion	30	30	30
Evaporative condenser	18	18	18
Water storage	6	28	49
Water pump	1	1	1
Power	17	17	17
	72	94	115

SYSTEM 6A



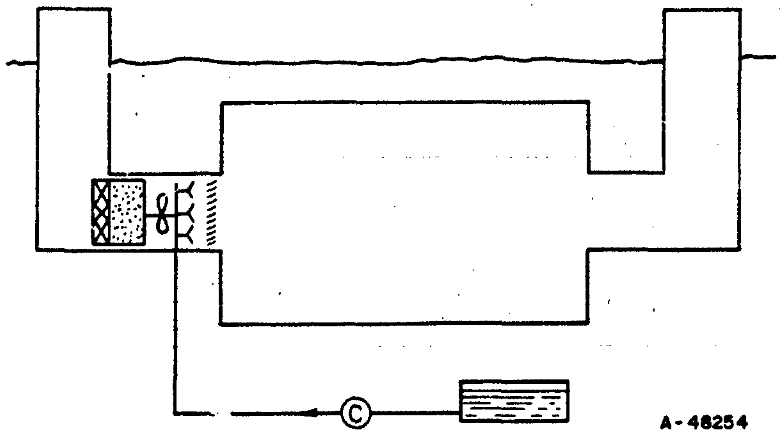
Description: Cool with ventilating air, filters optional
Heat sink: Ventilating air
Power: Engine-generator set

Cost:* $ per 1,000 Btu per hr

	Type of Filters		
	None	Particulate	CW/BW
Blast valves	7	7	7
Filters	0	2	21
Blower and power	8	11	14
	15	20	42

*Cost estimates made assuming: (1) 500-person shelter, (2) cooled by 20 cfm per person, (3) 20 cfm dissipates 600 Btu per hr. Costs vary directly with air-flow rate.

SYSTEM 6B



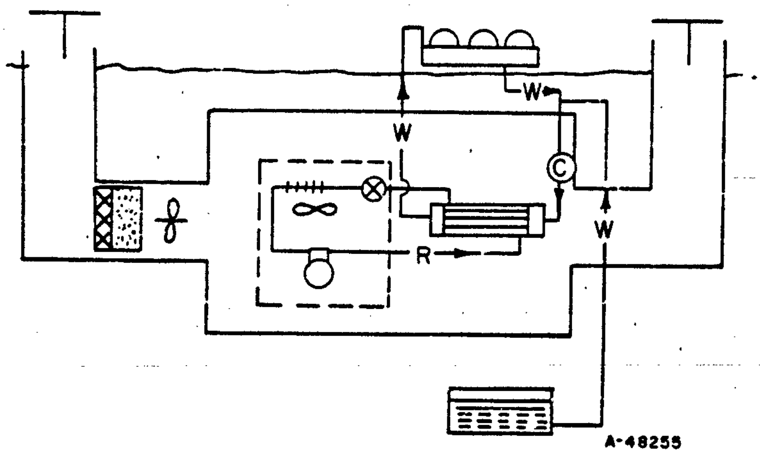
Description: Ventilating air evaporative cooled with stored water
Heat sink: Stored water
Power: Engine-generator set
Air-flow rate 1/2 of that required by ventilation only, 10 cfm per person, see System 6A
Evaporative cooler also an air cleaner

Cost: $ per 1,000 Btu per hr

	30-psi Blast Shelter		
	Type of Filters		
	None	Particulate	CW/BW
Blast valves	4	4	4
Filters	0	1	11
Blower and power	6	7	9
Evaporative cooler	7	7	7
Water storage*	23	23	23
Water pump	1	1	1
	41	43	55
	System For 100-psi Blast Shelter		
	58	60	72

*Assume sensible heat of inlet air equals that of outlet air.

SYSTEM 7



A-48255

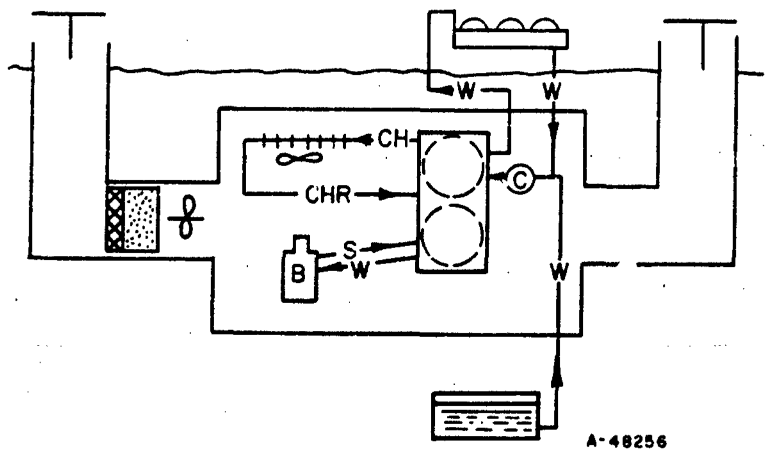
Description: Mechanical vapor compression with spray pond
Heat sink: Stored water
Power: Engine-generator set

Cost: $ per 1,000 Btu per hr

	Type of Water Storage	
	Fallout	30 psi
MVC, direct expansion, with condenser	30	30
Spray pond*	3	3
Water storage	6	28
Water pump	1	1
Power	16	16
	56	78

*10-inch concrete slab with spray equipment.

SYSTEM 7A



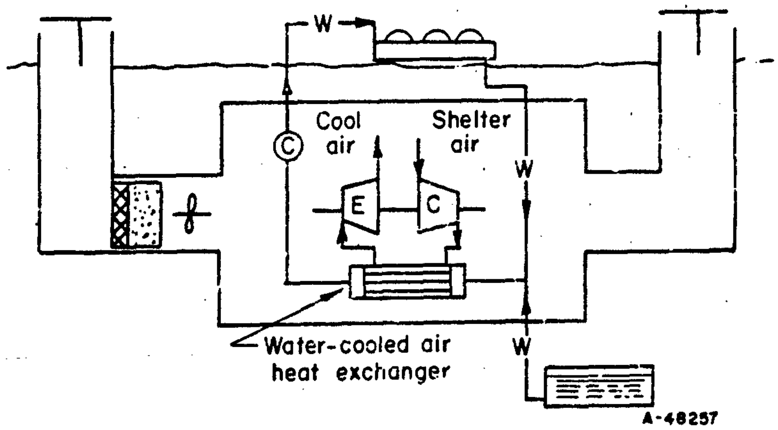
Description: Absorption with spray pond
Heat sink: Stored water
Power: Boiler and small quantity of electricity

Cost: $ per 1,000 Btu per hr

	Type of Water and Fuel Storage	
	Fallout	30 psi
Absorption	27	27
Air-handling unit	11	11
Boiler	14	14
Spray pond*	6	6
Water storage	12	56
Fuel and fuel storage	4	4
Pump	2	2
Power, electric	2	2
	78	122

*10-inch concrete slab with spray equipment.

SYSTEM 7B



Description: Air-cycle cooling with spray pond
Heat sink: Stored water
Power: Mechanical drive by engine
Equipment development required

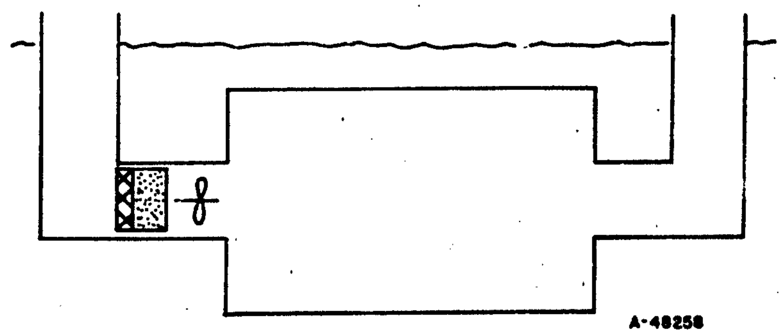
Cost: $ per 1,000 Btu per hr

	Type of Storage for Water	
	Fallout	30 psi
Compressor-expander	8	8
Heat exchanger	52	52
Spray ponds*	7	7
Water storage	14	67
Pump	3	3
Power	43	43
	127	180

*10-inch concrete slab with spray equipment.

P

SYSTEM 8



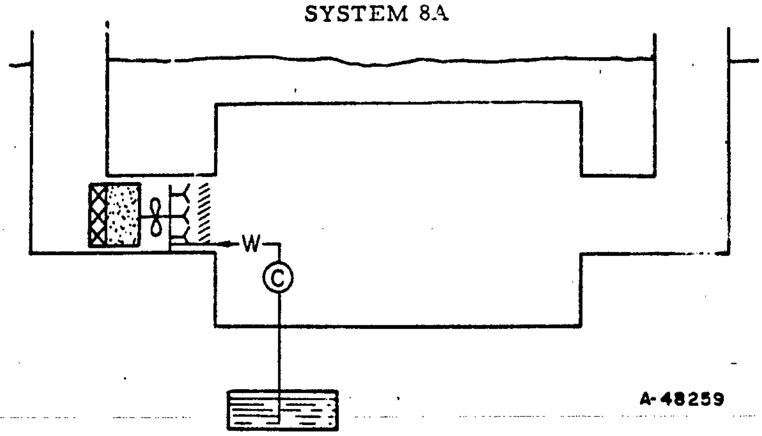
Description: Cool with ventilating air, filters optional
Heat sink: Ventilating air
Power: Engine-generator set

Cost* $ per 1,000 Btu per hr

	Type of Filter		
	None	Particulate	CW/BW
Filters	0	2	21
Blowers and power	8	11	14
	8	13	35

*Cost estimates made assuming: (1) 500 person shelter, (2) cooled by 20 cfm per person, (3) 20 cfm dissipates 600 Btu per hr. Costs vary directly with air flow.

SYSTEM 8A

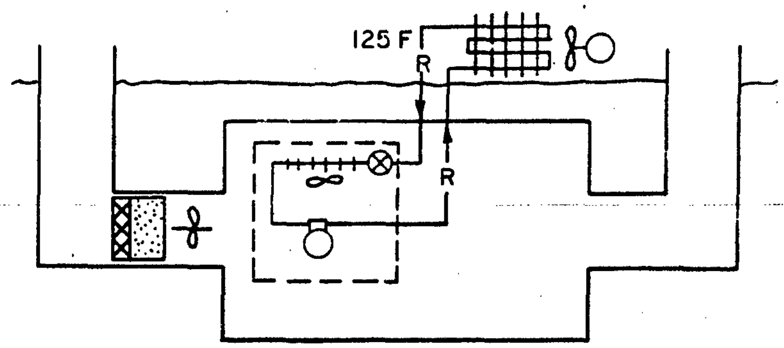


Description: Ventilating air evaporative cooled with stored water
Heat sink: Stored water and ventilating air
Power: Engine-generator set
Air flow one-half that required by ventilation only, 10 cfm per person, see System 8
Evaporative cooler also an air cleaner

Cost: $ per 1,000 Btu per hr

	Type of Filter		
	None	Particulate	CW/BW
Filters	0	1	11
Blower and power	6	7	9
Evaporative cooler	7	7	7
Water storage	5	5	5
Water pump	1	1	1
	19	21	33

SYSTEM 8B

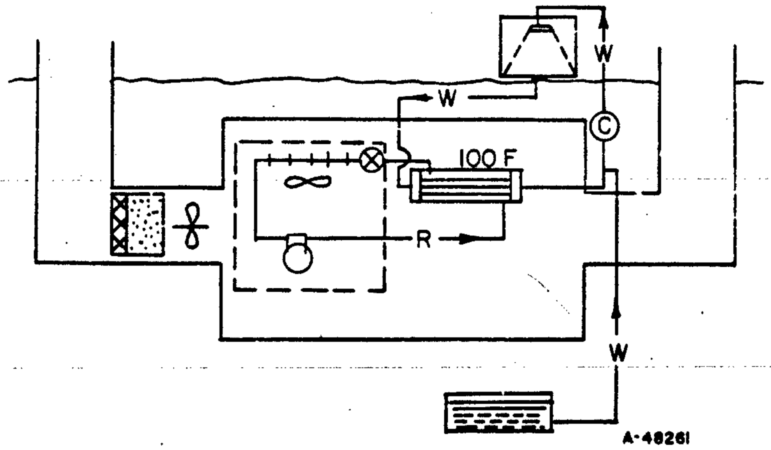


Description: Mechanical vapor compression with air-cooled condenser
Heat sink: Atmosphere
Power: Engine-generator set

Cost: $ per 1,000 Btu per hr

MVC, direct expansion, with condenser	40
Power	16
	56

SYSTEM 8C

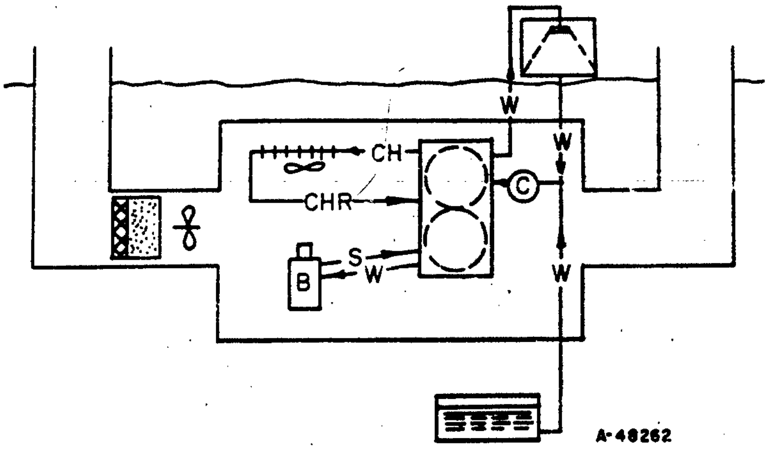


Description: Mechanical vapor compression with cooling tower
Heat sink: Stored water
Power: Engine-generator set

Cost: $ per 1,000 Btu per hr

MVC, direct expansion, with condenser	30
Cooling tower	9
Water storage	6
Water pump	1
Power	13
	59

SYSTEM 8D

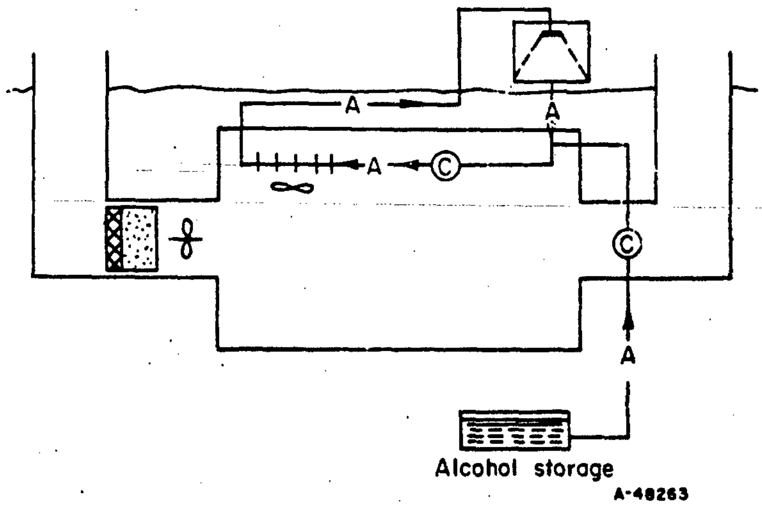


Description: Absorption with cooling tower
Heat sink: Stored water
Power: Boiler and small quantity of electricity

Cost: $ per 1,000 Btu per hr

Absorption unit	27
Air-handling unit	11
Boiler	14
Cooling tower	18
Water storage	12
Pump	2
Fuel and storage	3
Power, electric	2
	89

SYSTEM 8E



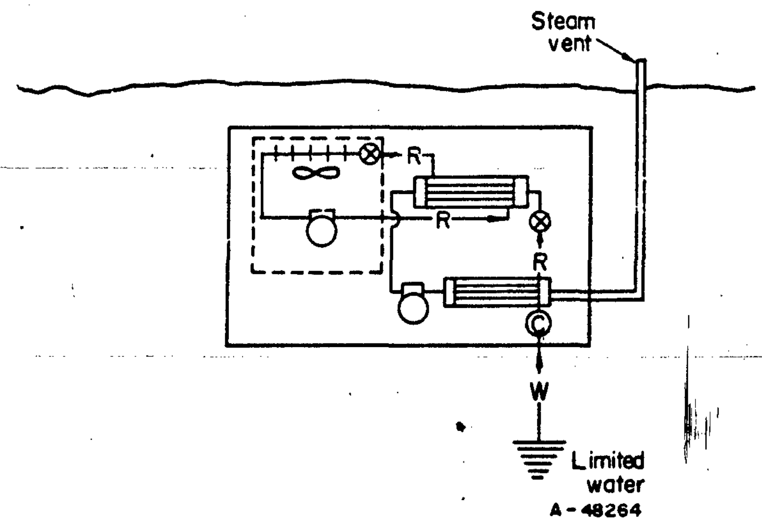
Description:	Chilled alcohol from cooling tower circulating in heat exchanger Heat sink: Alcohol Power: Engine-generator set Rejects heat to warm atmosphere without a refrigeration device

Cost: $ per 1,000 Btu per hr

Alcohol	52
Alcohol storage	20
Cooling tower*	9
Air-handling unit**	30
Power	6
Pumps	3
	120

*Cost of tower assumed to be same as water cooling tower.
**Air-handling unit sized for 75 F fluid.

SYSTEM 9



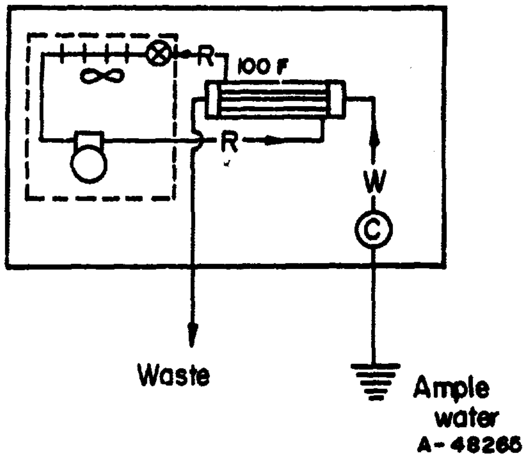
Description: Mechanical vapor compression, cascade cycle, 212 F condenser
Heat sink: Water, phase change
Power: Engine-generator set
Modified commercial equipment

Cost: $ per 1,000 Btu per hr

	Fallout	30 psi	100 psi
MVC, direct expansion*	60	60	60
Power	25	25	25
Water pump	1	1	1
	86	86	86

*Cost assumed to be twice that of a single-stage MVC unit.

SYSTEM 10



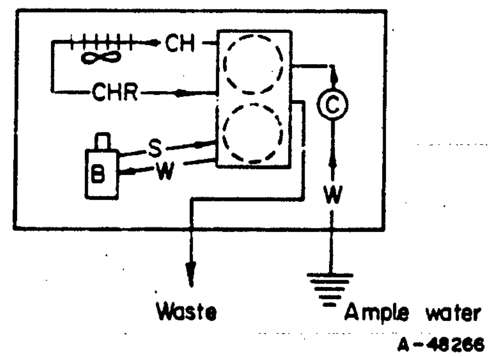
Description: Mechanical vapor compression, water-cooled condenser
Heat sink: Water
Power: Engine-generator set

Cost: $ per 1,000 Btu per hr

MVC, direct expansion, with condenser	30
Water pump	2
Power	11
	43

X

SYSTEM 10A

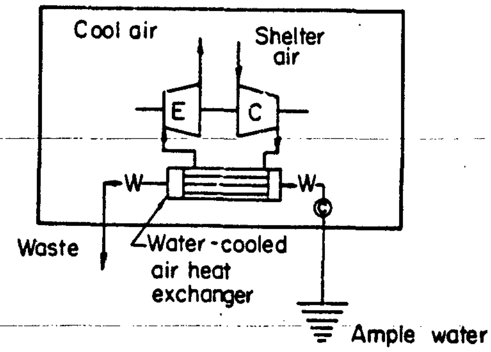


Description: Absorption, water-cooled absorber and condenser
Heat sink: Water
Power: Boiler and small source of electricity

Cost: $ per 1,000 Btu per hr

Absorption unit	27
Air-handling unit	11
Boiler	14
Fuel and storage	3
Pump	3
Power, electric	1
	59

SYSTEM 10B



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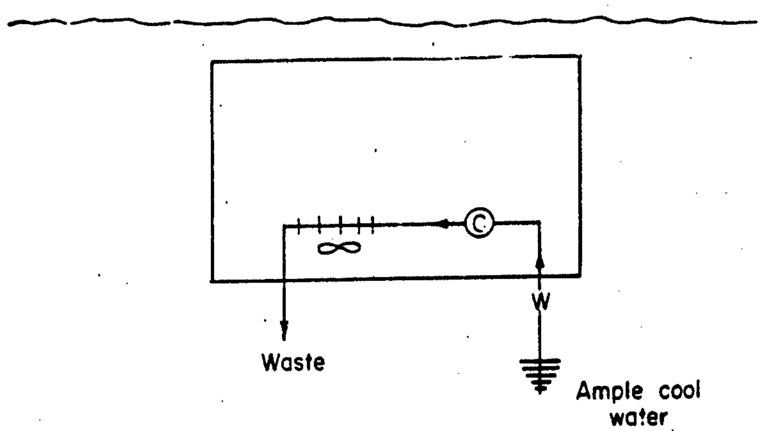
Description: Air-cycle cooling
Heat sink: Water
Power: Mechanically driven by engine
Development required

Cost: $ per 1,000 Btu per hr

Compressor-expander	8
Heat exchanger	52
Power	43
Pump	3
	106

SYSTEM 11



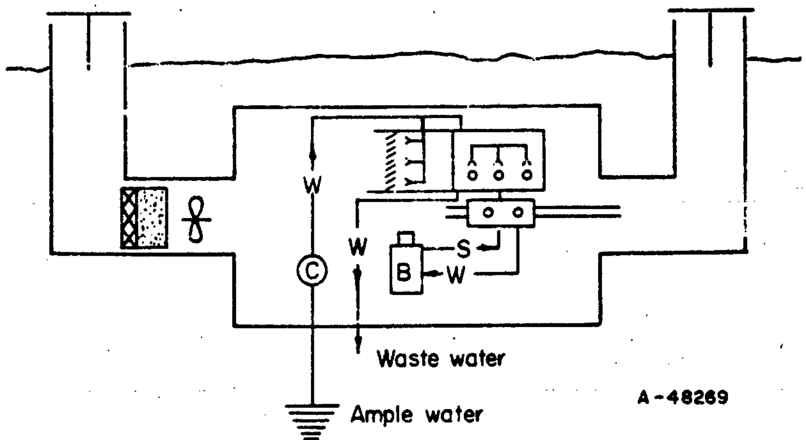
Description: Circulate cool water
Heat sink: Water
Power: Engine-generator set

Cost: $ per 1,000 Btu per hr

	Water Temperature, F	
	50	70
Air-handling unit	15	30
Well	1	1
Water pump	2	3
Power	1	2
	19	36

Aa

SYSTEM 13



Description: Dehumidify, evaporative cool, recirculate shelter air
Heat sink: Water
Power: Engine-generator set
Reject heat to relatively warm water
Remove bacteria and mold spores from air
Use ventilating exhaust air for regenerator

Cost: $ per 1,000 Btu per hr

Dehumidifier	27
Regenerator	13
Boiler	20
Evaporative cooler	5
Pump	1
Power	5
	71

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