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Final Technical Report

ELECTRIC AND HYBRID VEHICLES ENVIRONMENTAL CONTROL SUBSYSTEM STUDY

Prepared for:
Jet Propulsion Laboratory

Prepared Under:
Contract No. 955682

May 15, 1981

MECHANICAL TECHNOLOGY INCORPORATED
968 Albany-Shaker Road
Latham, New York 12110

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ENVIRONMENTAL CONTROL SUBSYSTEM STUDY

Prepared for:

Jet Propulsion Laboratory
California Institute of Technology
Pasadena, California
Contract No. 955682

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ABSTRACT

The design of environmental control subsystems (ECS) for the passenger compartment of electric and hybrid vehicles requires new approaches. Conventional as well as unconventional techniques were identified for providing environmental control suited for the unique characteristics of electric and hybrid vehicles. These techniques included various types of heat pumps, thermal energy storages, and reversible chemical reactions.

A novel technique called the Split Heat Pump appears to meet the requirements in a cost-effective manner, and, thus, has been selected as the most suitable element for long-term development. The split heat pump has no moving parts and requires no fuel on board the vehicle. Recharging can be accomplished with a suitable heat source in the garage, and hence can be operated with multitudes of energy sources such as electricity, oil, natural gas, or even wood.

Thermal energy storage utilizing sensible heat for heating and latent heat of freezing for cooling has the most merit for intermediate-term development. The gasoline-engine-driven heat pump is suitable for near-term product development.

As a by-product of this study, one version of the split heat pump also appears to have potential for application as an air-conditioner in conventional internal-combustion-engine vehicles. This version may provide a less expensive air-conditioning system with practically no penalty in mileage as compared to presently available systems.

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

LIST OF FIGURES

LIST OF TABLES

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EXECUTIVE SUMMARY

This final report contains the results of a nine-month study contract awarded to Mechanical Technology Incorporated (MTI) by the Jet Propulsion Laboratory, Pasadena, California. The program objective was to select an environmental control subsystem (ECS) suitable to the unique characteristics of electric and hybrid vehicles. The study, in addition to addressing the need for environmental control in the passenger compartments of these vehicles, also examined various methods of obtaining the desired temperature control for the battery pack.

Both conventional and innovative concepts were considered during the study, which began by defining the functional requirements of ECS equipment. Following categorization by methodology, technology availability and risk, all viable ECS concepts were evaluated. Each was assessed independently for benefits versus risk, as well as for its feasibility to short-, intermediate- and long-term product development. Selection of the preferred concept was made against these requirements, as well as the study's major goal of providing safe, highly efficient and thermally confortable ECS equipment. A summary of the major study tasks is presented below.

Functional Requirements Specification

The ECS function is to provide desirable environmental conditions within the controlled space (passenger compartment) when the ambient conditions take on a wide range of values. Thus, thermal comfort, safety and operator efficiency, coupled with an overall effort to minimize energy consumption, were key considerations throughout the study task to define ECS functional requirements.

Design criteria for the sizing of appropriate ECS elements were established from the following: demand thermal loads; controlled-space and ambient temperatures; time required to reach steady-state operation; relative humidity; number of air exchanges; safety (defogging and defrosting); and state-of-the-art surveys. The design point conditions for the passenger

compartment were derived from mathematical modelling of the physical, physiological and psychological processes involved in the determination of thermal comfort. The model, based on a thermal comfort equation developed by P.O. Fanger, accounted for:

- Thermal exchange with environment
- Change in stored heat
- Physiological response triggering vasomotor mechanisms
- Effects of blood circulation rate change.

The following chart represents the resulting design point specifications for the passenger compartments of electric vehicles:

Air Exchange > 5 cfm/person		
Parameter	During Heating Season	During Cooling Season
T_a, dry-bulb temp.	>68°F	<75°F
T_w, wet-bulb temp.	-	<75°F
Air velocity at the passenger	<0.5 meter/sec	<1.5 meter/sec
T_{mr}, mean radiant temp.	Limit Not Specified	

Design point ambient conditions were determined by integrating the weather data over the U.S.A. with the car population density distribution. The following philosophy was utilized: "Ambient conditions will be worse than design ambient conditions for less than 1% of the time for less than 1% of the total car population." The resulting design point specifications are:

Conditions	For Heating Season	For Cooling Season
T_a	-10°F	100°F
T_w	-	74°F
Air Velocity	45 mph	45 mph
Solar Insolation	-	326 Btu/hr/ft^2

Duration of environmental control was another important parameter in determining the ECS functional requirements. Travel scenarios depicting typical U.S. driving patterns were constructed in order to establish the following ECS design load specification:

- Continuous 17,000 Btu/hr (•5 kW)
- 2.5 hours of operation maximum (passenger compartment)
- 42,500 Btu (maximum)
- 10 hours of recharging
- 10 hours of unplugged operation for battery temperature controller.

- Battery Temperature Control

Key objectives in establishing the functional requirements for maintaining temperature control within the battery compartment were to:

- Ensure availability
- Enhance power density
- Enhance energy density
- Increase number of life cycles
- Improve charge/discharge efficienty.

Specific configurations, which used insulation to trap the by-product heat available from the batteries, were quantitatively examined. The various configurations were selected to see an optimum for volume devoted to insulation versus heat loss. Only well-established, state-of-the-art materials were considered.

The requirement for rejection of heat from the batteries is straightforward. In the calculations, the batteries are permitted to reach 130°F and it is assumed that cooling air at 100°F is available. With that temperature differential available, the dissipation from lead-acid batteries is within the capability of a small-capacity blower. Thus, the preferred method is to employ simple insulation with a small thermostatically controlled blower that uses ambient air for cooling. Attention should be given to securing the lowest available temperature, avoiding stagnant, under-hood heated air. An option of heating the batteries, using garage power when available and battery power when under way, may be useful for geographic areas with severely cold conditions.

The benefits of this approach appear significant. The trade-off of providing additional volume to accommodate the insulation has important implications to vehicle design and battery maintenance. No significant improvement was found on this simple configuration.

Identification and Ranking of ECS Elements

Because an electric vehicle is designed to significantly reduce the use of petroleum-based fuel, the maintenance of the thermal environment within the passenger compartment should use little or no fuel of this type. With this goal as a focus, over 80 elements for heating and/or cooling of electric vehicle passenger compartments were studied. Key considerations included component efficiencies, system COP, system capacity and system weight. Preliminary calculations were performed to determine each element's viability in gross terms such as weight and volume. Resulting data indicated that over 30 of these elements were capable of being developed into practical subsystems without imposing undue penalties on vehicle weight and propulsion energy usage and, hence, on vehicle range and performance.

For heating, a package weighing less than 60 lb and occupying less than 1 ft^3 was identified. This unit provides adequate heating at ambient temperatures as low as -10°F and requires no on-board storage of a petroleum-based fuel. For combined heating and cooling, a package weighing less than 200 lb and occupying less than 6 ft^3 was identified. Such a unit also does not require on-board storage of gasoline or energy from electric batteries.

ECS elements were divided into the following categories:

- Heat pumps
- Thermal storage
- Reversible thermochemical reactions.

Within the heat pump category, various types were considered; e.g., vapor compression, absorption cycle, thermoelectric, magnetic and split systems.

Thermal storage schemes under evaluation used either sensible heat or latent heat of phase change; i.e., salts, oils, paraffins, sand, liquified gases. Reversible thermochemical reactions were identified as having the potential for heat storage in excess of 3000 Btu/lb; however, published information is insufficient to enable a feasibility determination.

In order to be considered for ranking and possible recommendation, candidate ECS elements were screened for feasibility; each was required to meet the JPL-approved energy usage criteria and performance specifications. Criteria upon which each element was ranked included:

- First cost (25)*
- System life (5)
- Range impact (10)
- Energy efficiency (5)
- Storage period (5)
- Maintenance cost (10)
- Performance impact (10)
- Consumer-perceived risk (5)
- Noise level (10)
- Environmental impact (5)
- Packaging and volume (5)
- Development status (5).

Selection of Recommended ECS Elements

Based on the results of the ranking exercise, a gasoline-engine-driven vapor compression heat pump offers the greatest potential for near-term application. This system can readily be used because only state-of-the-art product development is required. First costs are low; no daily charging is required; and both heating and cooling can be provided with no preplanning.

For intermediate-term development, a system utilizing water thermal energy storage is the preferred configuration. Although this type of system offers

*Numbers in parentheses indicate weighting factor assigned to each criterion.

only a limited storage period, its other functional characteristics make it a superior choice for product development within the next three to four years. Such an ECS element requires no on-board use of petroleum fuel and can be effectively applied to both heating and cooling cycles. Other advantages include a simplicity and similarity with present automobile heating systems, low noise level and a short development period.

Preliminary calculations indicate than an ammonia-water split heat pump meets all functional requirements in a cost-effective manner; hence, its selection as the "best" configuration for long-term development. This system, which also can be applied to both heating and cooling cycles, requires no moving parts on board the vehicle and no on-board use of petroleum fuel. It offers low overall weight, as well as long storage periods that are comparable to gasoline engines.

In the split heat pump system, the thermodynamic process rates can be operated independently. It is thus possible to design the home-base equipment to perform the regeneration function over a 24-hour period, while the maximum operating time of the vehicle-base equipment is 2 1/2 hours. This design results in a considerable reduction in the size of the home-base equipment, as well as of the weight carried on board the vehicle. Furthermore, recharging can be accomplished with a variety of energy sources, including electricity, oil, natural gas or even wood.

1.0 INTRODUCTION

The development of propulsion systems for electric and hybrid vehicles has received considerable attention in the recent past. This development has now reached a stage where electric vehicles can satisfy most of the perceived needs of an average American family as far as the propulsion requirements are concerned. The electric vehicles built so far or those planned for the near future have not addressed the problem of passenger compartment environmental control to any significant extent. Most of the serious builders of electric vehicles provide a gasoline-fired heater for the passenger compartment. Only recently has prototype development of an electrically or gasoline engine operated air conditioner begun at Airco.

At present, electric vehicles cost far more than cars equipped with I.C. engines. Projections for the near future indicate this trend will continue. It is perceived that since the vehicle is expensive and is likely to continue to be so, the potential buyer will expect it to be comfortable year round. In any event, some minimal environmental control will be required by federal and state laws for reasons of safety; for example, defogging and defrosting of windshields.

A straightforward adaptation of the techniques used in vehicles equipped with I.C. engines is not possible due to the following differences in the propulsion systems between electric vehicles and I.C. engine vehicles:

1) I.C. engines produce large quantities of heat as a by-product, whereas electric vehicle motors generate relatively little heat as a by-product. Furthermore, what little heat is generated is distributed among components that are located far from each other. The temperatures of such heat sources are also not very high.

2) The I.C. engine in present-day cars runs continuously, even when the vehicle is not in motion such as at a red traffic light, whereas in many conceptual designs, the propulsion system of an electric or hybrid vehicle is turned off when the vehicle is at rest.

Thus, new approaches are required to address the need for environmental control of the passenger compartment of electric and hybrid vehicles.

The environmental control subsystem elements will have some impact on the performance of other subsystems. Any decrease in range or acceleration of an electric vehicle caused by its ECS will be of great concern as these performance factors are already substandard to what we are accustomed to.

The major objective of this study was to provide a basis for selection and to select, for the purpose of potential prototype development, those elements comprising the Environmental Control Subsystem (ECS) that are best matched to the unique characteristics and requirements of electric and hybrid vehicles. As a result of this study, three candidates were selected to address the needs in three different time spans: short (immediate), mid-term (within the next three to four years) and long-term (after five years). A program plan identifying various tasks, cost and schedule for prototype development was generated for each of these three systems.

Apart from the ECS for the passenger compartment, this study addressed the need for battery temperature control. The batteries for electric vehicles are undergoing major development. Some of them can function at their best at temperatures which are different from normally encountered ambient temperatures. For some battery types, higher than ambient temperature is most suitable; others, however, need lower temperatures. This study examined various methods of obtaining the desired temperature control for the battery pack.

1.1 Report Organization

The work on this study was divided into various subtasks. Detailed reports were issued on these subtasks or groups of subtasks. This report summarizes the work presented in those reports. However, for the sake of ready reference and completeness, the detailed reports are included here as appendices.

2.0 FUNCTIONAL REQUIREMENTS SPECIFICATION

As new approaches are required for designing the ECS for electric and hybrid vehicles, the first step is to identify the important parameters and specify their values. Figure 1 schematically shows the ECS function: to provide desirable environmental conditions within the controlled space (passenger compartment) when the ambient conditions take on a wide range of conditions. Literature surveys, as well as personal contacts with people connected with ECS's for passenger cars, revealed that no ECS design standards exist with the exception of windshield defogging and defrosting. A review of present practice indicated that the ground rules used for selection and design of ECS equipment are materially different from those applicable to electric vehicles. In the case of electric vehicles, utmost care must be taken to minimize the energy requirements for the ECS and also to minimize its weight. Significant reductions in thermal loads (and, hence, in energy requirements) can be achieved by suitable modification in the vehicle body construction. Such modifications will address themselves to reducing heat transfer rates from various parts of the body, viz., reducing cracks between doors and body to reduce unintentional air infiltration, using double thermopane glass for glazing, photo-chromic glass to reduce radiation loads, and use of selective coatings to exploit radiation cooling. However, such modifications have been considered to be out of the scope of this study. Present practice is to design air conditioning equipment to provide for 22,000 to 26,000 Btu/hr and 8,000 to 25,000 Btu/hr for heating in the case of a compact size automobile. For this study, assuming a thermally well-designed vehicle, cooling loads of 17,000 Btu/hr were used as the maximum value for steady-state conditions.

2.1 Design Conditions for Passenger Compartment

As it is desired to minimize ECS energy reqirements, it is necessary to keep the temperature difference between ambient and the passenger compartment as small as possible without sacrificing comfort. Thus, the parameters that affect thermal comfort were studied. Work on thermal comfort in automobiles is scarce in published literature; whatever is published is mostly experimental and empirical. However, considerable analytical work has been carried out for the home and work environment. A thermal comfort

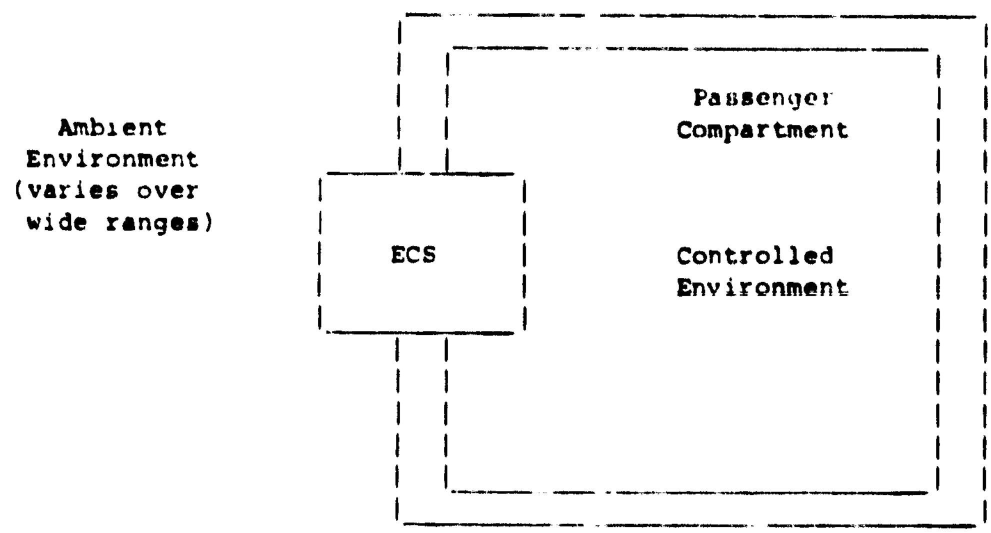

Figure 1. Schematic to Show the Function of ECS

equation, developed by P.O. Fanger, is based on mathematical modelling of various heat exchange processes of the human body coupled with the physiological control mechanisms such as vasomotor restrictions of blood circulation rate, etc. The criteria for comfort are based on values of measurable physiological variables such as heart rate, skin temperature and skin wettedness, etc., determined by statistical correlation of results of experiments performed on a large number of individuals under carefully controlled conditions. The results of the solution of the thermal comfort equation are presented in the form of lines of constant comfort in psychrometric charts. One such chart is shown in Figure 2.

Two design conditions are identified for the environment within the passenger compartment. These conditions are based on the fact that people wear heavy outdoor clothing during winter and light clothing during summer. Table 1 shows the parameters and respective values that define the thermal environment within the passenger compartment during winter and summer; these values are derived from the Fanger thermal comfort charts described above. It is recognized that a straightforward application of these equations derived for the home and office environment may involve inaccuracies, the major limitation being that the thermal comfort charts are based on an assumption of steady-state conditions, whereas transient effects play an important role in daily commuter car travel.

2.2 Design Ambient Conditions

The range of values of parameters that define ambient conditions (as far as the thermal load on the ECS is concerned) would be very large if the ECS design provided for the worst possible conditions over the continental U.S.A. Such an approach would result in an unduly large ECS for most consumers. Therefore, "design ambient conditions" were determined based on the following philosophy: "Ambient conditions will be worse than design ambient conditions for less than 1% of the time for less than 1% of the total car population."

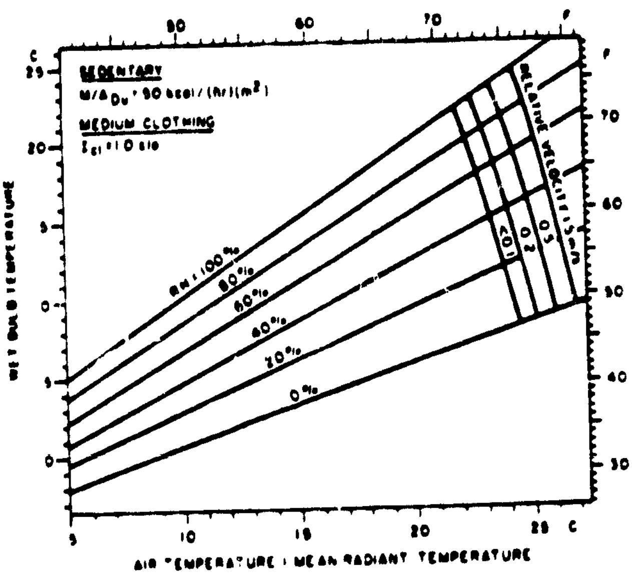

Figure 2. Lines of Comfort for Medium Clothing

TABLE 1

DESIGN POINT SPECIFICATIONS

PASSENGER COMPARTMENT

Air Exchange > 5 cfm/person		
Parameter	During Heating Season	During Cooling Season
T_a, dry-bulb temp.	>68°F	≤75°F
T_w, wet-bulb temp.	-	<75°F
Air velocity at the passenger	<0.5 meter/sec	<1.5 meter/sec
T_{mr}	Limit Not Specified	

Table 2 shows the parameters and their values for "design ambient conditions" for summer and winter. These values were determined by integrating the weather data over the U.S.A. approximately weighted by the car population density distribution. The details of the calculation procedure are presented in Appendix A.

2.3 Duration of Environmental Control

Vehicles must carry all the energy required on board. As the energy density of various energy storing devices is rather small, it is important to minimize the ECS on-board energy requirement so as not to impair mission profile. Batteries of many electric vehicles last only about two to three hours of driving before being totally discharged. Thus, it is clearly not necessary to carry energy for environmental control for a considerably longer period. In order to determine a more probable duration of environmental control, including the effect of events such as stopping at traffic lights, etc., typical travel scenarios were constructed for going to work and back, shopping and other chores, etc. These scenarios were based on a recent study that derived average U.S. driving patterns by surveying a large number of families. The results of these travel scenarios and other contractual requirements are shown in Table 3. The design value for the required duration of environmental control is taken to be 2.5 hours per vehicle charge/discharge cycle for this study. Table 3 shows that a more moderate value of 0.8 hours is adequate for most people's needs. The details of constructing travel scenarios are given in Appendix B.

2.4 Design Load Specifications

Table 4 shows the load specifications as determined from the above exercise.

TABLE 2

DESIGN POINT SPECIFICATIONS

AMBIENT CONDITIONS

Conditions	For Heating Season	For Cooling Season
T_a (Dry Bulb Temp.)	-10°F	100°F
T_w (Wet Bulb Temp.)	-	74°F
v (Wind Velocity)	45 mph	45 mph
Solar Insolation	-	326 Btu/ hr-ft^2

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TABLE 3

ACTUAL DRIVING TIMES FOR VARIOUS TRAVEL SCENARIOS

Scenario	Driving Time (hours)
SAE J227a, D Cycle Repeated 65 Times	1.75
Work-Related Travel	0.734
Shopping and Other Non-work-related Travel	0.584

TABLE 4

Design Load Specification

- Continuous 17,000 Btu/hr ($\approx$ 5 kW)
- 2.5 Hours of Operation Maximum (Passenger Compartment)
- 42,500 Btu (Maximum)
- 10 Hours of Recharging
- 10 Hours of Unplugged Operation for Battery Temperature Controller

3.0 IDENTIFICATION OF ECS ELEMENTS

3.1 Introduction

Environmental control of the passenger compartment of electric vehicles requires both heating and cooling subsystems. Because an electric vehicle is designed to significantly reduce the use of petroleum-based fuel, the maintenance of the thermal environment within the passenger compartment should use little or no fuel of this type. With this goal as a focus, a variety of heating and air conditioning elements, which use a variety of energy sources, are identified. In most cases, it is assumed that the energy carriers will be recharged at home during the period in which the propulsion batteries are being charged. Figure 3 schematically summarizes the various components in the ECS of an electric vehicle.

3.2 Energy Storage

Environmental control of the electric vehicle requires an adequate amount of energy on board the vehicle in some suitable form. Utilization of this energy at an appropriate rate then provides the desired heating and cooling.

Fundamentally, heat can be added or removed in two ways. The first method is addition/removal of heat in the form of heat itself. In this case, the amount of energy required to be stored is equal to or greater than the quantity of heat to be supplied/removed. In the second method, total added/removed heat is partly in the form of heat and partly in the form of work. Since the atmosphere may be used as a source/sink of heat, only the work component needs to be stored. Because most of the energy is obtained from the atmosphere, the work component represents only a fraction of the total energy requirement. However, unless the work can be stored in the form of potential energy, significant losses are bound to occur in converting any other form of energy, such as chemical or heat, to work. Thus, the advantage of a reduced energy storage requirement will be attenuated to a certain extent.

Energy can be stored in the form of heat (at high temperature for heating and at low temperature for cooling) or some other form such as chemical or

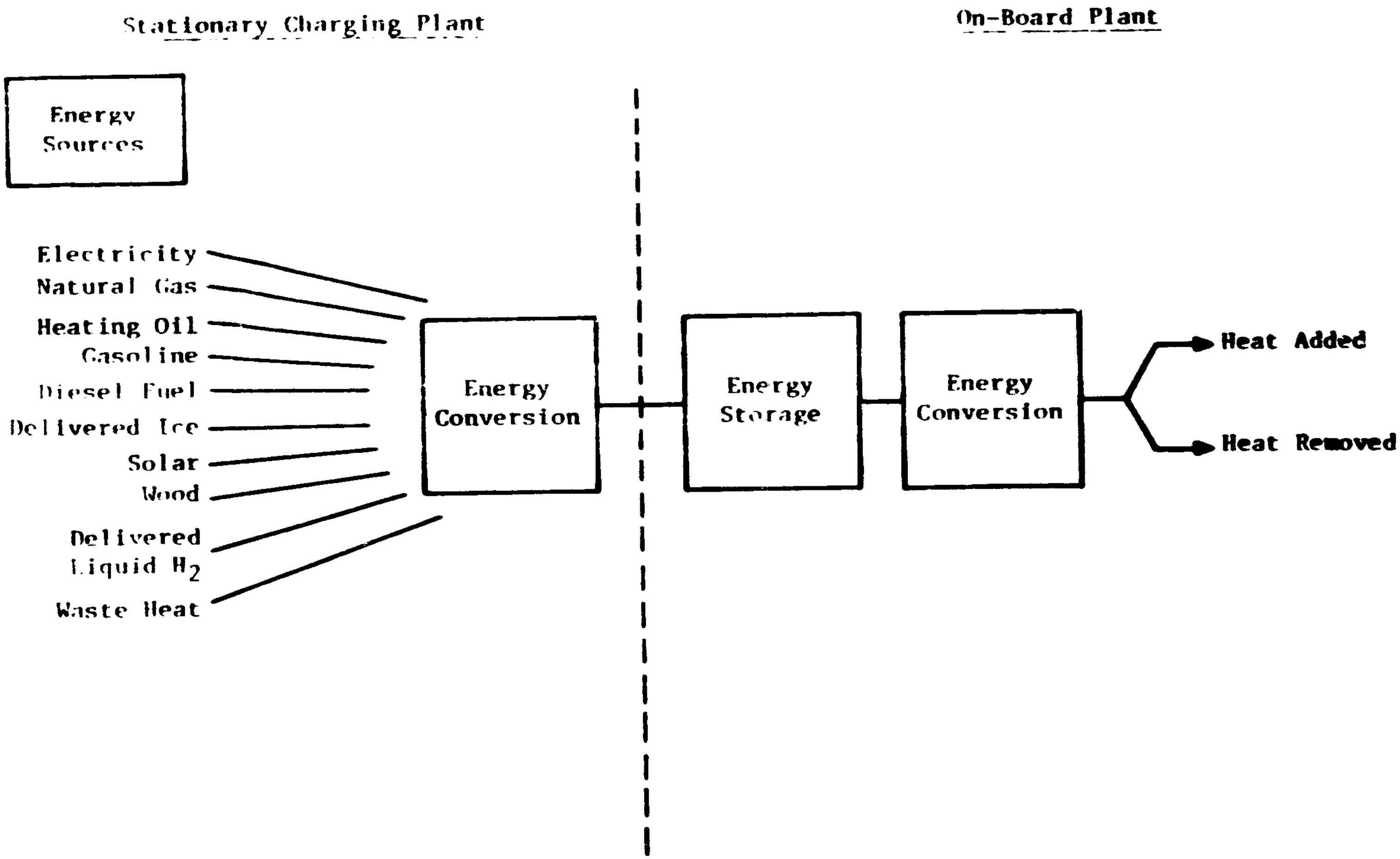

Figure 3. Schematic of Environmental Control Subsystem for Electric and Hybrid Vehicles

potential. When energy is stored directly in the form of heat, no energy conversion is required, and no conversion-associated losses occur. However, a certain amount of heat energy is bound to escape during the storage mode. The extent of the loss depends on the temperature differential between the storage and ambient temperatures, and on the level of insulation. When energy is stored in some other form, it must be converted to either heat and/or work. This conversion requires equipment and results in an energy loss that can be utilized only in the heating mode to a small degree. In the storage mode, however, energy losses are insignificant.

The length of storage time is limited by the form of energy storage used. Thus, for energy stored in the form of heat, storage time is on the order of only a few hours, unless elaborate insulation techniques are utilized. By using electric batteries, storage time can be extended to a few days. With gasoline, the storage time is unlimited.

3.3 Energy Source

The source of energy available presents another dimension to the selection of the appropriate scheme for the electric vehicle ECS. Energy could be obtained from private residences in the form of electricity, natural gas, heating oil or waste heat, and from service stations in the form of gasoline, propane, etc. Delivery of liquid hydrogen, nitrogen, oxygen or air from a suitable source is also conceivable. Furthermore, ice, dry ice (solid form of CO_2) or liquid ammonia could constitute suitable sources of energy.

3.4 Categories of ECS Elements

The various ECS elements considered can be divided into three general categories:

- Heat Pumps (2.0)*
- Thermal Storage (3.0)*
- Reversible Thermo-Chemical Reactions (3.0)*

*The numbers in parentheses show the section numbers of Appendix C in which these categories are discussed in detail.

3.4.1 Heat Pumps

Various types of heat pumps were considered in this study as follows:

- **Vapor-compression heat pumps driven by a gasoline engine or electric motor (2.1)***
- **Thermal engine heat pumps (2.2)***
- **Absorption-cycle heat pumps (2.3)***
- **Thermoelectric heat pumps (2.4)***
- Magnetic heat pumps (2.5)*
- Split heat pumps (2.6)*

In the first five systems, the complete heat pump hardware is on board the vehicle. The energy required to run the heat pump is also stored on board in electric batteries or petroleum-based fuel.

The last system, the split heat pump, is a novel design in which the refrigerant loop is split into two parts. An adequate quantity of refrigerant in state 1 is stored on the vehicle. When heat pumping is performed, this refrigerant passes to state 2 and is also stored on the vehicle.

In a conventional heat pump, the equipment that converts the refrigerant from state 2 to state 1 is part of the heat pump itself. In the split heat pump system, this equipment is not carried on board the vehicle but is kept, for instance, in a garage. Thus, when the vehicle's propulsion batteries are being recharged, the used refrigerant in state 2 is delivered to the other stationary half of the heat pump cycle equipment for reconditioning to state 1. Then, the refrigerant in state 1 is again stored on the vehicle.

3.4.2 Thermal Storage

Two types of thermal storage were considered: high-temperature thermal storage and low-temperature thermal storage. High-temperature thermal

*The numbers in parentheses show the section numbers of Appendix C in which these categories are discussed in detail.

storage systems act as heat sources from which heat can be extracted for heating the vehicle. Low-temperature thermal storage systems can be used as heat sinks to which heat can be rejected from the hot environment for cooling. Heat sources and sinks can be classified as either sensible heat or latent heat (of phase change), depending upon the mechanism employed for rejecting or absorbing heat.

If energy is added or removed in thermal form for the purpose of environmental control, the energy can be low grade in the sense that it does not have to be capable of delivering shaft work. Thus, from a storage point of view, only two parameters are of great significance, viz., gravimetric energy density and volumetric energy density. Table 5 shows the values of these two parameters for a variety of electric vehicle batteries. Many thermal storage schemes are identified that exceed the energy densities of batteries by orders of magnitude. Even relatively simple and readily available materials result in schemes that surpass energy densities of the batteries.

Thermal storage schemes were identified using:

- Sensible heat storage
- Latent heat of phase change.

Realization of practical schemes utilizing high energy densities depends on the temperature of storage, heat transfer problems and insulation problems. These systems are presented in further detail in Section 3.0 of Appendix C.

3.4.3 Reversible Thermo-Chemical Reactions

Certain chemical reactions have a property that when the reaction proceeds in one direction, say from state A to state B, heat is liberated (exothermic reaction) and the components of state B can be made to react with one another in a different set of conditions, resulting in products of state A. In many cases, the reaction from B to A requires heat to be supplied to the reactants. The materials for these chemical reactions, therefore, can be used as energy storage devices. Further, as the products in state B of reaction from state A to state B are not discharged out of the system, the same materials can be used over and over again many times.

TABLE 5

ENERGY DENSITIES FOR VARIOUS BATTERIES

Storage Type	Gravimetric Energy Density (Btu/lb)	Volumetric Energy Density (Btu/ft^3)
Lead-Acid Batteries		
Present	54	5,800
Advanced	77	8,700
Ni-Zn Batteries		
Present	108	11,100
Advanced	130	14,500

In the field of solar energy research, many reactions have been identified that have energy densities of thousands of Btu/lb and, thus, may provide a basis for compact electric vehicle ECS designs. However, this field of study has received attention only recently and information presently available in published literature is inadequate to make engineering decisions. Section 4.0 of Appendix C deals with this topic in further detail.

4.0 PROCESS OF SELECTION OF ECS

4.1 Elimination of Inappropriate ECS Elements

Many elements for providing heating and/or cooling are identified and described in Appendix C. Many of these have been eliminated from further consideration as they are judged not to have potential for development to a practical scheme. The reasons for eliminating any particular element are some combinations of the following:

1) Too expensive (above $1000)

2) Too heavy (above 500 lb)

3) Too bulky (over 10 cubic ft)

4) Too complex

5) Unacceptably high safety hazard

6) Not enough information available to enable determination of various trade-offs.

In case (6), significantly more work will be required to obtain the necessary information. Such work is outside the scope of this contract.

The elimination process is shown in Table 6. After elimination, 14 elements appeared to have sufficient merit to warrant their inclusion in the ranking scheme.

4.2 Ranking

The 14 selected elements were then ranked in three categories:

1) Applicable to heating only
2) Applicable to cooling only
3) Applicable to both heating and cooling.

The ranking was based on a scheme in which each system was given a score. This score was computed by a weighted sum of marks given to the system for various criteria. The weighting factors represent the relative importance of various criteria with respect to the overall selection process. The

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TABLE 6

ELIMINATION PROCESS

Scheme	Pg. No. In MTI 80TR53	Available for Heating	Available for Cooling	Remark*	Reasons (Number Code From Text)
Electric-Motor-Driven Heat Pump	2-21	x	x	E	1,2
Gasoline-Engine-Driven Heat Pump	2-20	x	x	R	
MTI Heat-Activated Heat Pump	2-23	x	x	(LTD) R	
Absorption Cycle Heat Pump	2-26	x	x	E	2,3
Thermoelectric Heat Pump	2-37	x	x	E	1,2 - battery
Localized Thermoelectric	2-41	x	x	(LTD) R	
Thermoelectric/Storage Combination	2-42	x	x	E	1
Magnetic Heat Pump	2-42	x	x	E	1,2
Water/LiBr	2-43		x	R	
Ammonia Water	2-50	x	x	R	
Water	3-1	x	x	R	
LiOH	3-7	x		R	
NaOH	3-7	x		R	
LiF	3-7	x		R	
$NaOH-NaNO_3$	3-13	x		R	
$K_2CO_3-Na_2CO_3-Li_2Co_3$	3-20,21	x		R	
Li_2CO_3	3-21	x		R	
Sodium Sulfate Decahydrate (Glauber's Salt)	3-21	x		E	2
Polyethylene Pellets	3-23	x		E	2
Other Phase-change Salts, Table 3-9	3-25	x		E	2
Liquified Gases	3-26		x	E	1,4,5
Paraffins	3-28	x		E	3,2
Organic Oils, Table 3-12	3-30	x		R	
Sand	3-33	x		E	2,1
Compressed Air	3-37	x	x	E	3
Ammoniated Salts	4-1	x		E	6
Reversible Thermochemical Reactions, Table 4-2	4-4	x		E	3
Reversible Thermochemical Reactions, Table 4-3	4-5	x		E	3,6
Ni_5LaH_x	4-6	x	x	E	6
$FeTiH_x$	4-6	x	x	E	2
MgH_x	4-10	x	x	(LTD) R	

*E=Eliminated; R=Retained for Further Consideration; (LTD)R=Retained Due to Long-Term Development Potential

Conditions

T_{amb} = -10°F   Heat Rate = 17,000 Btu/hr

Vehicle Space Temp. = 75°F   Total Quantity of Heat Added or Removed = 42,500 Btu marks given to a system for any given criterion represent the degree of "goodness" of that system for that particular criterion in comparison with the system used as a baseline reference system. The baseline reference system is taken as a gasoline engine driven heat pump.

The criteria and the associated weighting factors (shown in Table 7) were derived from a consensus of opinion of MTI engineers associated with this project. All criteria and weighting factors were approved by JPL. The reasoning for these factors is discussed in detail in Appendix D.

The marks given for any system for various criteria were arrived at by the following process. First, a schematic of the complete system was prepared which consists of all the components required for providing environmental control starting from energy sources available in the home, to providing hot or cold air within the passenger compartment. Estimates were then made of the cost, parasitic power required and the weight of each of the components.

Using these numbers, marks were computed for various criteria. For some criteria such as noise, marks were assigned only from judgement, while for other criteria such as range impact, a definite algorithm was available to compute the marks starting from the information about the ECS weight.

In case of storage period criterion, three systems inherently have a long period of the order of months, while for all the thermal storage schemes, the period of storage is dependent on the extent of the insulation provided on the storage device. Thus, for thermal storage schemes, the system cost and size is determined by assuming a storage period of 10 hours of out-of-home operation.

Table 8 summarizes the manner in which marks for various criteria were obtained. The detailed information is available in Appendix E.

TABLE 7

CRITERIA AND ASSOCIATED WEIGHTING FACTORS

Criterion	Weighting Factor (W_i)
Capital Cost Characteristics:	
1. First Cost	25
2. System Life	5
Use Characteristics:	
3. Range Impact	10
4. Energy Efficiency	5
5. Storage Period	5
6. Maintainance Cost	10
7. Performance Impact	10
Environmental and Safety Characteristics:	
8. Consumer Perception of Safety	5
9. System Noise	10
10. Other Environmental Impacts	5
Development and Manufacturing Characteristics:	
11. Ease of Packaging and Volume	5
12. Development Cycle Through Commercialization	5
TOTAL	100

TABLE 8

SUMMARY OF SYSTEM CRITERIA RANKING PROCESS

Criterion	Degree of Uncertainty of the Marks For Each System
First Cost	1
System Life	2
Range Impact	1
Energy Efficiency	1
Storage Period	1
Maintenance Cost	2
Performance Impact	1
Consumer-Perceived Risk	3
Noise	1.5
Environmental Impact	3
Packaging and Volume	2.5
Development Status	2.0

0 Calculated from precise system design.

1 Calculated from engineering data (engineering data are subject to variability due to personal preferences).

2 Engineering judgment based on engineering characteristics.

3 Judgment based on perceived socio-economic view of national goals.

Subjectivity is built into the entire ranking process. This comes at three different levels:

1) Selecting the criteria
2) Assigning weighting factors to the criteria
3) Assigning marks for any system for a given criterion.

Of these items, 3) is least prone to subjectivity as it enters mainly into the choice of a specific ranking algorithm from the engineering data. The very nature and scope of this study makes it impractical to perform detailed engineering calculations. Thus, a certain amount of judgement is involved in arriving at the engineering data such as cost, weight, etc. It is thus possible to arrive at very different sets of conclusions by persons with different perceptions as to the overall national or other socio-economic objectives.

4.3 Results of Ranking and Recommendations

The results of the ranking exercise are shown in Table 9. Based on these results, the following recommendations are made.

For heating only:

Thermal storage with water.

For cooling only:

Li-Br-water split heat pump

For both heating and cooling:

1) Near-term development: Gasoline engine-driven heat pump
2) Intermediate-term development: Thermal storage with water
3) Long-range development: Aqua-Ammonia split heat pump.

TABLE 9

RESULTS OF RANKING SYSTEM

System Type	System	Score
Both Heating and Cooling	Thermal Storage with Water	240
	Aqua-Ammonia Split Heat Pump System	195
	MTI Heat-Activated Heat Pump	136
	Gasoline-Engine-Driven Heat Pump	100
Heating Only	Thermal Storage with Water	377
	Thermal Storage with K_2CO_3-Na_2CO_3-Li_2CO_3	208
	Thermal Storage with Li_2CO_3	208
	Thermal Storage with Organic Oil	198
	Thermal Storage with LiOH	195
Cooling Only	LiBr-Water Split Heat Pump	213

5.0 DESCRIPTION OF RECOMMENDED SYSTEMS

5.1 Ammonia-Water Split Heat Pump

Basically this is a conventional absorption-type heat pump. The major modification is that the total absorption cycle is split into two subsystems. One of the subsystems of the cycle is located on board the vehicle and the rest of the components to complete the cycle are located at the home base. Thus different processes in the complete cycle are performed in different locations and at different times. Specifically, the refrigerant expansion and absorption processes are carried out on board the vehicle, while the regeneration of ammonia from the weak solution is carried out in the home base equipment. The process is shown schematically in Figure 4. Figure 5a shows a schematic arrangement of the system in the heating mode. Moving the baffles to another location makes it possible to use the system for cooling. Figure 5b shows the system in the cooling mode.

As the system is split in two subsystems, the rates of thermodynamic processes in the two subsystems can be independent of each other. It is thus possible to design the home base equipment to perform the regeneration function over a 24-hour period, while the maximum time of operation of the vehicle-based equipment is 2-1/2 hours. This results in a considerable reduction in the size of home base equipment. Moreover, splitting enables a minimization of the equipment and thus the weight carried on board the vehicle.

Preliminary calculations provide the following information about the system:

- On-board-the-vehicle equipment:

- Weight	316 lb
- Volume	6.8 cubic feet
- Heat transfer rate	17,000 Btu/hr
- Total quantity of heat transfer	42,500 Btu
- Cost	$365

Split Heat Pump System

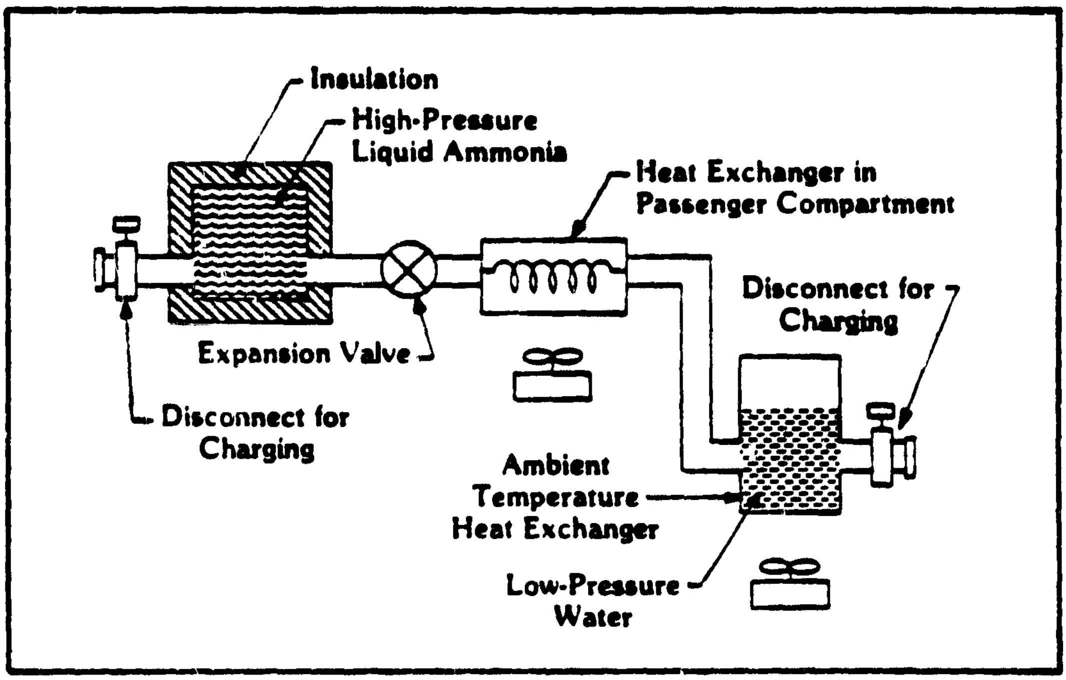

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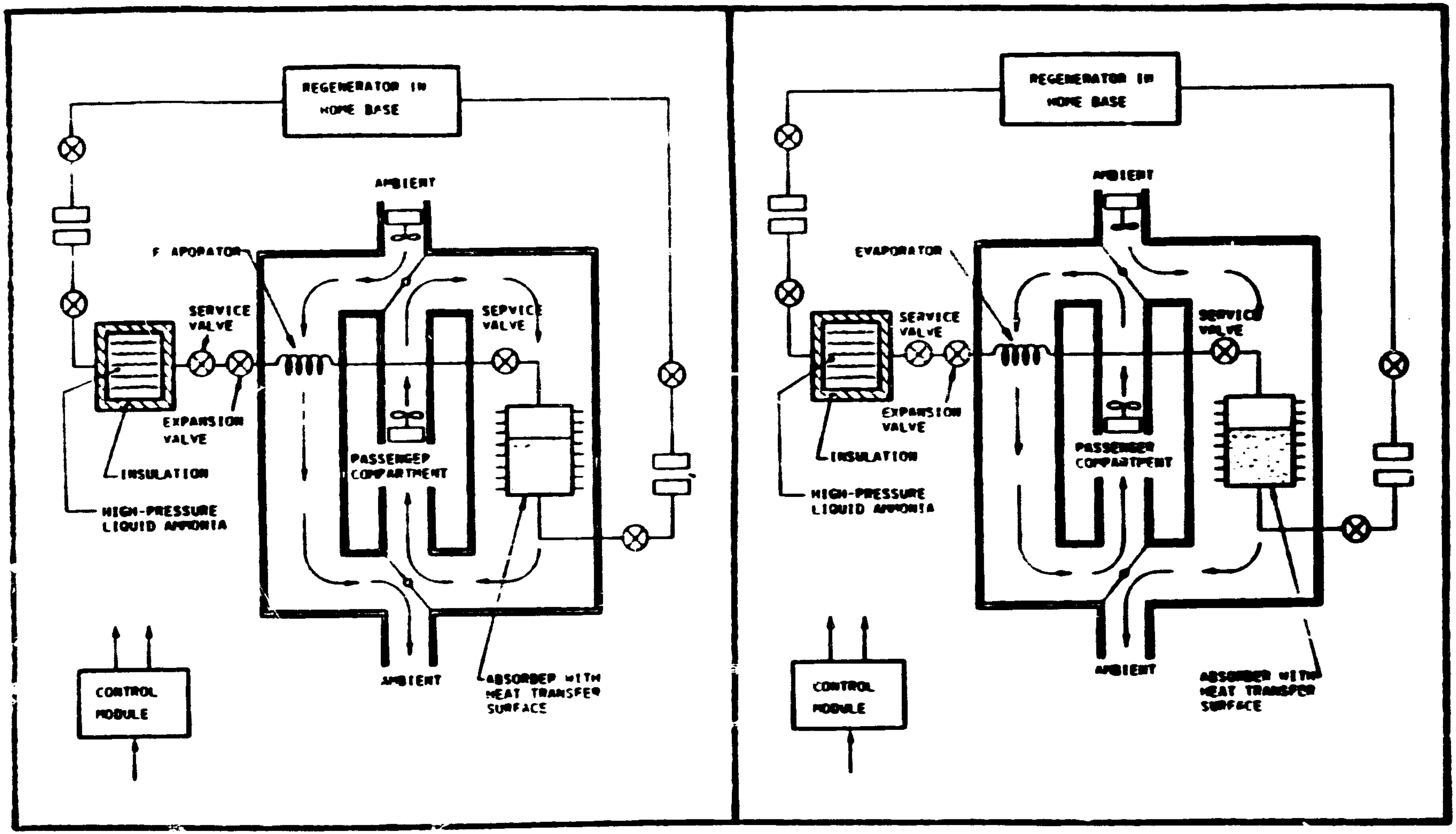
Figure 4.

Split Heat Pump System



Heating Mode

5a

Cooling Mode

5b

01352-1

Figure 5.

- **Home-base equipment:**
  - **Maximum heat transfer rate** **700 Btu/hr**
  - **Cost** **$361**

The salient points about the system are:

Benefits

- No moving parts on board the vehicle
- No on-board use of petroleum fuel
- Long storage periods of similar order of the system with gasoline engine
- Very little noise
- Low overall weight on board the vehicle
- Applicable to both heating and cooling.

Drawbacks

- Longer development period
- Perceived safety aspect.

5.2 Thermal Storage With Water

In this system, thermal energy is stored in water. For the heating season, water is heated to 250°F in a pressurized container resulting in a storage density of 150 Btu/lb of heat. This assumes that useful heat can be extracted until the temperature of water falls to 100°F to maintain the passenger compartment temperature of 68°F in winter. For the cooling season, water acts as a low-temperature heat sink. An energy storage density of 150 Btu/lb is obtained by letting the water freeze. This assumes that heat can be rejected to the low-temperature heat sink until the water temperature rises to 40°F to maintain the passenger compartment temperature at 75°F.

In order to take care of expansion during freezing and also heat transfer from passenger compartment to ice during frozen state, pure water is enclosed in many elastomer balls. These balls are then kept in a tank containing water with antifreeze like ethelyne glycol. The water with antifreeze is used as a heat transfer fluid and is circulated through the fluid to air heat exchangers located in the passenger compartment. Appropriate containers are adequately insulated to limit heat loss to 5% of the total stored energy over a period of 10 hours.

The thermal storage is charged overnight by circulating the water with antifreeze from the vehicle tank, through a reconditioning plant located in the home base. Such a reconditioning plant consists of a small freezer, refrigerator and a tank with an immersion heater. A schematic of the system is shown in Figure 6.

Preliminary calculations result in the following information:

- On-board-the-vehicle equipment:

- Weight (including water)	333 lb
- Volume	5 cubic ft
- Parasitic power	365 W
- Heat transfer rate	17,000 Btu/hr
- Maximum heat transferred	42,500 Btu
- Cost	$300

- Garage Equipment

- Heat transfer rate (24-hr basis)	700 Btu/hr
- Cost	$350

The pros and cons of this system are as follows:

Pros

- No on-board use of petroleum fuel
- Very little noise
- Applicable to both heating and cooling
- Simplicity and similarity with present automobile heating systems
- Short development period.

Thermal Storage System

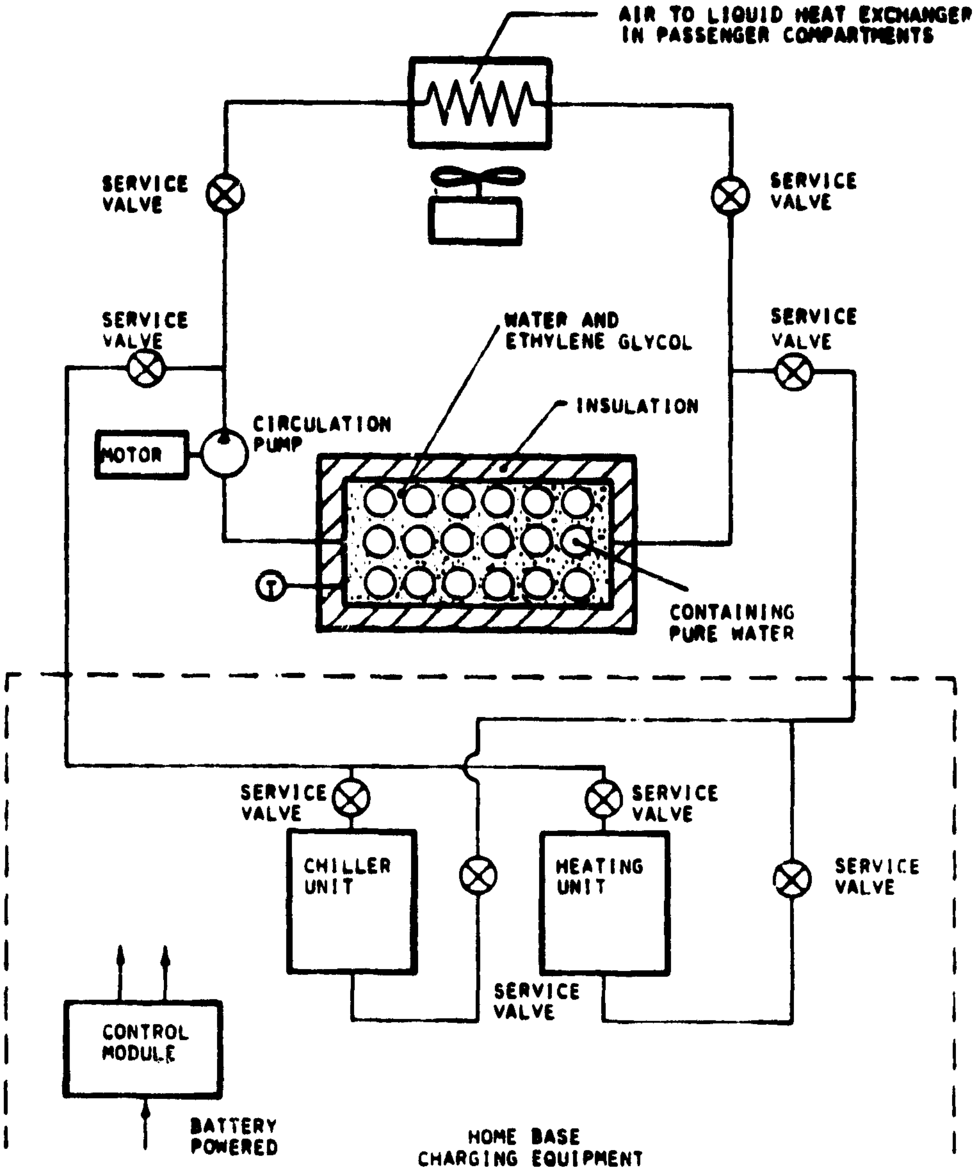


Figure 6.

81350

Cons

- Limited storage period
- Capable of providing heating or cooling only on a given daily charge.

5.3 Gasoline-Engine-Driven (Vapor Compression) Heat Pump

For the near-term application, this system has a potential for the least development period and cost. It is used as a reference system for comparative evaluation of various other schemes.

Figure 7 shows a schematic layout of the system. In winter, much of the energy content of the exhaust gases is made available to the passenger compartment by heating the evaporator with the exhaust gases. A simple movement of the damper is used to change from heating mode to cooling mode.

Preliminary calculations show the following information about the system:

- Weight 175 lb
- Cost $750

Benefits and disadvantages of this system are as follows:

Benefits

- Only state-of-the-art product development required
- No daily charging necessary
- Capable of providing either heating or cooling without any preplanning.

Disadvantages

- Use of on-board petroleum fuel
- Noise
- Hazardous fumes
- Maintenance.

Engine-Driven Heat Pump

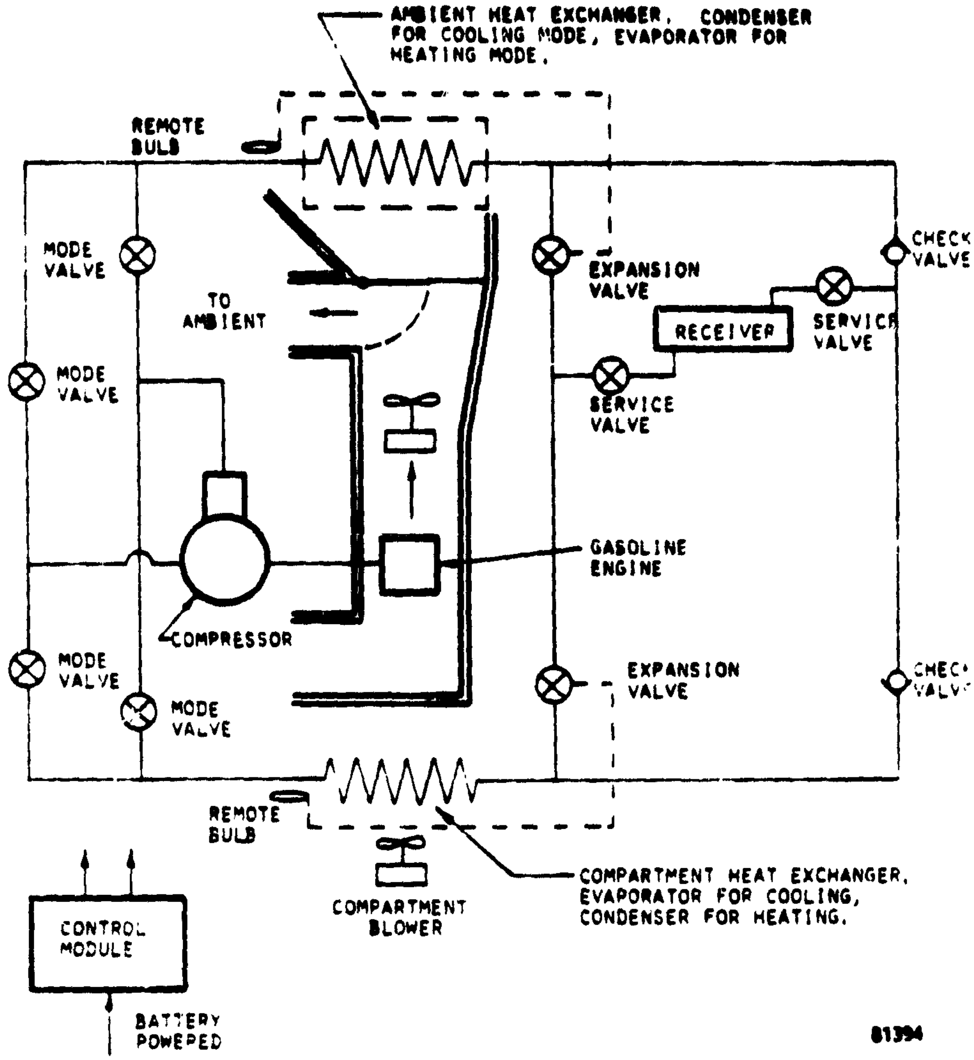


Figure 7.

MTI 80TR45

ELECTRIC AND HYBRID VEHICLES
ENVIRONMENTAL CONTROL SUBSYSTEM STUDY

FUNCTIONAL REQUIREMENTS SPECIFICATIONS

Prepared for:

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TABLE OF CONTENTS

ABSTRACT

The specification for design point conditions for designing Environmental Control System (ECS) elements has been derived to minimize energy consumption without sacrificing comfort.

The rationale for deriving values of various parameters defining conditions in the passenger compartment is based on the use of mathematical modelling of physical, physiological and psychological processes involved in the determination of comfort.

The rationale for deriving design point specification for ambient conditions is based on a suitable modification for vehicular application of the logic used in the ECS design for buildings.

INTRODUCTION

Since the energy stored on board electric vehicles is small, the energy demand for any purpose other than propulsion must be as low as possible, lest the vehicle range, which is already small, become even smaller. In the design of ECS equipment for current automobiles, the range penalty does not become a consideration due to the large range between refuelling stops and to the quick refuelling period of a few minutes as compared to a few hours for electric vehicles. Thus, the duplication of the practice of ECS equipment design used in current automobiles is not prudent for application to electric vehicles. (See Tables 1 and 2.) Therefore, Mechanical Technology Incorporated (MTI) studied various means of reducing the heating and cooling loads. The study focussed on the objectives of providing ECS equipment for electric vehicles. In order of priority, these objectives are:

- Safety
- Driver operation of the vehicle with maximum efficiency for accident avoidance
- Thermal comfort to occupants.

TABLE 1

MTI SURVEY RESULTS OF CURRENT ECS DESIGN PRACTICE FOR AUTOMOBILES*

• **Ambient Conditions (a/c)**	
Dry-Bulb Temp.	100° ↔ 110°F
Wet-Bulb Temp.	70° ↔ 75°F
Relative Humidity (RH)	30 ↔ 40%
• **Inside Conditions (a/c)**	
Steady State	
Dry-Bulb Temp.	75°F
RH	Not specified
Air Rate (fresh air and recirculated air)	100 ↔ 200 cfm
Air Exchange Rate (fresh air only)	10 ↔ 200 cfm
Cooldown	
Initial Temp.	140° ↔ 145°F
20 minutes	76° ↔ 86°F
• Capacity (a/c)	22,000 ↔ 26,000 Btu/hr (6.45 ↔ 7.61 kW)
• C.O.P. (a/c)	1.8
• Fuel Rate (a/c)	1 gal/hr
• Weight (a/c)	90 ↔ 100 lb
• Heating	
Ambient Conditions	
Dry-Bulb Temp.	-10° ↔ 2°F
Inside Condition	75°F
Capacity	8,000 ↔ 25,000 Btu/hr
(Steady State)	(2.35 ↔ 7.31 kW)
(Quick Heat)	up to 50,000 Btu/hr (14.6 kW)
Time to Warm Up	
40°F	10 minutes
75°F	25 minutes

*No mandatory standards exist for ECS design for present automobiles except for windshield defogging, defrosting and glazing transmission.

TABLE 2

IMPLICATIONS OF INFORMATION* ON ELECTRIC VEHICLE APPLICATION

At Design Points

- For a/c, about 2 batteries (Ni-Zn) of 60 lb each required per hour of operation
- For heating, about 1.8 batteries (Ni-Zn) of 60 lb each required per hour of operation

  Supporting calculations needed.

*Taken from Table 1

ENERGY CONSUMPTION FOR THERMAL COMFORT TRADE-OFF STUDY

Relatively little information is available in the published literature about characterising thermal comfort cause-effect relationship in the case of automobiles. Only two papers [1,2]* have been found which address this topic. However, extensive research has been performed in this field in the case of buildings, and substantial information is available in the published literature.

Some work has been recently done in the thermal comfort field to consider modifications in the space conditioning practice to reduce energy consumption. The research work examines, among other factors, human physiological responses for various activity levels, various clothing insulation levels, and effects of seasonal acclimatization. The research utilizes mathematical models of physical and physiological processes, and an extensive experimental data base of a statistical nature on the actual subjective feeling of thermal confort of a large number of individuals under various environmental conditions.

As energy conservation in space conditioning is of utmost importance in the case of electric vehicles, this work was examined to provide a rational basis for determining functional requirement specifications for ECS for the passenger compartment. These specifications would be consistent with the objective of providing thermal comfort with a minimum of energy consumption. A review of this study is presented in brief in the following section.

NUMERICAL EVALUATION OF THE STATE-OF-COMFORT FROM PHYSIOLOGICAL AND PHYSICAL PRINCIPLES

Thermal discomfort is expressed by descriptive words such as hot or cold and is associated with an index which historically was considered to be synonymous with dry-bulb temperature of surrounding air. As more understanding was gained, the importance of wet-bulb temperature, air movement and radiation was gradually recognized.

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*Numbers in brackets indicate references which can be found at the end of this report.

With every additional factor, attempts were made to maintain simplicity in the concept of "temperature" as an indicator of thermal comfort or the lack of it. Various indices or "effective temperatures" were defined, which combined the effects of the four environmental factors (dry-bulb temperature, wet-bulb temperature, air velocity and radiation) which affect the degree of thermal comfort. The definitions of some of these indices are given in the ASHRAE Handbook of 1977 Fundamentals, Chapter 8, pp. 8.16-8.18.

The subject of thermal comfort recently has been studied from a rational approach. In such an approach, a physically measurable definition of comfort is needed. Obtaining this definition is not a simple task due to the fact that the feeling of comfort is an integrated effect of a variety of social, psychological and physiological perceptions. The proposed ASHRAE standard 55-74 R for "Thermal Environmental Conditions for Human Occupancy" defines "thermal comfort" as that condition of mind which expresses satisfaction with the thermal environment.

Experimental data on a vast number of individuals indicate that the feeling of thermal comfort has a strong correlation with certain quantifiable and measurable physiological state variables (responses).

A partial list of these variables and their values during the state-of-comfort for most individuals is given in Table 3. This information has been adopted from Goldman [3]. Any departure in the state space from these value points represents a degree of discomfort. The extent of discomfort can be assigned a numerical value by the distance (a suitably defined metric) of the point from the point of comfort.

Numerical evaluation of the degree of comfort then reduces to first determining actual values of these state variables under any given conditions, and then computing the above-referred discomfort metric. The resulting numerical value will determine the extent of the "feeling of comfort." Determination of the values of state variables can be accomplished by mathematical modelling of the heat transfer processes between the body and the environment, and of the physiological processes involved in the thermoregulatory mechanisms of the body.

TABLE 3*

STATE-OF-COMFORT VALUES FOR MOST INDIVIDUALS

PARAMETERS	COMFORT
Mean Weighted Skin Temp. ($\bar{\bar{T}}_s$)	33.3 °C
% Wet Skin	≈ 20%
T_{finger}	≈ 20°C
T_{toe}	≈ 18.5°C
Deep Body Temp. (T_{re})	37 ± 0.5 °C
Change of Body Heat Content (ΔS)	0 Kcal
% H_2O LOSS	0%
WORK	100 kcal/hr
Heart Rate (HR)	60 - 80/min

where:

T_{finger} = temperature of the tips of the fingers

T_{toe} = temperature of the tips of the toes

---

*Adapted from Goldman [3].

MATHEMATICAL MODEL FOR EVALUATION OF PHYSIOLOGICAL RESPONSES

One mathematical model used by Azer et al. [4] is in the form of a differential equation, and hence can be used to determine the extent of comfort during transience. The essentials of this model are:

- **The body produces heat by the combustion of food. This heat rate is called metabolic rate, and is a function of the level of activity and other physiological factors.**
- The heat produced must be dissipated to the ambient through heat transfer processes to maintain the body temperature and the values of certain other physiological state variables within narrow limits.

The rates of heat transfer are governed by:

- Environmental conditions giving rise to different rates of convection, conduction, radiation and evaporative heat transfer
- Clothing insulation
- Physiological responses resulting in modification of skin conductance by blood flow regulation and/or rates of secretions by the sweat glands.

The environmental conditions affecting heat transfer processes can be completely characterized by:

T_a - dry-bulb temperature of air

T_w - wet-bulb temperature of air

T_{mr} - mean radiant temperature of the surroundings defined in the ASHRAE Handbook of 1977 Fundamentals

v - air velocity.

The effect of clothing on the heat transfer processes is rather complex. However, this effect is assumed to be represented by a single factor, I_{clo}, which has dimensions of thermal resistance.

The flow diagram of Figure 1 shows the schematic of the logic in the development of the model. This model shows that the physiological responses are the dependent variables, and are functions of six independent variables (T_a, T_w, T_{mr}, v, I_{clo}, and m). The variable m is the rate of heat generation within the body due to metabolic activity. For transient conditions, time, t, is one more independent variable. The following equation can then be **written:**

$$C_m = C_m(\vec{P}_n) \tag{1}$$

$$\vec{P}_n = \vec{P}_n(T_a, T_w, T_{mr}, v, I_{clo}, m, t) \tag{2}$$

where

$\vec{P}_n$ = vector of physiological response state variables

C_m = metric of comfort.

Combining the above two equations, the following can be written:

$$C_m = C_m'(T_a, T_w, T_{mr}, v, I_{clo}, m, t). \tag{3}$$

EQUATION FOR COMFORT AND ITS TYPICAL SOLUTIONS

For condition of optimal comfort,

$$C_m = 0.$$

Hence,

$$0 = C_m'(T_a, T_w, T_{mr}, v, I_{clo}, m, t) \tag{4}$$

represents an equation of comfort.

In steady state,

$$0 = C_m''(T_a, T_w, T_{mr}, v, I_{clo}, m). \tag{5}$$

This equation represents an equation for a surface in the six-dimensional space of the six independent variables. A similar equation used by Fanger [5] is known as the Fanger Comfort Equation. In the electric vehicle application, the passengers can be considered as sedentary, and hence,

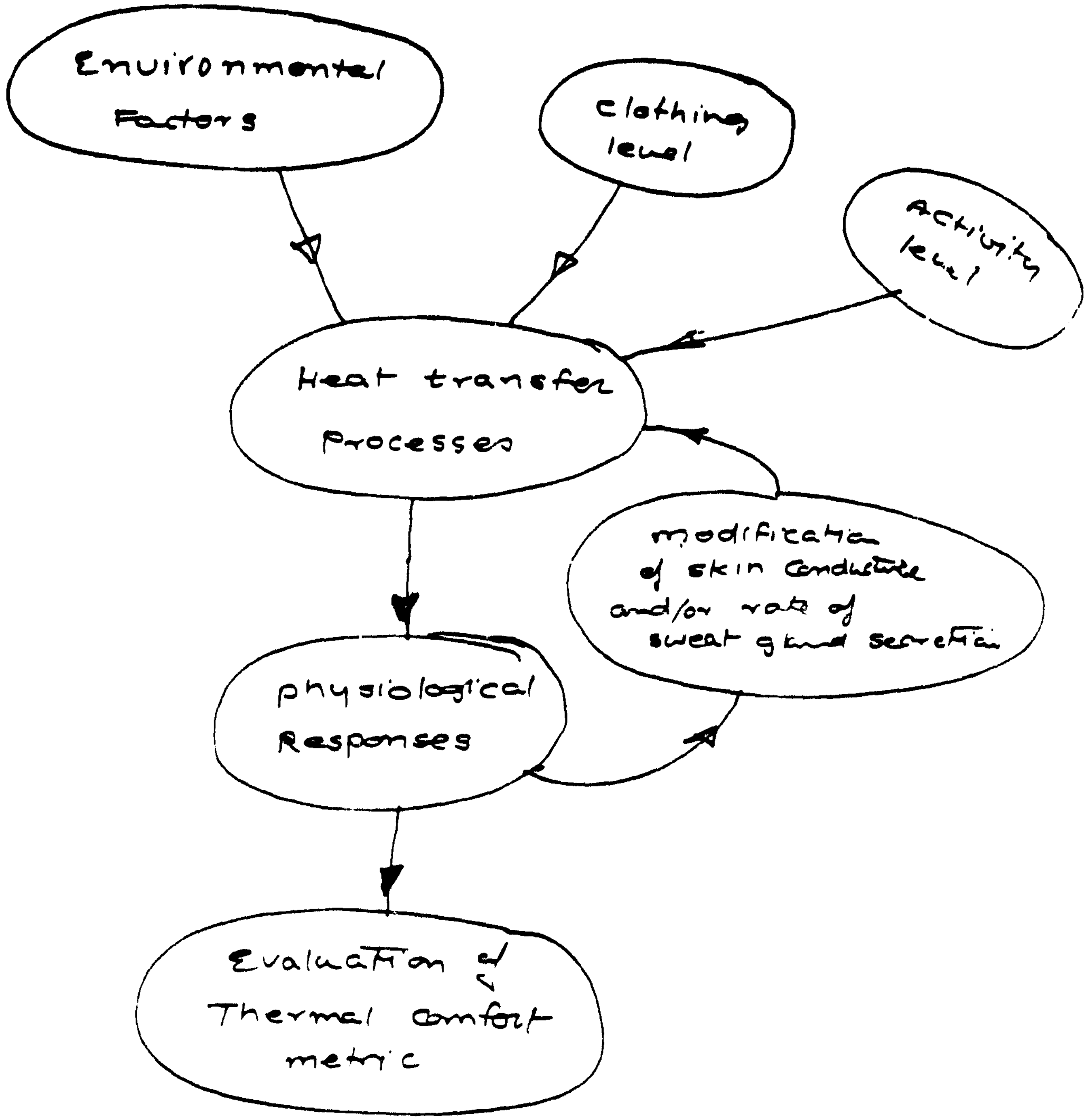

Fig. 1 Schematic of the Model for Numerical Evaluation of Extent of Thermal Comfort

$m = 58.2$ watts/meter^2.

Thus, Equation (5) is modified to

$$0 = C_m'''(T_a, T_w, T_{mr}, v, I_{clo}) \qquad (6)$$

with five independent variables.

The solution of Equation (6) can be graphically represented as lines of comfort shown in Figure 2. Here, the procedure for plotting the line of comfort is as follows.

Equation (6) is inverted to obtain:

$$T_w = T_w(T_a, T_{mr}, v, I_{clo}). \qquad (7)$$

Further constraints are introduced:

$$T_a = T_{mr}$$

and

$$I_{clo} = 0.5 \text{ clo}.$$

Thus, Equation (7) reduces to

$$T_w = T_w'(T_a, v). \qquad (8)$$

v is now assigned a specific value, as:

$$v = 1.5 \text{ meters/sec}.$$

Hence, Equation (8) reduces to

$$T_w = T_w''(T_a). \qquad (9)$$

Thus, the line shown on the map of T_w, T_a in Figure 2 for $v = 1.5$ meter/sec is a plot of Equation (9). Similar lines are drawn for different values for the velocity.

Following the line for $v = 1.5$ meter/sec, the effect of relative humidity on thermal comfort is shown. With 0% relative humidity, a condition of thermal confort exists at $T_a = 85.5°F$. Whereas, when relative humidity increases to

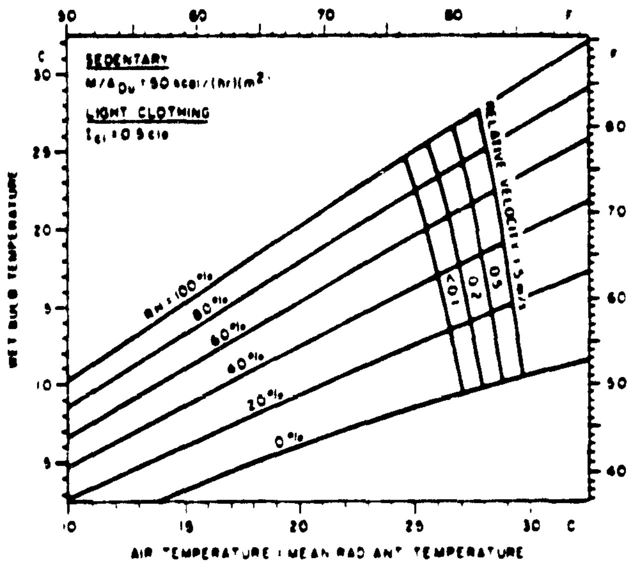

Fig. 2 Lines of Comfort for Light Clothing

100%, the value of T_a needs to be reduced to 82°F. A similar effect of high relative humidity in reducing T_a for thermal comfort can be seen at other air velocities.

The effect of air velocity can be seen by following the line of 100% relative humidity. At essentially standstill air, the T_a needs to be reduced to 76°F, although it can be as high as 82°F when air velocity is 1.5 meter/sec.

Figure 3 shows similar curves for a different level of clothing insulation. A comparison of the curves from Figures 2 and 3 shows that the wearing of a business suit requires the T_a to be reduced to 71°F for standstill air and 100% humidity, whereas with light, shirt-sleeve clothing, the T_a can be as high as 76°F for the same levels of air velocity and relative humidity.

Figure 4 shows the effect of mean radiant temperature. Following the line of standstill air shows that T_a needs to be reduced 5°F for every 10°F rise in T_{mr}.

Figures 2, 3 and 4 are called "Generalized Comfort Charts of Fanger" and are reproduced from the ASHRAE Handbook of 1977 Fundamentals, pp 8.25 and 8.26.

The utility of this approach in reducing cooling loads is realized when a comparison is made between extreme conditions of wearing a business suit (I_{clo} = 1), standstill air, 100% RH with a T_a of 71°F and wearing light clothing (I_{clo} = 0.5), air velocity of 1.5 meter/sec and 100% RH with a T_a of 82°F. The total temperature difference is 11°F. If outside air is assumed to be at 95°F, 60% RH, the reduction in cooling load required to cool fresh air is seen from the following table:

T_a (°F)	RH(%)	Enthalpy (Btu/lb of dry air)	Cooling Required (Btu/lb of dry air)
95	60	47.121	0
82	100	45.90	1.221
71	100	34.95	12.171

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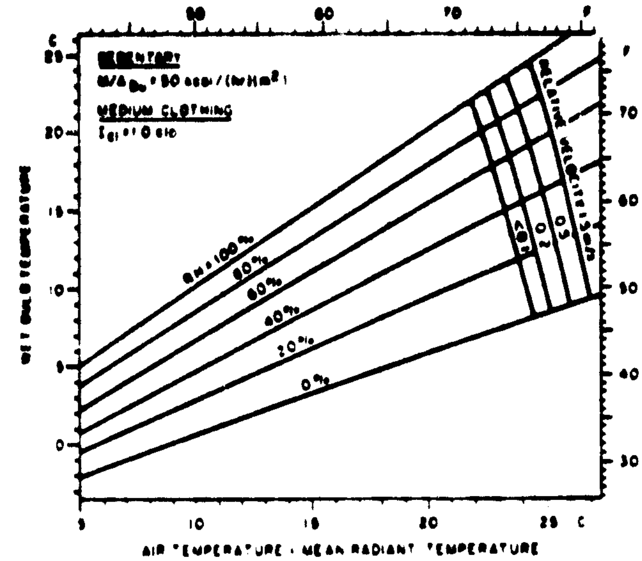


Fig. 3 Lines of Comfort for Medium Clothing

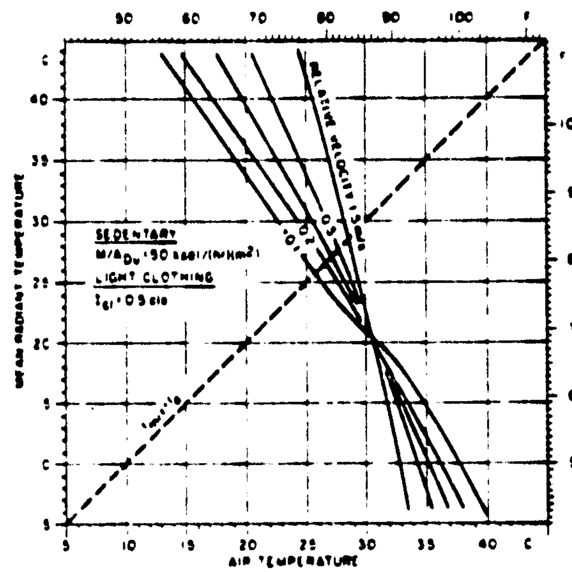


Fig. 4 Mean Radiant Temperature Effect

Thus, simply through the wearing of lighter clothes and the circulation of air, the cooling load imposed by the admission of fresh air can be reduced by a factor of 10.

FACTORS NOT CONSIDERED IN THE FANGER CHARTS

The described treatment has considered the gross problem of thermal comfort. The following factors, which affect the feeling of thermal comfort, are not considered.

Localized Cooling or Heating

The model assumes that clothing is uniformly distributed over the body, which is rarely the case. Also, the radiation load due to the sun would be highly localized in the case of a passenger in a vehicle. This radiation will produce a feeling of discomfort for the part of the body exposed, even if the body as a whole is in thermal comfort. Localized cooling of selected areas of the body where thermal receptors are located could result in a feeling of comfort at insignificant power requirements. This problem has been addressed by Temming et al. [1] of Volkswagenwerk. However, inadequate experimental data does not justify, at present, the inclusion of this information in the functional requirement specifications for the ECS.

Transient Effects

The usage pattern of an electric vehicle is expected to be many short trips of short duration (on the order of ten minutes). In fact, total time for which the car can be driven before the batteries are fully discharged is on the order of two to three hours. Thus, the steady-state comfort equation is of limited value in the study of electric vehicles.

A modification of the comfort equation to incorporate the time effect can be based on a model from Azer et al. [4]. However, various authorities have pointed out a lack of correlation in the predictions of comfort with experimental data during transient conditions lasting less than 10-20 minutes. Since only the steady-state modelling results can be relied on for information at the present time, more research is required to incorporate the effect of time.

AIR EXCHANGE CONSIDERATIONS

In the current automobile ECS practice, more than 40% of the thermal load is presented by the infiltration of outside air either coming in through cracks and leaks or intentionally introduced by blowers. A significant reduction in this load is possible if the amount of fresh air intake is reduced to a minimum. The purposes of fresh air intake are:

- To provide an adequate supply of oxygen for breathing and for any combustion taking place within the passenger compartment.
- To keep CO_2 concentration to a safe, low level by removing exhaled CO_2
- To remove any hazardous fumes leaking into the passenger compartment
- To remove undesirable odors, such as body odors, smoking, etc.

Relatively insignificant volumes of fresh air are required to satisfy the first two considerations. Hutchinson [6] states that 3.3 hours is the time limit before breathing becomes difficult for one person in a sealed environment of 100 cu ft due to the reduction of oxygen and the increase of CO_2. To quote Hutchinson,

> "This fact is even more clearly recognized when one realizes that a single occupant could be sealed in an airtight box of 10 feet by 10 feet by 10 feet for a week before loss of conciousness would be expected from oxygen deficiency. This same problem can be approached from a different point of view by assuming that it is desired to supply only sufficient outside air to an occupied enclosure to prevent the carbon dioxide concentration from exceeding 2 per cent. In order for equilibrium to exist, the volume of carbon dioxide introduced into the room in the incoming ventilation air plus the volume produced by the occupant must be equal to the volume of carbon dioxide leaving the room in the discharged air during the same time interval. By taking 1 hr as the time interval, the above balance gives [for CO_2]
>
> Volume in + volume produced = volume discharged"

Now, under standard atmospheric conditions, the following facts are noted:

1. **Amount of CO_2 in fresh air is 0.0003 cu ft per cu ft of fresh air.**
2. **The rate of CO_2 production per person is 0.6 cu ft/hr.**
3. **Breathing becomes difficult when the amount of CO_2 exceeds 0.02 cu ft per cu ft of air.**

To determine the rate of fresh air that needs to be supplied to an enclosed space to prevent breathing difficulty due to excessive CO_2 buildup, let:

$\dot{V}_{in}$ = rate of volume of fresh air in (cu ft/hr)

$\dot{V}_{in\ CO_2}$ = rate of volume of CO_2 coming in with the fresh air (cu ft/hr)

$\dot{V}_{p\ CO_2}$ = rate of volume of CO_2 produced by the occupants (cu ft/hr)

$\dot{V}_{d\ CO_2}$ = rate of volume of CO_2 discharged (cu ft/hr)

N = number of occupants.

Using these facts,

$$\dot{V}_{in\ CO_2} = 0.0003\ \dot{V}_{in}$$

$$\dot{V}_{p\ CO_2} = 0.6\ N$$

and

$$\dot{V}_{d\ CO_2} \le 0.02\ \dot{V}_{in}$$

(neglecting small changes in the volume of discharge air due to the generation of CO_2)

Hence, in steady state

$$\dot{V}_{in\ CO_2} + \dot{V}_{p\ CO_2} \le \dot{V}_{d\ CO_2}$$

$$0.0003\ \dot{V}_{in} + 0.6\ N \le 0.02\ \dot{V}_{in}$$

Hence,

$$\dot{V}_{in} \geq \frac{0.6\ N}{0.02 - 0.0003}$$

(i.e., $\dot{V}_{in} \geq 30.4$ cu ft/person-hr).

Thus, an air exchange of 0.5 ft^3/minute/person is all that is required to satisfy the first two considerations.

In electric vehicles, no hazardous fumes are generated due to the operation of the vehicle itself. However, if ECS elements utilize fuel combustion, such gases and fumes will be generated. Care will have to be taken to keep the fumes out of the passenger compartment either by making the compartment gas-tight or by ventilation.

Odor removal can be accomplished by dilution, masking, adsorption, or absorption. No mandatory requirements exist for automobile air exchange. The ASHRAE standard 62-73 for Natural and Mechanical Ventilation for volume of air intake will be used as follows:

- For electric vehicles utilizing ECS elements which will not result in any hazardous fumes or gases - 5 ft^3/min/person
- For electric vehicles utilizing ECS elements which may result in the generation of hazardous fumes or gases - 15 ft^3/min/person
- For hybrid vehicles - 15 ft^3/min/person.

AMBIENT CONDITION DESIGN POINT SPECIFICATION

The size and weight of the ECS is determined by the difference in conditions it has to produce between the ambient and the controlled space. The desirable conditions in the controlled space are relatively constant. However, the ambient conditions vary over a wide limit.

DETERMINATION OF DESIGN DRY-BULB TEMPERATURE

In the case of ECS design for buildings, the design-point, dry-bulb temperature and coincident wet-bulb temperature are selected in such a way that the resultant heating and cooling loads will be exceeded less than a certain number of hours in winter and summer, respectively. For example, the ASHRAE Handbook of 1977 Fundamentals, Chapter 23, has this temperature information on more than 1000 locations in the U.S.A. for three different numbers of hours: 5%, 2.5%, and 1% of total hours in winter and summer. Since the location is fixed in a building application, only local weather data statistics need to be considered. In an automobile application, different size units for different locations could be considered to better match the thermal loads with equipment capacity. However, for the purposes of this study, only one ECS is considered applicable all over the U.S.A. Thus, the same logic must be used as for the buildings, except that the effect of different locations in the percentile calculations must be integrated.

The algorithm for deciding design dry-bulb temperature is as follows:

Let

P_{ci} = population of cars in location i

$\theta_{\chi,i}$ = dry-bulb temperature which will be exceeded for χ% of the hours in a given season at location i

The car population percentage at location i can be determined by:

$$P_i = \frac{P_{ci}}{\sum_{i=1}^{N} P_{ci}}$$

where

N = number of locations in the data.

The car population percentage below certain temperature conditions is obtained by cumulative sums, as follows:

$$Q_\chi(\theta) = \sum_{i=1}^{N} \delta_i P_i$$

where

$\delta_i = 0$ when $\theta_{\chi,} \leq \theta$

$\delta_i = 1$ whe. ` , > ^

and

Q_χ = total percentage of cars which will experience temperatures higher than θ for χ% of hours in summer.

Weather data [7] are recorded at particular locations through time and cannot be meaningfully averaged to represent a large area such as a state. Weather stations, often located at airports, are generally assumed to be representative of the nearest city. Since car registration statistics [8] by city are not directly available, it is assumed in the calculations that per capita car registration for any city is not significantly different from per capita registrations for that entire state. P_{ci}, the population of cars in location i from page 18, is calculated for the 25 largest standard metropolitan statistical areas (SMSA's) in the U.S. as follows:

$$P_{ci} = \text{population of (SMSA)}_i \times \left(\frac{\text{cars registered in state(s)}}{\text{population of state(s)}}\right)$$

In the case of three SMSA's, more than one state was used to calculate per capita registration (portion of formula enclosed by parentheses). The ·al population of all 25 SMSA's considered is 34% of the total U.S. population; the total of all calculated car registrations ($\sum_{i=1}^{N} P_{ci}$) is ′ ‹ all registrations nationwide. Figure 5 shows the plots for χ = 1, 2.5 and 5.

DESIGN POINT SELECTION FOR AIR VELOCITY AND SOLAR RADIATION

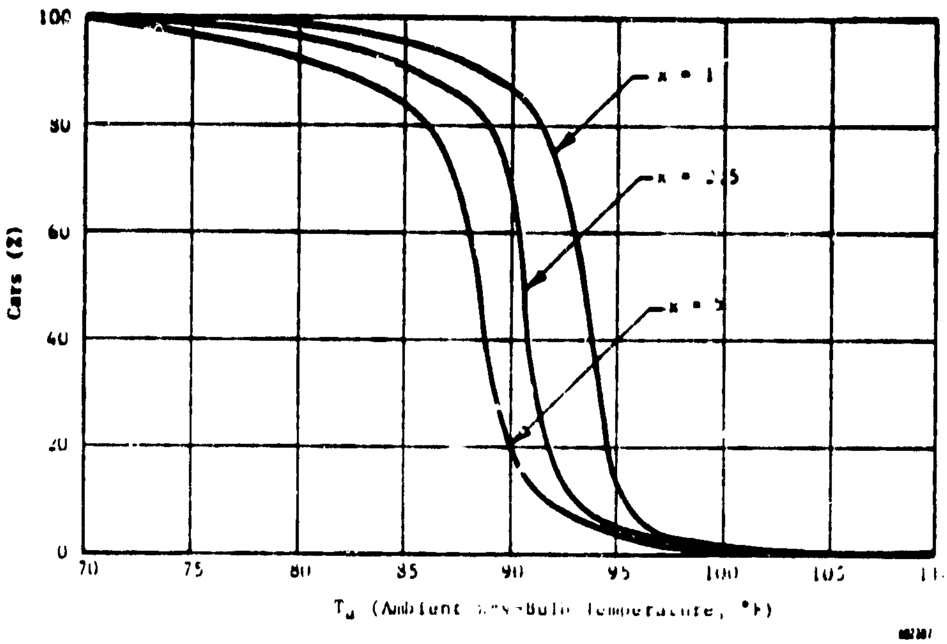
The ambient conditions of importance for heat load calculations are:

T_a - dry-bulb temperature

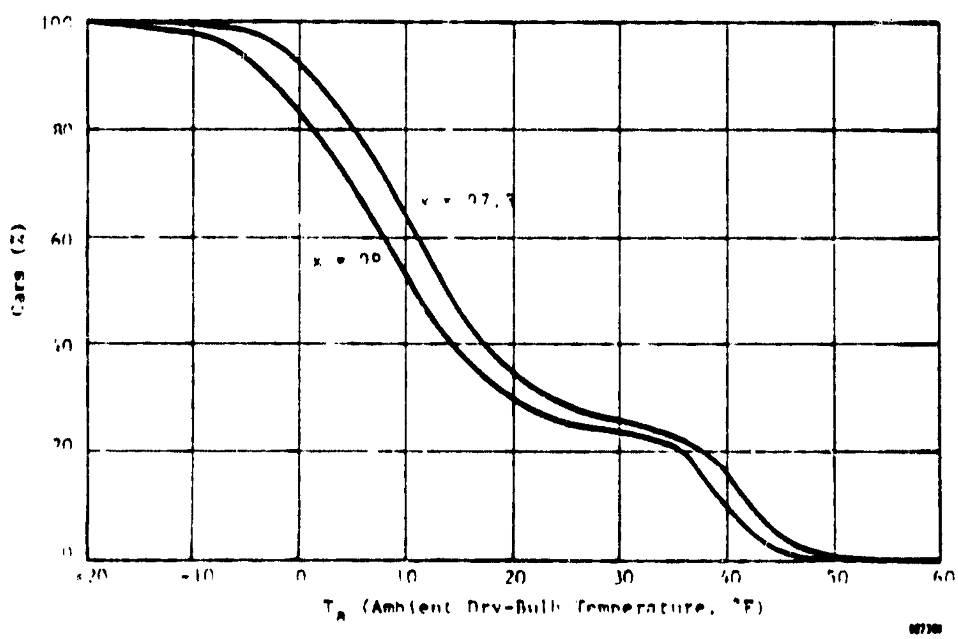
T_w - wet bulb temperature

v - air velocity

I - solar radiation (direct, diffuse and reflected).



(a) For Summer



(b) For Winter

Note: Curves are based on climatic data and car population statistics for 25 largest metropolitan areas in U.S.A.

Fig. 5 Percentage of Cars Experiencing Temperatures Higher Than T_a for x% of Hours

In the case of automobiles, the air velocity of importance is the relative velocity created by the car's own motion. Hence, the design value for this factor is taken as 45 mph (SAE J277 - D cycle top speed).

As the electric vehicle is considered to be operable at all times of the year and any time during the day, the selected design value of solar radiation should present a maximum load on the ECS. Thus, during the heating season, the benefit from solar radiation will be assumed to be zero. During the cooling season, the design value for solar radiation should be selected so that it will not be exceeded for 99% of the car population in the U.S.A. These calculations are made on similar lines as for the determination of dry-bulb and wet-bulb temperatures. **Figure 6 shows the plot of percentage** of cars for which a certain level of solar radiation will be exceeded in a year.

DESIGN POINT SPECIFICATION FOR PASSENGER COMPARTMENT

Based on the considerations for the various environmental parameters, such as dry- and wet-bulb temperatures, air velocity, and solar radiation, the ECS will be sized to provide the following conditions in the passenger compartment.

Air Exchange > 5 cfm/person		
Parameter	During Heating Season	During Cooling Season
T_a, dry-bulb temp.	>68°F	<75°F
T_w, wet-bulb temp.	-	<75°F
Air velocity at the passenger	<0.5 meter/sec	<1.5 meter/sec
T_{mr}	Limit Not Specified	

In the case of innovative solutions, the specification will refer to Fanger's generalized comfort charts, or comfort will be computed from the mathematical model of Azer, whichever is appropriate.

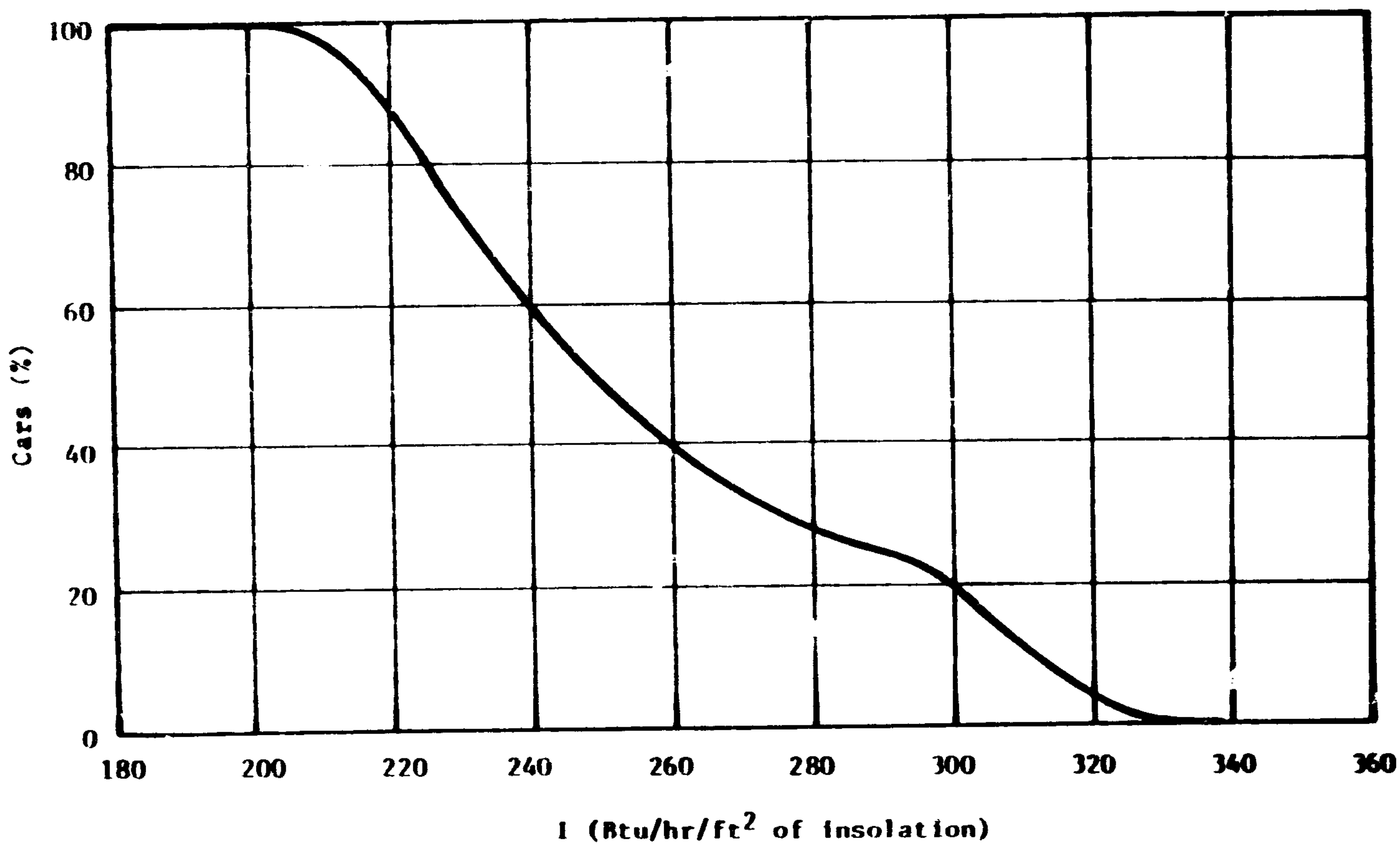


Fig. 6 Percent of Cars Experiencing Insolation Greater Than I

SELECTED DESIGN POINT SPECIFICATIONS FOR AMBIENT CONDITIONS

Based on the design point specification for the passenger compartment study, the following conditions are selected as the design point for sizing ECS.

Conditions	For Heating Season	For Cooling Season
T_a	-10°F	100°F
T_w	-	74°F
V	45 mph	45 mph
Solar Insolation	-	326 Btu/hr/ft^2

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MTI 80TR46

ELECTRIC AND HYBRID VEHICLES ENVIRONMENTAL CONTROL SUBSYSTEM STUDY

TRAVEL SCENARIO

Prepared for:

Jet Propulsion Laboratory
California Institute of Technology
Pasadena, California

Prepared under:

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Sponsored by:

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National Aeronautics and Space Administration

July 3, 1980

MECHANICAL TECHNOLOGY INCORPORATED
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TABLE OF CONTENTS

INTRODUCTION

The requirement for the scenarios used in this study are realistic sample missions in which the vehicles with and without ECS's are simulated. The scenarios also provide for exercising the simulation over the performance capability of the vehicle and meet the practical requirements for computational simplicity.

METHODOLOGY

References [1] and [2] have been identified at the end of this report. They catalog and interpret the travel behavior of American households. They are used to provide the factual basis for the travel scenarios used in this report. The following pages summarize material abstracted from the references.

- 8.9 hours per week average - is devoted to travel

  24.7% - work associated travel
  22.5% - leisure activity associated travel
  20.2% - shopping associated travel
  20.2% - personal travel
  12.4% - other non-work travel including educational, organizational, social leisure

- Work-associated travel, in averaged terms of travel time, stop time, and number of stops, is different from other travel activities

	DURATION OF TRIP SEGMENT	STOP TIME AFTER 1st SEGMENT
work associated travel	22 minutes	326 minutes
all non-work travel	18 minutes	24 minutes

- Distribution of personal automobile-driven-distance to work has a mean of 9.0 miles and 7.6 percentile for 25 or more miles.

A requirement set by the contract for this study is that one of the scenarios will be 65 repetitions of the SAE 227a - D driving cycle. This 65-cycle scenario demands total travel time of 131 minutes, maximum speeds of 72 km/hr (45 mph) and acceleration from 0 to 72 km/hr in 28 seconds. This scenario provides data with moderate speeds and emphasis on speed changes. It has the important benefit that there is a large data base of simulating the "D Cycle" for comparison.

The "work associated" scenario and the "shopping and all other non-work associated" scenario are designed to consider portions of the vehicle operating map that are not encompassed by the SAE 227a - D cycle and that are reasonable for the designated activity. The scenarios are also chosen to be practical in limiting the analytical effort and are, therefore, made simple.

Care was taken to justify the duration of the scenarios The energy expended for conditioning of the EV environment is determined most strongly by the time span that the vehicle is occupied (for any ambient conditions).

WORK-ASSOCIATED TRAVEL SCENARIO

The references indicate that the time for travel to work is less variable than the distance driven to work. That is, the shorter distances are driven more slowly than the longer distances. The average trip duration was used with higher- and lower-than-average driving speeds to exercise the high- and low-speed portion of the vehicle operating map for this study, and to have travel time and distance travelled be realistic. See Table 1.

SHOPPING- AND OTHER-THAN-WORK-ASSOCIATED TRAVEL SCENARIO

This scenario characterizes the remainder of the vehicle operating spectrum. The references indicate that the previous scenarios have not covered the less brief, multisegmented trips at moderate speeds. The elapsed time reasonably models the references. See Table 2.

TABLE 1

WORK-ASSOCIATED TRAVEL SCENARIO

Time-Seconds	Vehicle Speed - km/hr	(mph)
0	0	(0)
15	40	(25)
1,035	40	(25)
1,050	88	(55)
1,290	88	(55)
1,320	0	(0)
parked		
20,880	0	(0)
20,910	40	(25)
21,930	40	(25)
22,170	88	(55)
22,200	88	(55)
end	0	(0)

TABLE 2

SHOPPING- AND OTHER-THAN-WORK-ASSOCIATED TRAVEL SCENARIO

	Time-Seconds		Vehicle Speed - Km/hr	(mph)
Drive	0	*	0	(0)
	18		56	(35)
	160		56	(35)
	176		0	(0)
	180	*repeat five more times	0	(0)
Stop	1,080		0	(0)
	2,520		0	(0)
Drive	2,538		56	(35)
	2,680		56	(35)
	2,696		0	(0)
	2,700	*repeat five more times	0	(0)
	3,600 end			

SUMMARY

The travel scenarios used for the analysis in this study have the following characteristics:

SCENARIO	SAE 227a - D Cycle repeated 65 times		Workrelated travel		Shopping and other non-work related travel	
Maximum Speed - km/hr (mph)	72	(45)	88	(55)	56	(35)
Acceleration						
from 0 to --- km/hr (mph)	72	(45)	88	(55)	56	(35)
in --- seconds	28		40		12	
Distance per trip segment- km (miles)	1.5	(0.9)	18	(11)	2.5	(1.6)
Number of Segments	65		2		12	
Total Distance km (miles)	97	(60)	36	(22)	30	(19)
Total Driving Time - minutes	105		44		35	
Parking Time - minutes	0		325		24	

REFERENCES

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MTI 80TR53

ELECTRIC AND HYBRID VEHICLE ENVIRONMENTAL CONTROL SUBSYSTEM STUDY

IDENTIFICATION AND DESCRIPTION OF ENVIRONMENTAL CONTROL SUBSYSTEM ELEMENTS

Prepared for:

Jet Propulsion Laboratory
California Institute of Technology
Pasadena, California

Prepared under:

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MECHANICAL TECHNOLOGY INCORPORATED
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ABSTRACT

The report contained herein provides a description of the work completed for Tasks 3 and 4 of the Electric Hybrid Vehicles Environmental Control Subsystem Study under the Jet Propulsion Laboratory Contract No. 955682. The work involved in these tasks comprised the identification and description of environmental control subsystems for electric vehicle passenger compartments. Over 40 different heating and cooling schemes are presented including preliminary calculations used to determine their feasibility.

The discussion contains descriptions of subsystem characteristics, operating procedures, material storage methods, and sources of energy. The major parameters in the preliminary calculations include subsystem sizing and cost.

Many schemes appear to be attractive. For heating, a package less than 60 lb in weight and smaller than 1 cubic foot will be adequate to provide all the heating required without on-board storage of a petroleum-based fuel. Combined heating and cooling can be provided by a package as small as 200 lb and 6 ft^3, again requiring no gasoline or energy from electric batteries on board the vehicle. Determination of the most suitable subsystems will be established in future tasks.

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Mr. A. Rieger

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Mr. H. Chen

Mr. J. Wagner

Ms. Martha Swidersky of the Technical Publications Department carried out the technical writing and coordinating of the report.

The calculations for the Magnetic Heat Pump section were performed by Dr. G.V. Brown of NASA-Lewis Research Center, Cleveland, Ohio.

The project manager greatfully appreciates their excellent contributions.

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

LIST OF FIGURES

LIST OF FIGURES (Cont'd)

LIST OF TABLES

LIST OF TABLES (Cont'd)

1.0 INTRODUCTION

Environmental control of the passenger compartment of electric vehicles requires both heating and cooling subsystems. Because an electric vehicle is designed to significantly reduce the use of petroleum-based fuel, the thermal environment within the passenger compartment should use little or no fuel of this type. With this goal as a focus, this report has identified and described a variety of heating and air conditioning methods which utilize electricity. (Some schemes can use fuel oil or natural gas as well.) In most cases, it is assumed that the energy carriers will be recharged at home during the period in which the propulsion batteries are being charged. Figure 1-1 schematically summarizes the various components in the environmental control system (ECS) of an electric vehicle.

The materials and operating systems identified in the report are described in terms of their weight, volume, and cost. These factors become significant at a later stage of analysis when efficiency and range penalties are being evaluated. Since this study has the nature of a screening study, only preliminary calculations are carried out to determine the feasibility of each system. The calculations indicate that some of the systems are promising, while others are clearly impractical. Because no attempt has been made to arrive at a specific configuration, the design of ancillary components such as blowers and duct work is not included. More detailed calculations and specific ECS designs will be made in future tasks of this program.

1.1 System Sizing

The considered systems must be sized to a given set of performance specifications in order to assess their relative merits. Some of the parameters and their values for performance specifications, which are reproduced in Table 1-1, were derived in an earlier report [1]*. In addition to these parameters, two others have to be specified: duration of environmental control and thermal load.

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*Numbers in brackets indicate References found at the end of this report.

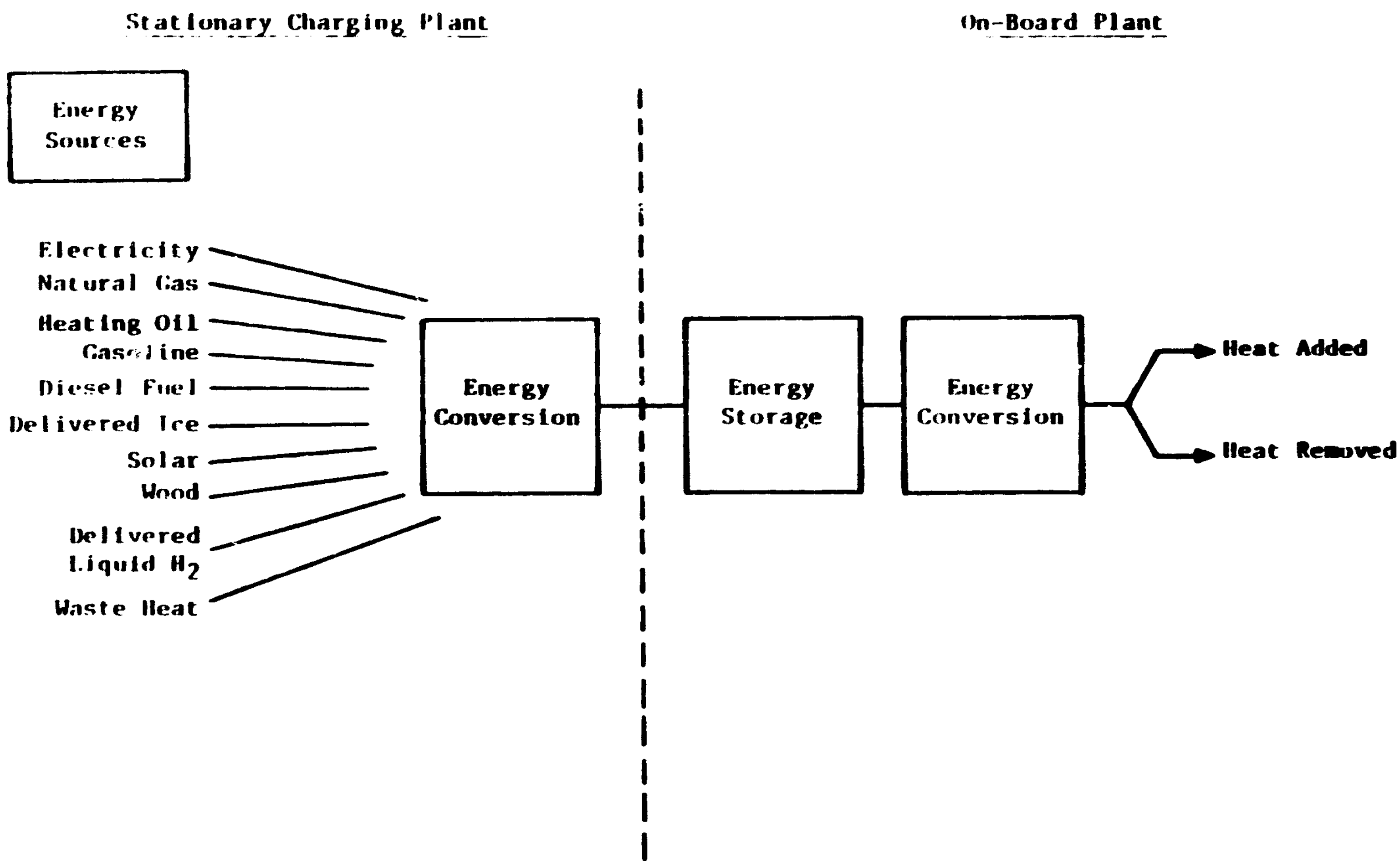

Fig. 1-1 Schematic of Environmental Control Subsystem for Electric and Hybrid Vehicles

TABLE 1-1

DESIGN POINT SPECIFICATIONS

PASSENGER COMPARTMENT

Air Exchange > 5 cfm/person		
Parameter	During Heating Season	During Cooling Season
T_a, dry-bulb temp.	>68°F	≤75°F
T_w, wet-bulb temp.	-	<75°F
Air velocity at the passenger	<0.5 meter/sec	<1.5 meter/sec
T_{mr}	Limit Not Specified	

AMBIENT CONDITIONS

Conditions	For Heating Season	For Cooling Season
T_a	-10°F	100°F
T_w	-	74°F
v	45 mph	45 mph
Solar Insolation	-	326 Btu/ hr-ft^2

1.1.1 Duration of Environmental Control

Attachment A of the study contract [2] specifies that the vehicle is provided with 18 batteries, each providing 1 kWh of deliverable energy. The General Electric Company (GE), under a contract from the U.S. Department of Energy, has built an electric vehicle which has the same number of batteries, and approximately the same characteristics as specified in Attachment D of this contract. The characteristics of the GE vehicle will be used in this study. Table 1-2 shows the data on the range of this vehicle as given in Reference 3. Table 1-2 shows the actual driving time under various conditions computed from the range data.

In an earlier report [4], Mechanical Technology Incorporated (MTI) developed probable-use scenarios for an electric vehicle. This report included the requirements specified on page 2 of Attachment B, "Functional Requirements" of the contract [2] for 65 repeats of the Jet Propulsion Laboratory (JPL) version of the SAE J227a Schedule D driving cycle. Actual driving times for these various scenarios are reproduced in Table 1-3.

A conclusion that can be reached from Table 1-3 is that for the most probable use of an electric vehicle, environmental control needs to be provided only for about one hour. For the most probable pattern of driving, viz. Table 1-2, SAE J227a Schedule D driving cycle, batteries will run out within 2.6 hours of driving. Thus, providing environmental conditioning for more than 2.6 hours is unnecessary, as the ECS can be recharged during the same period the propulsion batteries are being recharged. As the duration of environmental conditioning has a profound influence on the viability of some of the schemes considered, two sets of calculations will be considered, one for a one-hour duration and the other for a 2.5-hour duration.

1.1.2 Thermal Load

Thermal load is defined as a function of the design point specifications given in Table 1-1 and of the structural design of the vehicle. Ruth [5] has shown that significant reductions, up to 50%, are possible in cooling loads in summer through minor design changes. Various sources [5, 6, 7]

TABLE 1-2

ACTUAL DRIVING PERIODS UNDER VARIOUS DRIVING CONDITIONS FOR GENERAL ELECTRIC VEHICLE ETV-1

Cycle	Two Occupants 300 lb Payload		Four Occupants 600 lb Payload	
	Range (miles)	Driving Period (hours)	Range (miles)	Driving Period (hours)
35 mph constant speed	122	3.5	117	3.35
45 mph constant speed	102	2.27	97	2.16
SAE J227a Schedule D Driving Cycle	75	2.6	69	2.39

TABLE 1-3

ACTUAL DRIVING TIMES FOR VARIOUS TRAVEL SCENARIOS

Scenario	Driving Time (hours)
SAE J227a, D Cycle Repeated 65 Times	1.75
Work-Related Travel	0.734
Shopping and Other Non-work-related Travel	0.584

have used different values for the cooling loads, varying between 10,000 and 20,000 Btu/hr for compact cars. Similar variability exists in the heating loads during winter.

During the summer, the cooling load consists of solar, conductive, convective, human, and instrument loads. The relative values of these factors have been estimated by Ruth [5] and are reproduced in Table 1-4. During winter, the solar load is negative, but to allow for night conditions, this load is considered to be zero. The human load is also negative. The conductive and convective load are two to three times higher than in summer, due to the higher temperature differential that has to be maintained between the ambient and the passenger compartment. Therefore, the rate at which heat needs to be added during winter can be considered to be approximately equal to the rate at which heat needs to be removed during summer.

Various systems will be sized to provide a heat rate of 17,000 Btu/hr (5 kW) both in winter and in summer. Thus, the total quantity of heat that needs to be removed or added is given as:

- For one hour - 17,000 Btu
- For 2.5 hours - 42,500 Btu.

1.2 Energy Storage

Environmental control of the electric vehicle requires an adequate amount of energy on board the vehicle in some suitable form. Utilization of this energy at an appropriate rate then provides the desired heating and cooling.

Fundamentally, heat can be added or removed in two ways. The first method is addition/removal of heat in the form of heat itself. In this case, the amount of energy required to be stored is eq[illegible] or greater than the quantity of heat that needs to be supplied/re[illegible]. In the second method, total added/removed heat is partly in the form of heat and partly in the form of work. Since the atmosphere may be used as a source/sink of heat, only the work component needs to be stored. Because most of the energy

TABLE 1-4

BREAKDOWN OF COOLING LOAD

Load	Value (Btu/hr)	Percent of Total Load
Solar	4470	34.8
Conductive	1770	13.8
Fresh Air	5400	42.0
Passenger	1000	7.8
Instrument	200	1.6
Total	12,840	100

From Reference 5

C-2

is obtained from the atmosphere, the work component represents only a fraction of the total energy requirement. However, unless the work can be stored in the form of potential energy, significant losses are bound to occur in converting any other form of energy, such as chemical or heat, to work. Thus, the advantage of a reduced energy storage requirement will be attenuated to a certain extent.

Energy can be stored in the form of heat (at high temperature for heating and at low temperature for cooling) or some other form such as chemical or potential. When energy is stored directly in the form of heat, no energy conversion is required, and no conversion-associated losses occur. However, a certain amount of heat energy is bound to escape during the storage mode. The extent of the loss depends on the temperature differential between the storage and ambient temperatures, and on the level of insulation. When energy is stored in some other form, it has to be converted to either heat and/or work. This conversion requires equipment and results in an energy loss which can be utilized only in the heating mode to a small degree. In the storage mode, however, energy losses are insignificant.

The length of storage time is limited by the form of energy storage used. Thus, for energy stored in the form of heat, storage time is on the order of only a few hours, unless elaborate insulation techniques are utilized. By using electric batteries, storage time can be extended to a few days. With gasoline, a storage time of weeks and even months is possible.

1.3 Energy Source

The source of energy available presents another dimension to the selection of the appropriate scheme for the electric vehicle ECS. Energy could be obtained from private residences in the form of electricity, natural gas, heating oil or waste heat, and from service stations in the form of gasoline, propane, etc. Delivery of liquid hydrogen, nitrogen, oxygen or air from a suitable source is also conceivable. Futhermore, ice, dry ice (solid form of CO_2) or liquid ammonia could constitute suitable sources of energy.

An overriding factor in the selection of a suitable energy source is related to the end product after the utilization of the energy on board the vehicle. The end product can either be stored on the vehicle and reprocessed by a suitable technique after delivery to a reprocessing plant, or it can be released to the atmosphere. The first approach is clearly the more attractive from a local environmental consideration, but the total environmental impact must be kept in perspective.

1.4 Categories of ECS Elements

The various ECS elements considered can be divided into three general categories. These categories and their respective report section numbers are as follows:

- Heat Pumps - Section 2.0
- Thermal Storage - Section 3.0
- Reversible Thermochemical Reactions - Section 4.0

Within these categories, specific ECS elements are defined and described.

2.0 HEAT PUMPS

Various different types of heat pumps were considered in this study. The heat pumps described, and their corresponding subsection numbers are as follows:

- Vapor-compression heat pumps driven by a gasoline engine or electric motor - 2.1
- Thermal engine heat pumps - 2.2
- Absorption-cycle heat pumps - 2.3
- Thermoelectric heat pumps - 2.4
- Magnetic heat pumps - 2.5
- Split heat pumps - 2.6

In the first five systems, the complete heat pump hardware is on board the vehicle. The energy required to run the heat pump is also stored on board as electric batteries or petroleum-based fuel.

The last system considered, the split heat pump, is a novel design in which the refrigerant loop is split into two parts. An adequate quantity of refrigerant in state 1 is stored on the vehicle. When heat pumping is performed, this refrigerant passes to state 2 and is also stored on the vehicle.

In a conventional heat pump, the equipment which converts the refrigerant from state 2 to state 1 is part of the heat pump itself. In the split heat pump system, this equipment is not carried on board the vehicle but is kept, for instance, in a garage. Thus, when the vehicle's propulsion batteries are being recharged, the used refrigerant in state 2 is delivered to the other stationary half of the heat pump cycle equipment for reconditioning to state 1. Then, the refrigerant in state 1 is again stored on the vehicle.

2.1 Vapor-Compression Heat Pumps

This section describes a brief study of the current automotive air-conditioning or heat-pump systems. By blowing the conditioning air over either the condenser or the evaporator, the heating effect or the cooling effect is derived.

Cycle calculations are carried out for various ambient temperatures. Compressor mass flow capacity dependence on pressure ratio and ambient temperature is determined. Graphs are drawn for the system performance parameters: capacity and COP. A brief estimation of costs and weights is presented.

2.1.1 Basic Heat Pump Cycle

The system comprises a compressor, a condenser, an expansion valve, and an evaporator. Figure 2-1 shows a schematic diagram of the system. Freon vapor at a saturated state or slightly superheated state is compressed in the compressor. The compressed vapor is cooled and condensed in a condenser. The state of the fluid leaving the condenser is saturated or slightly subcooled liquid. From this state, the fluid undergoes a throttling process in the expansion valve. The low-pressure fluid is now in the mixture region and enters the evaporator. In the evaporator, the fluid receives heat and evaporates before admission to the compressor to recommence the cycle. Figure 2-2 shows the operation of the cycle on a temperature-entropy diagram (T-S) while Figure 2-3 shows the cycle on a pressure-enthalpy diagram (P-h).

Working fluids, such as the Freons; R-11, R-12, R-502, R-22, R-500, are used in heat pump systems. The following discussion will be, however, limited to R-12 which is commonly used in automobile systems.

A heat pump operates in the heating or the cooling mode. For this study, vehicle (compartment) temperatures of 68°F for winter and 75°F for summer are set. The ambient temperature is assumed to vary from -10°F to 65°F in winter and from 75°F to 100°F in summer. The range 68°F to 75°F is a "dead-band" when the heat pump is not required to operate.

2.1.2 Heating Operation

Figure 2-4 shows a typical illustration of an operating cycle in the heating mode. For this example, an ambient temperature of 0°F is chosen. Heat transfer rate is a function of the heat exchanger design parameters and the temperature difference across the evaporator. Obviously, smaller temperature differences increase the evaporator (heat exchanger) size and vice versa. Thus, the ambient temperature and the ΔT decide the evaporator temperature.

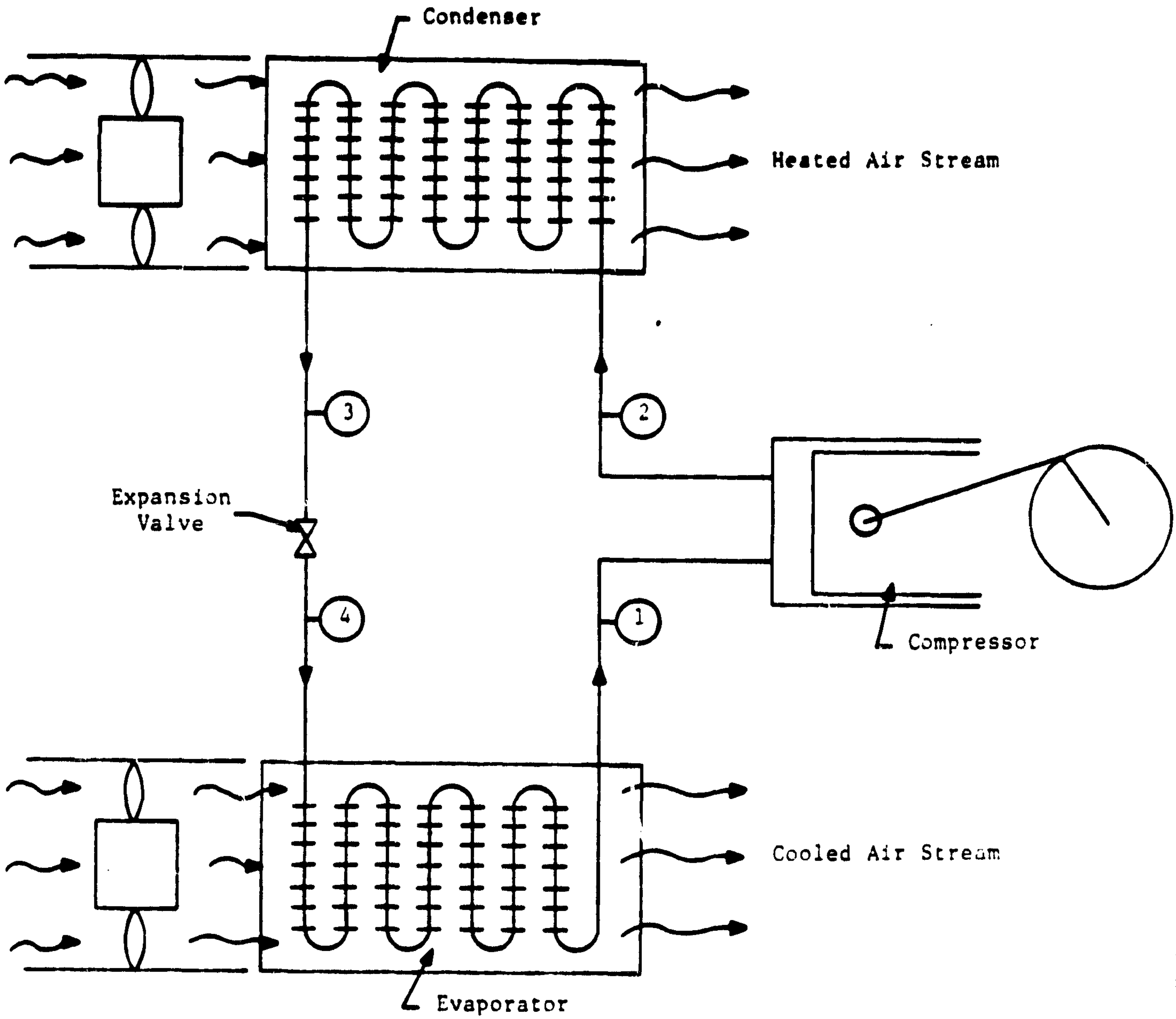

Fig. 2-1 Schematic Diagram of a Heat Pump

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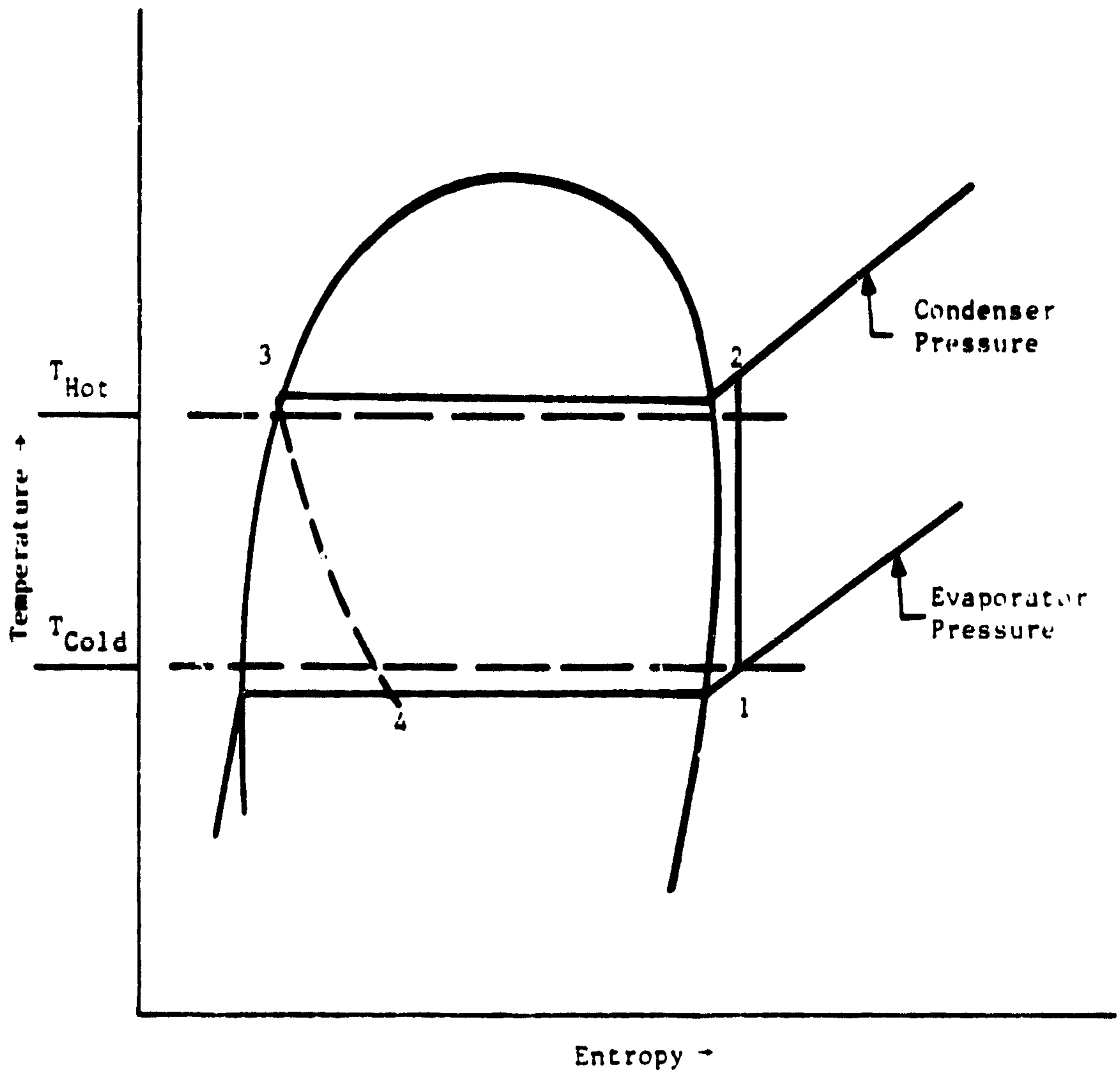

Fig. 2-2 Temperature-Entropy Diagram for a Heat Pump

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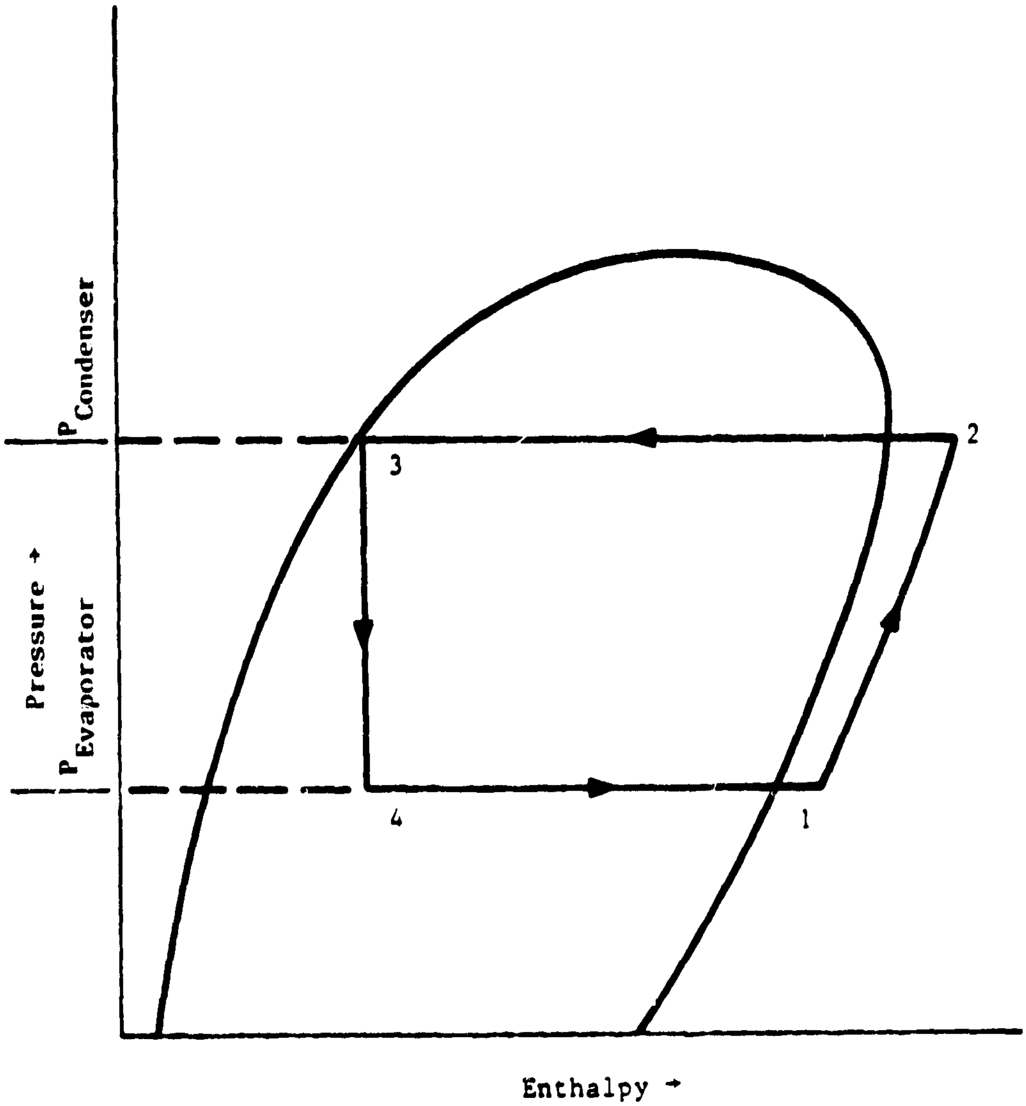

Fig. 2-3 Pressure-Enthalpy Diagram for a Heat Pump

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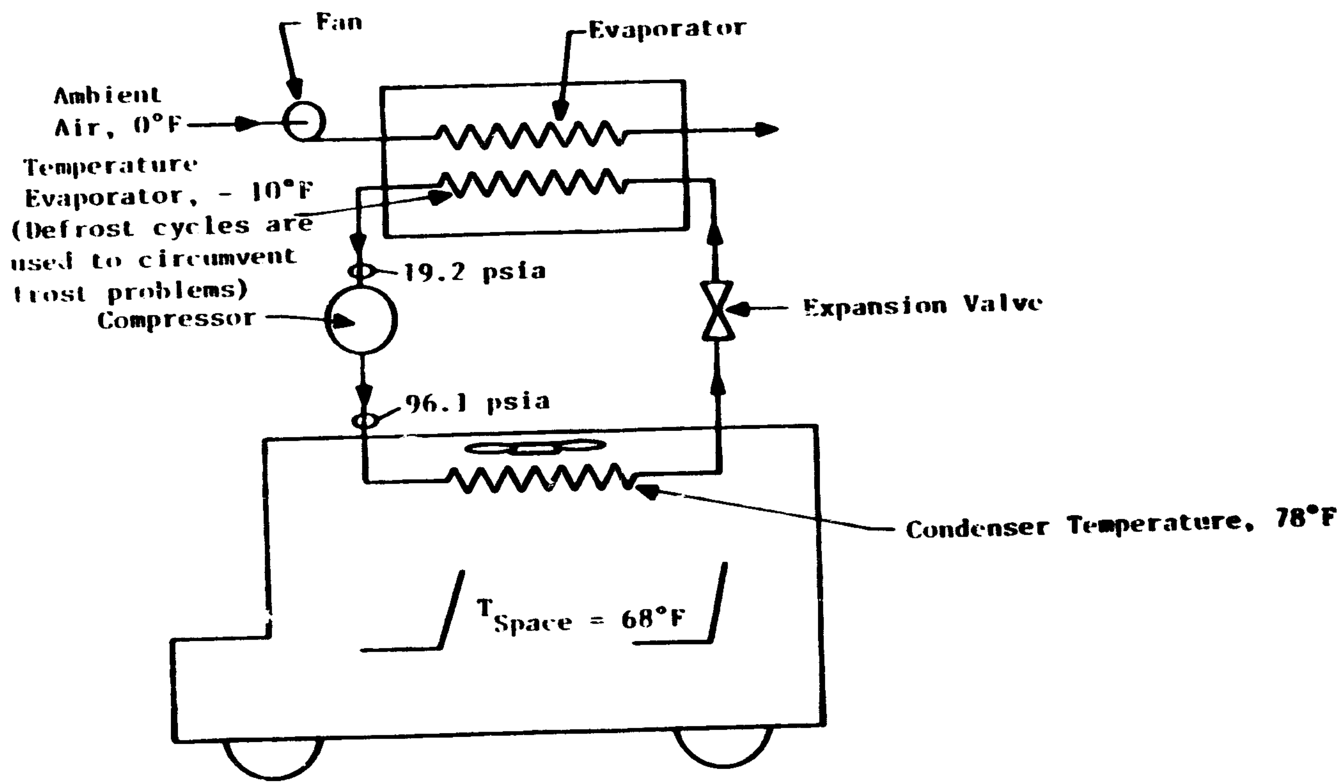


Fig. 2-4 Schematic Arrangement of a Typical Heating Cycle

For the chosen working substance, this temperature fixes an evaporator pressure (i.e., the saturation pressure corresponding to the evaporator temperature).

Compressor suction pressure is slightly lower than the evaporator pressure due to pressure losses in the connecting lines and the suction valves. The vapor is compressed and is delivered to the condenser, which is placed in the conditioned space (compartment). The required condenser pressure and the pressure losses in the lines and delivery valve determine the compressor discharge pressure. The vapor is condensed and heat is transferred to the compartment. Once again, the heat transfer rate is a function of ΔT and the heat exchanger design parameters. The condenser temperature is the saturation temperature corresponding to the condenser pressure. Lastly, the fluid flows through the expansion valve to effect the required pressure change between the condenser and the evaporator.

2.1.3 Cooling Operation

The operation of the cycle in the cooling mode is similar to that in the heating mode except that the positioning of the condenser and evaporator with respect to the conditioned space is altered. Figure 2-5 is illustrative of a typical arrangement.

2.1.4 Compressor Operation

Inspection of the cooling and heating operations described above shows that the compressor operating pressures vary with the mode. The pressure ratio over which compression is carried out varies with the ambient temperature. Figure 2-6a shows the pressure-volume (P-V) diagram for a reciprocating compressor. The compression line of the cycle on a T-S diagram is shown in Figure 2-6b for an understanding of the temperature correspondence. As the source and sink (compartment and ambient or vice versa) temperatures vary, the compressor operation varies over a certain pressure range.

Ideally, the volume delivered by the compressor is equal to the stroke volume. But in an actual compressor, the mass of gas in the clearance volume expands (process 3-4) and the volume of gas sucked is given by V_1-V_4. Thus, the

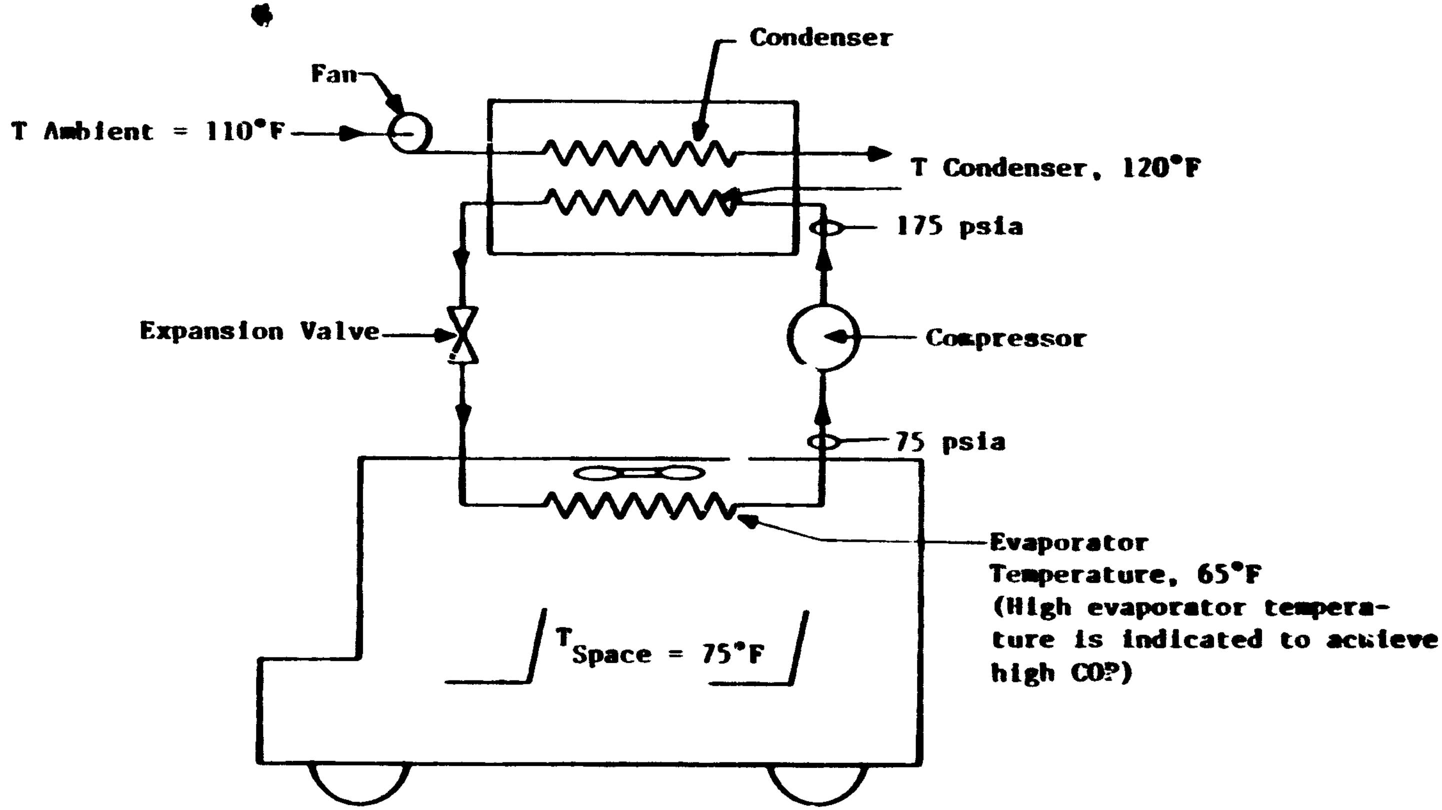
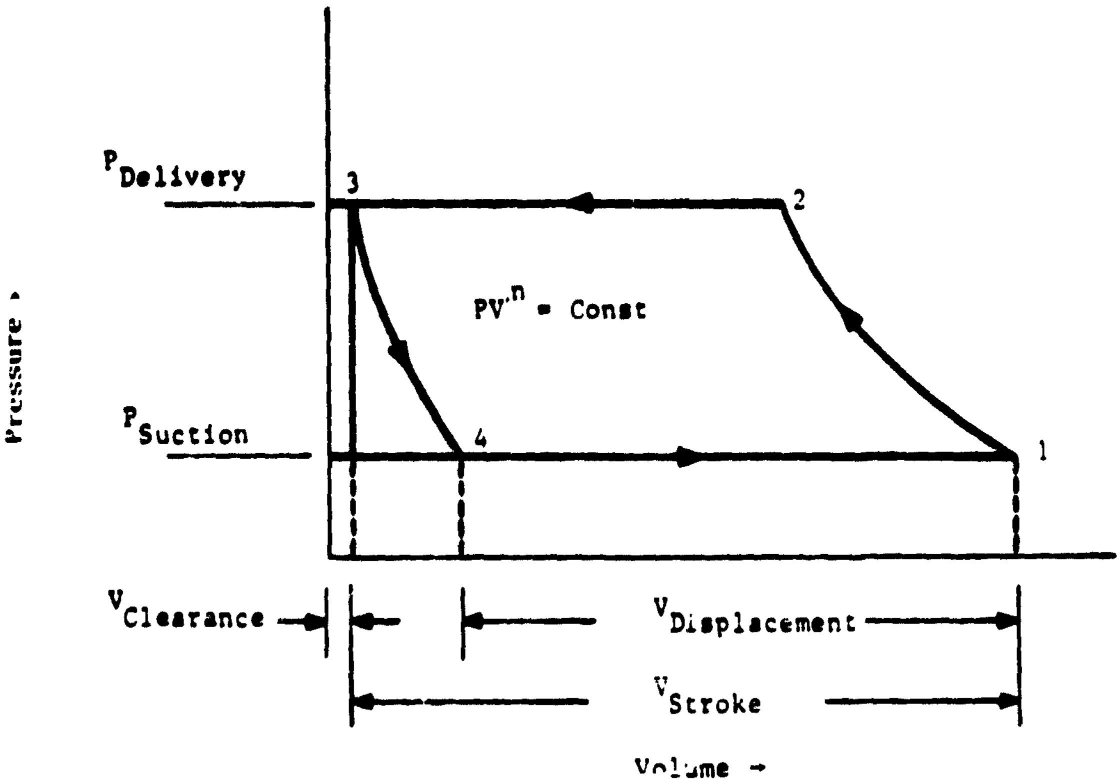
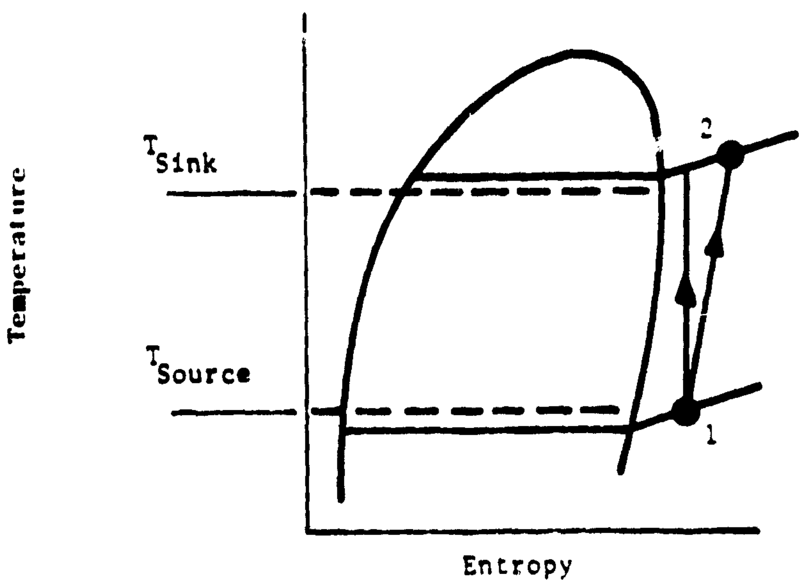


Fig. 2-5 Schematic Arrangement of a Typical Cooling Cycle



(a) P-V Diagram



(b) T-S Diagram

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Fig. 2-ı) Compressor Oneration

volumetric efficiency of the compressor is a function of the pressure ratio and the clearance volume. Thus, in the heating mode, when the ambient temperature is very low (such as -10°F), the compressor volumetric capacity falls off considerably. Moreover, the density of the gas is low at low temperatures resulting in very poor mass flows through the compressor.

2.1.5 Performance

Heat pump performance is measured by the amount of heat transferred and the coefficient of performance (COP).

In the heating mode, these two parameters are the heat transferred from the condenser to the conditioned space and the heating COP. The heating COP (COP_H) is defined as the heat added divided by the total work input to the heat pump system.

In the cooling mode, the corresponding parameters are the heat transferred from the conditioned space to the evaporator and the cooling COP. The cooling COP (COP_C) is defined as the heat removed divided by the total work input to the heat pump system.

The following relationships define these parameters:

$$COP_H = (h_2-h_3)/(h_2-h_1)$$

$$COP_C = (h_3-h_1)/(h_2-h_1)$$

For definitions, see Figure 2-6.

As presented in the previous discussion, the COPs vary with the ambient temperature. Such a variation may be calculated. Table 2-1 shows an illustrative calculation. Figures 2-7 and 2-8 show graphs of COP variation with ambient temperature.

Argonne National Laboratories (ANL) conducted a study [8] which included heat pump performance variation with temperature and presented polynomial equations. The results incorporating the ANL study are also shown in Figures 2-7 and 2-8 and reasonable agreement is apparent.

TABLE 2-1

ILLUSTRATION OF CALCULATIONS OF THE COP_H VARIATION WITH AMBIENT TEMPERATURE

T Ambient, °F	-10
ΔT Evaporator, °F	10
T Evaporator, °F	-20
P Evaporator, psia	15.27
ΔP Suction, psia	2.0
P Suction, psia	13.27
T Conditioned Space, °F	68
ΔT Condenser, °F	10
T Condenser, °F	78
P Condenser, psia	96.07
ΔP Delivery, psia	5.0
P Delivery, psia	101.07
r compressor	7.62
c (clearance), %	3.0
Capacity/Displacement	0.84
V_G, (Specific Volume at Compressor Inlet), ft^3/lbm	2.44
H_S (Enthalpy at Compressor Suction), Btu/lbm	74
H_D (Enthalpy at Compressor Delivery), Btu/lbm	90
η_C (Adiabatic Efficiency of Compressor)	0.7
W_C, Compressor Input, Btu/lbm	23
ΔH, Enthalpy Drop in Condenser, Btu/lbm	75
COP_H	3.26

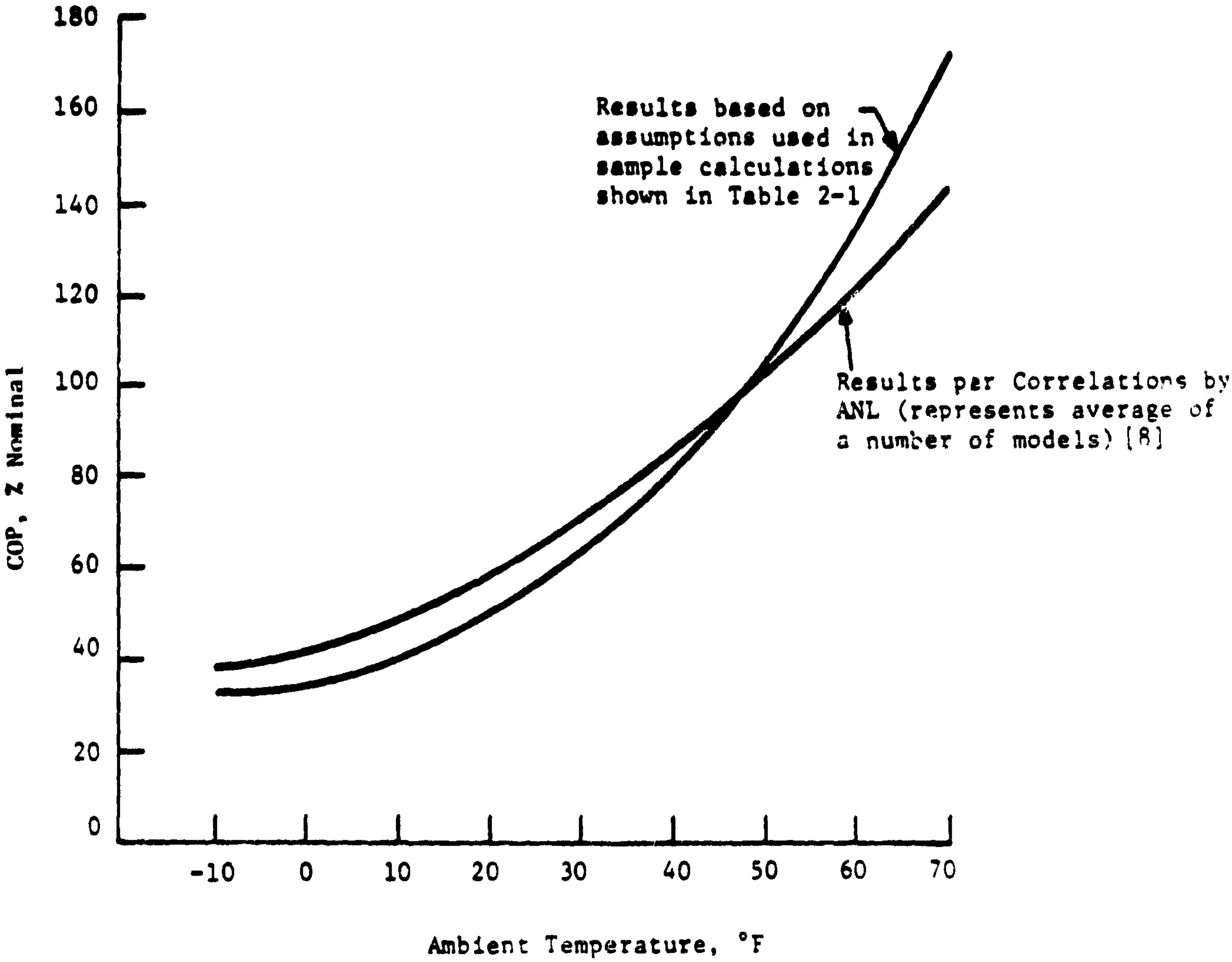

Fig. 2-7 Effect of Ambient Temperature on the COP of a Heat Pump (Heating Mode)

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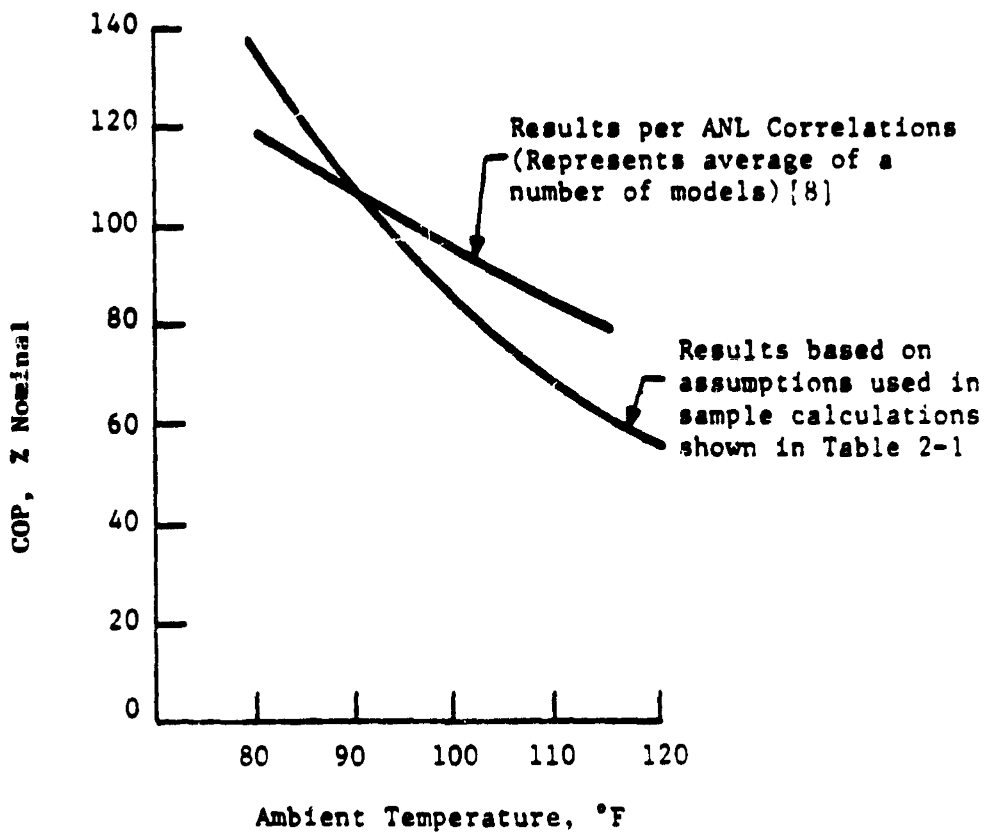

Fig. 2-8 Effect of Ambient Temperature on the COP of a Heat Pump (Cooling Mode)

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The volumetric efficiency of a compressor is a function of the pressure ratio and the clearance volume. Furthermore, the mass flow capacity of a compressor is a function of the compressor size, the volumetric efficiency, and the specific volume at the compressor suction. Hence, the heat pump capacity is a function of the ambient temperature. In this analysis, this capacity variation is obtained by calculations as well as by the use of ANL correlations, and is presented in Figures 2-9 and 2-10. The methods for calculations are standard and are obtained from References 9-13.

2.1.6 Costs

ANL analysis presents correlations for equipment costs. Since heat pumps are not presently being used in automotive applications, the values given are for residential units. For the 1-1/2-ton capacity unit, the cost may be obtained from the following equation:

$$\text{Equipment Cost} = 1400\ (CAP/3)^{0.86}$$

Where CAP = capacity in tons

The resulting costs are \$736 for the 5-kwt* unit, and \$184 for the 1-kwt unit.

The automotive units do not require a number of items of equipment (such as motors and cabinets) that the residential units require. Also the scale of manufacture and marketing are very favorable to the automotive units. These considerations are estimated to result in the automotive units costing only 25 to 50% of the corresponding residential units. Thus, the costs are estimated to be \$200 for the 5-kwt unit and \$90 for the 1-kwt unit. A more accurate estimation of these costs, although desirable, is not attempted in this task.

2.1.7 Weight

The weight of a heat pump unit is estimated from the data available for a commercial unit and is expected to be 30 to 50 lb per kwt.

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*kwt stands for thermal kilowatt.

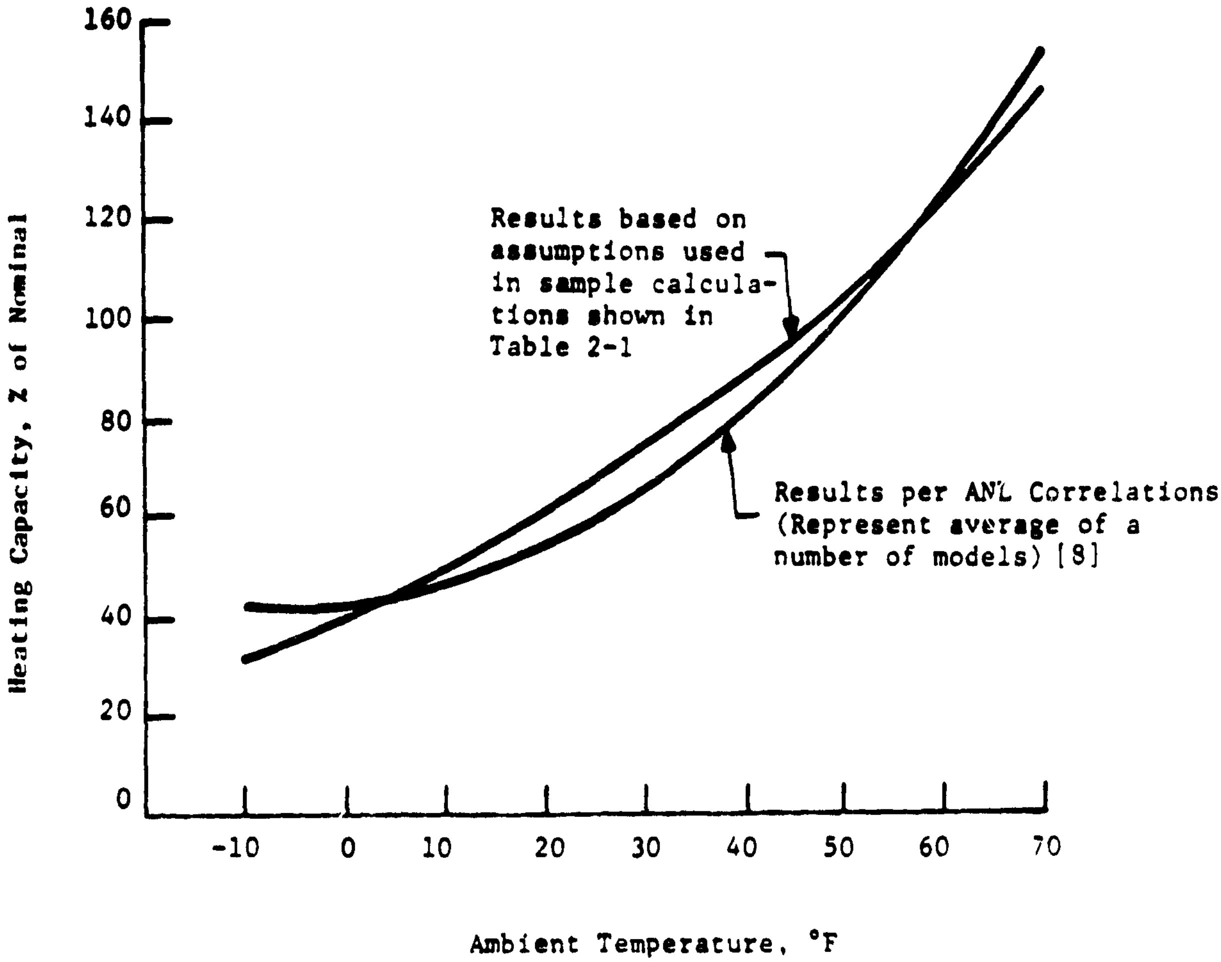

Fig. 2-9 Effect of Ambient Temperature on the Heating Capacity of a Heat Pump

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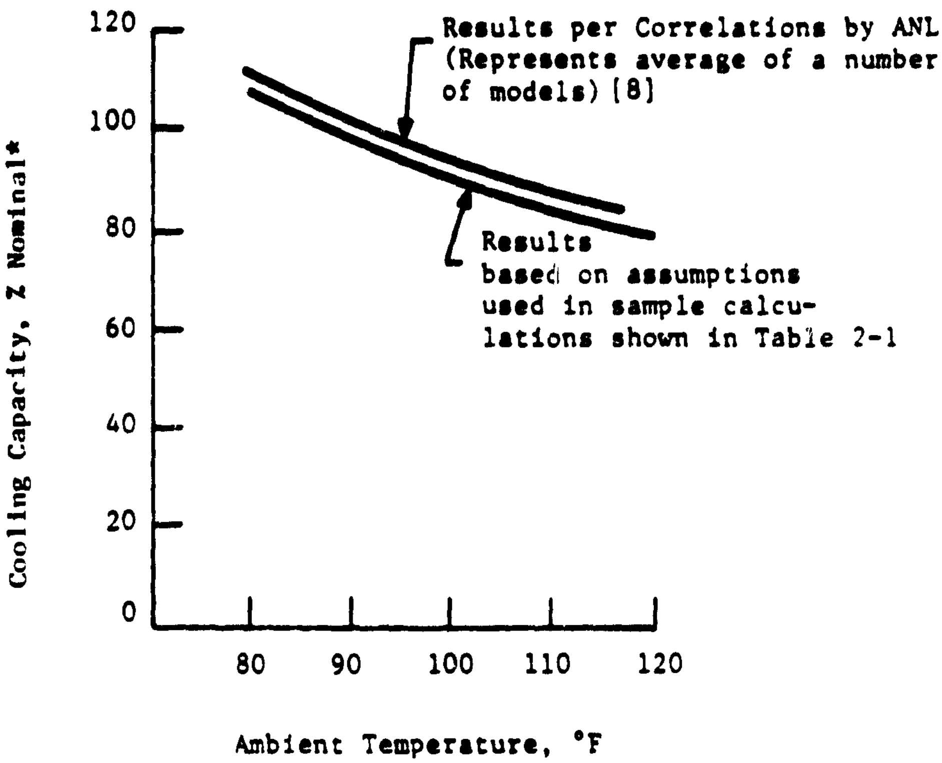

*Nominal cooling capacity is taken as equal to the heating capacity at 47°F, as done by ANL,

Fig. 2-10 Effect of Ambient Temperature on the Cooling Capacity of the Heat Pump

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2.1.8 Example of a Commercial Unit

Data on compressor specifications are available for York [14] units, and are shown in Table 2-2. Performance calculations are carried out and the capacity of the models is shown in the same table. The COPs of some commercial units are shown in Figure 2-11 which is adapted from Reference 15.

2.1.9 Limitations and Conclusions

In this brief study, reasonable values (based on current practice) are assumed for pressure drops and temperature drops. Detailed component analysis is not carried out to estimate the performance, weights or costs. The effects and types of controls are not addressed.

The results are, however, compared with correlations by ANL and are found to be satisfactory.

When these results are used for electric vehicles with motor driven compressors, motor efficiency should be included. If variable-speed, dc drives are used, the speed can be so chosen that the capacity is augmented to compensate for the ambient temperature effects.

2.1.10 Drive for the Heat Pump

From the above discussion and a few further calculations, a conclusion has been reached that to meet both heating and cooling requirements at the appropriate design temperatures, the drive for the heat pump compressor should have:

- 1.5 to 2.5 hp output power
- Variable speed capability over a speed range of 3:1.

The two types of drives that can be considered to meet the above requirements are:

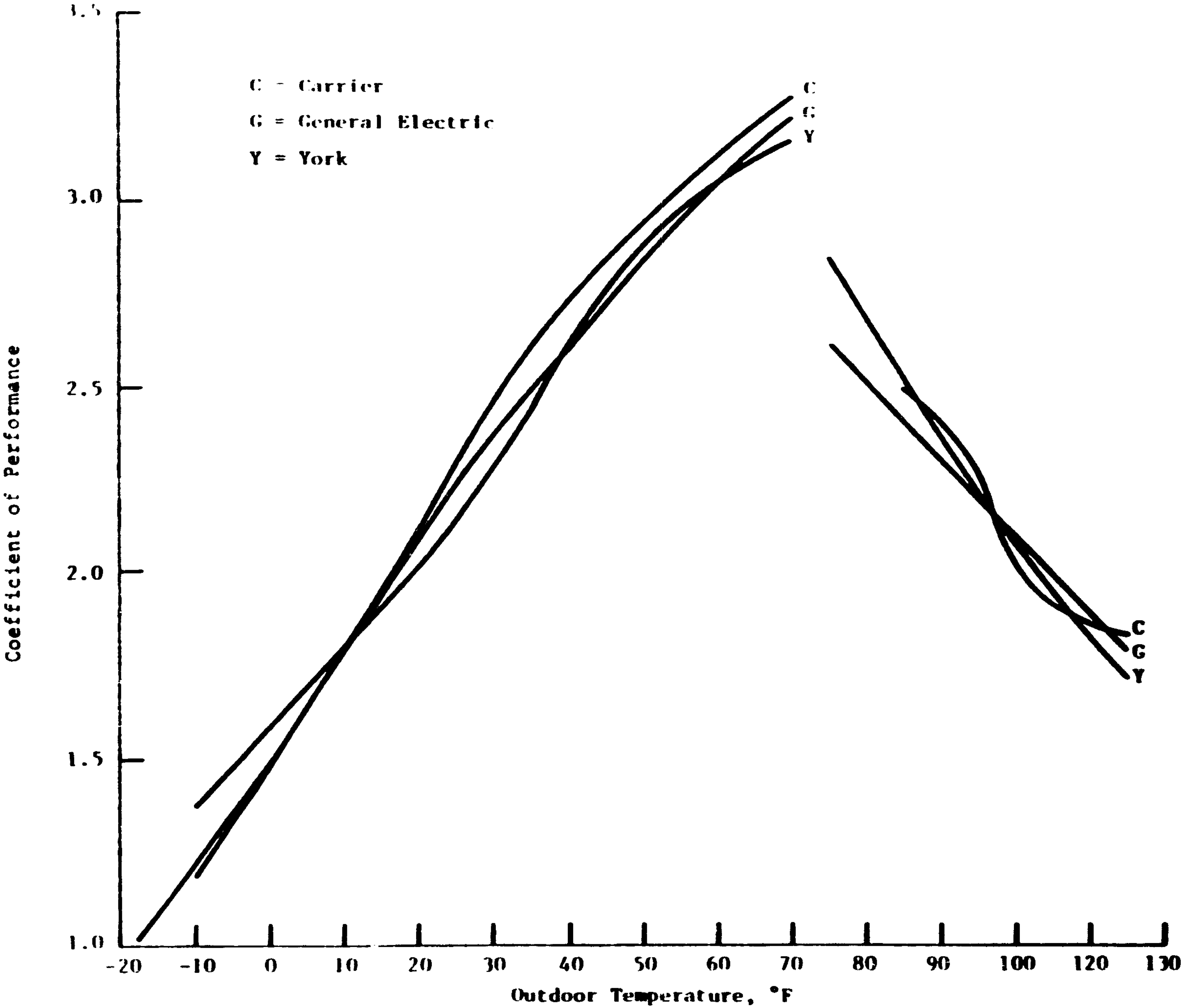
- Gasoline engine
- Electric motor, driven from the battery.

TABLE 2-2

YORK AUTOMOTIVE AIR CONDITIONING COMPRESSORS DATA AND PERFORMANCE CALCULATION (EXAMPLE)

Data			
Model Number	206	209	210
No. Cylinder	2	2	2
Bore, in.	1.875	1.875	1.875
Stroke, in.	1.105	1.573	1.866
Disp., $in.^3$/Rev.	6.11	8.7	10.3
rpm - Maximum	6000	6000	6000
Refrigerant	12	12	12
Initial Oil Chg., oz.	10	10	10
Weight, lb	14.6	14.6	14.6
Lubrication	Positive Pressure		

Performance (95°F Ambient)			
Maximum Capacity, cfm	21.2	30.2	35.8
Specific Volume at Compressor Suction, ft^3/lb	0.518	0.518	0.518
Maximum Mass Flow, lb/m	41	58	69
Volumetric Efficiency	0.84	0.84	0.84
Mass Flow at Maximum rpm, lb/m	34.4	48.7	58.0
ΔH Evaporator, Btu/lbm	54.0	54.0	54.0
Refrigeration Effect, Tons at			
Max. rpm	9.3	13.1	15.7
2000 rpm	3.1	4.4	5.2
1000 rpm	1.55	2.2	2.6
Power Input, hp at			
Max. rpm	5.5	7.7	9.2
2000 rpm	1.8	2.6	3.1
1000 rpm	0.9	1.3	1.5



Reproduced from Reference 15

Fig. 2-11 Heat Pump Heating and Cooling COP

Based on information given for characteristics of gasoline engines and dc motors available in the marketplace today, approximate numbers for total system weight and cost to provide 2-1/2 hours of environmental conditions at the rate of 17,000 Btu/hr can be derived. These numbers are given in Table 2-3.

2.2 Thermal Engine Heat Pumps*

Although the electric heat pump can be designed to have an electric motor hermetically sealed into the refrigerant loop to drive the vapor compression cycle compressor, clearly other means of providing compressor shaft power can be used as well. Indeed, using a prime mover such as a thermal engine as a power source has the considerable advantage in that engine waste heat rejected in coolant or exhaust gas can be used to supplement the refrigeration cycle output in the heating mode, thus substantially increasing the overall on-site system COP. On the other hand, the extra heat produced becomes a liability in the cooling mode in that it has to be rejected without significantly diminishing the refrigerating capacity of the system.

This section discusses a number of new heat pump concepts using an on-site thermal engine as a prime mover to drive a refrigeration machine. In most cases of interest, the latter is of the usual vapor compression cycle type; in some cases, a cycle operating entirely in the gas phase is treated.

2.2.1 Aspects of Design and Operation

Some generally applicable considerations in the design and operating characteristics of thermal engine heat pumps will be noted before discussion of specific heat pump concepts in Section 2.2.2. Use has been made of a number of comprehensive discussions on this subject published by Wurm and Rush (1975), Wurm et al. (1976), Colosimo (1976) and an AGA research project report edited by Wurm (1974).

*Much of the material in this section has been adopted from Reference 15.

TABLE 2-3

HEAT PUMP SYSTEM WEIGHTS AND COST

Drive Type Weights	Gasoline Engine	dc Motor
Heat Pump System	150 lb	150 lb
Drive	30 lb	60 lb
Energy Storage	10 lb	470 lb*
Total System	190 lb	680 lb
Costs		
Heat Pump System	$200	$200
Drive	100	200
Energy Storage	10	400
Total First Cost	$310	$800

* Ni-Zn batteries.

2.2.1.1 Advantages

Depending on the particular prime mover or refrigeration cycle contemplated, thermal engine heat pumps have a number of significant advantages in comparison to, say, electric heat pumps. The major advantages are listed below:

- High overall efficiency (in a primary energy sense) in comparison to state-of-the-art electric heat pumps in a heating mode and gasoline burners
- Ability to utilize prime mover waste heat to supplement refrigeration cycle on the heating mode
- Capability to modulate heat pump capacity by varying fuel input rate to the prime mover
- Availability of on-site waste heat for evaporator defrost
- Adaptability to total-energy systems or for operation independent of utility lines.

Some of these characteristics are discussed further below:

The major inducement to thermal engine heat pump development is, of course, the high combination engine efficiency and refrigeration cycle coefficient of performance obtainable in principle. Waste heat utilization on-site, facilitated defrost, and ability to modulate capacity all add to high seasonal performance in the heating mode. In the cooling mode, electric heat pumps have certain advantages, but a thermal engine heat pump designed to "hold its own" on cooling as well is also possible.

Prime movers, especially turbomachinery, are relatively easily speed-modulated by controlling fuel flow to the combustor or bypassing some of the power fluid around the prime mover.

Having a source of waste heat on site not only makes supplemental heat available for augmentation of the output but allows its use for efficient evaporator coil defrosting in the heating mode. This consideration may potentially eliminate what is frequently a major reliability problem with electric heat pumps as well as a cause of some efficiency loss.

2.2.1.2 Disadvantages

Significant disadvantages applicable in general to thermal engine heat pumps, in contrast to electric heat pumps or gas or oil burners, are listed below. As before, these disadvantages apply to varying degrees, depending on the particular system being considered.

- Requirement to reject prime mover waste heat in the cooling mode
- Greater system complexity which could result in potentially higher system cost and lower reliability
- Difficulty in designing totally hermetic refrigeration system
- Aggravated noise and pollution problems due to on-site prime mover operation and fuel combustion
- Dependence on availability of scarce fossil fuels (natural gas, fuel oil).

In elaboration of some of the disadvantages listed above, a note should be made that the act of having a fuel-burning prime mover on site, while highly advantageous in the heating mode, becomes a decided liability on cooling in that additional heat transfer sur.ace, and pump or fan energy is required to reject engine waste heat.

With an on-site prime mover, design of a hermetically sealed refrigerant loop becomes difficult.

2.2.2 MTI Heat-Activated Heat Pump

At MTI the problem of a hermetically sealed refrigerant loop has been solved by using an interesting concept. In this diaphragm concept, a free-piston Stirling engine drives a linear oscillating Freon compressor heat pump operating on a vapor compression cycle. The Stirling-cycle engine has been selected because of its potentially high level of efficiency. R. Ackermann of MTI [16] has described this system as follows.

The diaphragm concept revolves around separating the working fluids in the engine and compressor with a flexible metal diaphragm, and hydraulically transferring the power output from the engine to the compressor. A schematic layout for this system is shown in Figure 2-12. The system will consist of two moving subassemblies: the engine displacer, and the Freon compressor assembly which consists of the engine power diaphragm, the compressor, a gas spring diaphragm, and the intervening hydraulic oil. These two subassemblies form a coupled resonant system that functions as a conventional Stirling engine; i.e., the displacer shuttles the working fluid between the hot and cold spaces generating the driving pressure wave for the system, and the power piston (compressor) extracts work from the gas to power the load. In the diaphragm system, the power is extracted from the engine and delivered to the compressor through the hydraulic fluid which provides a force/displacement transfer path between the engine and compressor. This coupling also applies the proper dynamics to the engine to give it its resonant characteristics and volumetric phase relationships.

The benefits to be derived from this system are:

- The separation of the working fluids eliminates the need for the spring tube, improving both the cost and performance of the compressor.
- The use of the diaphragm eliminates the power piston and the intricate gas bearing design associated with it, and replaces it with a less costly lubricated bearing on the compressor.
- The use of the diaphragm has reduced the system from a three-degree-of-freedom system to a two-degree-of-freedom system, thus improving the operating control of the system.

In order to realize these benefits, diaphragm fabrication must be reliable and less expensive than the spring tube assembly. A diaphragm development program has been started that will:

- Identify industrial applications and diaphragm manufacturers
- Determine the fatigue characteristics of diaphragms

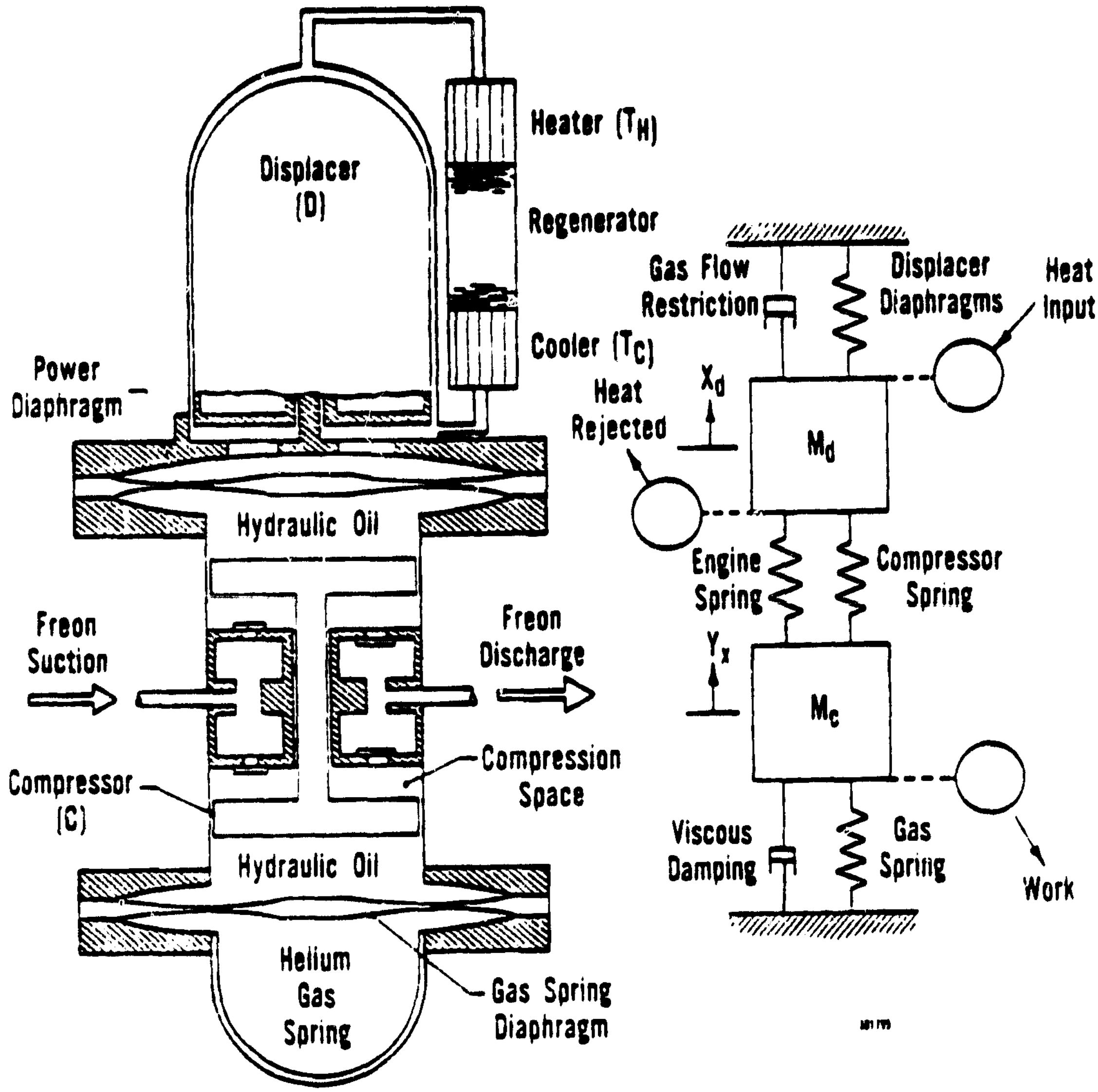


Fig. 2-12 Diaphragm Stirling Engine Heat Pump Power Module

- **Establish design procedures and develop computer design codes.**

The progress that has been made on this program is that several applications ranging from reciprocating compressors to transmission couplings for high-speed rotating machines have been identified and several diaphragm manufacturers have been consulted. Subcontracting arrangements have been established with two of these companies to manufacture the power diaphragm.

The first of these diaphragms was scheduled for delivery in June 1980 at which time a life testing program was initiated at MTI.

Ackermann has made calculations to verify the suitability of this kind of heat pump for electric vehicle application. Figure 2-13 shows the system weight and cost. Figure 2-14 shows the volume of the engine compressor assembly as a function of design point cooling capacity based on the following assumptions.

- Ambient temperature - 95°F
- Passenger compartment temperature - 72°F

At a design point capacity of 17,000 Btu/hr (5 kW) some of the operating characteristics of this system are shown in Table 2-4.

2.3 Absorption Cycle Heat Pumps

The absorption refrigeration cycle is similar to a conventional refrigeration cycle in its principle, i.e., a two-pressure domain system.

Figure 2-15 illustrates the basic vapor compression refrigeration cycle consisting of four components: compressor, condenser, evaporator and expansion valve. The condenser operates at a higher pressure than the evaporator, condensing the refrigerant at higher temperature, and thus rejecting heat. The refrigerant then passes through an expansion valve to the evaporator which is at a lower pressure than the condenser and therefore at a lower temperature, thus enabling the refrigerant to absorb heat. The two pressure domains are separated by the compressor and the expansion valve.

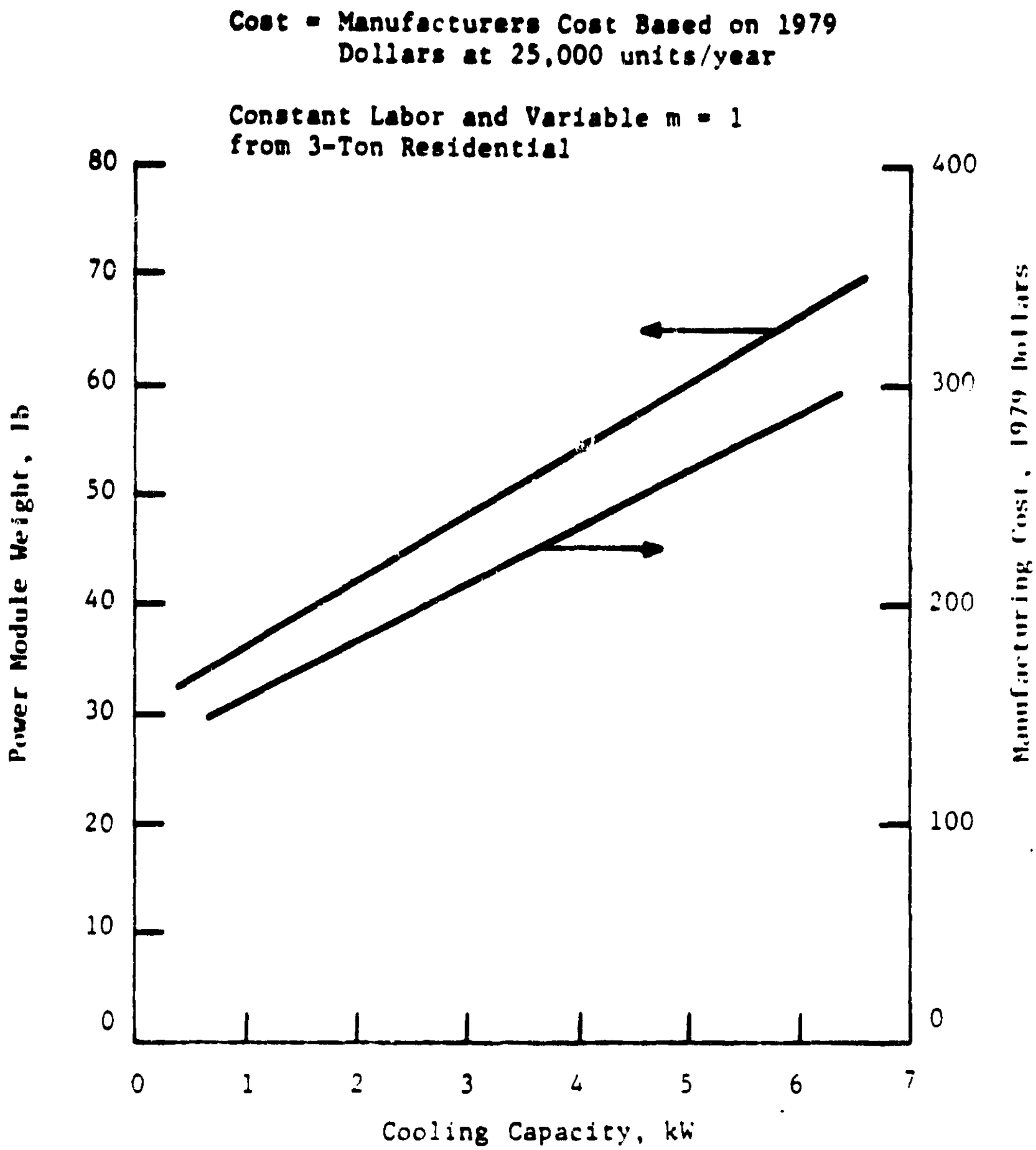


Fig. 2-13 Weight and Cost of Stirling Engine Automobile Heat Pump Power Module

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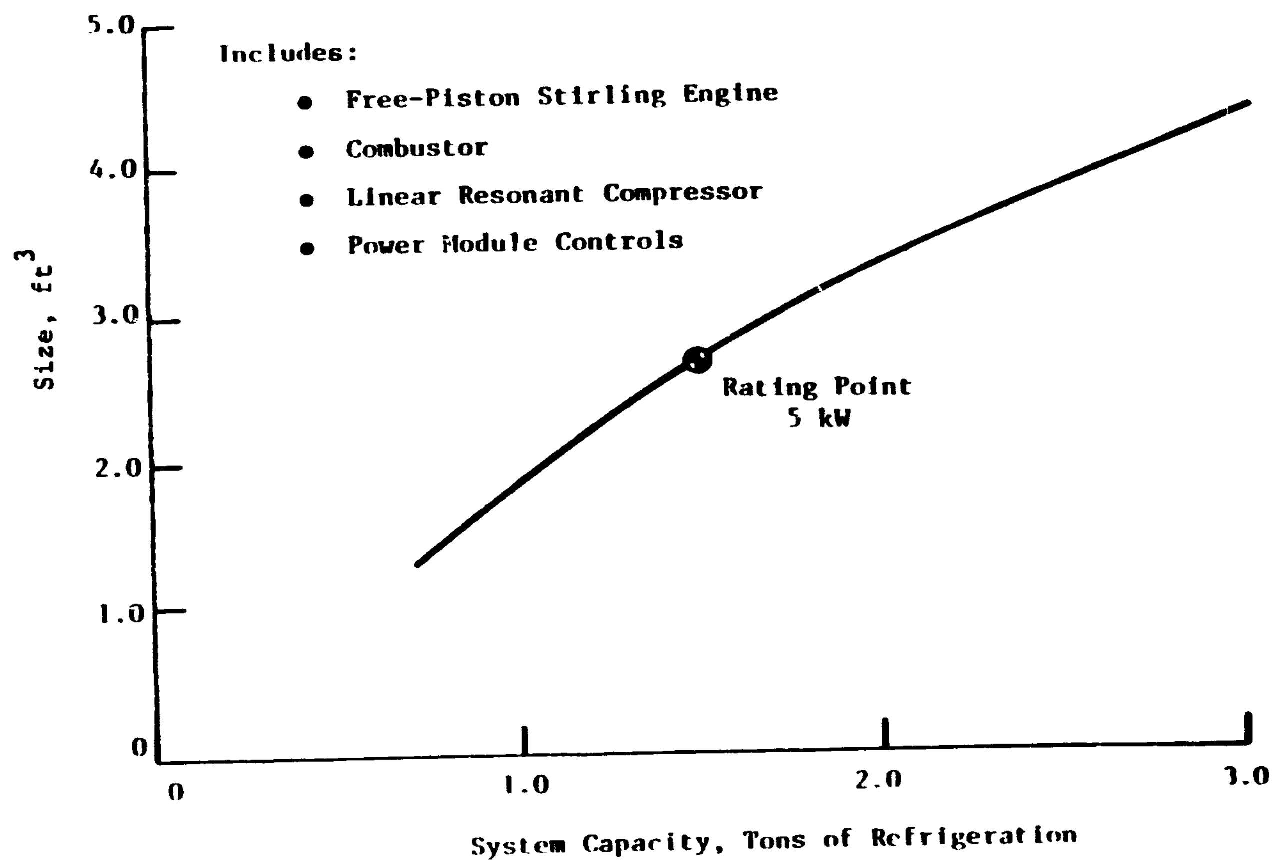

Fig. 2-14 Free-Piston Stirling Engine Heat Pump Power Module Size

TABLE 2-4

DUAL MODE OPERATING POINT DATA FOR A 5-kW RATED HEAT-ACTIVATED HEAT PUMP

Operating Point (°F)	Heat Exchanger ΔT (°F)		Efficiencies			Fan Power (watts)
	Indoor	Outdoor	Engine	Combustor*	Compressor	
95	25	30	0.35	0.75	0.80	500
80	12.5	16	0.40	0.80	0.85	250
45	18	12	0.40	0.80	0.85	250
25	28	5	0.35	0.75	0.80	500

*Lower Heating Values

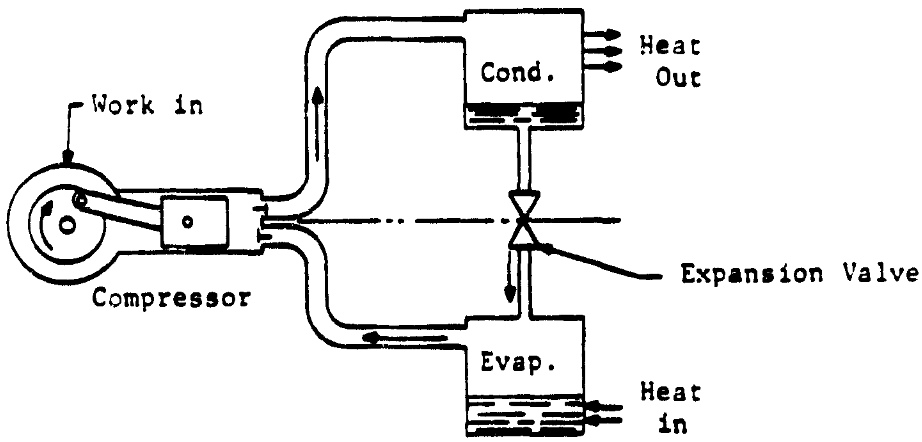

Fig. 2-15 Basic Vapor Compression Refrigeration Cycle

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The absorption cycle, Figure 2-16, consists of four basic components: absorber, generator, condenser, and evaporator. The condenser and evaporator are identical to the vapor compression cycle, while the generator and absorber replace the compressor in the vapor compression cycle. Compression of the refrigerant operates on the physical chemistry principle that certain liquids and liquid solutions have the capability of absorbing a vapor or gas. The weight of the gas or vapor which can be absorbed is directly proportional to the pressure and inversely proportional to the temperature of the liquid. Therefore, the cool absorber absorbs the refrigerant vapor and the solution is pumped into the generator where heat energy (instead of the work in the compressor) is applied, thus raising the temperature of the solution. Consequently, the refrigerant is relieved into the condenser at higher pressure.

Many combinations of refrigerant-absorbent are available commercially. The most used combinations in the industry today for which data are available are the ammonia-water combination, where the ammonia is the refrigerant, and the water-lithium bromide combination, where the water is the refrigerant. The latter of the two is used in this feasibility study since it is the only one operating successfully in cooling a small residential unit, and as such, is the closest to the motor vehicle requirement.

2.3.1 Analysis of Absorption Cycle

The analysis of a practical absorption cycle consists of heat balance and material balances around the various components as well as around the system as a whole (See Figure 2-17).

The following equations are set per unit mass of refrigerant flowing per unit time [17].

Heat balance of the whole system:

$$q_e + q_g + q_p - q_a - q_c = 0. \qquad (2.1)$$

Heat balance around evaporator:

$$q_e + h_2 - h_4 = 0. \qquad (2.2)$$

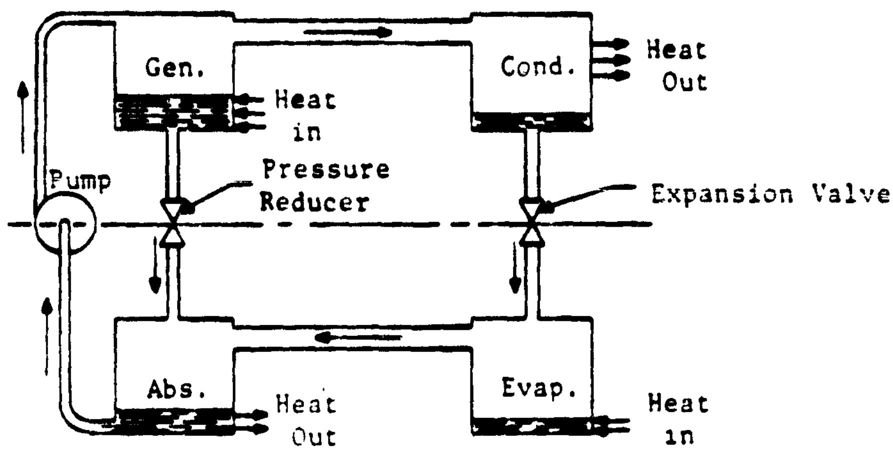

Fig. 2-16 Basic Absorption Refrigeration Cycle

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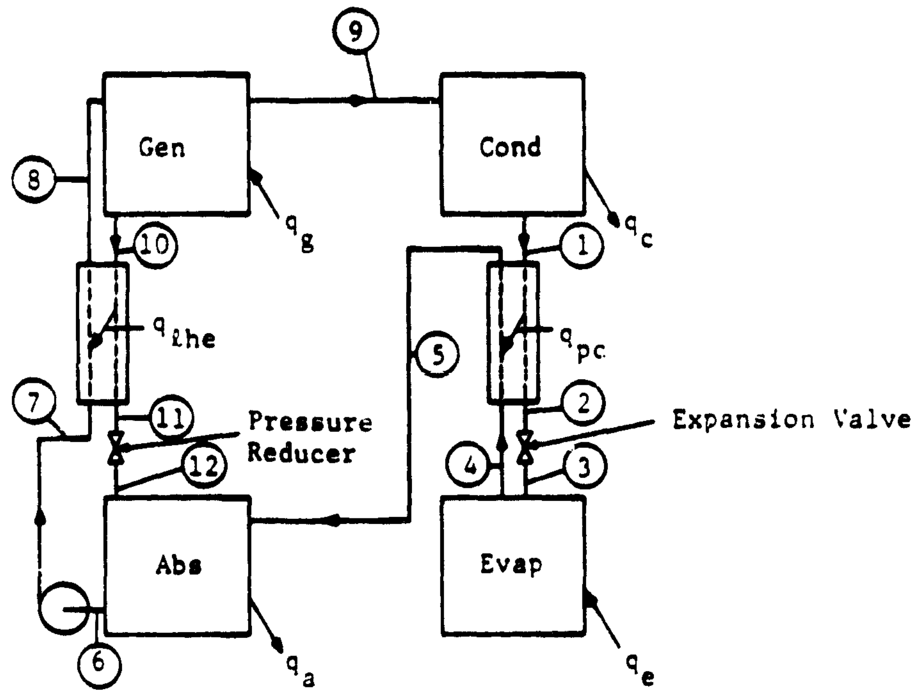

Fig. 2-17 Basic Absorption Refrigeration Cycle for Cycle Analysis

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Heat balance around absorber:

$$-q_a + h_5 + Mh_{12} - (M+1)\ h_6 = 0 \quad . \tag{2.3}$$

Heat balance around generator:

$$q_g + (M+1)\ h_8 - Mh_{10} - h_9 = 0 \quad . \tag{2.4}$$

Heat balance around condenser:

$$-q_c - h_1 + h_9 = 0 \quad . \tag{2.5}$$

Heat balance around liquid heat exchanger:

$$q_{\ell he} = M(h_{10} - h_{11}) = (M+1)\ (h_8 - h_7) \quad . \tag{2.6}$$

Heat balance around refrigerant precooler:

$$q_{pc} = h_1 - h_2 = h_5 - h_4 \quad . \tag{2.7}$$

Where:

q_e = heat flow to evaporator

q_g = heat flow to generator

q_p = heat equivalent of pump work

q_a = heat flow from absorber

q_c = heat flow from condenser

h_i = enthalpy of i th stream (heat per unit refrigerant mass)

M = mass of absorbent solution entering absorber (mass per unit mass of refrigerant flowing through the evaporator)

$q_{\ell he}$ = heat exchange flow of liquid

q_{pc} = heat exchange flow of refrigerant

Calculating the coefficient of performance (COP) of water-lithium bromide is accomplished by using Equations 2.1 through 2.7, charts from Reference 17 and steam tables. The following assumptions were made based on industry components design:

- **The absorbent solution leaves the absorber at a temperature of 80°F (point 6).**
- **The absorbent solution enters the absorber (point 12) at a concentration of 60% lithium bromide and has been cooled in the liquid heat exchanger within 10°F of the temperature of solution leaving the absorber.**
- The condensing temperature is 120°F (points 9 and 1).
- The vapor leaving the precooler is warmed to within 10°F of the temperature of the liquid refrigerant leaving the condenser.

Table 2-5 shows the conditions for the cycle analysis. Using this table, the following values can be found:

$$q_e = 1077 - 45 = 1032 \text{ Btu/lbm}$$

$$-q_a = -698 - 1109 + 615 = -1192 \text{ Btu/lbm}$$

$$q_g = 395 - 311 + 1142 = 1226 \text{ Btu/lbm}$$

$$-q_c = 88 - 1142 = -1054 \text{ Btu/lbm}$$

Therefore, the COP is:

$$\frac{q_e}{q_g} = \frac{1032}{1226} = 0.842$$

2.3.2. Industry Survey

Several companies in the USA manufacture absorption cooling systems. However, most of them are involved in large units, beyond the requirement of an automotive vehicle. The only one - to MTI's knowledge - that manufactures small units with the smallest capacity of two tons of refrigerations is ARKLA of

TABLE 2-5

CONDITIONS OF CYCLE ANALYSIS

Point*	Temp. (°F)	Pressure (mm Hg)	Weight Fraction (LiBr)	Enthalpy, h (Btu/lbm)	Mass Flow Rate(lbm/unit time)	Heat Flow Rate (Btu/unit time)
1	120	90	0	87.92	1.0	87.92
2	77	90	0	45.3	1.0	45.3
3	40	6.5	0	45.3	1.0	45.3
4	40	6.5	0	1077	1.0	1077
5	110	6.5	0	1109	1.0	1109
6	80	6.5	0.508	-76	9.19	-698
7	80	90	0.508	-76	9.19	-698
8	145	90	0.508	-43	9.19	-395
9	190	90	0	1142	1.0	1143
10	190	90	0.60	-38	8.19	-312
11	90	90	0.60	-75	8.19	-615
12	90	6.5	0.60	-75	8.19	-615

*Refer to Figure 2-17.

Indiana. Their units have a COP of around 0.8 with a water-lithium bromide combination. The physical sizes of the smallest ARKLA absorption units are:

- 2-Ton Unit - Weight, 450 lb and Volume, 55 ft^3
- 3-Ton Unit - Weight, 680 lb and Volume, 60 ft^3

2.3.3 Absorption Cycle Conclusions

Absorption cycle systems are used primarily in areas where heat is available as a by-product which will be wasted otherwise. In electric vehicles, where energy is at a premium, the absorption cycle low COP is very unattractive. In the industry, the vibration of the absorber unit is known to reduce substantially its absorptivity, which will further reduce the COP. Furthermore, the system's physical size and weight will further deteriorate the vehicle performance on the road.

2.4 Thermoelectric Heat Pumps

Thermoelectric devices are widely used for cooling electronic equipment. These devices are also used in various other equipment such as beverage coolers, mobile refrigerators, medical instruments and cooling of infrared detectors. In all such applications, heat rejection rates are small, of the order of a few hundred Btu/hr. Commercial devices can produce ΔT of 70°C between the hot and cold face.

2.4.1 Operating Principle

The following description of the thermoelectric system operating principle is taken from Reference 18.

> Thermoelectricity is a specialized branch of solid-state technology, and thermoelectric heat pumps have only come into their own as practical, mass-producible devices in the last few years. The basic cooling element employed in these devices is the Peltier couple; a thermoelectric cooler's cooling capacity is proportional to the number of such couples it contains and the magnitude of the current passing through them.
>
> A Peltier couple consists of two semiconductor elements, alloyed from bir h, tellurium and other compounds that are doped to form p- and pe units. At the top end, the elements are soldered to a ..non copper strap, while at the bottom, they are soldered

to individual copper straps to which electrical connections are made. The common top strap functions as the cold end in the cooling mode.

In an actual device, thin ceramic insulating plates of alumina or beryllia, which prevent shorting of the copper connections yet provide good heat-transfer capability, are soldered to the copper straps. (Except in special high-efficiency devices, manufacturers are discontinuing the use of beryllia because it is toxic in powder form.) In a cooling application, the thermoelectric device's cold-side insulating plate mates to a heat sink or other heat-transfer element, using standard heat-sinking techniques such as thermal grease and clamping pressure. The hot-side plate mates to the load to be cooled.

When dc current is applied to the Peltier couple, it passes from the n- to the p-type semiconductor material; the temperature of the common copper strap decreases, and heat is absorbed. Junction cooling occurs because the electrons pass from a low energy level in the p-type material, through the connecting copper tab, to a higher energy level in the n-type element; the heat is pumped from the cold end through the elements by the means of electron tansport to the opposite ends, which grow hotter.

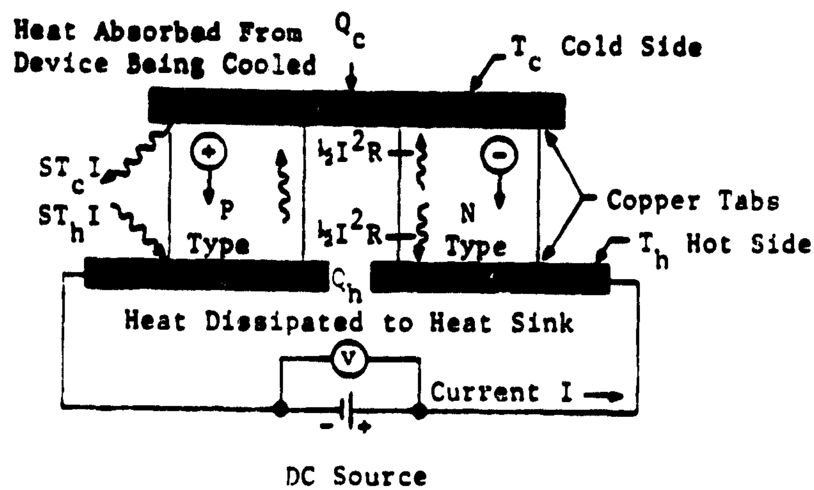
Ideally, 100% of the cold junction's heat-absorbing capacity, which is proportional to the product of the Peltier coefficient and the current, is available to soak up the heat from the load. But in practice, not all of this capacity is actually available because some of it gets used up in absorbing the heat from two internal sources: the Joule (I^2R) heat (Q_j) generated by current flow through the Peltier-couple resistances, and the heat transmitted by thermal conduction (Q_t) from the hot end back to the cold end. Thus, the net heat (Q_c) that can be usefully absorbed is $Q_p - (Q_j + Q_t)$.

The heat dissipated at the cooler's hot side (Q_h) is the sum of the Peltier heat plus half the Joule heat, minus the thermally conducted heat: $Q_h = Q_p + 1/2Q_j - Q_t$. The cooler's electrical input power (P_i) is the difference (measured in watts) between the amount of heat dissipated at the hot side and the net amount of heat absorbed at the cold junction: $P_i = Q_h - Q_c$. [See Figure 2-18.]

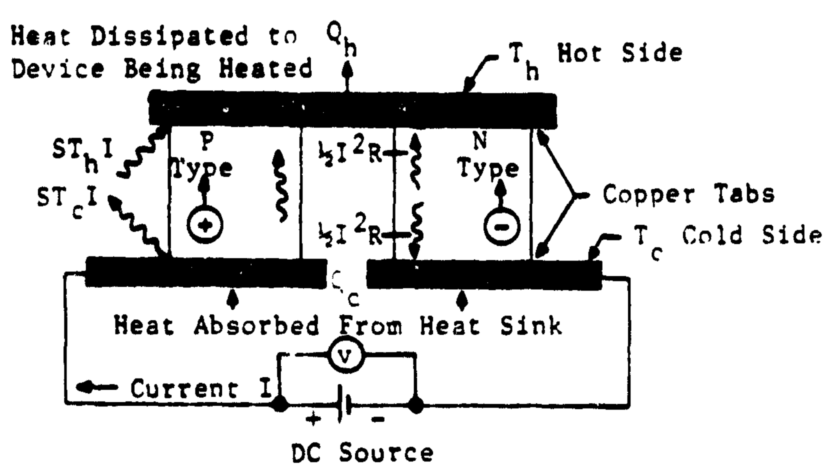
2.4.2 EHV Application

For electric/hybrid vehicles, the areas of most interest are heat rejection rates in the range of 1 to 5 kW (3413 to 17,065 Btu/hr).

Westinghouse has developed a module for providing 24,000 Btu/hr of air conditioning for environmental control of remote shelters. The complete system



(a) Cooling Mode



(b) Heating Mode

Fig. 2-18 Thermoelectric System Operation package consisting of thermoelectric modules, heat exchangers, fans etc. weighs 262 lb. The package's overall dimensions are 40 in. x 40 in. x 20 in. The COP for this system is 1 for T_{cold} = 95°F and T_{hot} = 140°F. The specifics of this package are described in detail by Percupile et al. [19].

Westinghouse has also developed thermoelectric air conditioning (heating and cooling) for shipboard use in capacities of 6000 Btu/hr. However, these units are expensive; on the order of $10,000 to $15,000. These cost figures include the cost of heat exchangers, power supplies, etc.

A more realistic idea of the cost of thermoelectric devices is obtained by examining the catalogs for these devices. For example, Melcor company's device #CP5-31-06 has a heat pumping capacity of 200 Btu/hr with T_{cold} = 40°F and T_{hot} = 122°F. This device has a list price of $32 each for quantities of 1000 or more. The price does not include the cost of heat exchangers.

The characteristics of thermoelectric devices to provide required air conditioning capacity would be:

- T_{hot} - 122°F
- T_{cold} - 40°F
- Capacity - 5 kW (17,065 Btu/hr)
- Cost - $2720
- Volume - 0.0525 ft^3
- Weight < 150 lb
- COP ≈ 0.244 .

The above cost figures are based on present production rates. With mass-production techniques, cost may be significantly lower than the above figures.

Therefore, the thermoelectric system will not be cost competitive if it is to produce identical heat rejection rates as required by conventional systems.

2.4.3 Localized Cooling/Heating

The small size of thermoelectric devices can be exploited for providing localized cooling which will provide some level of comfort to individuals. Cooling would be achieved with a heat rejection rate requirement reduced to a tenth or less of the conventional space conditioning technique. In such an application, thermoelectric devices will be superior to any other technique by the following:

- Least cost
- Least energy consumption
- Least weight
- No effect on vehicle drag.

One such technique would be to make a "weather coat", which has thermoelectric devices spread throughout. Such a coat would provide heating or cooling as desired, by passing currents of appropriate polarity through the thermoelectric elements. Alternatively thermoelectric devices could be incorporated in seat belts and seat cushions.

Due to closer contact with the source of heat (i.e., human body), the temperature differential over which heat needs to be pumped is small. Assuming T_{cold} = 80°F and T_{hot} = 110°F, a heat pumping rate of 600 Btu/hr per person would be adequate, as only the heat generated by the human body (plus a little heat flowing in due to conduction when ambient temperature is higher than 80°F) is to be pumped out. At these conditions, the thermoelectric device system will have the following characteristics:

- Capacity - 176 watts/person (600 Btu/hr)
- T_{hot} - 110°F
- T_{cold} - 80°F
- ΔT - 30°F (16.7°C)
- Weight - 0.4 lb
- COP - 1.32
- Electric power required - 0.23 kW/person
- Cost ≈ $30/person .

Additional equipment required would be blowers, fans, control system, etc.

2.4.4 Combination Systems

The small size and weight of the thermoelectric devices can be exploited to provide for charging of thermal energy storage systems. As charging time could easily be of the order of 10 hours to charge (either heat or cool), the storage system with 42,500 Btu of heat would require an average charging rate of 4250 Btu/hr. The characteristics of thermoelectric devices for providing such rates will be:

- Capacity - 4250 Btu/hr (1.241 kW)
- $T_{hot} = 95°F$
- $T_{cold} = 0°F$
- $\Delta T = 100°F$ (55.5°C)
- Weight = 60 lb
- Volume = 0.026 ft^3
- Cost ≈ $1360

2.5 Magnetic Heat Pump

The working principle of magnetic heat pumps, known for more than three decades, is based on the fact that many magnetic materials become warmer when subjected to a magnetic field, and cooler when the field is removed. This reaction is called the thermomagnetic effect. Recently, NASA physicist Dr. G.V. Brown used a material called Gadolinium to make this reaction produce heating and cooling effects near room temperature. The device - a magnetic heat pump - is believed to be a more efficient alternative to existing methods. Its use promises significant potential savings in equipment, energy, and peripheral operating costs for many industrial, commercial, and residential applications. The physics behind the operation of such a magnetic heat pump are well described by Dr. Brown [20].

Dr. Brown made a few sample calculations to determine the feasibility of a magnetic heat pump utilizing Gadolinium for electric vehicle applications. These calculations were not attempted to obtain an optimized engineering design but rather to obtain order of magnitude estimates.

Assumptions (for summer):

- **Temperature at which heat is absorbed by the heat pump = 50°F**
- **Temperature at which heat is rejected to the ambient = 110°F**
- **Cost of Gadolinium = $75/lb**
- **Cost of ferrite permanent magnet material = $0.20/lb**
- **Rate of cooling required = 17,000 Btu/hr.**

The results of these calculations are given in Table 2-6.

Significant opportunities exist to reduce the weight of the ferrite permanent magnet required by optimizing the magnetic circuit configurations and also increasing the frequency of operation.

2.6 Split Heat Pump

Two split heat pump systems are described here. One system uses water as the refrigerant, the other uses ammonia.

2.6.1 Water-Lithium Bromide System for Air Conditioning

Water has the highest latent heat of vaporization per unit mass of any known substance. This property can be utilized to obtain a very lightweight air-conditioning system for electric vehicle application. Figure 2-19 shows the schematic of the arrangement.

Water is stored in a tank at ambient temperature and pressure. This water is passed through an expansion valve and its pressure is dropped to a very low absolute pressure, e.g., 0.12 psia. This decrease in pressure results in flash boiling of water and a temperature drop to 40°F. The mixture of water vapor and water is then passed through a heat exchanger located in the passenger compartment. Here, the low-pressure, steam-water mixture receives heat from the passenger compartment, and the water is completely evaporated (and even partly forms superheated steam). The water vapor is further passed into a container holding lithium bromide. Lithium bromide absorbs the water vapor, thus retaining constant pressure on the low-pressure side

TABLE 2-6

RESULTS OF SAMPLE FEASIBILITY CALCULATIONS

Material	Weight (lb)	Cost ($)
Gadolinium	11	825
Ferrite	319	64
Total	330	889

Performance

Frequency	Capacity	COP
5 Hz	17,000 Btu/hr	3.7
2 Hz	7,850 Btu/hr	6.0

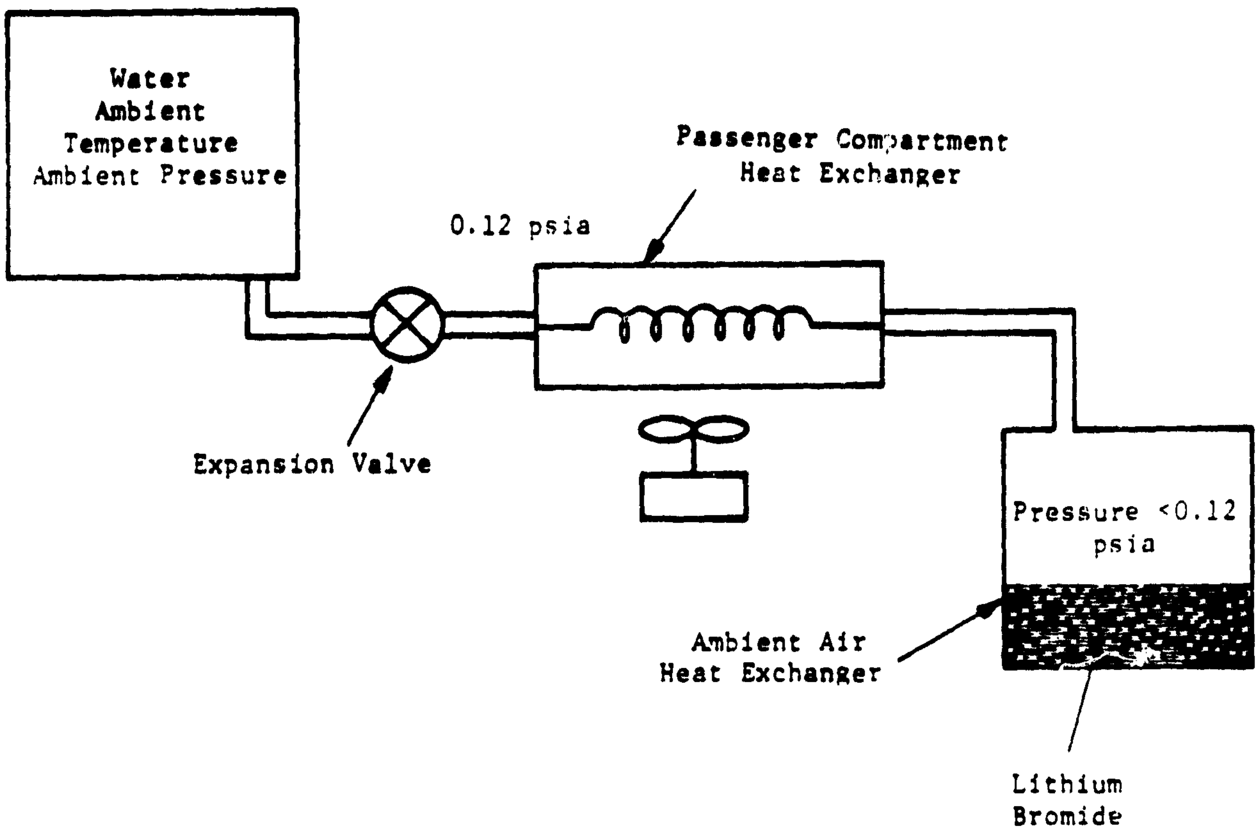


Fig. 2-19 Schematic of Water-Lithium Bromide Arrangement

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of the system. During absorption, the heat of the solution is liberated, raising the temperature of the solution and hence the container. This container is made in the form of a heat exchanger, and therefore, the liberated heat of solution is removed by blowing ambient air on the container. As a result, the solution is at a few degrees above ambient and at a very low pressure, such as 0.12 psia.

The various pressures, and amounts of water and lithium bromide are calculated using data from steam tables and the data on the properties of lithium bromide aqueous solutions from the ASHRAE Handbook, 1977 Fundamentals. These calculations are shown in Section 2.6.1.1.

For recharging, the lithium bromide aqueous solution is heated to a temperature that essentially drives off all the water from the solution. The tank is then sealed, resulting in a vacuum of appropriate magnitude when the tank is cooled.

Calculations in Section 2.6.1.1 show that the weight of the system is very small. These calculations will have to be modified to provide adequate rates of absorption. Certain problems will arise due to vibration and the high vacuum needed.

2.6.1.1 Calculations for System Size Determination

Assumptions

• Ambient temperature	100°F
• Ambient pressure	14.7 psia
• Temperature of cooling coil in the passenger compartment	40°F
• Temperature of absorber heat exchanger	120°F
• Storage	
- Temperature	100°F
- Pressure	14.7 psia
- Enthalpy	68 Btu/lb from steam tables [21]

- **After Expansion**
  - Temperature — 40°F
  - Pressure — (6.3 mm Hg) 0.121 psia } From steam tables [21]
  - Enthalpy — 68 Btu/lb
- After Evaporation
  - Temperature — 40°F
  - Pressure — 0.121 psia
  - Enthalpy — 1079.0 Btu/lbm } (Assuming saturated vapor). From steam tables [21]
  - Volume — 2445.8 ft^3/lbm
- After Absorption
  - Temperature — 120°F
  - Pressure — 0.121 psia

From Figure 2-20:

Equilibrium concentration, 63% by weight of LiBr.

Hence, the amount of heat removed from the passenger compartment per lbm of water

= Enthalpy after evaporation - enthalpy before evaporation

= 1079 - 68

= 1011 Btu/lbm of water

Thus, the amount of water required to provide a total cooling of 42,500 Btu is

$$\frac{42,500}{1011} = 42 \text{ lb.}$$

The amount of lithium bromide required is

$$\frac{42 \times 0.63}{(1 - 0.63)} = 71.5 \text{ lb.}$$

EQUATIONS

1. $t = At' + B$
2. $t' = (t - B)/A$
3. $A = -2.00755 + .16976X - (3.133362E\text{-}3)X^2 + (1.97668E\text{-}5)X^3$
4. $B = 321.128 - 19.322X + .374382X^2 - (2.0637E\text{-}3)X^3$
5. $\log P = C + D/(t'+459.72) + E/(t'+459.72)^2$
6. $t' = \dfrac{-2E}{D + [D^2 - 4E(C - \log P)]^{0.5}} - 459.72$

TEMP. RANGE (REFRIGERANT) $0 \geq t' \leq 230$ F
TEMP. RANGE (SOLUTION) $40 \geq t < 350$ °F
CONCENTRATION RANGE $45\% \geq X < 70\%$

$C = 6.21147$
$D = -2886.373$
$E = -337269.46$
t' = REFRIGERANT TEMP °F
t = SOLUTION TEMP °F
X = PERCENT LIBR
P = PSIA

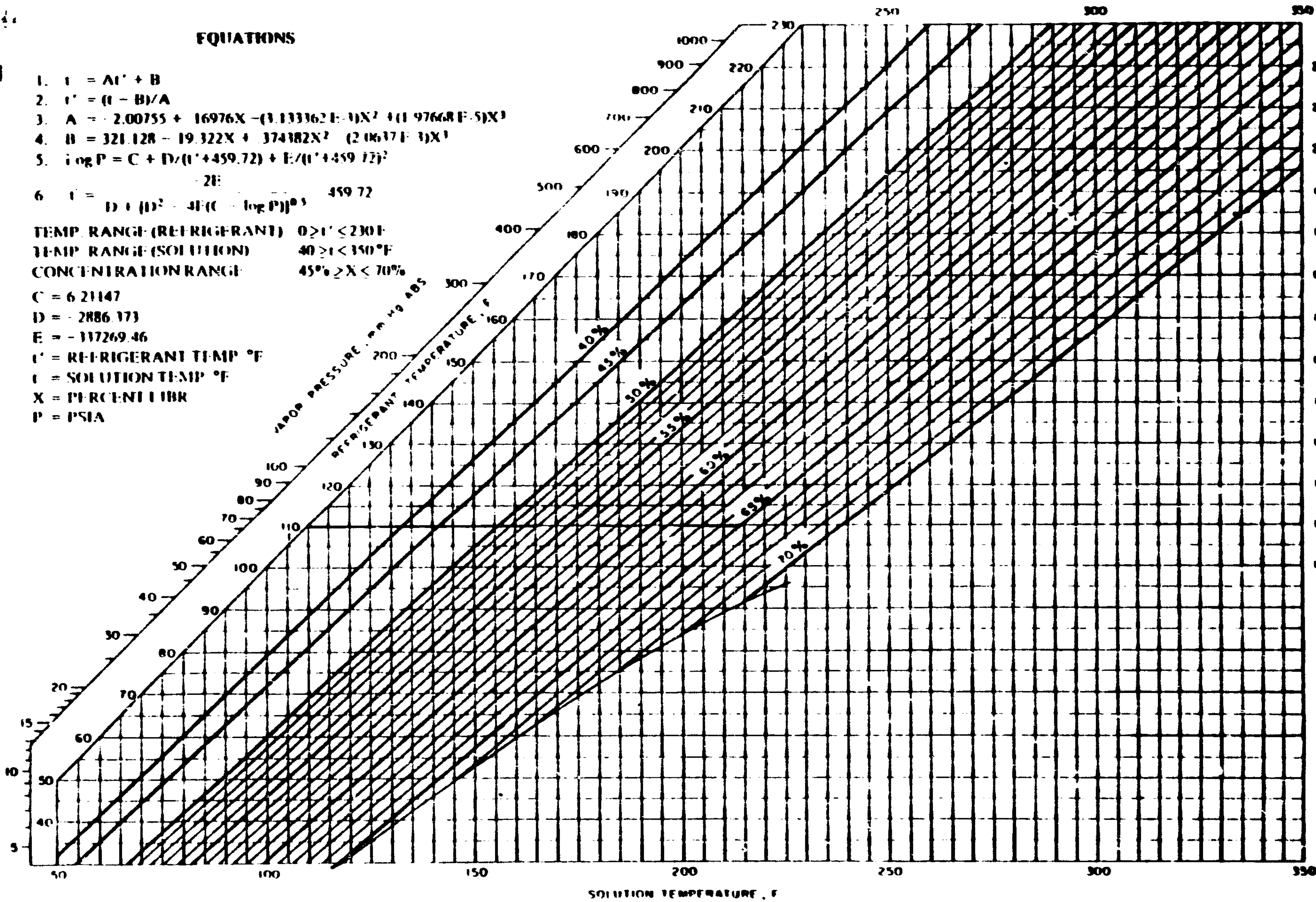

Fig. 2-20 Equilibrium Chart for Aqueous Lithium Bromide Solutions*

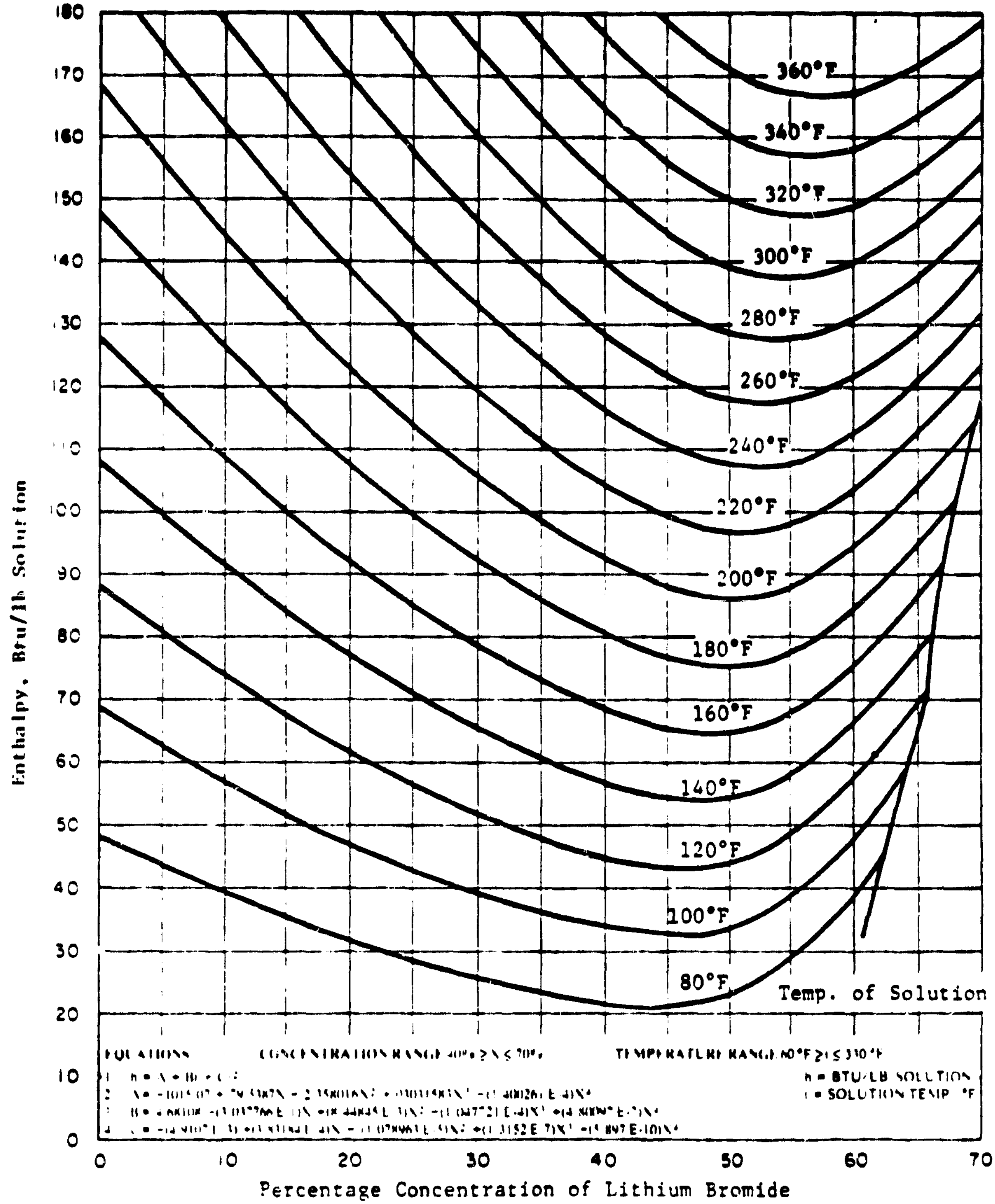

Fig. 2-21 Enthalpy-Concentration Diagram for Water-Lithium Bromide Concentration

*Reproduced from Reference 22.

The total refrigerant + absorbent weight

= 42 + 71.5

= 113.5 lb.

2.6.2 Ammonia - Water Scheme

Ammonia has the highest heat of vaporization with the exception of water. A system utilizing ammonia is shown in Figure 2-22. High-pressure, liquid ammonia is stored in an insulated container. This ammonia liquid is passed through an expansion valve, lowering the pressure and temperature. This low-temperature fluid is passed through an ammonia-to-air-heat exchanger located in the passenger compartment. Here, the ammonia liquid absorbs heat from the passenger compartment and is thus vaporized. This vapor is absorbed in the water contained in the second heat exchanger. As the vapor is absorbed, heat of vaporization is given off by the ammonia, causing the mixture temperature to increase. This heat is removed by the ambient air flowing on this heat exchanger.

Recharging of this system is accomplished externally, e.g., in a garage. For recharging, the mixture of ammonia and water is heated to drive the ammonia vapor off. The water is returned to the ambient heat exchanger while the ammonia vapor is compressed and cooled. In this manner, saturated liquid ammonia is obtained, as in its initial condition, to charge the ammonia tank.

2.6.2.1 Cooling Mode

Typical calculations are shown below:

- Ambient temperature: 100°F
- Condition of ammonia liquid in the ammonia tank:
  - Temperature, 55°F
  - Pressure, 98.06 psia
  - Enthalpy, 103.5 Btu/lb
  - Specific volume, (1/38.75) ft^3/lb

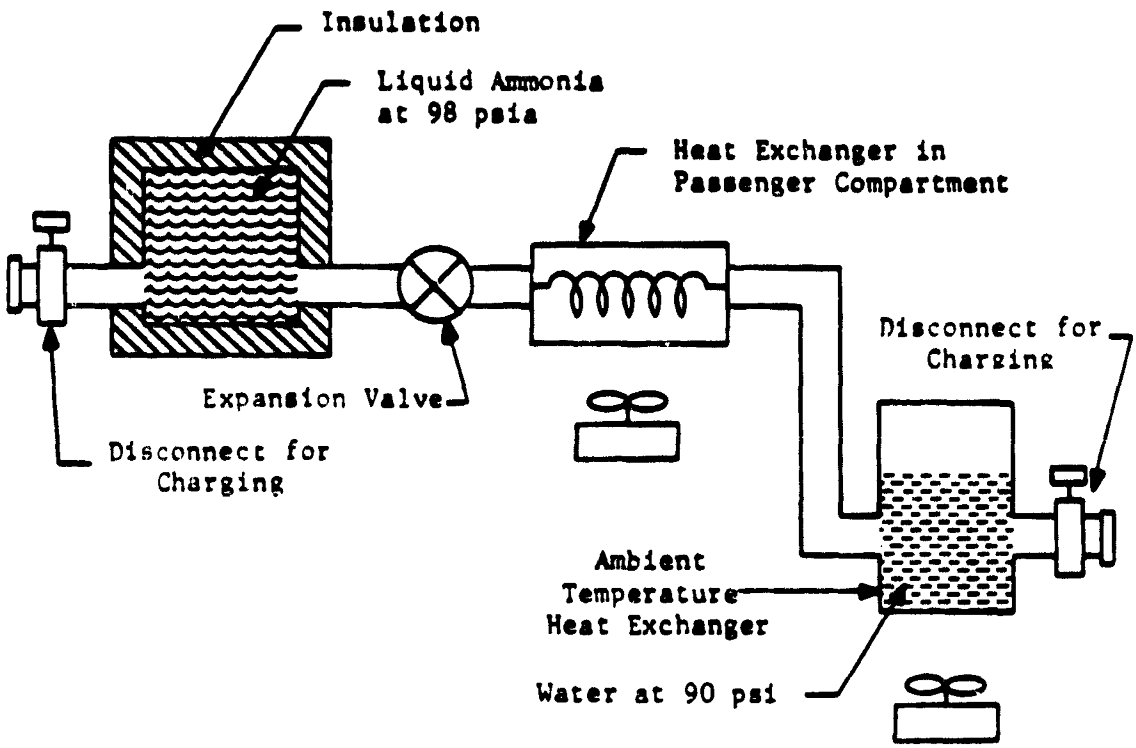

Fig. 2-22 Ammonia System

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- **Condition of ammonia vapor at the exit from the passenger compartment heat exchanger:**
  - **Temperature, 50°F**
  - **Pressure, 90 psia**
  - **Enthalpy, 625.2 Btu/lb**
  - **Specific volume, 3.3 ft^3/lb.**

Thus, heat absorbed from the passenger compartment = 625.2 - 103.5

= 521.7 Btu/lb.

When the ammonia vapor is absorbed by the water in the second heat exchanger, the conditions are:

- Temperature, 110°F
- Pressure, 90 psia

The amount of water needed to absorb the ammonia vapor at these conditions is found to be 1 lb of water for 1 lb of ammonia. This amount is derived from Figure 2-23, which is reproduced from the ASHRAE Handbook, 1977, Fundamentals.

Thus, for providing a storage capacity of 42,500 Btu, the following is needed:

$$\frac{42,500}{521.7} = 81.5 \text{ lb of ammonia}$$

and 81.5 lb of water.

Thus, a total storage fluid weight equals 163 lb.

The volume of the ammonia tank

$$= \frac{81.5}{38.75} = 2.1 \text{ ft}^3.$$

The volume of the ambient-temperature heat exchanger

= weight of solution x specific volume of solution

= 163 x 0.0197

= 3.21 ft^3.

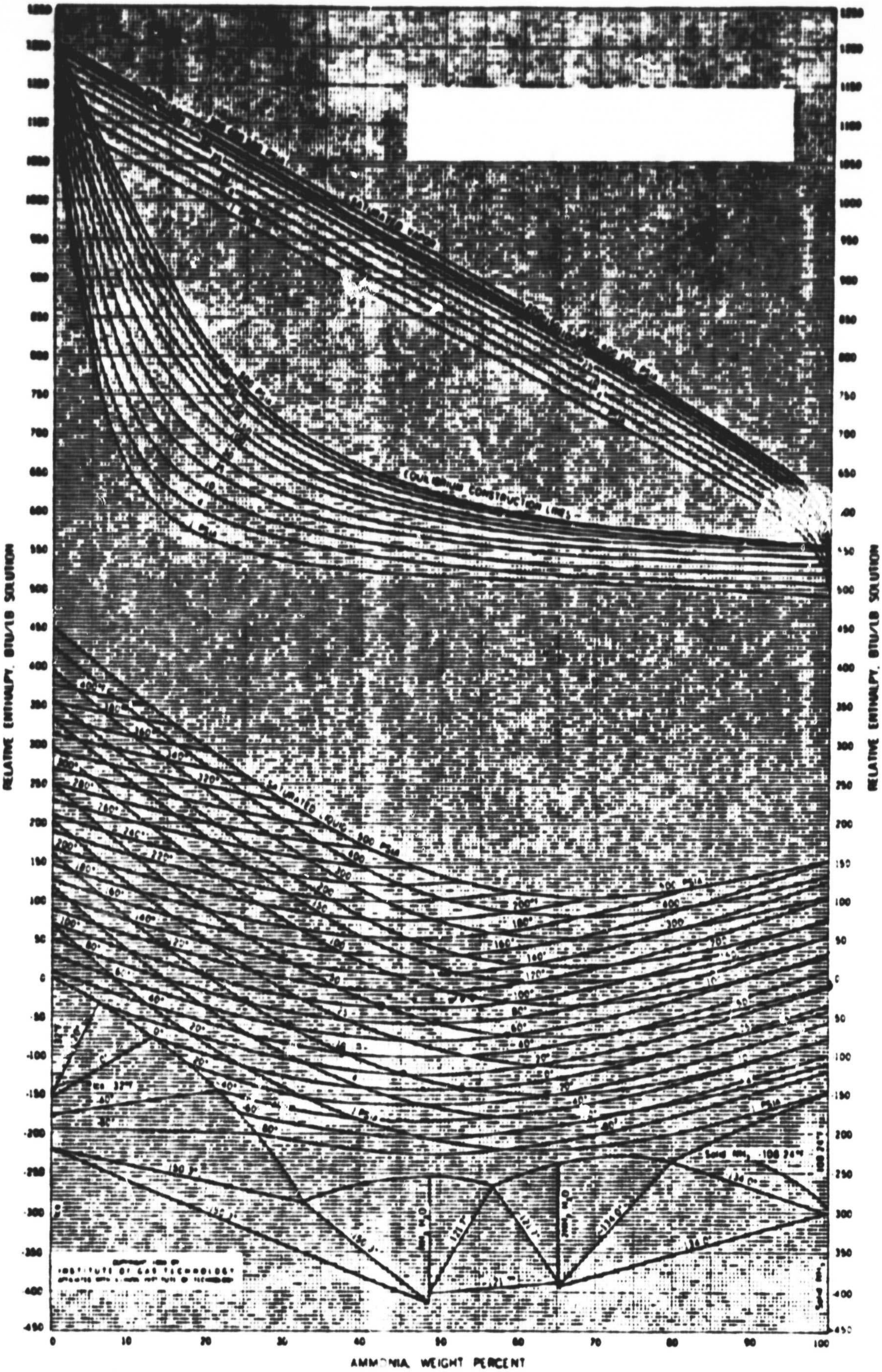

Fig. 2-23 Enthalpy Concentration Diagram for Ammonia Water Mixture [22]

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Total storage volume

= 3.21 + 2.1

= 5.31 ft^3.

2.6.2.2 Heating Mode

The same system can be used for providing heating. The liquid ammonia is passed through an expansion valve, lowering its pressure and temperature. The lower pressure value selected is such that after expansion, the fluid temperature will be below ambient. Thus, when this fluid is passed through the air-to-ammonia heat exchanger, it can absorb heat from ambient air. Hence, in the heating mode, the first heat exchanger exchanges heat with the ambient instead of with the passenger compartment as in the cooling mode. With the heat absorbed from the ambient, the low-temperature liquid ammonia is vaporized and is further passed on to the second heat exchanger which contains water. Here the ammonia vapor is absorbed by the water, and heat is liberated. This heat will cause the temperature of the heat exchanger to rise to well above the temperature required inside the passenger compartment. Hence, when air is blown over this second heat exchanger, it can be used to heat the passenger compartment.

A set of typical calculations for the heating mode are shown below. The data about the properties of ammonia and aqueous solutions of ammonia are obtained from the ASHRAE Handbook, 1977 Fundamentals and from Figure 2-23.

- Storage
  - Temperature -10°F
  - Pressure 23.74 psia
  - Enthalpy 32.1 Btu/lb
  - Volume 1/41.78 ft^3/lb
- After Expansion
  - Temperature -30°F
  - Pressure 13.90 psia
  - Enthalpy 10.7 Btu/lb
  - Volume 1/42.65 ft^3/lb

- **After Evaporation**
  - **Temperature** **-30°F**
  - **Pressure** **13.90 psia**
  - **Enthalpy** **601.4 Btu/lb - 77.9 Btu/lb* = 523.5 Btu/lb**
  - **Volume** **18.97 Btu/lb**
- **After Absorption**
  - **Temperature** **80°F**
  - **Pressure** **13.90 psia**
  - **Enthalpy** **-40 Btu/lb of solution**
  - Solution Concentration **30% by weight ammonia**

    $= \frac{-40}{0.3}$ x Btu/lb of ammonia

    = -133 Btu/lb of ammonia
- Total Heat Given Off = 523.5 - (-133)

  = 656.5 Btu/lb of NH_3

  - Weight of Ammonia Required $= \frac{42,500}{656.5} = 64.5$ lb
  - Weight of Water Required $= \frac{64.5 \times 0.7}{0.3} = 151$
  - Total Weight to be Carried $= \frac{64.5}{0.3} = 216$ lb

*The enthalpies of pure ammonia are based on 0 for the saturated liquid at -40°F, and enthalpies of the mixture are based on 0 for water at 32°F and 0 for ammonia at 32°F. The enthalpy of pure ammonia at 32°F based on 0 at -40°F is 77.9 Btu/lb.

3.0 THERMAL STORAGE

Two types of thermal storage are considered: high-temperature thermal storage and low-temperature thermal storage. High-temperature thermal storage systems act as heat sources from which heat can be extracted for heating the vehicle. Low-temperature thermal storage systems can be used as heat sinks to which heat can be rejected from the hot environment for cooling. Heat sources and sinks can be classified as either sensible heat or latent heat (of phase change), depending on the mechanism employed for rejecting or absorbing heat. The following subsections describe various thermal storage systems.

3.1 Water

3.1.1 Heating

Liquid water has the highest specific heat, with the exception of gaseous hydrogen and helium. In the heating mode, water can be heated to temperatures in excess of 212°F, either by increasing pressure and/or adding other substances such as glycols or salts (NaCl, $CaCl_2$, etc.). If the pressure-increase method is chosen, the heat storage capability can be determined from using steam tables.

Assumption:

T_{cold} = 100°F

Capacity required = 42,500 Btu

where

T_{cold} = lowest temperature of the storage material at which useful heat can be extracted.

Table 3-1 shows the results of calculations using various pressures. This table indicates that a practical heat storage system can be obtained with water as a storage medium. Other advantages of water are:

- The same water can be used as a heat transfer medium.

TABLE 3-1

STORAGE SYSTEM WEIGHT FOR
DIFFERENT PRESSURES FOR STORING 42,500 Btu

Pressure (psia)	Temperature (°F)	ΔT (°F)	Sensible Heat (Btu/lb)	Weight of Water Required (lb)
14.698	212	112	112	380
29.82	250	150	150	283
66.98	300	200	202	210
134.55	350	250	253	167

- **Temperatures and pressures are reasonably small and hence can be handled with inexpensive insulation and containment materials.**

- **Water is very inexpensive. Hence, its weight can be reduced during those days when heating requirements are low, reducing overall weight of the vehicle, with little cost consequences.**

3.1.2 Cooling

For the cooling mode, the following assumptions are made:

$T_{hot} = 50°F$

Capacity required = 42,500 Btu.

where

T_{hot} = highest temperature of the storage material at which heat can be rejected to the storage material.

A maximum ΔT of 18°F can be obtained before solidification occurs. Hence, a maximum of 18 Btu/lb can be stored in a sensible heat mode. A large amount of water will be necessary to store the required capacity. The storage capacity can be increased in two ways:

- Using latent heat of fusion of ice
- Lowering the freezing temperature by adding suitable materials such as propylene or ethylene glycols.

In the case of the first alternative, an additional 144 Btu/lb, which is the latent heat of fusion of ice, could be obtained, thus bringing the total storage capacity to 162 Btu/lb at 32°F. The formation of ice is accompanied by an increase in volume and also by a reduction of thermal conductivity. This combination will necessitate a special heat exchanger design and some heat transfer medium. One such design is shown in Figure 3-1.

In this design, the problem of freezing expansion is solved by filling elastomer balls with water. These balls are kept in the container and the rest of the space in the container is filled with water containing some freezing

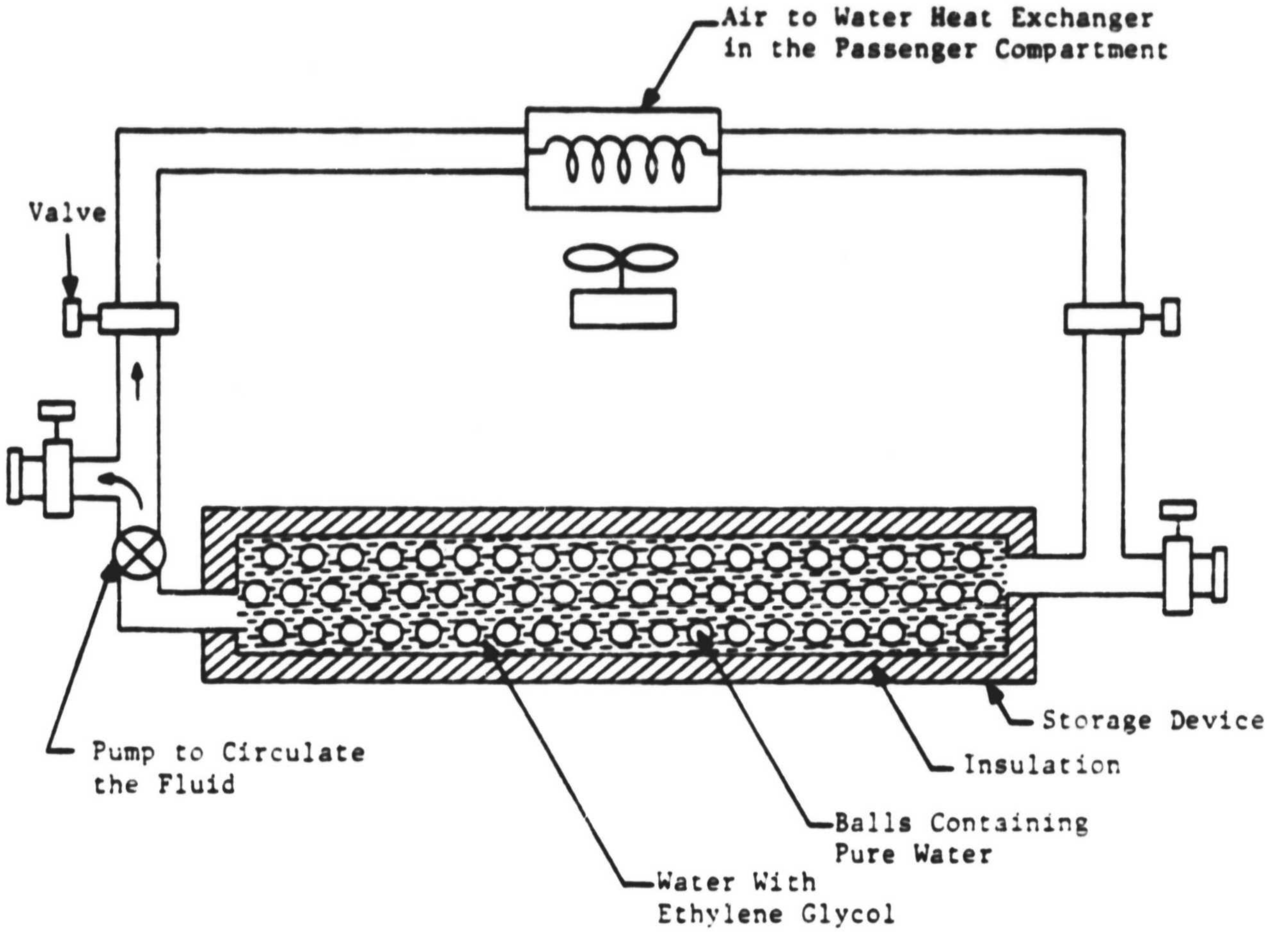

Fig. 3-1 Thermal Storage with Water

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temperature depressant such as ethylene glycol. This fluid acts partly as a storage medium but mostly as a heat transfer medium. The fluid is pumped through the water-to-air heat exchanger located in the passenger compartment.

During recharging time, the fluid is circulated through a refrigerating apparatus located externally, e.g. in a garage, and the temperature of the storage system is brought down to about 20°F. This reduction in temperature causes the water in the balls to freeze, storing 144 Btu/lb of latent cold for freezing, and about 30 Btu/lb of sensible cold. Thus, a total of about 174 Btu/lb of cold is stored. The heat transfer medium also stores some sensible cold of the order of 26 Btu/lb while the metal containing device stores about 6 Btu/lb. Therefore, obtaining an overall storage density of 150 Btu/lb is conceivable.

The same system can be used for heating the vehicle provided the elastomer balls can stand a pressure of about 70 psi at 300°F, so that, in the heating mode, identical storage density is achieved.

Alternative forms of heat exchanger/storage designs can be built, using the basic idea of water in an expandable container to utilize latent heat of fusion of ice. Such schemes have an added advantage of requiring minimal changes during heating and cooling seasons.

In the second alternative discussed earlier, of using brines to lower the freezing temperatures, the latent heat of fusion of ice can also be used, but not nearly to the same extent as in the first alternative. If, for example, a mixture of ethylene glycol (30% by weight) and water is used, it has a freezing temperature of 0°F. However, in brines, the freezing does not result in complete solidification of the liquid until the temperature is reduced to the freezing temperature of a eutectic mixture. In the case of ethylene glycol and water, this temperature is below -60°F.

Between 0°F and -60°F more and more ice crystals drop out as the temperature is lowered. However, a significant portion of the total mixture remains liquid, leaving the whole mixture in the form of slush. This situation is ideal, as the liquid can be circulated to obtain the desired heat transfer and no artifices are required to prevent bursting due to freezing. The only drawback is that since all the ice is not formed at any constant temperature, only a fraction of the latent heat can be utilized, until the temperature is reduced to a very low value. Furthermore, addition of ethylene glycol reduces the specific heat and hence, storage of sensible heat is reduced. Calculations show that to obtain 150 Btu/lb of cooling, a temperature of -50°F will have to be attained. This low temperature will require better insulation to limit heat leaks to a reasonable value.

The system utilizing ethylene glycol thus is simpler, at the expense of requiring very low temperature, than the system utilizing the expandable containers. In either case, the cost of storage material is insignificant. The total cost of the storage device is expected to be in the neighborhood of $50.00.

3.2 Salts

Most salts have a large sensible heat capacity but relatively low density (about 1.7 kg/liter), giving better heat capacity per unit mass than materials like cast iron or magnetite, and comparable capacity per unit volume. Single salts or a salt eutectic mixture exist to provide melting temperature ranges anywhere between 130°C and 900°C. Salts have large latent heats of melting (comparable to water in heat/volume) and can provide large amounts of heat at essentially constant temperature.

Salts have low viscosity, poor heat transfer as stagnant liquids, and poor thermal conductance as solids (copper = 10 x steel = 100 x NaCl). Salts have very large expansion on melting (typically 20%) with attendant large pressure differentials and risk of container rupture. However, this characteristic could be used to provide strong convection and stirring, enhancing thermal transfer by as much as tenfold.

Heat transfer will not be critical during melting since the heat flux is through the convecting liquid to the solid boundary. Freezing will be much more serious because the solid-liquid interface provides the heat at constant temperature. This heat has to pass through the growing layer of poorly conducting solid.

Corrosion over periods of 20 years or more is likely to be a serious problem with molten salts. Whereas the solid is a relatively inert, dense-packed crystal, the liquid is almost completely dissociated and fully ionized - hence the large latent heat and expansion on melting. Most pure salts are not expected to be corrosive, but small traces of water - parts per million or less - could be extremely corrosive to steel or even stainless steels.

Dehydration before melting will be essential at levels unknown at present in commercial use of such materials. Cathodic protection of the containers will be necessary above 400°C (or even lower), removing residual OH and also any H_2O which may enter in protective gases such as argon.

3.2.1 Solids for Heat Storage

Table 3-2 gives the properties of a number of eutectic salts, insofar as those properties relate to thermal storage. Only the solid state of these salts is considered. Using the common NaOH and the uncommon LiOH as examples from Table 3-2, the constants for these salts are given in Table 3-3.

The amount of solids required to store enough heat to provide 42,500 Btu of heating is computed as follows:

- Assume that the lowest temperature at which heat can be extracted from the solid storage is 100°F
- Hence, heat extracted per pound of solid equals
  $C \times (T_{max} - 100°F)$

Using the information in Table 3-3, the following quantities are computed for NaOH and LiOH.

	W(lb)	V(ft^3)	Cost($)	Btu/lb
NaOH	177	1.33	21	240
LiOH	94	1.04	186	450

3.2.2 Lithium Fluoride

Lithium fluoride (LiF) is potentially an excellent storage material for two reasons: it has a high energy density, and its specific heat increases with increasing temperature. Thus, most of its energy is released over the early

TABLE 3-2

PROPERTIES OF SOLIDS FOR HEAT STORAGE

Salt*[2]	Melting Point (°F)	Heat Capacity at Melting Point (Btu/lb-°F)	Thermal Conductivity at Melting Point (Btu/hr-°F-ft)	Specific Weight at 25°C (lb/ft^3)	Containment Material	Cost ($/lb)
LiOH	864	0.59		91.[illegible]	Mild Steel[3]	[illegible]
LiBr	1017	0.25		[illegible]	SS	3.00
NaOH	606	[illegible]		[illegible]	Mild Steel[3]	[illegible]
B_2O_3	842	[illegible]	0.90	[illegible]	SS	[illegible]
34KCl-66$ZnCl_2$	810	0.26		150.[illegible]	SS	[illegible]
61KCl-39$MgCl_2$	815	0.19		131.[illegible]	SS	0.06
68NaCl-32$MgCl_2$	842	0.22		139.[illegible]	SS	0.08
[illegible]KCl-[illegible]$MgCl_2$	878	0.20		137.0	SS	[illegible]
33NaCl-67$CaCl_2$	932	0.20		134.8	SS	[illegible]
37$MgCl_2$-63$SrCl_2$	995	0.16		[illegible]	SS	[illegible]
47Li_2CO_3-53K_2CO_3	910	0.26		137.[illegible]	SS	[illegible]
44Li_2CO_3-56Na_2CO_3	925	[illegible]		166.8	SS	[illegible]
28Li_2CO_3-72K_2CO_3	928	0.35		119.8	SS	[illegible]
51K_2CO_3-49Na_2CO_3	1310	0.40		167.8	Protected SS	[illegible]
33LiF-67KF	918	0.32	[illegible]	157.7	316 SS[3]	[illegible]
67NaF-33MgF_2	1530	0.34	[illegible]	133.4	316 SS[3]	[illegible]
45NaBr-55$MgBr_2$	808	0.12		217.[illegible]	316 SS[3]	[illegible]
20LiF-80LiOH	799	[illegible]		97.7	300 Series SS	2.06
35KCl-27$CaCl_2$-[illegible]8$MgCl_2$	909	0.19		157.[illegible]	SS	0.08
5KCl-29NaCl-66$CaCl_2$	939	0.28		114.2	SS	[illegible]
13KCl-19NaCl-68$SrCl_2$	939	0.16		171.[illegible]	SS	[illegible]
28KCl-19NaCl-53$BaCl_2$	1008	[illegible]		189.[illegible]	SS	[illegible]
24KCl-47$BaCl_2$-29$CaCl_2$	1024	[illegible]		182.[illegible]	SS	[illegible]
32Li_2CO_3-35K_2CO_3-33Na_2CO_3	747	0.40		163.6	SS	[illegible]
12NaF-59KF-29LiF	849	0.32	[illegible]	157.9	316 SS[3]	[illegible]
40KCl-23KF-37K_2CO_3	482	0.24		142.5	SS	[illegible]
17NaF-21KCl-62K_2CO_3	968	0.28		148.9	SS	[illegible]
35Li_2CO_3-65K_2CO_3	941	0.32		141.5	SS	[illegible]
20Li_2CO_3-60Na_2CO_3-20K_2CO_3	1022	0.38		148.[illegible]	SS	[illegible]
22Li_2CO_3-16Na_2CO_3-62K_2CO_3	1022	0.43		146.[illegible]	SS	[illegible]
68K_2CO_3-32$MgCO_3$	860	-		-	-	-

*Reference [23]

[1] Values underscored are experimental. Other values are estimates.

[2] Mixture composition on weight basis.

[3] When a corrosion inhibitor is added

TABLE 3-3

PROPERTIES OF NaOH AND LiOH

Salt	NaOH	LiOH
c, Btu/lb-°F	0.48	0.59
w, lb/ft^3	133	91.1
Cost, $/lb	0.12	1.97
T_s max, °F	600	860

part of the heating cycle. In the past, it has been necessary to bring LiF to temperatures exceeding 1558°F under vacuum conditions to eliminate impurities which are a major contributor of this material's corrosiveness. However, the same benefits can be achieved by adding about 0.3% alumina to the LiF. LiF can store more heat than any other known material as its temperature is varied in the range from 1022°F to 1558°F. Approximately half the energy stored is in the form of latent heat of fusion (LiF melts at 1554°F , and the remainder is sensible heat.

The relevant constants for LiF are:

- Heat of Fusion: 774 Btu/lb at 1554°F
- Specific Weight: 147 lb/ft^3
- Specific Heat at 59°F: 0.373 Cal/g°C = 0.373 Btu/lb-°F.

Thus, assuming for purposes of calculation that the material, after solidifying, can be cooled down to 100°F, then,

$$Q = W\,[H_{\ell g} + c\,(\theta_s - 100)]$$

$$42{,}500 = W\,[774 + 0.373\,(1554 - 100)] = 1318\,W$$

or

$$W = 33 \text{ lb}$$
$$V = 33/147 = 0.22 \text{ ft}^3$$

These values are extremely attractive numbers.

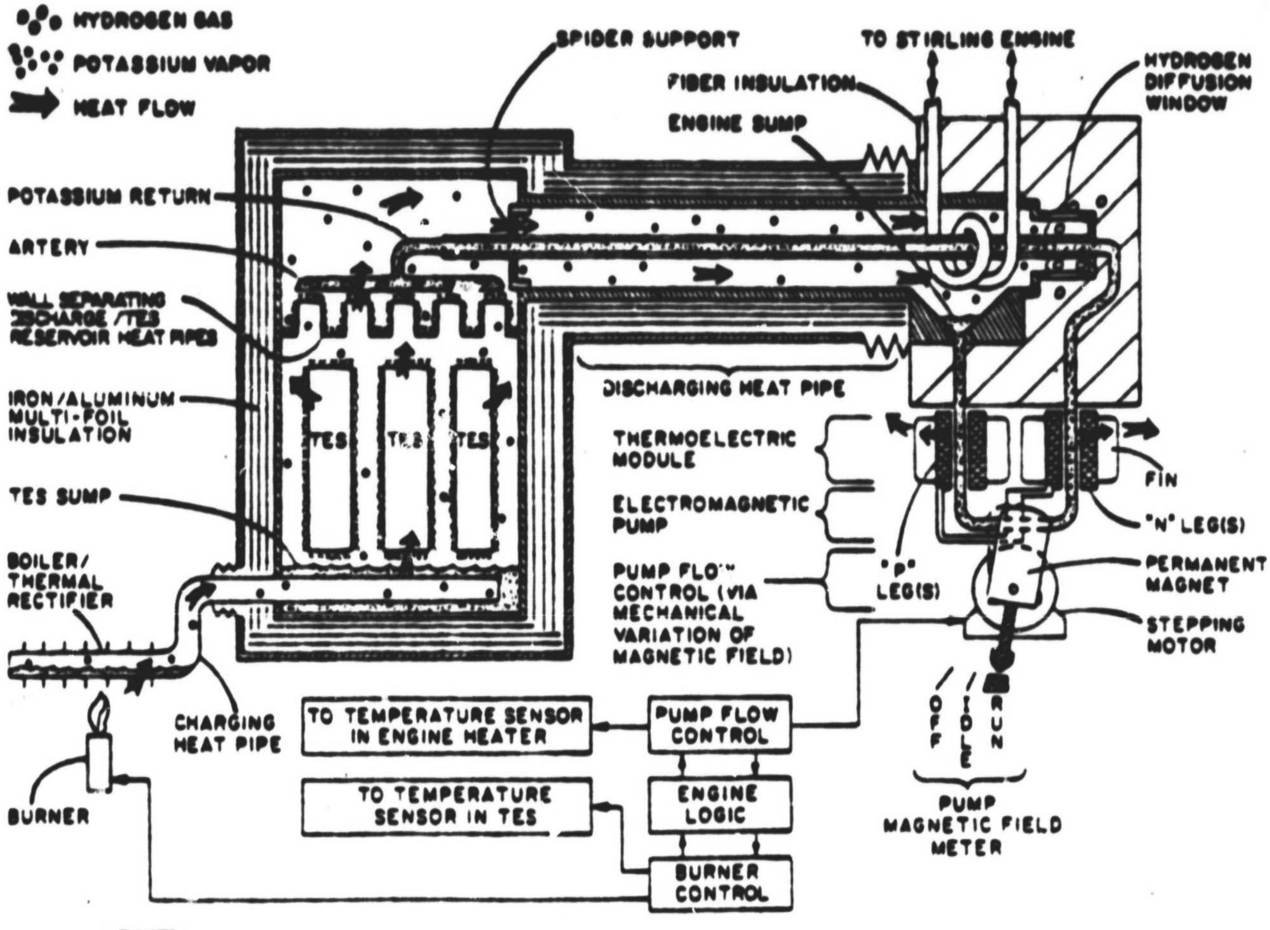
Molten lithium fluorides have been tested as storage materials using heat transport fluids ranging from gas to sodium vapor. However, a number of problems exist with this material. Due to its high storage temperature, LiF would have to be extremely well insulated and placed in some heat resistant connecting fluid for circulation through the vehicle. The storage systems will require delivery of energy either by electrical resistance heaters or through some other circulating thermal conduction device, and the supply, containment, and transport of the heat in and out of the system will present serious difficulties. The toxic and combustible properties of the material itself, and combinations of materials which might be used in a device, have to be examined for any design case. Figure 3-2 shows a schematic of a complete system designed by Thermo Electron Corporation [24] for an automotive propulsion energy storage system. Table 3-4 shows some of the numbe. ‘f interest in this design, also derived from the same source. Prelimina. rmal cycling tests have shown an initial tendency of LiF to degrade, and whether or not stabilization occurs with prolonged cycling is not known.

Production of storage units using salts will have to be conducted under controlled atmospheres, since moisture and air will initiate corrosive attack of materials even at room temperature. At high temperatures and in operation of the units, complete sealing and/or inert blanket gas cover of the salt is required. Consequently, dealing with this hazardous condition in vehicular use requires a high-temperature device or provision for the escape of LiF.

Burn and fire hazards are lower than those associated with the use of gasoline fuels, should that flammable liquid escape. Molten salts freeze rapidly on exposure to cold surfaces or air, and would not decompose unless exposed to extremely high temperatures or contacted by acid or acid fumes, in which case toxic fumes could be released. Removal of spilled LiF after cooling should present no problem.

3.2.3 Modified Caustic Soda ($NaOH + NaNO_3$)

Sodium hydroxide (caustic soda) is a widely used, i nsive, industrial inorganic chemical which has two phase changes in the region of 450 to 630°F, one associated with fusion at about 616°F and one associated with a crystalline transformation at about 574°F [25]. The addition of about 8% sodium nitrate has the effect of depressing the fusion phase change so that all the latent heats of both the NaOH and the $NaNO_3$ are available within a narrow temperature range. Some of the characteristics of this material are shown in Figure 3-3. Unlike some inorganic media, the material can be contained in inexpensive mild steel vessels; the heat exchangers can be fabricated from this material as well. Moreover, the medium has a low vapor pressure and can operate in a vented vessel. Therminol oil, described in Section 3.5, can be used as circulating fluid for storing and extracting heat.



Dual-Mode Operation. Burner On. Pump Control Set at Run.

The burner is on; therefore, heat is supplied to the TES and/or the engine. At low engine power the TES charged (i.e., melted) and at high engine power the TES is discharged. Thus the combustion system can be sized for average power, rather than peak power, operation. From Reference [24].

Fig. 3-2 Thermal Energy Storage for Automobile Propulsion

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TABLE 3-4

ENERGY DENSITIES FOR VARIOUS ENERGY STORAGES

Storage Type	Gravimetric Energy Density (Btu/lb)	Volumetric Energy Density (Btu/ft^3)
LiF (Complete System Between 1554°F and 100°F	1545	58,000
Lead-Acid Batteries		
Present	54	5,800
Advanced	77	8,700
Ni-Zn Batteries		
Present	108	11,100
Advanced	130	14,500

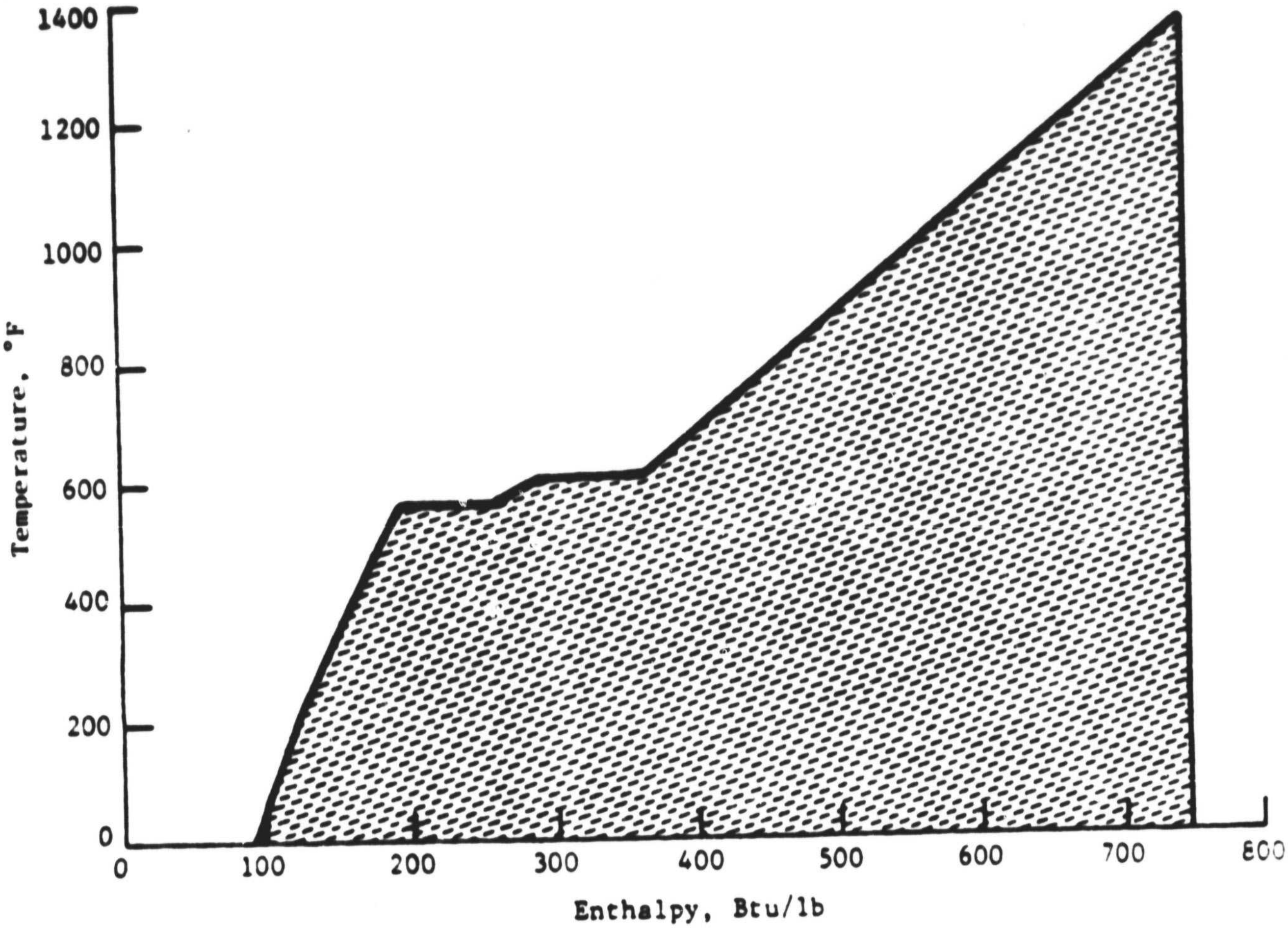

Fig. 3-3 General Thermal Properties of $NaOH \cdot NaNO_3$

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Referring to Figure 3-3, the change in enthalpy between 620°F and 100°F is approximately 276 Btu/lb.

- **Specific Weight - 110 lb/ft^3**
- **Cost - \$0.15/lb**

The above information yields

$$m = \frac{42,500}{276} = 154 \text{ lb}$$

$$V = 154/110 = 1.4 \text{ ft}^3$$

The mass of material required is on the high side, though its volume is reasonable. The cost of the caustic soda, above, would amount to 154 x 0.15 = \$23, which is insignificant.

3.2.4 Alkali Carbonates

Alkali carbonates are attractive as latent-heat storage materials due to their relatively high storage capacity and thermal conductivity, low corrosivness, moderate cost, and safe and simple handling requirements. Test data on a number of these salts are given by Petri et al. [26]. Some of their results are reproduced in Tables 3-5 and 3-6, as well as in the following summary.

- Alkali/alkaline earth carbonates can be used as latent heat thermal storage materials over the temperature range of 747 to 1636°F.
- $LiKCO_3$ (melting point: 941°F) exhibited excellent chemical, physical, and thermal stability for over 5600 hours and 129 cycles, as well as good compatibility with type 316 stainless steel.
- Pure Li_2CO_3 (261 Btu/lb, \$0.93/lb) and Na_2CO_3 (113 Btu/lb, \$0.03/lb) appear attractive in approximate temperature regimes of 1300°F and 1500°F respectively.

TABLE 3-5*

SALTS SELECTED FOR LAB-SCALE TESTING

Salt System	Melting Point (°C)	Melting Point (°F)	$H_{s\ell}$** (Btu/lb)	Heat Capacity At Melting Point (Btu/lb-°F) c (s)	c (ℓ)	Thermal Conductivity k(ℓ) (Btu/hr-°F-ft)	Specific Weight at 25°C (lb/ft^3)	Cost ($/lb)	Capacity Cost ($/10^6 Btu)
$(Li/K/Na)_2CO_3$	397	747	119	0.40	0.39	1.17	143.6	0.33	2741
$CaCO$-Li_2CO_3	662	1224	118					0.53	
$CaCO_3$-Na_2CO_3	686	1267	74					0.08	
K_2CO_3-Na_2CO_3-Li_2CO_3	706	1303	70	1.00	0.40	0.37	149.8	0.11	1597
K_2CO_3-Na_2CO_3	710	1310	70	1.00	0.40	0.37	149.8	0.11	1597
KF-NaF	721	1330	252				154.9	0.56	2223
Li_2CO_3	726	1339	261	0.63	0.60	1.13	131.6	0.93	3563
$CaCl_2$	772	1422	110.1				134.1	0.04	363
Na_2CO_3-K_2CO_3	790-737	1454-1360	109				156.9	0.06	551
Na_2CO_3	858	1576	114	0.24	0.24	1.06	157.8	0.03	264
K_2CO_3	891	1636	86	0.36	0.36	1.00	151.6	0.19	2209

*Ref. [26]
**s - solid, ℓ - liquid

TABLE 3-6*

SALT TESTS EXPERIMENTAL RESULTS

Salt System	Melting Point °C (°F)	ΔH_f, Btu/lb	Discharge Solidification Range °F	- °F	Time To Discharge Salt mp +50°C-mp -50°C min	$\bar{Q}$ mp -50→ mp +50°C Btu/hr-ft^2
K_2CO_3-Na_2CO_3	710 (1310)	70	1306	1292	34	11,715
K_2CO_3-Na_2CO_3-Li_2CO_3	706 (1303)	70	1303	1283	46	14,091
Na_2CO_3-K_2CO_3	790-737 (454-1360)	109	1454	1360		14,629
Li_2CO_3	726 (1339)	261	1341	1337	48	
Na_2CO_3	858 (1564)	113	1594	1584	27	14,478
K_2CO_3 (CO_2 Blanket)	891 (1636)	86	1681	1674	21	23,002
$(Li/K/Na)_2CO_3$	397 (747)	119	770	754	86	2,206
$(Li/K/Na)_2CO_3$ w/Duorel	397 (747)	119	766	759	53	3,216
$BaCO_3$-Na_2CO_3	686 (1267)	(74)	1323	1314	31	13,220
$CaCl_2$	772 (1422)	110				
$CaCO_3$-Li_2CO_3	662 (1224)	(118)	1224	1215	48	12,817
Li_2CO_3 (CO_2 Blanket)	726 (1333)	261	1353	1346	54	24,293
KF-NaF	721 (1330)	252				

*Ref. [26]

- **Na_2CO_3-$BaCO_3$ mixture is an attractive low-cost ($0.08/lb) mixture for 1300°F applications.**

- **85 mole-percent Na_2CO_3 and 15 mole-percent K_2CO_3 salt solidified incongruently over a range of 1445 to 1364°F.**

- **$CaCO_3$ and $NaCO_3$ were chemically stable as mixtures with alkali carbonates over a range of 1200 to 1300°F.**

- **Carbonates can be melted and safely loaded in air environments.**

- Chlorides and fluorides require more sophisticated handling, fabrication and operation procedures. Their vapors are toxic in air and they must be handled and cycled in dry, inert environments to avoid accelerated containment corrosion by oxygen/moisture contamination.

- 304 and 316 stainless steel demonstrated good compatibility over a range of 700 to 1200°F.

- Prolonged operation above 1200°F results in salt creepage and increased oxidation, carburization, grain boundary carbide precipitation and brittle sigma phase formation in austenitic stainless steels. Use of superalloys may be required.

- Baseline thermal discharge characteristics and heat fluxes at the order of 12,000 to 24,000 Btu/hr-ft^2 with storage capacity of 340 Btu units were established.

Selecting two salts from Table 3-5, one with the highest heat of fusion, the other with the highest value of c, gives,

Salt	$H_{s\ell}$ Btu/lb	c, Btu/lb-°F	w, lb/ft^3	Cost, $/lb	Melting Point, °F
$[K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3]$	70	1.0	149.8	0.11	1310
Li_2CO_3	261	0.63	131.6	0.93	1339

The required masses and volumes of these two salts (for a final salt temperature of 165°F) are as follows:

Salt	Storage Capacity, Btu/lb	Mass, lb	Volume, ft^3	Cost of Salt, $
[$K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$]	1280	33	0.22	3.6
Li_2CO_3	1041	41	0.31	38

These numbers, like the lithium fluoride previously considered, are practical and these salts are thus potential candidates. However, a point to be kept in the foreground is that extraction of heat from these salts down to 100°F may not be practical and storage at 1300°F may be a demand difficult to meet due to excessive insulation levels. Also, while the cost of salts is low, the cost of the total system as projected from a solar heating system estimate, given in Table 3-5, would be $67 for the $K_2CO_3 \cdot Na_2CO_3$ and $155 for the Li_2CO_3.

3.2.5 Sodium Sulfate Decahydrate*

Sodium sulfate decahydrate [27] offers many advantages over competitive materials for storage of thermal energy in the temperature range from 32°F to approximately 100°F. The heat of hydration is comparable to that of the best of the paraffins; yet its thermal conductivity is approximately five times greater. Containment of the material is simplified because the solid phase is more dense than the liquid, so the substance contracts upon solidification. Sodium sulfate offers the added advantage of safety and low cost, which is on the order of 3¢ - 5¢/lb.

To obtain working temperatures for cooling, other salts must be added to the sodium sulfate to depress its freezing point from the nominal 90°F. A mixture, consisting of sodium sulfate decahydrate and five other salts, as shown in Table 3-7, was added to decrease the solidification point to 55°F (13°C). The mixture does not behave as a eutectic: the concentration of the solid phase increases as the temperature decreases, with complete solidification occurring at approximately 9°F (-13°C). The effective heat capacity was found to be approximately 2 Btu/lbm-°F over the temperature range from 50 to 60°F.

*Known as Glauber's salt

TABLE 3-7*

COMPOSITION OF SALT MIXTURE TESTED

Chemical	Molecular Weight	Fraction by Weight	Moles Per Mole Na_2SO_4
Water(1)	18.0	0.414	10
Thickening Agent (Clay)	-	0.087	-
Borax ($Na_2B_4O_7 - 10H_2O$)	-	0.026	-
Boric Acid (H_3BO_3 Anhydrous)	-	0.0173	-
Sodium Chloride (NaCl Anhydrous)	58.5	0.0666	0.5
Ammonium Chloride (NH_4Cl Anhydrous)	53.5	0.0616	9.5
Sodium Sulfate (Na_2SO_4 Anhydrous)	142.0	0.325	1
Sodium Phosphate Tripoly ($Na_5P_3O_{10}$ Anhydrous)(2)	-	0.0025	-
		1.0000	

1. The ratio of sodium sulfate to sodium sulfate and water is 0.440, which is the exact ratio for $Na_2SO_4 - 10H_2O$.

2. Wetting agent

*Ref [27]

Tests indicate that the salt performs as if the undissolved anhydrous sodium sulfate is not able to combine with its water of hydration during the hydrating process. As a result, the heat of transformation is less than the theoretical 75 Btu/lb. In fact, the salt behaves as if it had a variable specific heat of approximately 2 Btu/lb-°F rather than as a phase-change material. The phase-change process occurs continually over a temperature range of some 55°F, although most of the heat of transformation is realized over a range of 30°F.

Even taking the maximum possible value of 75 Btu/lb, and using a temperature range of 140°F to 68°F, the following weight requirement is obtained:

$$\frac{42,500}{75 + 2(140-68)} = \frac{42,500}{219} = 194 \text{ lb}$$

No data for specific weight are available, but whatever the weight's exact value, the volume required would be of the order of 1 to 2 ft^3.

For the purposes of comparison, Table 3-8 gives information on some of the commercially available phase change materials. This table is reproduced from "Solar Age" [28]. Using the information given in this table, the weights for storing 42,500 Btu are calculated and are shown in Table 3-9.

3.3 Polyethelene Pellets

A highly crystalline polymer, such as high-density polyethylene offers advantages as a thermal storage material, if it is rendered form-stable so that it does not flow on melting. Uncrosslinked pellets deform and flow into flattened discs upon melting, whereas properly crosslinked pellets maintain their original shape during the melting-freezing cycle. Long-term durability tests showed excellent hydrogen retention up to 1000 cycles. In terms of producing the least inter-particle adhesion, the best product is the silane-grafted/crosslinked polyethylene resin [29]. As a heat transfer fluid, glycerin is found to be a suitable medium.

The relevant properties of this polyethylene material are:

- Melting point - 270°F

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TABLE 3-F

COMMERCIALLY AVAILABLE PHASE-CHANGE MATERIALS

No.	Manufacturer	Product Name	Phase Change Material	Container	Total Weight (lb)	Phase Change Temperature (°F)	Heat Storage Capacity (at phase change temperature) (Btu)	Warranty
1	Addison Products Co. Addison, MI 49220 (517) 567-6131	Solar Therm	paraffin	steel, round, 1 gallon	7.25	115	40	1 year
2	Architectural Research Corp. 40 Water St. New York, NY 10004 (212) 943-3160	Sol-Ar-Tile™	Glauber's salt (1)	polymer resinous concrete tile, 2 feet by 2 feet	44	73	1,000	2 year
3	Blue Lakes Engineering Pace Corp. P.O. Box 1033 Appleton, WI 54912 (414) 733-0961 (a distributor for PSI Energy Systems, Inc.)	Thermol-81	calcium chloride hexahydrate, with additives	black polyethylene tube, 6-feet long, 3 1/2-inch diameter	35	81	2,460	10 year
4	Boardman Energy Systems, Inc. 3720 Kennett Pike P.O. Box 4198 Wilmington, DE 19807 (215) 388-7454	product available July 1980	sodium sulphate (1)	plated steel tubes, 30-inch long and 4-inch diameter; built-in spacers, selective coating available	22	65°, 66°, 74°, 78°, 81°, 89°	1,000-2,000, varies with phase-change temperature	will be available
5	Collodial Materials, Inc. P.O. Box 696 Andover, MA 01810 (617) 475-3276 (a licensee of Cabot Corp.)	Heat Pac™	sodium sulphate (2)	3-ply aluminum foil laminate pouch, 3/4 inch by 2 feet by 2 feet	10	73 (may be varied between 65° and 89°F)	350	5 year
6	Energy Materials, Inc. 2622 South Zuni Englewood, CO 80110 (303) 934-2444 (under OEM agreement with Dow Chemical Co.)	Thermalrod-27	calcium chloride hexahydrate	black polyethylene pipe, 6-feet long, 3 1/2-inch diameter	35	81	2,542	10 year limited
7	PSI Energy Systems, Inc. 1533 Fenpark Drive St. Louis, MO 63026 (314) 343-7666 (under OEM agreement with Dow Chemical Co.)	Thermol-81	calcium chloride hexahydrate, with additives	black polyethylene tube, 6-feet long, 3 1/2-inch diameter	35	81	2,460	10 year
8	Texxor Corporation 9910 North 48th St. Omaha, NE 68152 (402) 453-7558	Texxor Heat Cell	calcium chloride hexahydrate (Bisol II)	steel cylinder, 7-inch long, 4.26-inch diameter	4.36	81	345	5 year
9	Valmont Energy Systems, Inc. Valley, NE 68064 (402) 359-2201		Glauber's salt (2)	polyethylene rectangular tube, 2 feet by 1 foot by 2 inch	16	99	1,294	5 year limited

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TABLE 3-9

PHASE CHANGE MATERIALS WEIGHTS FOR STORING 42,500 BTU

	Phase-Change Material	Heat Storage Capacity (Btu/lb)	Storage Weight for Storing 42,500 Btu (lb)
1	Paraffin	11	3860
2	Glauber's Salt (1)	22.8	1865
3	Calcium Chloride Hexahydrate with Additives	70.2	605
4	Sodium Sulfate (1)	66 → 91	645 → 468
5	Sodium Sulfate (2)	35	1215
6	Calcium Chloride Hexahydrate	73	581
7	Calcium Chloride Hexahydrate with Additives	70	606
8	Calcium Chloride Hexahydrate (Bisoi II)	76	560
9	Glauber's Salt (2)	81	525

- Heat of Fusion - 90 Btu/lb
- Specific Gravity - 0.96
- Cost -$0.2-0.3/lb

For these properties, the mass and volume of storage material required are respectively 472 lb and 7.9 ft^3.

A cost study for a solar storage system given in Reference 29 yields the following estimates:

0.94¢ per Btu for a liquid heat transfer medium
0.79¢ per Btu for air circulation

These numbers were obtained for an application different from the present one, but if these numbers are projected to a passenger vehicle, the cost of the system would be of the order of $400.

3.4 Liquefied Gases

Liquefied gases such as air or hydrogen can, from a practical standpoint, be considered only as candidates for vehicle cooling. Liquefied gases can be stored at atmospheric pressure at their respective liquefaction temperatures. Since these temperatures are extremely low, the gases would have to be kept in special, highly insulated containers. In addition, in the case of hydrogen, special safety precautions would have to be taken if the gas were to be boiled off to the atmosphere, or if a road accident occurred.

Table 3-10 gives some of the liquefied gas properties relevant to the present survey.

As shown, hydrogen has the highest heat of vaporization and also a very high specific heat. Of the gases, hydrogen will be discussed first. The refrigeration requirement is given as 42,500 Btu.

Assumptions:

- $T_{hot} = 40°F$
- $T_{cold} = -422°F$

TABLE 3-10

PROPERTIES OF LIQUEFIED GASES

Liquid	Liquefaction Temp. (°F) at 1 Atmosphere	Specific Gravity	Specific Weight (lb/ft^3)	Heat of Vaporization (Btu/lb)	Average Specific Heat at Constant Pressure (Btu/lb-°F)
Air	-318	0.92	54.6	88	0.22
N_2	-320	0.804	50.2	86	0.25
O_2	-297	1.14	71.4	92	0.21
H_2	-422	0.07	4.3	194	3.4

- **Latent heat of vaporization = 194 Btu/lb**
- **Average specific heat over the temperature range $\approx$ 2.9 Btu/lb-°F**
- **Density of liquid hydrogen = 4.3 lb/ft^3**

Hence, the total cooling capacity over the temperature range

$$= 194 + [40-(-422)] \times 2.9$$
$$= 1534 \text{ Btu/lb}$$

The weight of liquid hydrogen required is:

$$\frac{42,500}{1534} = 27.7 \text{ lb}$$

The volume of liquid hydrogen required is:

$$\frac{27.7}{4.3} = 6.4 \text{ ft}^3$$

Performing the same calculation for the other gases listed in Table 3-10, the respective weights and volumes are given in Table 3-11.

3.5 Organic Oils

Reference [25] contains a brief discussion of the potential use of special high-temperature oil labelled Therminol. The properties of these oils are given in Table 3-12. Table 3-13 shows the weights and volumes required to store 42,500 Btu, utilizing the highest usable temperature and assuming that heat is extracted until the oils reach a temperature of 100°F in the heating mode. In the cooling mode, the oils are assumed to be precooled to the lowest usable temperature and cooling is provided until their temperature rises to 50°F.

3.6 Paraffins

The material considered here is Shellwax 700, a commercial grade paraffin. Data indicate that paraffins have approximately the same thermal conductivity in the solid and liquid phases. Their thermal properties are shown in Figure 3-4. Calculations [30] performed for a solar storage system using a heat of fusion of 63 Btu/lb showed the following trends:

TABLE 3-11

WEIGHTS AND VOLUMES REQUIRED FOR PROVIDING COOLING OF 42,500 Btu

Liquid	Mass (lb)	Volume (ft^3)
Air	254	4.66
N_2	241	4.73
O_2	255	3.84
H_2	28	6.5

TABLE 3-12

WEIGHTS AND VOLUMES OF OILS REQUIRED TO STORE 42,500 Btu

Trade Name	Heating			Cooling		
	Capacity (Btu/lb)	Weight Required (lb)	Volume Required (ft^3)	Capacity (Btu/lb)	Weight Required (lb)	Volume Required (ft^3)
Therminol-55	360	118	2.62	36	1179	26.2
Therminol-66	360	118	2.52	16.4	2581	55.3
Dowtherm-A	215	198	3.72	20.4	2008	39.2
Humbletherm 500	358	119	2.88	39.5	1072	26.0

TABLE 3-13*

ORGANIC HEAT TRANSFER OILS

Trade name	Composition	Producer	Usable Temp. Range (°F) Low	Max.	Fire Point (°F)	Auto-Ignition Temp. (°F)	Thermal Properties at Max. Usable Temp.: Specific Heat (Btu/lb °F)	Viscosity (lb/ft-hr)	Thermal Conductivity (Btu/hr-ft-°F)	Density (lb/ft)	Comments
Therminol 55	Alkylated aromatic	Monsanto	0	600	410	670	0.72	1.1	0.065	45	Currently in use; requires nonoxidizing cover gas
Therminol 66	Hydrogenated terphenyls	Monsanto	25	650	380	705	0.655	0.649	0.0612	46.8	Currently in use; requires nonoxidizing cover gas
Caldria HT-43		Exxon		600		759					Currently in use; requires nonoxidizing cover gas
Dowtherm-A	Diphenyl-diphenyl oxide eutectic	Dow Chemical	12	500	275	1,150	0.537	0.65	0.0645	53.0	Requires pressurization for temperatures exceeding 500°F
Humbletherm 500	Aliphatic petroleum oil	Humble Oil	-5	600	475	NA	0.7165	1.01	0.0655	41.3	

*Ref. [25]

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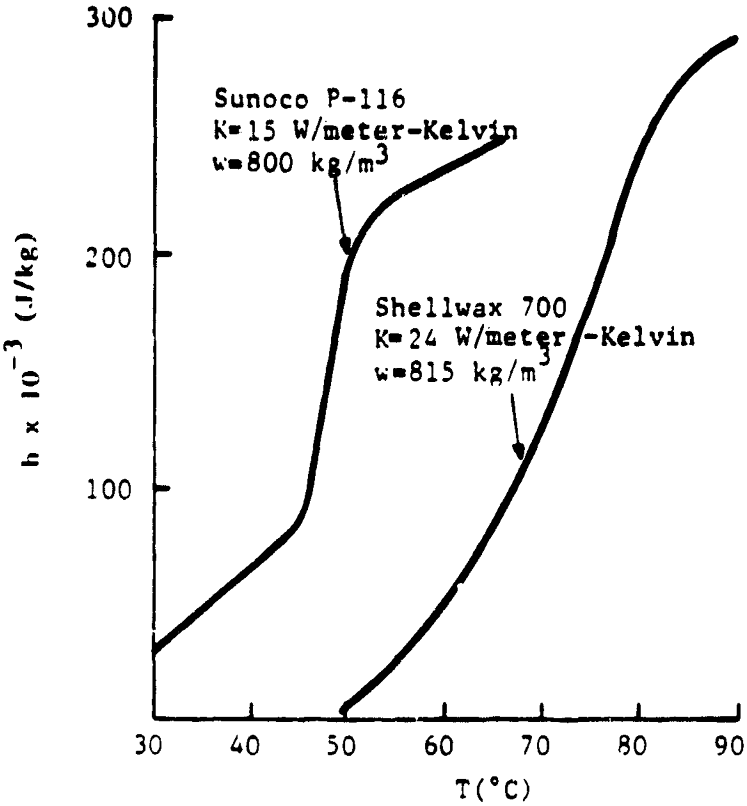

NOTE: Enthalpy Data Chosen For Convenience in Plotting

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Fig. 3-4 Thermal Properties of Commercial Grade Paraffins

- **The cost of the system (total) is twice as much as that of a comparable sensible heat water storage system. The main item of expense is the heat exchanger.**
- **The optimum occurs when the heat stored is 85% of the total storage capacity.**
- **The volume required is 2/3 less than an equivalent water storage system.**

For this case, considering again a range of 190°F to 120°F, the following is obtained from Figure 3-4.

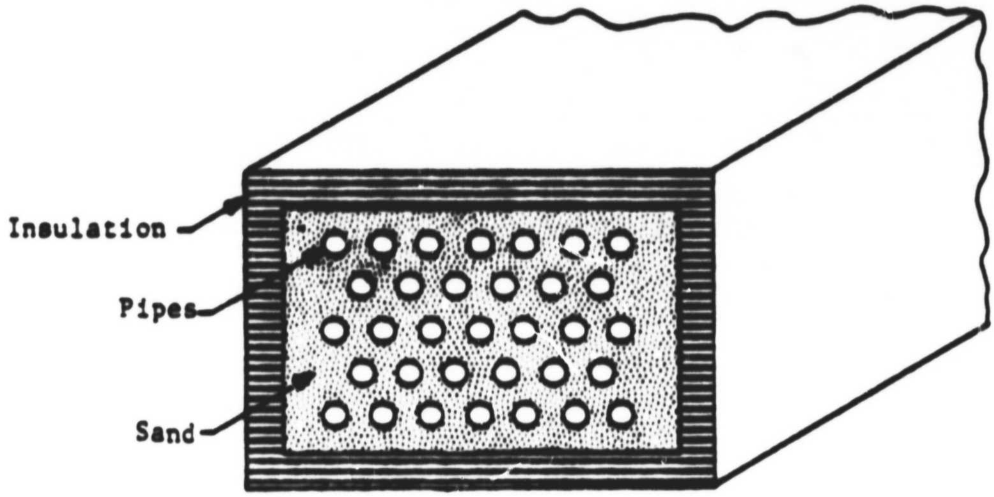
$$W = \frac{42,500}{129} = 329 \text{ lb.}$$

3.7 Sand

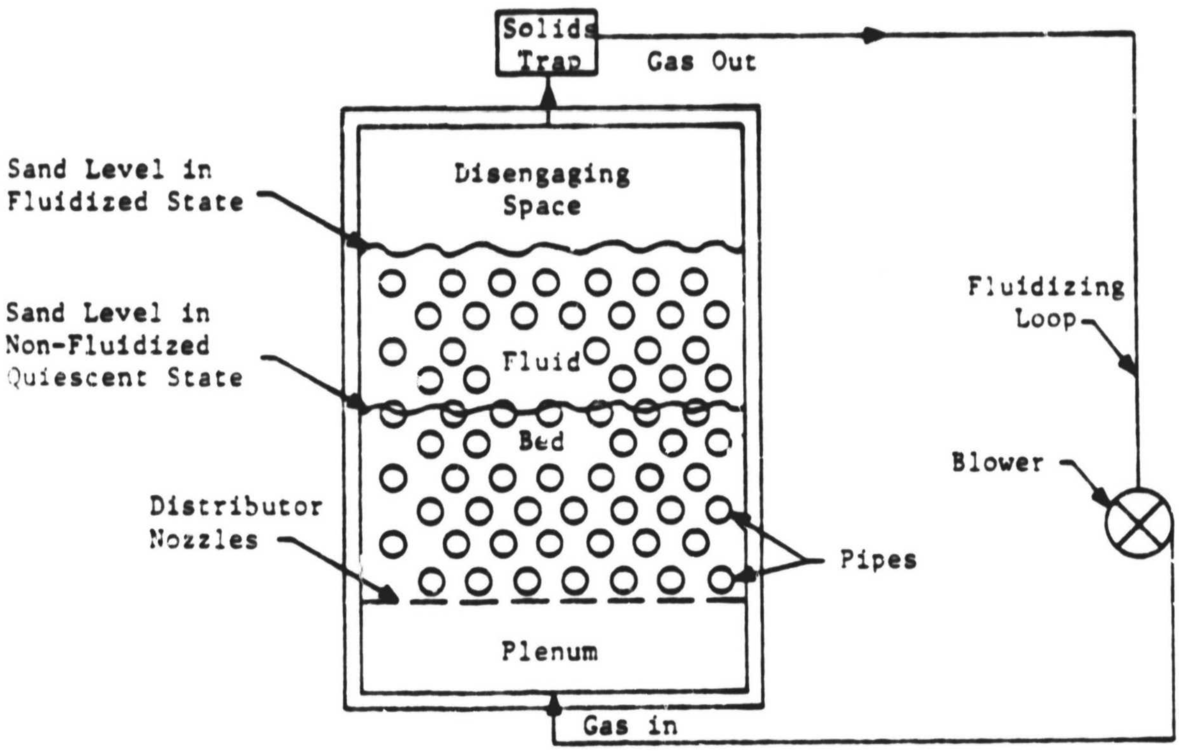
Figure 3-5a taken from Reference [31] shows a possible scheme for a fixed-bed sand storage system. The sand is retained by concrete walls which are the least expensive method for such containment, except for the two opposing walls made of corrugated metal steel plates. Holes in the steel plate are spaced to align hot water pipes passing through the sand volume. Header pipes are welded to the tubes just outside the steel retaining walls which allows for easy replacement. In the charging mode, pressurized hot water enters the system, the heat is transferred from the hot water through the pipe walls and diffuses through the sand volume, heating the sand. In the extraction mode, cool, pressurized water enters the system via the same tubes. Heat is transferred by conduction from the sand to cool fluid which is circulated through the heat exchanger in the vehicle.

In the cooling mode, cold, pressurized water is first circulated through the sand, cooling it. Subsequently, when cooling is required, water is circulated through the cold sand bed which is cooled, and in turn cools the vehicle.

A limiting factor in the extraction (or input) or thermal energy from (or to) the fixed-bed sand system is the low thermal diffusivity of the storage material. One way to counteract this low diffusivity is to increase the



(a) Fixed Sand Bed



(b) Fluidized Sand Bed

Fig. 3-5 Sand Storage System

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surface conductance between the pipes and the sand by moving the sand with respect to the pipes. The sand is fluidized or otherwise agitated into motion, thereby increasing the heat transfer potential; this concept is shown in Figure 3-5b. In the absorption or charging mode, the sand in fluidized by an inert gas such as air or nitrogen, and will absorb heat from the pipes which transport energy in the form of hot pressurized water. After sufficient time has passed to store the desired quantity of thermal energy, the bed is defluidized with the sand stationary until energy extraction commences. The low thermal diffusivity of quiescent sand retards heat loss, in addition to the fact that the shell of the fluidized bed is insulated. In the discharge or extraction mode, the sand which has remained at a high temperature will again be fluidized and the heat will flow from the sand to the pipes, which now have cool fluid running through them. The electrical energy consumption of the blower necessary to fluidize the bed is on the order of 5% of the thermal storage that the system may contain.

The advantages and disadvantages of fixed and fluidized beds are summarized in Table 3-14.

The following are used as a feasibility check of using sand [31, 32]:

$$b = 50 \frac{\text{Btu}}{\text{hr-ft}^2\text{-°F}} \quad \text{(for fluidized bed)}$$

$$c = 0.195 \frac{\text{Btu}}{\text{lb-°F}}$$

$$w = 65 \text{ lb/ft}^3$$

$$\text{Cost} = \$10/\text{ton}$$

$$\theta_{max} \sim 900°\text{F}$$

With these numbers allowing 100°F at the end of the cycle,

$$Q = W \times c\ (900 - 100)$$

$$42{,}500 = W \times 0.195\ (800) = 156\ W$$

TABLE 3-14

COMPARISONS OF SAND STORAGE SYSTEMS

Systems	Advantages	Disadvantages
Fixed Beds	• Uncomplicated; few moving parts • High reliability • Low development risk • Low containment costs	• High tube density required • Probable high system cost
Fluidized Beds	• Lower pipe cost • Improved performance	• Parasitic power losses due to large blower • Large containment cost

yielding

$$W = 272 \text{ lb}$$

$$V = 4.2 \text{ ft}^3$$

Considering the fact that a 900°F sand temperature may not be advisable and that the bulk of piping, insulation, etc., may equal or exceed the weight of the sand, sand would be an impractical system for vehicle applications.

3.8 Compressed Air Energy Storage

A general physical arrangement scheme for the utilization of adiabatic air compression for purposes of both heating and cooling is shown in Figure 3-6. The charging part of the cycle consists, in all cases, of the adiabatic compression of air and its storage in compressed form in a well insulated container aboard the vehicle. The external power could be delivered to the system either by simply hooking on the storage vessel to an external air compressor, located in homes, gas stations, supermarkets, etc.; or the vehicle could have a motor compressor aboard, in which case only an electrical connection to the outside would be required. The latter is, of course, more convenient, but its price is the cost, installation, and weight of additional equipment.

As portrayed in Figure 3-6, in the heating mode, a circulating fluid would extract the heat from the hot compressed air and pass it on to the interior of the vehicle via a heat exchanger. As heat is extracted, pressure and temperature in the constant volume storage vessel would drop. When the ΔT would fall to a sufficiently low level, the system would have to be recharged to its original thermodynamic state of high pressure and high temperature. This recharging can be done in one of two modes. A resistance heater can be used to restore the temperature to its original level (heating at constant volume); or the low pressure gas can be exhausted and a compressor used to charge the vessel with newly compressed air (adiabatic compression).

In the cooling mode, the compressed air can be throttled to ambient pressure and exhausted to the atmosphere. On expansion, the temperatures would drop, drastically cooling the interior of the car via a heat exchanger. For

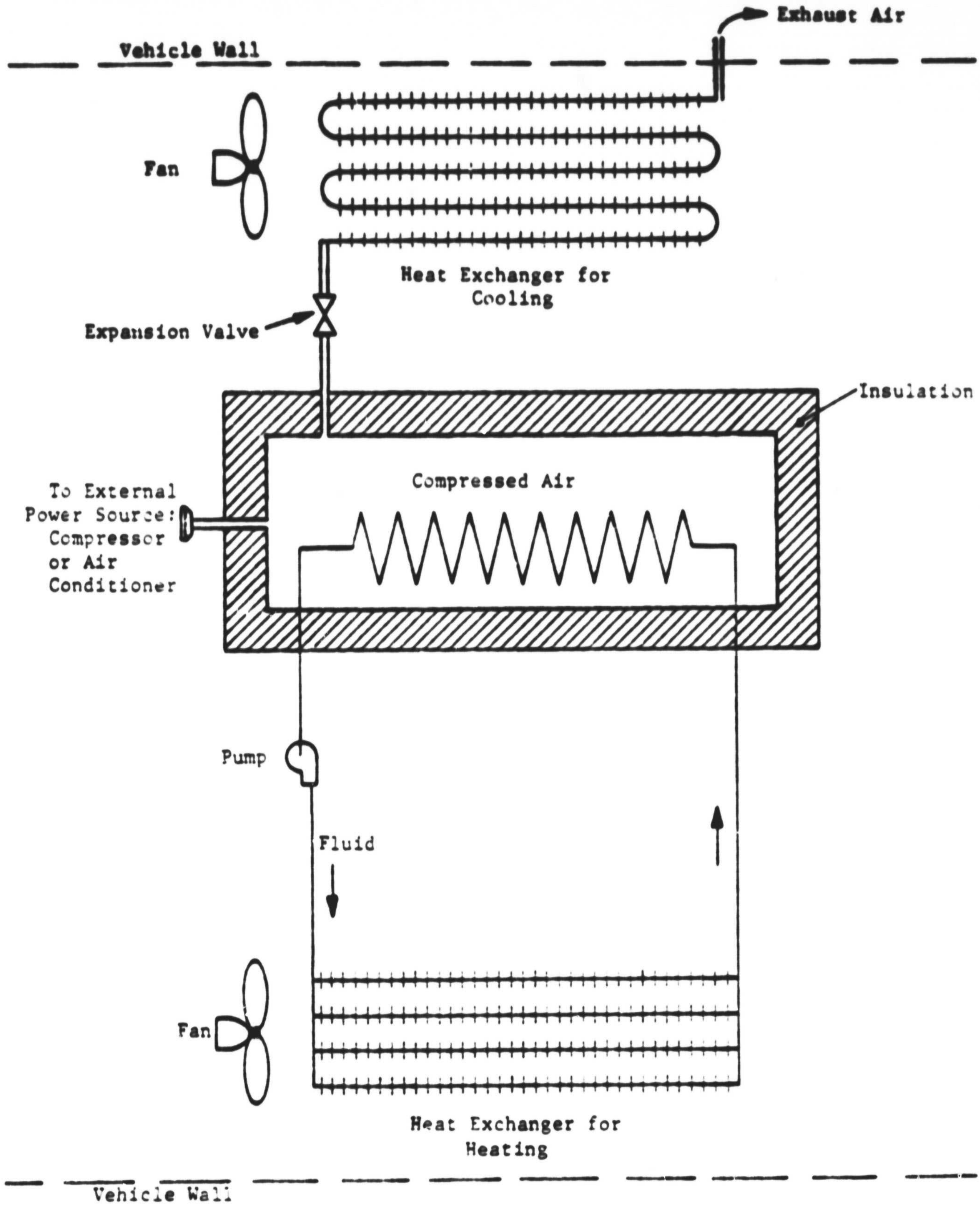

Fig. 3-6 General Scheme for Compressed Air Energy Storage

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cooling purposes, storing air at a sufficiently low temperature so as to yield adequate temperature differentials during throttling may be preferable to compressing the air adiabatically. To simplify the arrangement, a possibility may exist to construct a single heat exchanger to serve both the heating and cooling modes. Since it is unlikely that both systems would be needed during a given period, it would only be a matter of connecting or switching the heat exchanger from one mode of operation to the other.

3.8.1 Feasibility of Systems

The relation between the thermodynamic properties of air for adiabatic compressors is given by:

$$\left(\frac{p}{p_a}\right) = \left(\frac{T}{T_a}\right)^{\frac{\gamma}{\gamma-1}}$$

Figure 3-7 gives this relation for air being compressed from atmospheric conditions of p_a = 14.7 psia and 80°F. Keeping generally to the temperature levels of the previous systems, the conditions of 400 psia at a temperature of 920°F can be used for the compressed state. The amount of heat that would be extracted from compressed air (at a constant volume) would be given by:

$$Q = Wc_v\ (\theta_s - \theta_a)$$

For our requirements,

$$42,500 = W \times 0.171(920 - 100)$$

or

$$W = 304 \text{ lb}$$

From pV = W RT, the volume required is:

$$V = \frac{304 \times 53.3 \times 1380}{400(144)} = 352.9 \text{ ft}^3$$

Thus, compressed air is not likely to be a candidate for energy storage aboard a passenger vehicle.

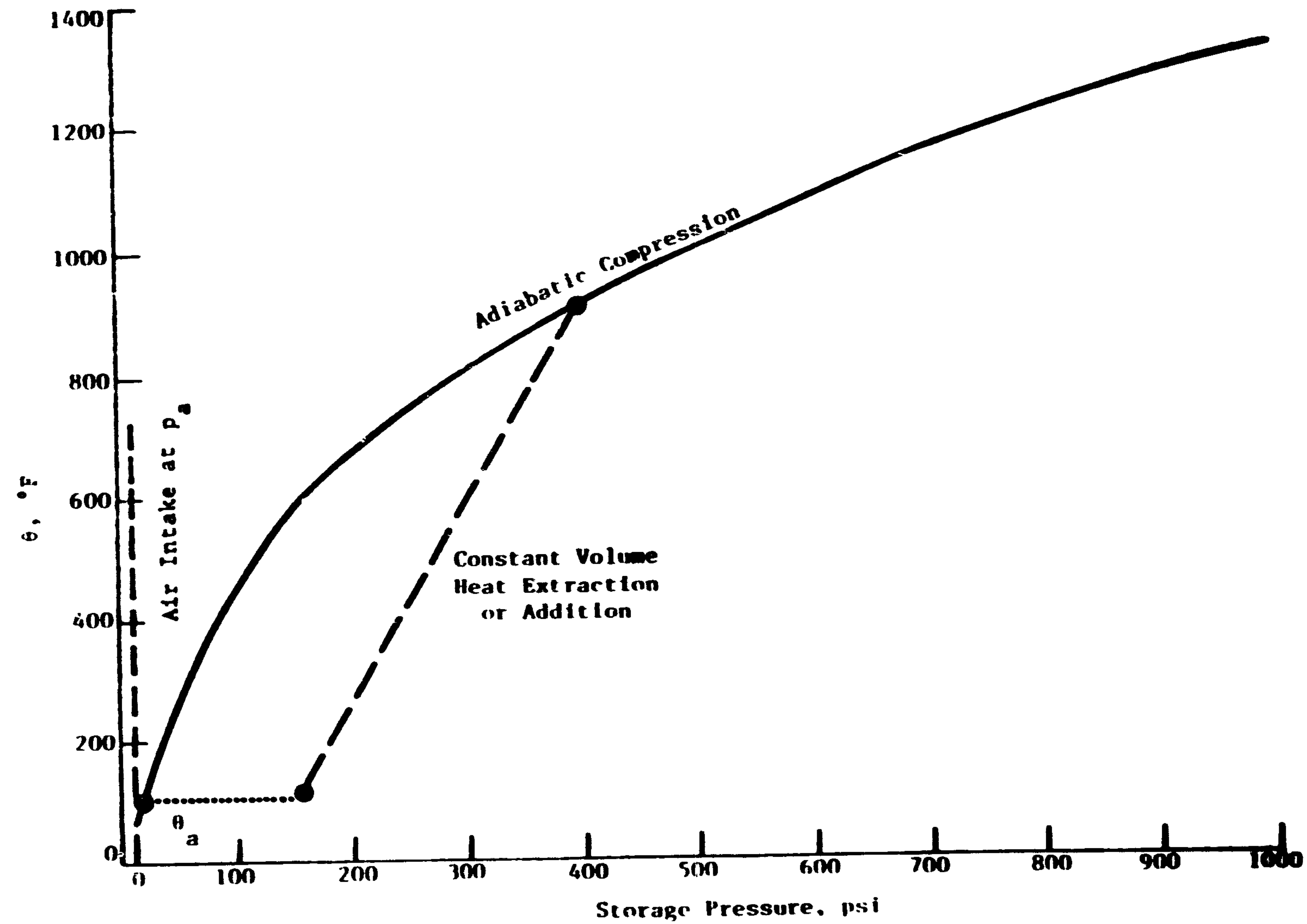

Fig. 3-7 Pressure-Temperature Relationship for Compressed Air

4.0 REVERSIBLE THERMOCHEMICAL REACTIONS

The write-up on thermochemical storage systems will be modest for the simple reason that it is a new field and relatively little information is available. This limited information pertains not only t the systems as a whole but even with regard to some of the chemical and thermal features of utilization of these reactions.

4.1 Ammoniated Salts

The system utilizes a pair of reversible ammoniated salt reactions; one operates at ar elevated temperature (170°F to 620°F) and the other operates at a near-ambient (or below) temperature [33]. The reactions are reversible and the process reproducible. Because the system can operate close to equilibrium and has no moving parts, it can possess an efficiency close to that of a Carnot cycle. The two reactions always operate at essentially the same pressure in a range from 0.5 to 4 atmospheres. The system accepts thermal energy at an elevated temperature, converts part of it to chemical energy and the remainder to near-ambient temperature thermal energy. In reverse, the system absorbs near-ambient temperature thermal energy, adds the stored chemical energy and regenerates the original thermal energy at close to the original temperature. Because the system stores only temperature-independent chemical energy and ambient-temperature thermal energy, losses to the environment can be made negligibly small and the storage time can be infinitely long.

Values for the heats of reaction and the reaction temperature at a pressure of 1 atmosphere for each of these seven re[illegible]s are listed in Table 4-1. The respective vapor pressure curves are s[illegible]m [illegible] Figure 4-1. As can be seen from the figure and table, any s[illegible]rce temperature between 170 and 620°F can be utilized by pairing one of th[illegible] six reactions in Table 4-1 with the near-ambient temperature $CaCl_2$ [illegible]action.

4.2 Liquid-Solid Reactions

Tables 4-2 and 4-3 give a listing o[illegible] a number of other possible thermochemcal reactions. What is of p[illegible]inent interest here, is that the average heat

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TABLE 4-1*

AMMONIATED SALT REACTIONS

Reaction	T, (°F) at 1 atm	Heat of Reaction (Btu/lb of Reactants)
1) $MgCl_2 \cdot 2NH_3 \rightleftarrows MgCl_2 \cdot NH_3 + NH_3$	529	511
2) $MnCl_2 \cdot 2NH_3 \rightleftarrows MnCl_2 \cdot NH_3 + NH_3$	478	375
3) $CaCl_2 \cdot NH_3 \rightleftarrows CaCl_2 + NH_3$	413	229
4) $CaCl_2 \cdot 2NH_3 \rightleftarrows CaCl_2 \cdot NH_3 + NH_3$	356	359
$MgCl_2 \cdot 6NH_3 \rightleftarrows MgCl_2 \cdot 2NH_3 + 4NH_3$	275	676
6) $MnCl_2 \cdot 6NH_3 \rightleftarrows MnCl_2 \cdot 2NH_3 + 4NH_3$	196	565
7) $CaCl_2 \cdot 8NH_3 \rightleftarrows CaCl_2 \cdot 4NH_3 + 4NH_3$	89	571

*Ref [33]

TABLE 4-1*

AMMONIATED SALT REACTIONS

Reaction	T,(°F) at 1 atm	Heat of Reaction (Btu/lb of Reactants)
1) $MgCl_2 \cdot 2NH_3 \rightleftarrows MgCl_2 \cdot NH_3 + NH_3$	529	511
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4) $CaCl_2 \cdot 2NH_3 \rightleftarrows CaCl_2 \cdot NH_3 + NH_3$	356	359
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7) $CaCl_2 \cdot 8NH_3 \rightleftarrows CaCl_2 \cdot 4NH_3 + 4NH_3$	89	571

*Ref [33]

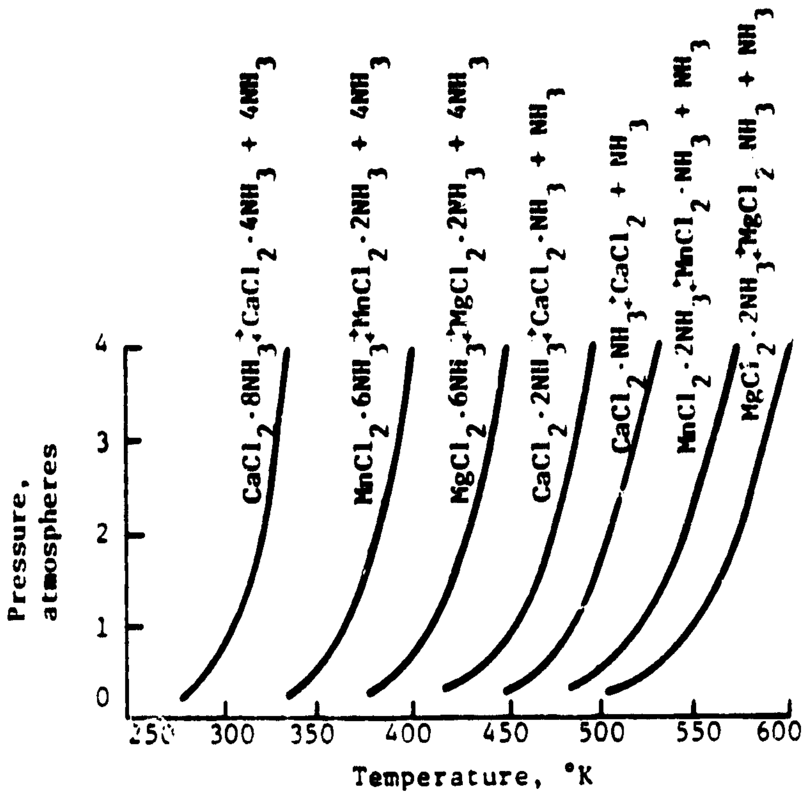

Fig. 4-1 Vapor Pressure Curves for Ammoniated Salts [33]

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TABLE 4-2*

ENERGY CONTENT OF CHEMICAL ENERGY STORAGE REACTIONS EXCLUDING THOSE CONTAINING SOLIDS

Reaction Exothermic ⇄ Endothermic	Reaction Enthalphy at 298°K (77°F)		Temperature (°K) at which	
	J/g	Btu/lb	90% Formed	90% Dissociated
$CO(G) + 3H_2(G) \rightleftharpoons CH_4(G) + H_2O(L)$	7,345	3,160	-	-
$CO(G) + 3H_2(G) \rightleftharpoons CH_4(G) + H_2O(G)$	6,053	2,604	754	1,466
$C_2H_4(G) + H_2(G) \rightleftharpoons C_2H_6(G)$	4,561	1,962	841	1,205
$2CO(G) + 2H_2(G) \rightleftharpoons CH_4(G) + CO_2(G)$	4,118	1,772	778	1,252
$CO(G) + 2H_2(G) \rightleftharpoons CH_3OH(L)$	3,996	1,718	345	434
$N_2(G) + 3H_2(G) \rightleftharpoons 2NH_3(L)$	3,861	1,661	-	-
$N_2(G) + 3H_2(G) \rightleftharpoons 2NH_3(G)$	2,695	1,159	346	528
$2NO(G) + O_2(G) \rightleftharpoons N_2O_4(L)$	1,750	753	549	930
$SO_2(G) + Air \rightleftharpoons SO_3(G)$**	1,544	644	806	1,270
$SO_2(L) + 1/2\ O_2(G) \rightleftharpoons SO_3(L)$	1,517	652	792	1,235
$SO_2(G) + 1/2\ O_2(G) \rightleftharpoons SO_3(G)$	1,235	531	792	1,235
$NO(G) + 1/2\ O_2(G) \rightleftharpoons NO_2(G)$	1,243	535	549	930
$CO(G) + Cl_2(L) \rightleftharpoons COCl_2(L)$	1,172	504	628	881
$NO_2(G) + NO_2(G) \rightleftharpoons N_2O_4(L)$	932	401	288	381
$SO_3(L) + H_2O(L) \rightleftharpoons H_2SO_4(L)$	885	381	535	723
$SO_2(G) + Air \rightleftharpoons SO_3(G)$	727	313	806	1,270
$NO(G) + 1/2\ Cl_2(L) \rightleftharpoons NOCl(L)$	695	299	425	519
$H_2O(L) + H_2SO_4(L) \rightleftharpoons H_2SO_4 \cdot H_2O(L)$	230	99	-	-
For comparison:				
$H_2(G) + 1/2\ O_2(G) \rightleftharpoons H_2O(G)$	13,423	5,775	2,830	5,600

*Ref [23]

**Based on SO_2 weight only. Air open cycle.

TABLE 4-3*

ENERGY CONTENT OF CHEMICAL ENERGY STORAGE REACTIONS, SYSTEMS WITH SOLID CONSTITUENTS

Reaction Exothermic ⇄ Endothermic	Reaction Enthalpy at 298°K (77°F)		Temperature at which $P_{Diss.}$ =		
	J/g	Btu/lb	0.1 bar	1 bar	$P_{Cond.}$
$Li(S) + 1/2\ H_2(G) \rightleftarrows LiH(S)$	11,403	4,906	1,181	1,223	-
$NaF(S) + (HF)_n(L) \rightleftarrows NaHF_2(S)$	4,442	1,911			
$Li_2O(S) + CO_2(G) \rightleftarrows Li_2CO_3(S)$	3,029	1,303			
$Na_2O(S) + CO_2(G) \rightleftarrows Na_2CO_3(S)$	3,014	1,296		2,445	
$Mg(S) + H_2(G) \rightleftarrows MgH_2(S)$	2,893	1,245	~500	560	-
$CaO(S) + SO_3(L) \rightleftarrows CaSO_4(S)$	2,539	1,092			
$CaO(S) + CO_2(G) \rightleftarrows CaCO_3(S)$	1,776	764	1,028	1,171	
$MgO(S) + CO_2(G) \rightleftarrows MgCO_3(S)$	1,387	597		670	
$BaO(S) + CO_2(G) \rightleftarrows BaCO_3(S)$	1,353	582		1,473	
$NiCl_2(S) + 6NH_3(L) \rightleftarrows [Ni(NH_3)_6]Cl_2(S)$	1,301	560			
$NH_3(L) + H_2SO_4(L) \rightleftarrows NH_4HSO_4(S)$	1,256	540	-	-	-
$KF(S) + (HF)_n(L) \rightleftarrows KHF_2(S)$	1,031	444			
$CaO(S) + H_2O(L) \rightleftarrows Ca(OH)_2(S)$	880	378	722	820	676
$MgO(S) + H_2O(L) \rightleftarrows Mg(OH)_2(S)$	644	277	614	649	598
$BaO(S) + H_2O(L) \rightleftarrows Ba(OH)_2(S)$	598	257	1,052	1,271	961
$FeCl_2(S) + 6NH_3(L) \rightleftarrows [Fe(NH_3)_6]Cl_2(S)$	302	129			
$CaCl_2(S) + 6NH_3(L) \rightleftarrows [Ca(NH_3)_6]Cl_2(S)$	210	90			

*Ref [23]

of reaction is of the order of 1,000 Btu/lb. Thus, any of these systems could be a candidate for vehicle environmental control. The weight required would be of the order of 40 lb. The volume required for the gaseous phase would force the exclusion of nearly all of the reactions given in Table 4-2. In Table 4-3, a number of reactions involve only the solid-liquid phases. From among these, the potential candidates would have to be examined. As can be seen, some of these reaction belong to the familiar ammoniated salt families, e.g.

$$CaCl_2\ (s) + 6NH_3\ (\ell) \rightleftarrows [CO(NH_3)_6]\ Cl_2\ (s) \quad .$$

However, this formula's reaction enthalpy is only 90 Btu, requiring a weight of the order of 450 lb, which is forbidding.

4.3 Hydrides

4.3.1 Lanthanide Hydrides - Ni_5LaH_x

Intermetallic lanthanide compounds such as Ni_5La absorb large quantities of hydrogen near room temperature and at a few atmospheres of pressure. A compound Ni_5LaH_x is formed where at fixed temperature the pressure is virtually constant as x varies from 1 to 6. If the compound is in contact with hydrogen at a pressure above the absorption value, hydrogen is absorbed when the heat of absorption is removed until only the hydrogen-rich hydride is present. If the pressure is reduced, hydrogen is liberated at constant pressure until the system is almost exhausted, in a manner shown in Figure 4-2. At 4 atmospheres (and below about 110°F) such compounds store hydrogen at a density which would only be obtained under 1000 atmospheres for gaseous storage.

4.3.2 Iron Titanium Hydrides - $FeTiH_x$

Iron titanium hydride ($FeTiH_x$) is a decomposable compound that can serve as a hydrogen carrier at ordinary temperatures and moderate pressures. Table 4-4 shows the compound's properties. As shown in Figure 4-3, the value of x in this intermetallic compound approaches two, under equilibrium conditions, using ultrahigh purity hydrogen ($\geq$99.999%) at 1000 psia.

The manner in which the equilibrium pressure of hydrogen varies with hydride composition is shown in Figure 4-3 for a system temperature of 104°F.

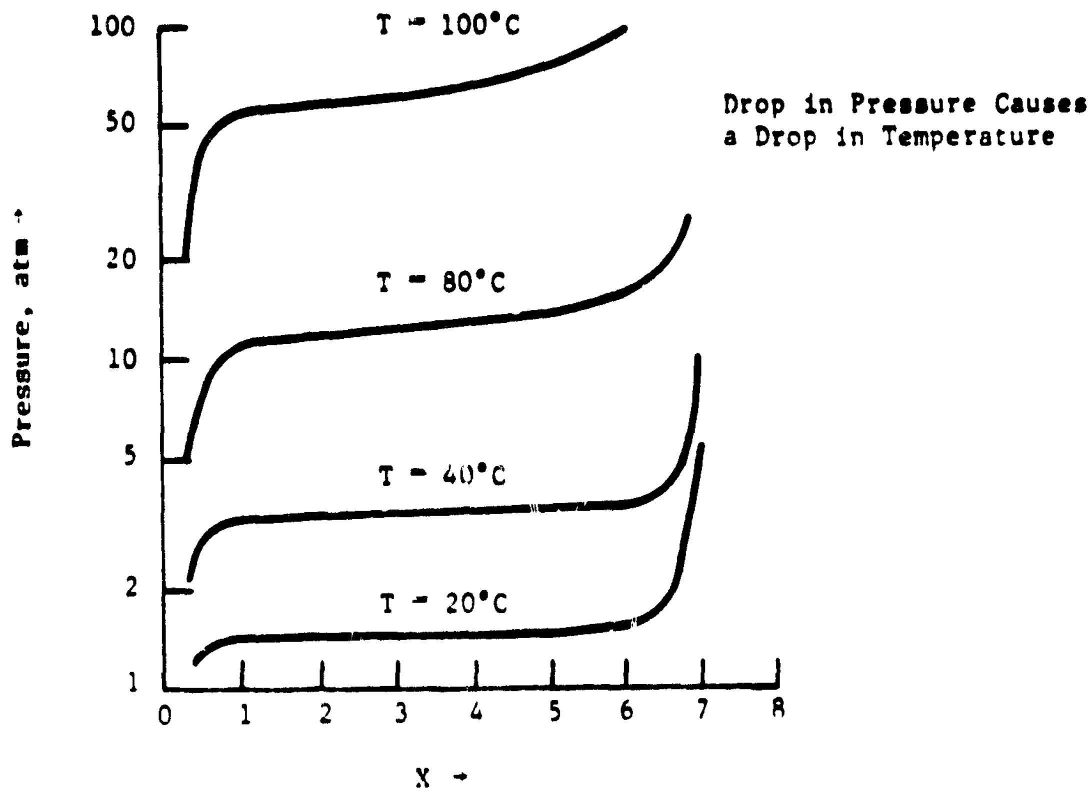

Fig. 4-2 Storage of H_2 in Ni_5LaH_x (Lanthanide Compound) [?3]

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TABLE 4-4

PROPERTIES OF HYDRIDES

Property	$FeTiH_{1.6}$	MgH_x (10 wt% Ni)
Hydrogen content (wt%)	1.5	5.2 (6.9 equilibrium)
Bulk density (lb/ft^3)	200	56
Heat of dissociation (Btu/lb H_2)	6,300	16,650
Heat capacity (Btu/hr·°F)	0.15[a]	0.25[a]
Thermal conductivity (Btu/hr-°F-ft)	1.0[b]	0.3[b]

[a]Estimated

[b]Granular solid in hydrogen at 1 atmosphere

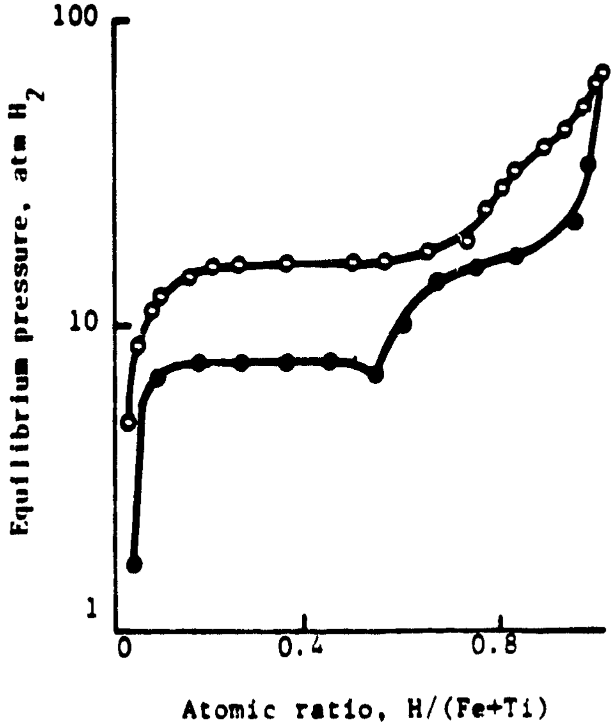

Fig. 4-3 Pressure-Composition Relationship for $FeTiH_x$ at 40°C (104°F) [34]

Separation of the upper curve for hydrogen reaction, and the lower curve for hydrogen dissociation, describes a hysteresis effect characteristic of the system. During hydriding, the supply pressure must exceed the reaction pressure to provide a driving force for the reaction and similarly, during dehydriding, the dissociation pressure must exceed the pressure of the external system.

The low thermal conductivity of the hydride is the restricting factor in the overall heat transfer coefficient. Various other facets of $FeTiH_x$ behavior complicate its use as a storage medium for hydrogen. Impurities in the hydrogen, such as oxygen or water vapor, can poison the bed and reduce its storage capacity. Particle attrition occurs during hydride-dehydride cycling, and volumetric expansion during hydriding offers additional complications.

4.3.3 Magnesium Nickel Hydride - MgH_x

The hydride of Mg (10 wt% Ni) alloy stores hydrogen reversibly and is a potential hydrogen carrier. As shown in Table 4-4, its density is considerably lower than that of $FeTiH_x$ and its hydrogen content much higher (5.5% by weight). MgH_x operating temperature is close to 570°F, and its heat of dissociation, approximately twice that of $FeTiH_x$.

The best known alloys are MgCu and MgNi. Both require heating to 572°F to function as storage media. The alloy recommended for practical use is MgNi (10% Ni by weight). This alloy contains about 6.9% hydrogen by weight at equilibrium, and can probably store about 5.2% hydrogen under dynamic conditions. The Mg alloy's heat of dissociation and other properties of interest are listed in Table 4-4.

The alloy containing 10% copper by weight has similar properties and behavior; two isotherms for its dissociation are shown in Figure 4-4. Both the nickel and copper in these alloys act as catalysts in the formation of MgH_2 over the composition range corresponding to the pressure plateau.

Iron titanium hydride is a relatively benign material whereas magnesium-based hydride requires materials capable of withstanding its pyrophoric tendency. Hydrogen gas release from these materials is a potential hazard.

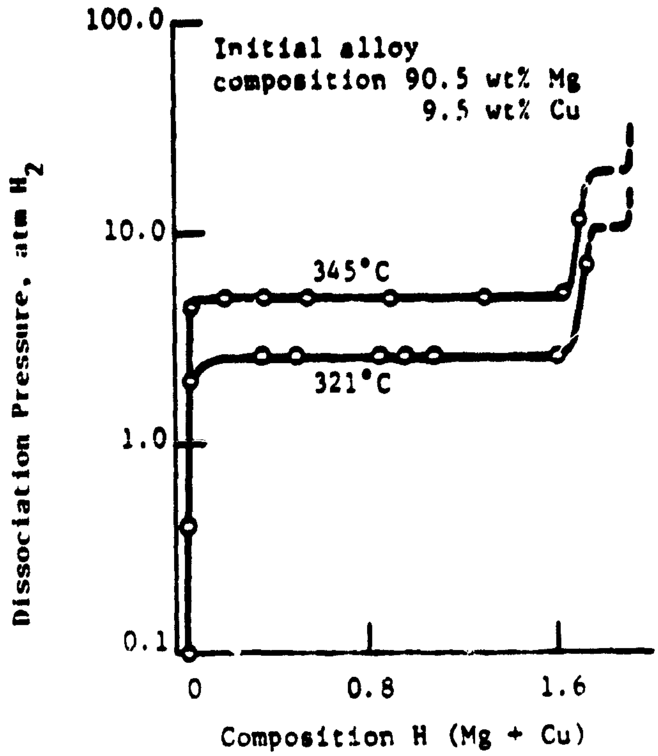

Fig. 4-4 Pressure-Composition Relationship for Dissociation of MgCu Hydride

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Considering only the heat of dissociation of these hydrides, their potential storage capacity is as follows:

Hydride	Amount, lb	Hydrogen, lb	Heat of Dissociation, Btu	Volume, ft^3
$FeTiH_x$	450	6.75	42,500	2.25
MgH_x	50	2.60	42,500	0.90

Thus, the magnesium hydride is one of the most attractive materials for purposes of car heating or cooling, provided the previouslv mentioned complications can be overcome.

5.0 CONCLUSIONS

Over 80 schemes were studied to determine their suitability for providing heating and/or cooling of electric vehicle passenger compartments. Preliminary calculations indicate that more than 30 of these schemes are capable of being developed into practical schemes without undue penalties on vehicle weight and propulsion energy usage, and hence, on vehicle range and performance.

More schemes appear to be available for heat than for cooling. However, few heating schemes have been identified which will weigh less than 60 lb and will be smaller than 0.5 ft^3 to provide adequate heating at ambient temperatures as low as -10°F. Some of the schemes are attractive enough to consider installing one for cooling and one for heating while retaining a practical package. Six schemes are capable of providing both cooling and heating, with essentially the same hardware, by only a change in mode of operation.

On-board storage of petroleum fuel is required only in three cases. In most of the other cases, energy is stored directly in the form of heat (or cold). In some cases the energy is stored in chemical form.

The nature of these tasks in the study was only to identify and describe various schemes. Hence, only preliminary calculation were performed to determine their viability in gross terms such as approximate weight and volume. In the next task, many of these schemes will be eliminated, and a few of the more promising schemes will be pursued to develop more detailed information.

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1. General Approach

The Environmental Control Systems (ECSs) for electric vehicles discussed in Appendix C are to be ranked with respect to a number of criteria selected for their importance to system selection and use. A scheme for determining relative, quantitative scores, derived from the generalized formula set forth by JPL in its RFP for this work, is:

$$\text{Score (system)} = \sum_{i=1}^{M} W_i \; S_i \; / \sum_{i=1}^{M} W_i, \text{ Where} \qquad (1)$$

W_i = the weighting factor assigned to the "i"th criterion to define its importance relative to the other criteria

S_i = actual score given to the ith criterion for this system

The ranking scores are to be developed from this formula by the following process:

- Defining the criteria to be applied to the rankings.
- Assigning relative weighting factors (W_i) to the criteria. For each criterion, a W_i is thus established that describes the relative importance of the criterion to the systems' total score with respect to all other criteria.
- Quantifying for each system, a set of values for the S_is (i=1, ..., M) from the system evaluation data.

In order to apply equation (1) correctly it is necessary that scores S_i be non-dimensional quantities. This is achieved as follows:

- Select one particular system as a reference system.
- Give a score of 1 for each of the criterion for this reference system. Thus for this reference system:

  $S_i = 1$ (For i=1, 2.....M)

  and

  $$\text{Score System} = \frac{\sum_{i=1}^{M} W_i}{\sum_{i=1}^{M} W_i} = 1 \qquad .(1)$$

- For any other system obtain the score for "i" th criterion by comparing the system under consideration with the reference system for that particular criterion. This comparison is made to determine how much "better" (or more desired) a given system is from the consideration of "i" th criterion. Thus, for the cost criterion, the lesser the cost the better; while for the storage period criterion, larger values are considered more desirable. For example, let us consider that we want to arrive at scores S_1 and S_2 for criterion cost (criterion 1) and volume (criterion 2) for system A. The following table shows the values for system A and another system B which is selected to be the reference system.

System	Cost	Storage period
A	\$500	2 days
B (Reference)	\$700	5 days

From this information the scores S_1 and S_2 for system "A" are worked out as follows:

$$S_1 = \frac{700}{500} = \frac{\text{Value for reference system}}{\text{Value for system A}}$$

$$= 1.4$$

$$S_2 = \frac{2}{5} = \frac{\text{Value for system A}}{\text{Value for reference system}}$$

$$= 0.4$$

Note: Both S_1 and S_2 are non-dimensional quantities. For the cost criterion, smaller costs are desirable. Hence in working out the nondimensional score S_1, the cost of the reference system is in the numerator. Thus, systems with smaller costs will receive scores greater than 1, indicating that they are better; while the system with higher cost will receive a score less than unity, indicating that it is a less desirable system. The magnitude in either case is an indicator of the degree of desirability.

In the case of the storage period, the reverse is true. Hence, the value of the reference system is in the denominator. This gives a system with higher storage period a score higher than 1, indicating that it is a more desirable system.

- For each system, establish a score from equation (1).
- Rank the systems according to their overall scores. The process is structured such that the better scores will be associated with the higher total values.

Technologies evaluated in this report are for the most part not sufficiently developed to allow for immediate design, manufacture and incorporation into electric or hybrid vehicles. There are thus two major groups of focus in judging the relative value of the ECSs: the manufacturers (automobile, and ECS where different) that would incorporate the systems into the vehicles; and the vehicle purchasers. The purchasers are ulitmately the more important group, since they will make the purchase decision. However, the choices and concerns of the manufacturer are important, since this is the group that will select the system or systems that will be offered in the vehicles. Over any given period, a small number of systems are likely to be developed and offered in the commercial vehicle.

In addition to addressing the concerns of these two groups, the ranking must consider the impact on national concerns (energy management, environmental quality, vehicle safety, etc.) that the ECS options present. At this stage, the choice of technology, may have the chance to affect energy management patterns (ECS system efficiencies, fuel type used on - and off - board, etc.). Use of such a criterion as energy efficiency is important for the ranking scheme for this standpoint, although the energy efficiency of ECS alone might not appreciably affect purchaser or manufacturer choice preference. Other national concerns such as vehicle safety and environmental quality must also be considered: regulatory programs may place limits on acceptability that must be met before a system can be futher considered; purchaser and manufacturer preference may have an additional impact on relative ECS ranking from these standpoints.

The choice of ranking criteria and weighting factors (W_i) is thus multi-dimensional. Several viewpoints must be considered on their development. The newness of the technologies of ECS and vehicle, also limits the ability to apply meaningful quantitative techniques without much more substantive data. In viewing this situation within the limited scope of ranking tasks available at this time, MTI has chosen to use an approach for quantifying the scores based on its experience gained during the project.

In theory, the evaluation of a wide variety of ECS options as they offer potential for effective use in the vehicles has afforded the project team with sufficient experience to structure a meaningful set of criteria, to weigh their relative importance to the ECS's overall score, to develop quantified information on the criteria, and to then rank the ECS's based on the overall scores from Equation (1). The strength of the results, however, depends mainly on whether the reader agrees with the approach MTI has taken. It is possible to take another approach that would provide an entirely different set of rankings.

2. Criteria Definition and Weighting

The ranking criteria and their weighting factors are shown in Table 1. They are grouped into four major groups of criteria that identify separate types of system characteristics:

- Capital Cost Characteristics, including first cost and system life.
- Use Characteristics, including storage period, range impact, energy efficiency and maintenance cost.
- Environmental and Safety Characteristics, including consumer perception of safety, noise, and other environmental impacts.
- Development and Manufacturing Characteristics, including weight, ease of packaging and volume, and development cycle through commercialization.

2.1 Capital Cost Characteristics

The first cost (W_i = 25) is viewed as the most important single criterion. All else being equal, both the consumer and the manufacturer will select the least cost alertnative. For at least the first years of heavy electric or hybrid vehicle use, the consumer will be presented with a relatively expensive vehicle compared to ICE vehicles. The first cost of components will thus be important to control. The vehicle manufacturer will take a similar view with the intent of being competitive with the products of other vehicle manufacturers. For a technology at the stage of development of most ECS options, the focus of the manufacturer, who will have a large input into system selection, is very important. In addition, a number of the ECS alternatives require auxiliary equipment (e.g., heaters or refrigerant equipment in the garage) that can add a significant amount to first cost. Out of a total of 100 points for all weighting factors, the first cost is given 25 points.

System life is also an important consideration for ECS options whose projected useful life is significantly less than the useful life of the vehicle. Such a discrepancy can add to the consumer's perception of capital cost. A good example has been the short (2-3 years) projected life of currently commercial power batteries for electric vehicles. This short life is a significant factor in increased market penetration of these vehicles. The system life is given a weighting factor of 5 points.

TABLE 1

CRITERIA AND ASSOCIATED WEIGHTING FACTORS

Criterion	Weighting Factor (%)
Capital Cost Characteristics:	
1. First Cost	25
2. System Life	5
Use Characteristics:	
3. Range Impact	10
4. Energy Efficiency	5
5. Storage Period	5
6. Maintainance Cost	10
7. Performance Impact	15
Environmental and Safety Characteristics:	
8. Consumer Perception of Safety	5
9. System Noise	10
10. Other Environmental Impacts	5
Development and Manufacturing Characteristics:	
11. Ease of Packaging and Volume	5
12. Development Cycle Through Commercialization	5
TOTAL	100

2.2 Use Characteristics

Five criteria are included in this group. They are:

Impact on Range

Energy Efficiency

Storage Period

Maintainance Cost

Impact on Acceleration Performance

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2.2.1 Impact on Range

On the surface the range of electric vehicle appears to be considered as a predominant barrier for successful introduction of electric vehicles. However the statistics which we had cited in our report on "Travel Scenarios" MTI Report Number MTI 80TR46 July 3, 1980, shows that work related travel is of the order of 22 miles a day and shopping and other non-work related travel is of the order of 20 miles a day. Most electric vehicles under development now and in particular D.O.E.'s ETV-1 has a range far exceeding these requirements.

The impact of ECS on reducing the range of electric vehicles comes from two reasons:

1. Some ECS schemes would require using energy from on-board batteries and hence less energy will be available for propulsion unless additional batteries are provided for taking care of ECS needs. If in fact the additional batteries provided for this purpose are compatible with those used for propulsion, they will increase the range on most days as ECS energy requirement is only a fraction of the maximum on most days in a year.

2. The added weight requires more energy for propulsion per mile. However, the reduction in range due to this is relatively small unless the ECS weight is prohibitively high.

It therefore appears, that adequate range will be available for routine needs of an average family even though a slight reduction in available range occurs due to installation of ECS.

However, from a consumer perception the reduced range might be a serious handicap. Thus, the weighting factor of 10 is considered to represent the relative importance of the impact of range reduction.

2.2.2 Energy Efficiency

The number of hours of utilization of an electric vehicle or for that matter any family automobile is very low. Our report on "Travel Scenarios" cited above in the Section 2.2.1 refers to a statistics showing a weekly use of car in the order of 10 hours. This amounts to only 500 hours a year. Combined with this is the fact that the full capacity of ECS is used only on a rare occasion. Such a rare occasion being longest driving time occuring at the worst weather conditions. This being so, annual energy delivered by ECS for space conditions of the passenger compartment can be estimated to be of the order of 1 to 1.5 million Btu a year.

If one assumes reasonable values for efficiencies, the yearly energy budget for providing space conditioning is in the order of 2 to 2.5 million Btu.

Additional energy consumption will result from carrying added weight around all year. However, the effect of this is to be considered elsewhere.

It is thus clear, the the energy efficiency of the ECS is a consideration of relatively minor importance. The annual energy variation between 100% efficient and 50% efficient system being at the most equivalent to 10 gallons of gasoline. Hence, a weighting factor of 5 is assigned to this criterion.

2.2.3 Storage Period

Some of the ECS schemes utilize thermal storage at higher than ambient temperatures for heating schemes and lower than ambient temperatures for cooling schemes. In such schemes the stored heat is slowly lost from the thermal storage device. The rate of such heat loss depends on the extent of thermal insulation provided. The level of insulation that can be provided is limited for reasons of space and cost. In such schemes, therefore, the period over which useful amount of energy can be extracted is rather limited. For normal use a period of 10 to 12 hours is all that is needed, the energy being recharged at night. However, for exceptional uses a longer period of storage might be desirable. An example of such use is parking the car at the airport when one is going on a business trip. It is conceivable though, to be able to take care of such oddities, once an appropriate infra-structure particularly suitable for electric vehicles is developed. Thus the storage period is important only in the early phases of electric vehicle commercialization. The relative importance of this criterion is considered to be represented by assigning a weighting factor of 5.

2.2.4 Maintainance and Operating Cost

A rational buying decision, in theory, requires looking at life cycle costs - in which time value of money is appropriately considered. In such analysis the first cost has higher weighting than the recurring costs during the lifetime of equipment. A life cycle cost analysis can be performed with reasonable accuracy only in the case of equipment which has a minor design modification from the equipment which has been in service for an extensive period. For newly developed equipment, the accuracy of life cycle cost is such as to not warrant detailed work as a large number of quantities required for such analysis are uncertain. We can take into consideration this effect equally well by assigning a weighting factor of 10.

2.2.5 Impact on Acceleration Performance

The weight of ECS affects the vehicle in three ways:

- Added structural weight to support the ECS.
- Reduced range due to increased energy consumption.
- Reduced acceleration performance.

The first item will result in increased vehicle cost and will also affect the other two. The impact of second item is considered a criterion in 2.2.1.

As regards the third item we observe that the electric vehicle acceleration performance is already very poor. Addition of weight will directly reduce it further. This is therefore an important consideration for selection between various alternatives for ECS and is given a weighting factor of 10. The relative scores for this criterion can be determined from the relative weights of on-board components of different ECS systems.

2.3 Environmental and Safety Characteristics

It is assumed that all feasible ECS's will meet minimum government environmental and safety standards.

Consumer perception of safety may affect acceptance of an ECS and is given a weighting factor of 5 points. In some cases regardless of the actual safety of equipment, the perception of safety by the consumer may affect ECS market penetration. This particularly applies to technologies that have not, in their general form, been used in on-the-road vehicles to date.

System noise is also a consideration. Focus is placed primarily on internal noise levels, although external emissions will become a consideration for particularly noisy ECS options. System noise is given a weighting factor of 10 points.

Other environmental impacts, such as air pollution and hazardous materials release at ECS failure, are given a weighting factor of 5 points. Most such impacts will be alleviated by meeting regulatory requirements. The criterion thus measures such incremental environmental impacts as air emissions below regulated levels.

2.4 Development and Manufacturing Characteristics

Ease of packaging and volume are treated in this group as considerations applying to the development and manufacturing. The impact to the consumer of these effects is included above in other criteria such as range impact. This criterion thus measures the limits placed on development and production design by the ECS. This criterion is given a weight of 5 points.

The volume of an ECS is important only if it is excessive. More important than volume is the ability to distribute the volume throughout the vehicle in a manner that can make use of available space within the vehicle envelope.

Each ranked ECS option is considered to be a commercial possibility within the near term. The ECS options, however, would need different development periods before commercialization would be possible. Thus, within the general limits of acceptability, each ECS is given a ranking for this criterion. The criterion is given a weighting factor of 5 points.

E.0 INTRODUCTION

A ranking scheme is developed in Appendix D. To follow that ranking scheme, marks had to be assigned to different criteria for every system. These marks for any given system were obtained as follows.

A conceptual schematic was developed for the system reflecting all the functions the system is required to perform. Very preliminary calculations were performed to determine the sizes of various components of the system. In such calculations, reasonable engineering judgment was employed, based on previous experience with similar systems.

The weight and volume of the part of the system carried on the car was evaluated from these preliminary calculations. The weight information was used in the formula for determining the reduction of acceleration and range.

The costs were determined next. In some cases, costs were determined with reasonable certainty due to the similarity with systems available in the market today. In one case, a detailed proprietary cost study of a similar system in a different rating was available. This study was used to project the cost of the system with projections and modifications based on system characteristics, engineering judgment, and experience with similar systems' manufacturing cost studies. In other cases, cost was determined by obtaining information on materials cost from telephone inquiries, made with volume production in mind. The total cost of materials was obtained by summing the costs of materials for the individual components. The labor cost with burden was assumed to be $1.50/lb of hardware material, which is typical of similar equipment. A profit of 20% was then added to obtain the retail selling price in terms of 1981 dollars.

Assigning marks to the other criteria was based on judgement, since determining them more accurately will require work which is beyond the scope and nature of this study.

In the following paragraphs, calculations and the rationale used for each of the systems retained for ranking are presented.

E.1 GASOLINE-ENGINE-DRIVEN, HEAT-ACTIVATED HEAT PUMP

This system was chosen as the base line system for the Rankine study mainly because accurate data were available for quantifying each of the rating parameters.

E.1.1 First Cost

Cost was determined in the following fashion. First the heat exchanger refrigeration loop costs were determined and then the engine driver costs were determined. Total costs were assumed to be the sum of these two. The cost quotes for this and all other systems are at the uninstalled retail price level. Refrigeration loop costs were established from conventional car air-conditioning system prices.

The price of an after-market packaged air-conditioning system for the capacity size of interest is quoted from a local dealer at $425.00. This price was increased to reflect the difference between refrigeration loop costs for residential heat pump versus refrigeration loop costs for a residential central air-conditioning system. The ratio is typically 1.5. This led to a refrigeration loop cost of $650.00. Assuming a refrigeration loop COP of 3.0 in the cooling mode led to a required engine power of about 2.25-hp shaft output power.

An air-cooled Onin gasoline engine model OJA1 was selected as a typical prime mover for power requirement in this size range. The engine efficiency is equal to 20%, and the engine weight, including fuel tank and control system, is equal to 80 lb. A local dealer quote of $100.00 for this engine package was established.

Therefore, the total cost for the gasoline-engine-driven, heat-activated heat pump was set at $750.00. Both the refrigeration loop and the engine are rated for operation of greater than 3000 hours (typical of vehicle operating life). Therefore, system life was assumed to be adequate and a rating of 1.0 was given.

E.1.2 Range Impact

The impact of the ECS on vehicle range was established from two separate considerations: additional weight and parasitic power required. Total weight for this system was equal to 175 lb. The formula that was used to indicate the reduction in range due to weight is given by: per unit range reduction = 0.00026 X ECS weight. Total weight impact for this system is 4.6% reduction in range.

Parasitics are assumed to run continuously during the operation of the ECS. Under maximum utilization, the ECS would be operated for about 2 1/2 hours. The impact on vehicle range due to the parasitics was determined by multiplying the parasitic power requirement by 2 1/2 hours. It was further assumed that the parasitic power was supplied by the vehicle battery pack. Therefore, the reduction in range was equal to total parasitic energy demand divided by 18 battery packs of 1 kilowatt-hour energy each.

Parasitic power to drive the evaporator and condensor fans was established by looking at typical power requirements for standard heat pump equipment. In the residential class equipment under 3 tons, parasitic power requirements of about 243 watts per ton are typical. Based on nearly a 1 1/2-ton requirement, a parasitic power requirement of 365 watts was established for this system.

Parasitic power consumed for this vehicle was based on a parasitic load of 365 watts yielding a range impact of 5%. The total range impact for the gasoline engine heat pump was determined to be 9.6%.

E.1.3 Energy Efficiency

The correct evaluation for comparative purposes should be based on seasonal performance factors (SPF) using appropriate weather data, equipment efficiency characteristics, and daily utilization periods. Such calculations are considered beyond the scope of this study.

For the purpose of this study, an approximate equipment energy efficiency value is obtained by taking the average COP during heating and cooling seasons at the equipment design points and reflecting this value to the source of energy. For electric energy, COP will refer to fossil fuel sources, utilizing 33% conversion efficiency from fuel to wall-plug electricity. For the gasoline-engine-driven heat pump system, the energy efficiency number is computed under the following assumptions:

- Thermal efficiency of the engine = 0.2
- COP summer = 3.0
- COP winter = 2.0

Hence, the energy efficiency number

$$= \left(\frac{3 + 2}{2}\right) \times 0.2 = 0.5 \quad .$$

E.1.4 Storage

For the gasoline-engine-driven system, a storage period was considered to be an indefinite period of greater than six months.

E.1.5 Maintenance Costs

An assumption was made that the gasoline engine would have to be maintained at least once a year in terms of standard oil change and lubrication. The assumed charge for this was $20.00 for a one-year checkup. It was further assumed that the levelized maintenance costs for recharging the refrigeration loop would be an additional $15.00 per year, totaling $35.00 per year maintenance costs for this system.

E.1.6 Performance Impact

The acceleration performance is mainly affected by the added weight. The reduction in the acceleration can be assumed to be directly proportional to the ratio: original weight ÷ (original weight + the weight of the ECS system). Using a curb weight without the ECS for DOE's electric vehicle ETV-1 of 3320 lb, the reduction in acceleration for this system is computed as:

$$\text{acceleration reduction} = \frac{3320}{3320 + 175} = 0.95 \quad .$$

E.1.7 Consumer Risk

The components involved in this system are commonly known and recognized by most consumers. Therefore, the consumer-perceived safety risk was considered to be low and a value of 2 was designated for this system.

E.1.8 Noise

Noise levels for small gasoline (single-cylinder engines) are notoriously high. The noise level for this system was expected to be higher than any of the others considered, and a value of 10 out of 10 was given.

E.1.9 Environmental Impact

Electric vehicles are thought of as being inherently clean and quiet. Therefore, noxious exhaust fumes that would be given off by the gasoline engine could have a negative and undesirable impact upon consumer opinion. Therefore, a negatively high value of 10 out of 10 was given to this system.

E.1.10 Packaging Volume

Problems that would be encountered in packaging this system would be different that those encountered in standard automotive ai -conditioning systems. A score of 5 was given to this system.

E.1.11 Development Status

Development status was based on the six-point military development ranking: 6.1 is basic research, 6.2 is exploratory development, 6.3 is advanced development, 6.4 is engineering development, 6.5 is field test, and 6.6 is full procurement. It was expected that for this system development activity would center around controls and system integration. Therefore, a 6.35 advanced development status was assigned to this system.

E.2 STIRLING-ENGINE-DRIVEN, HEAT ACTIVATED HEAT PUMP

E.2.1 First Cost

The same refrigeration loop costs as for the gasoline-engine-driven system were assumed here. Based on MTI's current development goals of the Stirling engine, a cost of $170.00 was established for the prime mover in this size range, yielding a total system cost of $820.00.

E.2.2 System Life

Life for this system was assumed to be greater than 3000 hours. A ranking equal to that of a gasoline-engine-driven system was given.

E.2.3 Range Impact

The total weight penalty of 200 lb led to a range impact of 5%. A parasitic power requirement of 415 watts led to a range impact of 6%, yielding a total impact of 11%. (The power requirement for the combustor blower is assumed to be 50 watts.)

E.2.4 Energy Efficiency

The assumptions made concerning efficiency were as follows:

- Engine efficiency = 0.4
- Average COP of refrigeration loop = 2.5.

Hence,

energy efficiency number = 2.5 x 0.4 = 1.0 .

E.2.5 Storage Period

The storage period for this system, like the gasoline engine system, would involve storage of a liquid fuel, leading to an extremely long storage period.

E.2.6 Maintenance Cost

Another feature of the Stirling engine is the requirement for little or no maintenance during the short life required in this application. Since the

entire system is hermetically sealed, the maintenance cost is assumed to be $10/year.

E.2.7 Performance Impact

Performance impact is computed by the formula given for the gasoline-engine-driven heat pump system. For this Stirling engine system, the ratio = 3320/(3320 + 200) = 0.945.

E.2.8 Noise

Engine noise would be extremely low. However, some noise would be associated with the combustion and evaporator air fans. The value of 5 out of 10 was given.

E.2.9 Environment

Emissions from the Stirling engine, both chemical and noise, are known to be relatively low. Therefore, a relative value of 5.0 was assigned.

E.2.10 Packaging and Volume

The packaging for this system would be typical of air-conditioning designs. However, as the volume is expected to be larger than that of the gasoline-engine-driven system, a relative score of 6 was given to this system.

E.2.11 Development Status

Stirling engine component development has to be established before this system could be viable in a system sense. The status is similar to 6.2 exploratory development in a military ranking. A score of 6 was assigned to this system, relative to the 2 assigned to the reference system.

E.3 AQUA-AMMONIA SPLIT HEAT PUMP SYSTEM

E.3.1 First Cost

Table E.1 is reproduced from an unpublished cost study of absorption systems for heat pumps. This study was performed in 1975 for an aqua-ammonia based, 3-ton heat pump.

A split heat pump system is different from the one used for the study in the following respects:

- Some parts of the system are in the car, while the remaining parts are in the garage.
- The rating of the parts on the car is 17,000 Btu/hr, while the rating of the parts in the garage is 700 Btu/hr based on a 24-hour reprocessing period.
- Storage facilities have to be provided on the car as well as in the garage.

In order to arrive at the cost, the following procedure is used.

1) List the components from Table E.1 which will be required on the car.

2) For each of these parts, assess how much of the total material cost is capacity dependent and how much is independent of capacity.

3) Add up the capacity-dependent costs and multiply by $\frac{17,000}{36,000}$ to reflect the smaller size required on the car, compared to the one used in the cost study of Table E.1.

4) Add the following material costs to the number arrived at in Step 3.

   - Capacity-independent costs of parts on the car from Table E.1.
   - Cost of storage tanks for ammonia and weak solution. (Both tanks are assumed to be made up of steel.)
   - Cost of charging ammonia.

TABLE E.1

COST ESTIMATE OF A 3-TON AQUA-AMMONIA ABSORPTION HEAT PUMP*

Component	Material Cost (1975 dollars)
Generator	25.0
Absorber and Absorber Heat Exchanger	18.0
Liquid Heat Exchanger	4.5
Condenser	15.0
Evaporator	24.0
Precooler	4.0
Solution Pump	52.0
Water Pumps	17.0
Motor, Pump	31.0
Outdoor Coil	85.0
Fan, Motor and Mount	24.0
Burner System	9.0
Controls and Wiring	38.0
8-Way Valve and Motor	18.0
Valves, Service and Relief	6.0
Sheet Metal and Grill	34.0
Charge, NH_3, H_2O, etc.	1.5
Capillaries, Connecting Tubing	10.0
Insulation	9.0
Miscellaneous	10.0
Crating	8.0
Total Material	443.0
Labor	75.0
Burden 225%	169.0
Grand Total	687.0

*Production quantity of 30,000 to 50,000 per year.

This procedure gives the material costs of the parts on the car. The cost of material of the parts in the garage is worked out in identical steps, except that the heat transfer rate of 700 Btu/hr is used for determining the material cost of capacity-dependent parts.

E.3.1.1 Labor Cost

The labor cost is worked out with the following assumptions:

- 50% of the labor cost of the 3-ton system of Table E.1 is capacity independent.
- The rest of the labor cost is directly proportional to the capacity
- The labor cost of the split system is 50% higher than a single system.
- The total labor cost of the 3-ton system of Table E.1 is divided in two part: the car system and the garage system. The division of labor cost is assumed to be in the same ratio as the material costs.

Now, it is shown later that the cost of materials of the 3-ton system can be divided as follows:

Garage System = \$297.50

Car System = \$132.00

Hence,

$$\frac{\text{Material Cost for Car System}}{\text{Total Material Cost}} = \frac{132}{443} \approx 0.3$$

and,

$$\frac{\text{Material Cost for Garage System}}{\text{Total Material Cost}} = \frac{298}{443} \approx 0.7$$

From Table E.1 the total labor cost with burden

= 75 + 169 = \$244 .

Hence, applying the above assumptions, the labor cost of the car system

$$= (244 \times 1.5) \times (0.5 + 0.5 \times \frac{17,000}{36,000}) \times 0.3$$

$$= 366 \times (0.5 + 0.236) \times 0.3 \approx \$81$$

and, the labor cost of the garage system

$$= (244 \times 1.5) \times (0.5 + 0.5 \times \frac{700}{36,000}) \times 0.7$$

$$= 366 \times 0.51 \times 0.7 = \$130 \quad .$$

E.3.1.2 Effect on Inflation

The costs worked out under the above procedures are in terms of 1975 dollars. These costs are converted to 1981 dollars by multiplying by 1.7.

E.3.1.3 Profits

A profit of 20% is added to arrive at the retail selling price.

E.3.1.4 Car Parts

E.3.1.4.1 Ammonia Storage Tank. Required volume (from Appendix C) = 2.1 ft^3.

Assume

- Maximum pressure = 200 psig
- Cylindrical tank length = 2.1 feet
- Sheet thickness = 0.05 inch
- Cost of material (steel) in 1975 dollars = \$0.18/lb.

Hence, the following can be determined:

- Diameter = 1.125 ft
- Surface area = 8.6 ft^2
- Weight of the tank = 17.4 lb
- Cost of steel = \$3.13.

The ammonia storage tank needs to be insulated to prevent excessive pressure rise during hot days. The size of the insulation is determined under the following assumptions:

- **The pressure in the tank is not to exceed 200 psig. Thus, the temperature of the stored ammonia is not to exceed 101°F.**
- **The ambient temperature surrounding the tank space will not exceed 140°F.**
- **The starting temperature of stored ammonia is 55°F.**

Thus, the average temperature of ammonia

$$= \frac{101 + 55}{2} = 78°F \quad .$$

The temperature difference between maximum ambient temperature and the average ammonia temperature

$$= \Delta t = 140 - 78 = 62°F \; .$$

The amount of heat that ammonia can absorb before reaching 101°F

ΔQ_{max} = (Enthalpy at 101°F - Enthalpy at 55°F) x weight of liquid ammonia stored (from Appendix C) = (156.4 - 103.5) x 81.5 = 52.9 x 81.5 = 4300 Btu.

Hence, the number of hours of 140°F ambient as a function of "R" value of insulation is obtained from

$$\text{Hrs} = \frac{\Delta Q_{max} \times R}{\Delta t \times \text{Area of Tank}} = \left(\frac{4300}{62 \times 8.6}\right) R = 8.05 \; R.$$

A 3-inch layer of urethane insulation will have an "R" value of 7 x 3 = 21. This will insure prevention of excessive pressures even when the tank is relatively empty.

The cost of insulation is obtained from a retail price of $\$1/ft^2$ (1981 dollars). Hence, in terms of 1975 dollars and assuming a difference of 20% between OEM and retail price, the cost of insulation for the ammonia storage tank

$$= \frac{\text{Area} \times \text{Cost}/ft^2 \text{ (retail 1981 dollars)}}{1.7 \times 1.2} = \frac{8.6 \times 1}{1.7 \times 1.2} = \$4.2 \quad .$$

Thus, the total material cost for the ammonia storage tank is:

Cost of steel	= $3.13
Cost of insulation	= 4.20
Total	= $7.33

E.3.1.4.2 Weak-Solution Storage Tank. Required volume (from Appendix C) = 3.21 ft^3.

Assume

- Cylindrical tank length = 3.2 ft
- Sheet thickness = 0.05 inch
- Cost of sheet material (steel) in 1975 dollars = $0.18/lb.

Hence the following can be calculated:

- Diameter = 1.125 ft
- Surface area = 12 ft^2
- Weight of the tank = 24.2 lb
- Cost of steel = $4.36.

E.3.1.4.3 Sum of Components. Table E.2 shows the cost of the components of the 3-ton system of Table E.1 which are assignable to the car system. From Table E.2 the cost of the materials for these car parts for a system of 17,000 Btu/hr is obtained as follows:

$$\text{Cost} = 125 \times \frac{17{,}000}{36{,}000} + 7 = \$66 \quad .$$

Thus, the total cost of materials for the equipment on the car is:

• Ammonia storage tank	=	$ 7.5
• Weak-solution storage tank	=	$ 4.5
• Parts from Table E.1 assignable to car equipment	=	66.0
• Sheet metal and grill*	=	10.0
• Charge of ammonia*	=	10.0
Total material cost of equipment on the car		$98.0

*Estimate by judgement.

TABLE E.2

COST OF THE COMPONENTS OF THE 3-TON SYSTEM OF TABLE E.1 ASSIGNABLE TO CAR EQUIPMENT

Item	Capacity-Dependent Cost	Capacity-Independent Cost
Absorber and Absorber Heat Exchanger	18	
Evaporator	24	
Outdoor Coil[1] (car part only)	42	
Fan, Motor and Mount	24	
Controls and Wiring[2]	12	7
Valves, Service and Relief[3]	3	
Capillaries, Connecting tubing, etc.[3]	2	
Total of Parts on the car (obtained from Table E.1 for a 3-ton system)	125	7

[1] In the 3-ton system of Table E.1, the outdoor coil is assumed to be designed for heat transfer from absorber and condenser. In the split system, half the cost is assigned to the car equipment, as only the absorber is on the car The other half is assigned to the garage equipment.

[2] Total cost of controls and wiring of the 3-ton system of Table E.2 is assumed to be divided into two halves: one half is assigned to equipment on the car, and the other half to the garage equipment. The car portion is further subdivided between capacity dependent and capacity independent. The division here is strictly an engineering judgement.

[3] The total cost of these components on the 3-ton system of Table E.1 is divided for the split system into two parts. One part is assigned to the equipment on the car and the other part is assigned to the equipment in the garage. The division is based on engineering judgement.

The total cost of the car equipment in terms of 1975 dollars is obtained as follows:

Direct material cost	=	\$ 98
Labor with burden	=	81
Total		\$179

Hence, the retail price of the car equipment in terms of 1981 dollars

$= 179 \times 1.7 \times 1.2 = \$365$.

E.3.1.5 Garage Parts

The material cost of the ammonia tank (identical to the one on the car) is \$7.5. The material cost of the weak-solution storage tank (identical to the one on the car) is \$4.5.

Table E.3 shows the cost of the components of the 3-ton system of Table E.1 which are assignable to the garage equipment. From Table E.3, the cost of materials for these parts for a system with a capacity of 700 Btu/hr is worked out as follows.

$$\text{Cost} = 283.5 \times \frac{700}{36000} + 14 = \$19.5.$$

Thus, the total cost of materials for the garage system is worked out as follows:

• Ammonia storage tank	=	\$ 7.5
• Weak-solution storage tank	=	4.5
• Parts from Table E.1 assignable to garage equipment	=	19.5
• Charge of ammonia	=	10.0
• Couplings for connecting to the car equipment	=	5.5
Total material cost		\$47.0

Hence, the cost of garage equipment

= material cost + labor cost with burden

= 47 + 130 = \$177 (1975 dollars).

TABLE E.3

COST OF THE COMPONENTS OF 3-TON SYSTEM OF TABLE E.1 ASSIGNABLE TO THE GARAGE EQUIPMENT

Item	Capacity-Dependent Cost	Capacity-Independent Cost
Generator	25	
Liquid Heat Exchanger	4.5	
Condenser	15	
Precooler	4	
Solution Pump	52	
Water Pumps	17	
Motor, Pump	31	
Outdoor Coil[1]	43	
Burner System	9	
Controls and Wiring[2]	9	10
8-Way Valve and Motor	18	
Valves, Service and Relief[3]	3	
Sheet Metal and Grill	34	
Capillaries, Connecting Tube	4	4
Insulation	9	
Miscellaneous	6	
Total	283.5	14

[1] In the 3-ton system of Table E.1, the outdoor coil is assumed to be designed for heat transfer from absorber and condenser. In the split system, half the cost is assigned to the car equipment, as only the absorber is on the car. The other half is assigned to the garage equipment.

[2] Total cost of controls and wiring of the 3-ton system of Table E.2 is assumed to be divided into two halves: one half is assigned to equipment on the car, and the other half to the garage equipment. The car portion is further subdivided between capacity dependent and capacity independent. The division here is strictly an engineering judgement.

[3] The total cost of these components on the 3-ton system of Table E.1 is divided for the split system into two parts. One part is assigned to the equipment on the car and the other part is assigned to the equipment in the garage. The division is based on engineering judgement.

The retail price of the garage equipment in terms of 1981 dollars

= 177 x 1.7 x 1.2 = $361 .

Hence, the retail price of the entire split heat pump system

= retail price of car system + retail price of the garage system

= 365 + 361 = $726 .

E.3.2 Range Impact

The total vehicle-based weight of the ECS system will vary depending on the weather conditions. A greater quantity of water and ammonia will have to be carried during weather extremes, i.e., high heat and humidity in summer and low temperature in winter. For the purposes of calculating range impact the average value of summer and winter weight is assumed.

From Appendix C:

Weight of fluid for summer = 163 lb

Weight of fluid for winter = 216 lb

Average weight of fluid = 189 lb

Constant weight of the hardware = 100 lb

Hence, the total weight for range impact calculations is 289 lb.

The parastic power requirements to run the air blower and the control system are considered to be identical to those in the gasoline-engine-driven heat pump system.

Hence, parasitic power requirements are 365 watts. Thus, the range impact is:

Weight contribution = 289 x 0.00026 = 0.075

Parasitic power effect = 0.050

Total range reduction = 0.125

Hence, effective range = 87.5% of original.

E.3.3 Energy Efficiency

The split heat pump basically is an aqua-ammonia absorbtion system. Hence, in the cooling mode a COP of 0.584 can be used. (This value is derived by detailed cycle analysis in the ASHRAE Handbook of Fundamentals, 1972, pages 23-24.) In the heating mode, the split heat pump cannot take advantage of waste heat as it is removed during reprocessing of the fluid in the garage equipment. However, some of the heat applied during reprocessing can be used for home heating as it will be at a high enough temperature.

The actual heat to be allocated for ECS operation is worked out as follows:

Let

h_{sw} = enthalpy of water at 120°F (Btu/lb of water)

h_{sa} = enthalpy of liquid ammonia at 120°F (Btu/lb of ammonia)

h_{fs} = enthalpy of solution being returned to the garage equipment for reprocessing (Btu/lb of solution)

m_w = mass of water

m_a = mass of ammonia

Q = quantity of heat supplied by ECS to the passenger compartment

Q_{in} = quantity of input heat for reprocessing to be allocated to the ECS.

(The temperature of 120°F is selected for the starting point since the rest of the heat of ammonia and water will not be useful for home heating.)

Hence,

$$Q_{in} = (m_w h_{sw} + m_a h_{sa}) - (m_w + m_a)\, h_{fs}$$

and

$$COP = \frac{Q}{Q_{in}} .$$

Now, using 32°F as a reference:

$$h_{sw} = 120 - 32 = 88 \text{ Btu/lb of water}$$

$$h_{sa} = 179 - 77.9^{*} = 101.1 \text{ Btu/lb of } NH_3.$$

From Appendix C, page 2-55, we have:

$$h_{fs} = -133 \text{ Btu/lb of ammonia}$$

$$m_w = 151$$

$$m_a = 64.5$$

$$Q = 42{,}500.$$

Hence,

$$Q_{in} = (151 \times 88 + 64.5 \times 101.1) - (151 + 64.5) \times (-133) = 48{,}470 \text{ Btu}.$$

Hence,

$$COP = \frac{42{,}500}{48{,}470} = 0.88 \quad .$$

The average of summer and winter is:

$$COP = \frac{(0.584 + 0.88)}{2} = 0.732 \quad .$$

If the heat is obtained from gas or oil, the energy efficiency number** to be used in the ranking study is 0.732. If, however, heat is obtained from resistance heating the energy efficiency number to be used in the ranking study will be 0.732 x 0.33 = 0.242.

E.3.4 Storage Period

Since no loss of fluids due to evaporation or otherwise occurs, the system has a very large storage period. This period is considered to be indefinite, greater than six months.

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*See footnote on page 2-55 of Appendix C.

**Refer to Section E.1.3 for definition of energy efficiency number.

E.3.5 Maintenance Cost

Industry experience reveals aqua-ammonia absorption systems to be relatively free of maintenance problems. Quality control in manufacturing and attention to details in design are major factors in alleviating maintenance costs. A maintenance cost of $15.0 per year is assumed, considering the split nature of the system.

E.3.6 Performance Impact

Performance impact is computed by the formula given for the gasoline-engine-driven heat pump system. For this split system, the weight of the ECS is considered as the average of summer and winter as explained in the subsection on range impact (E.3.7). Hence, the performance impact number is computed to be

$$= \frac{3320}{(3320 + 289)} = 0.92.$$

E.3.7 Consumer-Perceived Risk

Ammonia is considered to be a dangerous material and the quantity of ammonia that can be carried in homes is limited by law. However, on the vehicle, the aqua-ammonia system is far safer than gasoline for the following reasons:

- Ammonia is less flammable.
- The pungent order associated with ammonia gives early notice to the presence of a leak.

Nevertheless, consumer perception of such a system in the initial introduction period is bound to be that the system is more risky than the gasoline-engine-driven heat pump system for two reasons:

- The system is unproven.
- Ammonia is considered a dangerous material.

Hence, a score of 4 on a scale of 10 was assigned to this system.

E.3.8 Noise

Very little or no noise is associated with the system. The only noise expected is that due to the evaporator blower fan. Hence, a rating of 2 on a scale of 10 was assigned for this system.

E.3.9 Environmental Impact

The system is much cleaner; no hazardous fumes are emitted during operation. Therefore, a score of 1 on a scale of 10 is assigned to this system.

E.3.10 Packaging and Volume

The volume of system is higher than for the gasoline-engine system. However, the temperature of the system is not very high, and no provision needs to be made for combustion-air intake and high-temperature exhaust. Thus, a score of 6 on a scale of 10 was given to this system as compared to the score of 5 for the gasoline-engine-driven heat pump system.

E.3.11 Development Status

The basic system is not new: only the splitting of the components in two parts is new. Thus, the system can be built with well proven components. A score of 5 was given to this system in comparison with 2.5 for the gasoline-engine-driven heat pump system, indicating a greater development period involved.

E.4 WATER THERMAL STORAGE FOR BOTH HEATING AND COOLING

System requirements were based upon both heating and cooling requirements for the system. Heading mode considerations dictated that the design approach for a high-pressure system would result in the most favorable configuration. An operating pressure of 30 psi and a storage temperature of 250°F was selected as the design point for the heating mode storage.

E.4.1 System Cost

On the car system, costs totalling $300 were established from the following component costs. A Young brand hot water radiator for 17,000 Btu/hr was quoted at about $150. The four valves involved in the system are estimated at $40, the fluid pump at $25, the storage tank (consisting of a 4-ft long by 1-ft diameter tank) was estimated to cost about $50. System antifreeze was estimated to about $6, and the cold-water storage elastomer balls were estimated to be about $30.

In any of the thermal storage concepts, stationary support equipment would be required to provide the refrigeration or the heating effects at the garage site. The cost of these elements was estimated to be $200 for a small refrigerator. This figure was estimated by comparing prices of small, home bar refrigerator units, and was averaged to be about $200 for the size range of interest. An immersion heating element was estimated to be about $50. The requirement for stationary equipment for the storage systems implies that an installation cost over and above that required for vehicle installation would be involved. Therefore, an installation cost estimate of $100 was included in this system cost. The total system cost was, therefore, $650.

E.4.2 System Life

A water storage system is expected to have a life equivalent to, or even better than, any of the heat-activated heat pumps. A value of 1 was designated for this system.

E.4.3 Range Impact

The impact due to the fluid weight of 283 lb and a hardware weight of 50 lb was calculated to be 8.7%. A parasitic power loss was estimated to be accountable for an additional 5%, totalling 13.7% range impact. The effective range is thus 86% with this system as compared to the range without the system.

E.4.4 Energy Efficiency

In the summer, the water has to be cooled to below 32°F. In this aspect, the garage unit operates like an icemaker. A typical COP of heat pumped to wall-plug electricity can be assumed to be between 1.5 to 2.0, depending on the design and cost of the equipment. The thermal input at the power plant has a summer COP of 0.5 to 0.67.

In the winter, the COP will be 0.85 or 0.33, depending on whether heating is accomplished using natural gas, oil, or wall-plug electricity.

If the use of cheaper first cost equipment is assumed, the summer COP is 0.5 and the winter COP is 0.33. Hence, the energy efficiency number for this scheme is

$$(0.5 + 0.33) \div 2 = 0.415 \quad .$$

E.4.5 Storage Period

During the winter the water in the storage tank will be at 250°F, while the ambient design temperature is -10°F. Insulation will have to be provided so that over a period of 10 hours, only 5% of stored heat will be lost. The insulation thickness is determined as follows.

The area of the 4-foot-long, 1-foot-diameter tank is:

$$\text{area} = 4\,\pi + 1.57 = 14.2\ \text{ft}^2$$

$$\Delta\theta = \text{temperature difference} = 260°\text{F}.$$

Hence, the required "R" value of the insulation is:

$$R = \frac{260 \times 10 \times 14.2}{42{,}500 \times 0.05} = 17.4 \quad .$$

Urethane insulation 2 1/2 inches thick will provide an "R" value of 20. This thickness provides a reasonable level of insulation both in cost and space. Thus, a storage period of 10 hours is assigned to this system.

E.4.6 Maintenance Costs

This system has no regular maintenance schedule other than replacement and checking of antifreeze, which would be considered minimal. A maintenance value of $5 per year is estimated for this system.

E.4.7 Performance Impact

The weight of this system is 333 lb. The reduction in acceleration is computed following the same procedure as for the gasoline-engine-driven heat pump system.

$$\text{Acceleration reduction} = 3320/(3320 + 333) = 0.91 \quad .$$

E.4.8 Consumer Risk

For this system, the risk would be lowest on a technical basis, and in fact, the perceived risk by consumers would probably be lower than for any of the other systems. Therefore, a very low value of 2 was assigned in comparison to a value of 2 for the gasoline-engine-driven heat pump system.

E.4.9 Noise

This system would involve little, if any, noise except for the fan operation. The fan noise would be equivalent to that in heating systems used in conventional automobile heating equipment. A value of 2 was given.

E.4.10 Environmental Impact

This system would be one of the cleanest, both in terms of real pollution and accidental pollution due to breakage of the storage container, etc. Therefore, a value of 1 out of 10 was given.

E.4.11 Packaging and Volume

The volume and weight of the system are higher for this system than for the reference system (gasoline-engine-driven heat pump system). However, neither exhaust ducting nor very high temperatures are involved. Furthermore, relatively large freedom exists to distribute and shape the system components as compared to the reference system. A score of 6 is given to this system in comparison to 5 for the reference system. (The higher number indicates a greater degree of difficulty).

E.4.12 Development Status

The design of the components in this system is not new. The only novelty is the use of neoprene balls to allow for expansion during freezing. Alternative techniques are easily available for ice making. In fact, ice making is a very well-known and old art - with the first applications of man-made refrigeration being applied to ice making. Thus, the development status of this system is comparable to that of the reference system. A score of 3 is assigned to this system as compared to 2.5 for the reference system.

E.5 THERMAL STORAGE WITH $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$

The following properties of this salt have been obtained from Appendix C.

- Melting temperature - 1310°F
- Heat of fusion - 70 Btu/lb
- Specific heat - 1 Btu/lb-°F
- Thermal conductivity - 0.37 Btu/hr-°F-ft.

From page 3-21 of Appendix C, the following information is available about the system for storing 42,500 Btu.

- Maximum temperature - 1310°F
- Minimum temperature - 200°F
- Volume of salt required - 0.22 ft^3
- Cost of salt required - $3.6.

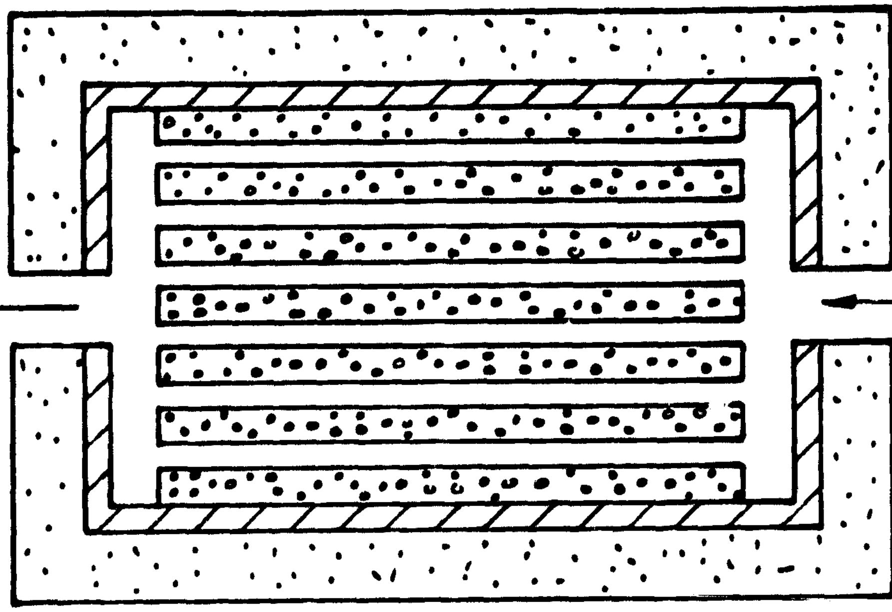
Consider a storage system consisting of a cylinder made up of stainless steel with high-temperature insulation outside, and an air-to-salt heat exchanger consisting of stainless steel tubes inside, as shown in Figure E.1. The salt is filled in the remaining space inside the cylinder. An electric heater in the form of a coil of nichrome wire is also installed inside the cylinder to facilitate heating of the salt overnight from wall-plug electricity.

The salt is heated to 1310°F at night. During the day when heat is needed in the passenger compartment, air is blown through the stainless heat exchanger and the outcoming air is diluted by adding low-temperature air before passing it into the passenger compartment. The amount of low-temperature air added varies depending on the temperature of the air coming out of the heat exchanger. As more and more heat is extracted, the temperature of the storage falls. Useful heat is assumed to be practically extractable as long as the temperature of the storage is above 200°F.

E.5.1 Design Calculations for Heat Exchanger

Assumptions:

- Salt temperature · 200°F
- Air temperature in · 60°F
- Air temperature out · 100°F.



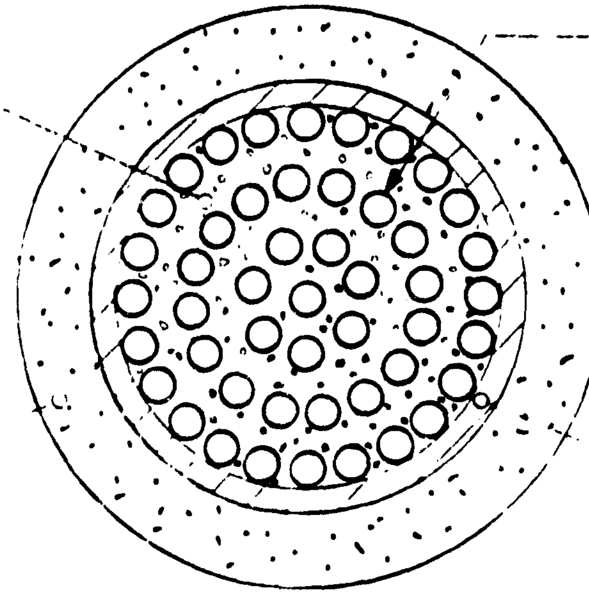


Fig. E. 1 Schematic of Thermal Storage Device for Use With High-Temperature Solids

Hence,

$$\Delta t_1 = 200 - 60 = 140$$

$$\Delta t_2 = 200 - 100 = 100$$

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \frac{\Delta t_1}{\Delta t_2}} = \frac{40}{\log_e 1.4} = \frac{40}{.346} = 116°F \ .$$

Desired heat transfer rate = 17,000 Btu/hr.

Hence, $17{,}000 = \frac{116 \times \text{Area}}{R}$.

Assume R = 0.1 for heat transfer at inside of tubes with air flowing at high velocity. Hence, the required tube area is given by:

$$\text{Area} = \frac{17{,}000 \times 0.1}{116} = 14.65 \text{ ft}^2 = 14.65 \times 144 \text{ in.}^2 = 2.11 \times 10^3 \text{ in.}^2.$$

Assume pipes with 1-in. diameter and 1-foot long. Hence, the number of pipes required

$$= \frac{2.11 \times 10^3}{12 \times \pi} = 56$$

Cross sectional area of 56 pipes = $56 \times \frac{\pi}{4} = 44$ in.^{2}.

The volume required for salt storage = 0.25 ft^3 (allowing for 10% heat loss during stored mode).

The total cross sectional area of the stainless steel tank

$$= (44 \times 1.44 + \frac{0.25}{1} \times 144) \times 1.1 = 110 \text{ in.}^2.$$

Where the stainless steel tubes are assumed to be 0.1 inch thick. The diameter of the stainless steel tank

$$= \left(\frac{110 \times 4}{\pi}\right)^{1/2} = 12 \text{ in.}$$

The insulation thickness required is determined from the condition that within 10 hours of storage, only 10% of stored heat will be lost. Hence,

$$Q = 42{,}500 \times 0.1 = 4250 \text{ Btu.}$$

Surface area of the tank

$$= (2 \times 110) + (12 \times 12 \times \pi) = 673 \text{ in.}^2 = 4.7 \text{ ft}^2.$$

$\Delta\theta$ = temperature difference between storage and ambient

$$= 1310 - (-10) = 1320°\text{F}.$$

Hence,

$$R = \frac{1320 \times 4.7 \times 10}{4250} = 14.6.$$

High temperature insulation like Min-K 2000 (made by the Johns-Manville Corporation, Xen-Caryl Ranch, Denver, Colorado 80217) can be used. The "R" value of this insulation material is 3 per inch of thickness. Hence, a thickness of 5 inches will be adequate. The retail price of $\$167/\text{ft}^2$ is quoted by Johns-Manville for this material in a 5-inch thickness. For mass production, the OEM price is estimated to be one-third of the retail price. Hence, a price of $\$56/\text{ft}^2$ will be assumed for this study.

The weight of stainless steel in the tank is worked out as follows. Assume:

Wall thickness = 0.125 inch

$$\text{Volume} = 673 \times 0.125 = 84 \text{ in.}^3$$

$$\text{Weight} = 84 \times 0.28 = 23.6 \text{ lb.}$$

The weight of the heat exchanger tubes is calculated as follows. Assume:

Wall thickness of 0.1 inch

Surface area of 12 inches long, 1-inch inside diameter coil

$$= \pi \times 12 = 37.7 \text{ in.}^2$$

Volume of stainless steel in one pipe

$$= 37.7 \times 0.1 = 3.77 \text{ in.}^3$$

Volume of stainless steel in 56 pipes

$= 56 \times 3.77 = 211 \text{ in.}^3$

Weight of heat exchanger tubes

$= 211 \times 0.28 = 59$ lb.

E.5.2 First Cost

The total cost of the system is worked out as follows.

Data:

- Cost of stainless steel - \$1.80/lbm
- Cost of insulation (R value 15) - \$56/ft^2
- Weight of stainless steel - 59 + 24 = 83 lb
- Insulation area - 5 ft^2
- Cost of salt (from Appendix C) - \$3.6.

Calculations:

• Cost of stainless steel	-	\$150.0
• Cost of insulation	-	280.0
• Cost of salt	-	3.6
• Cost of miscellaneous material	-	19.4
Total Material Cost	=	\$453.0.

Labor cost is estimated to be \$1.50 per lb of hardware. This is about the rate of labor cost for products with equivalent technological levels.

The total weight of the system consists of the following:

• Weight of stainless steel	-	83
• Weight of insulation (density 20 lb/ft^3)	-	42
• Weight of salt (from Appendix C)	-	33
• Miscellaneous	-	32

The total system weight is 190 lb. However, the weight of the salt will not enter into the labor cost calculations.

Hence,

Labor cost = 157 x 1.5 = $236.

Thus,

Total cost = 236 + 453 = $689.

Retail price with 20% profit = $827.

E.5.3 System Life

This system has no moving parts. However, the corrosive effects of the salt at high temperatures are not known. It will always be possible to obtain a container material where corrosion problems are eliminated. The cost of such a system cannot be guessed at present. In view of this uncertainty, system life is considered to be half that of the reference system. Thus, the estimated life for the purpose of this study will be assumed to be 1500 hours.

E.5.4 Range Impact

The technique for calculating range impact is the same as for the reference system (gasoline-engine-driven heat pump system).

Data:

- Total system weight - 190 lb
- Parasitic power - 365 watts.

Hence, the range reduction due to weight

$= 0.0026 \times 190 = 0.0495 \approx 5\%$.

The range reduction due to parasitic power = 5%.

Hence, total effective range = 0.9.

E.5.5 Energy Efficiency

The system is provided with electric heating for recharging the thermal energy storage. Hence, the energy efficiency number for this system is 0.33.

E.5.6 Storage Period

By design, the storage period is 10 hours.

E.5.7 Maintenance Cost

As this system has no moving parts, no regular maintenance is expected. However, the high temperature of the system may result in some maintenance requirement. Hence, a maintenance cost of 20% of the reference system is assumed for this system. Thus, the maintenance cost for this system is $7/year.

E.5.8 Performance Impact

The performance impact is calculated as for the reference system.

Data:

- System weight - 190 lb.

Hence, the performance impact number $= \frac{3320}{3320 + 190} = 0.95$.

E.5.9 Consumer-Perceived Risk

As the temperature of the stored material is high, it will pose some danger in the event of an accident. This system is therefore considered to be slightly more risky than the reference system. Hence, a score of 3 is given to this system in comparison with 2 for the reference system.

E.5.10 Noise

A score of 2 is assigned to this system. For the rationale used in assigning this score, see Section E.4.9.

E.5.11 Environmental Impact

A score of 1 is assigned to this system. For the rationale used in assigning this score, see Section E.4.10.

E.5.12 Packaging and Volume

The total volume is 5.3 ft^3. Furthermore, high-temperature requirements, which dictate system shape, leave relatively few options for configuring this system to fit into odd-shaped places. A rating of 8 is given to this system in comparison to 5 for the reference system.

E.5.13 Development Status

The properties of the salt are not fully known, the temperatures are high. However, basic heat exchanger design is well known. Hence, a rating of 5 is given to this system in comparison to 2.5 for the reference system.

E.6 THERMAL STORAGE WITH LiOH

E.6.1 Design Calculations

The following information is available from Appendix C, page 3-2, and Table 3-3 on page 3-9 of Appendix C.

- Maximum Temperature - 860°F
- Specific heat - 0.59 Btu/lb-°F
- Density - 91.1 lb/ft^3
- Cost - $1.97/lb.

For storing 42,500 Btu, based on a 200°F minimum temperature for practical heat extraction, the following is obtained:

Heat storage density = (860 - 200) x 0.59 = 390 Btu/lb.

Allowing a 10% loss of stored heat over a period of 10 hours, initially the quantity of heat to be stored

= 42,500 x 1.1 = 46,750 Btu

where 42,500 Btu is the amount of useful energy required to be delivered to the passenger compartment.

To store 46,750 Btu:

$$\text{Weight of salt required} = \frac{46,750}{390} = 120 \text{ lb}$$

$$\text{Volume of salt required} = \frac{120}{91.1} = 1.32 \text{ ft}^3.$$

The structure of this system is similar to the one used for the $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$ system described in Section E.5. Hence, 56 pipes with the following specifications will be used for making the heat exchanger:

- Material - stainless steel
- Length - 12 inches
- Inside diameter - 1 inch
- Wall thickness - 0.1 inch
- Outside diameter - 1.2 inches.

Hence, the area occupied by the pipes

$$= 56 \times \frac{\pi}{4} \times (1.2)^2 = 63.4 \text{ in.}^2.$$

The area occupied by salt

$$= \frac{\text{Volume Required}}{\text{Length of the Container}} = \frac{1.32}{1} = 1.32 \text{ ft}^3 = 190 \text{ in.}^2.$$

Hence, the total area of the container

$$= 63.4 + 190 = 253.4 \text{ in.}^2.$$

Hence, the inside diameter of the container

$$= \left(\frac{253.4 \times 4}{\pi}\right)^{1/2} = 18 \text{ inches.}$$

The cylindrical surface area of the container

$$= \left(\frac{18}{12}\right) \times \pi \times 1 = 4.7 \text{ ft}^2.$$

The area of the circular face

$$= \frac{253.4}{144} = 1.76 \text{ ft}^2.$$

The total surface area of the container, allowing 1/2 in. for header space on each side.

$$= \left(4.7 \times \frac{13}{12}\right) + (2 \times 1.76) = 5.1 + 3.52 = 8.62 \text{ ft}^2.$$

To calculate the weight of the stainless steel in the container, assume a wall thickness of 0.125 inch.

$$\text{Volume} = 8.62 \times 144 \times 0.125 = 155 \text{ in.}^3$$

$$\text{Weight} = 155 \times 0.28 = 43.4 \text{ lb.}$$

Insulation requirements are computed to meet the following requirements:

- Period - 10 hours
- Heat lost - 4250 Btu
- Area of the container - 8.62 ft^2
- Temperature difference = 860 - (-10) = 870°F.

Hence,

$$R = \frac{870 \times 8.62 \times 10}{4250} = 17.7.$$

Suitable insulation will be Min-K 2000 made by the Johns-Manville Corporation, Denver, Colorado. This insulation has an R value of 3 per inch at 860°F. Hence, a 6-inch thickness will meet the storage requirement.

Weight of the insulation

$$= (8.62 \times \frac{6}{12}) \times 20 = 86.2 \text{ lb.}$$

Total weight of the system is worked out as follows:

Weight of stainless steel heat exchanger tubes	= 59
Weight of stainless steel container	= 43
Weight of the salt	= 120
Weight of the insulation	= 86
Miscellaneous	= 22
Total	= 330 lb.

E.6.2 System Cost

The cost of the system is work out as follows.

Assumptions:

- Cost of stainless steel - $1.80/lb
- Cost of insulation (R value of 18) - $68/ft^2
- Cost of Salt - $1.97/lb
- Cost of Labor - $1.50/lb of hardware
- Weight of stainless steel - 59 + 43 = 102 lb
- Area of insulation - 8.6 ft^2
- Weight of salt - 120 lb
- Weight of hardware - 210 lb

Hence,

Cost of stainless steel	=	**$ 184**
Cost of insulation	=	**585**
Cost of salt	=	**236**
Miscellaneous	=	**50**
Total material cost	=	**$1055**
Cost of labor with burden	=	315
Total cost of manufacture	=	$1370

Adding a profit of 20%, the **retail selling price is $1645.**

E.6.3 Range Impact

Weight contribution = 0.00026 x 330 = 0.086

Parasitic power (same as for $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$ system) = 0.05

Total effective range = 0.86.

E.6.4 Performance Impact

Performance impact = 3320/(3320 + 330) = 0.91.

This system has identical scores with those for the $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$ system for the following criteria because of similarities between the two systems:

- System Life - 1500 hours
- Energy Efficiency - 0.33
- Maintenance Cost - $7/year
- Consumer-Perceived Risk - 3
- Noise - 2
- Environmental Impact - 1
- Development Status - 5

E.6.5 Packaging and Volume

The volume is higher than the volume of the $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$ system. Other features regarding this item are identical. Hence, a score of 10 is assigned to this system.

E.7 LiBr-WATER SPLIT HEAT PUMP SYSTEM

This system is available for cooling only. The system requires practically no garage equipment. For recharging overnight, the solution of LiBr water is heated until most of the water is driven off as water vapor and the water tank is filled with water. For heating purposes, the absorber tank is provided with electric heaters.

In Appendix C, the following information is computed to provided cooling capacity.

- Weight of water required - 42.5 lb
- Weight of LiBr required - 72.5 lb

Hence, the volume of the water storage tank

$$= \frac{42.5}{64} = 0.665 \text{ ft}^3.$$

The volume of the solution storage tank

$$= \frac{42.5 + 72.5}{64} = 1.8 \text{ ft}^3.$$

The water storage tank is considered to be a 1-foot long cylinder. Hence, the area of the circular face = 0.665 ft^2.

$$\text{The diameter} = \left(\frac{0.665 \times 4}{\pi}\right)^{1/2} \text{ ft} \approx 12 \text{ inches.}$$

Hence, the surface area of the tank

$$= (\pi \times 12 \times 12) + 2 \times \left(\frac{\pi \times 12 \times 12}{4}\right) \text{ in.}^2 = 4.7 \text{ ft}^2.$$

For a 0.030-inch-thick container, the weight will be 5.7 lb.

A solution storage tank with following specifications will satisfy the requirements.

- Diameter - 12 inches
- Length - 33 inches

- **Wall thickness - 0.03 inch**
- **Surface area - 10 ft^2**
- **Weight - 12.2 lb.**

Thus, the total weight of the system is calculated as follows:

Weight of water storage tank	=	5.7
Weight of solution storage tank	=	12.2
Weight of evaporator, absorber, valves, etc.	=	42.1
Weight of water	=	42.5
Weight of LiBr	=	72.5
Total		175.0 lb.

E.7.1 System Cost

The cost of this equipment can be assessed by assuming \$3/lb of hardware. Hence, the cost of the system

= 3 x 60 = \$180.

E.7.2 System Life

As there are no moving parts, the system life can be assumed to be 3000 hours.

E.7.3 Range Impact

The weight of the system with fluids is 175.0 lb. Parasitic power is the same as for the gasoline-engine-driven heat pumps. Hence, the effective range is 0.92.

E.7.4 Energy Efficiency

A summer COP of 0.78 is assumed for a **LiBr-water absorption system.** The heat is provided electrically to reprocess overnight. Hence, the energy efficiency number for this system is 0.78 x 0.33 = 0.26

E.7.5 Storage Period

The absorber and the solution tank have to be maintained at a high vacuum level. Hence, a storage period of 10 days is assumed.

E.7.6 Maintenance Cost

The expansion valve will have to maintain a high vacuum. Hence, a maintenance cost of $20 per year is assumed.

E.7.7 Performance Impact

The weight of the system is the same as that for the gasoline-engine-driven heat pump. Hence, a performance impact number of 0.95 is calculated for this system.

E.7.8 Consumer-Perceived Risk

LiBr is a toxic substance. Thus, in case of an accident it will pose some problem. Hence, a score of 3 is assigned to this system.

E.7.9 Noise

Very little or no noise is associated with this system. Hence, a score of 2 is assigned, which is the same as for the aqua-ammonia system.

E.7.10 Environmental Impact

During recharging, the solution is heated until the water from the solution is driven off. Ideally, no LiBr should escape to ambient. However, with the cheapest equipment, there may be a danger of minute quantities of LiBr escaping along with the water vapor. This problem is an unknown at this time. Hence, a high score of 20 is assigned, to indicate a potential for problems in this area.

E.7.11 Packaging and Volume

Packaging and volume are similar to that of the gasoline-engine-driven heat pumps since some means will have to be provided to vent water vapor during reprocessing in the garage. A score of 5 is assigned to this system.

E.7.12 Development Status

The development status is the same as for the aqua-ammonia system. Hence, a score of 5 is assigned to this system.

E.8 THERMAL STORAGE WITH ORGANIC OILS

The following information is available from Appendix C, pages 3-28 through 3-31. For Therminol-66:

- Maximum useful temperature - 650°F
- Specific heat - 0.655 Btu/lb-°F
- Density - 46.8 lb/ft^3.

Allowing a 10% heat loss during the 10-hour storage period, 46,750 Btu must be stored to obtain a useful heat quantity of 42,500 Btu. As the temperature is high, the storage structure will be considered to be similar to the one for thermal storage with $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$. The weight of the fluid required is calculated as follows. The minimum storage temperature is 200°F. Hence,

$$\Delta T = 650 - 200 = 450$$

$$\text{Heat storage density} = 450 \times 0.655 = 295 \text{ Btu/lb}$$

$$\text{Required fluid quantity} = \frac{46,750}{295} = 158 \text{ lb}$$

$$\text{Fluid volume} = \frac{158}{46.8} = 3.38 \text{ ft}^3.$$

The heat exchanger tubes are assumed to be of the following specifications:

Outside diameter	= 1.2 inches
Inside diameter	= 1.0 inch
Wall thickness	= 0.1 inch
Length	= 24 inches
Number of tubes	= 28
Area occupied by pipes	= 32 in.^{2}
Area occupied by fluid	$= \frac{3.38 \times 1728}{2 \times 12} = 243$ in.^{2}
Total area of container's circular face	= 32 + 243 = 275 in.^{2}
Container diameter	= 18.8 inches
Cylindrical area of container	$= 24 \times \pi \times 18.8 = 1420$ in.^{2}
	= 9.85 ft^2

Total container wall area $= 9.85 \times \frac{25}{24} + 2 \times \frac{275}{144} = 14.1\ ft^2$.

The wall thickness for the container is assumed to be 0.125 inch. Hence,

Container weight = 14.1 x 144 x 0.125 x 0.28 = 70 lb.

The total steel weight equals the weight of container plus the weight of the heat exchanger tubes:

Steel weight = 70 + 59 = 129 lb.

Insulation requirement:

$\Delta\theta = 650 - (-10) = 660°F$

$\Delta Q = 4250$ Btu

Hours = 10

Area = $14.1\ ft^2$

$$R = \frac{660 \times 10 \times 14.1}{4250} = 21.9 = 22$$

A sandwich insulation with Mink-2000 will be used near the walls of the container, along with an outer layer of urethane foam with an R value of 7 per inch. As the maximum temperature the urethane can withstand is 200°F, the thickness of the Mink-2000 insulation is determined as follows.

Let R_1 be the thermal resistance of Mink-2000 insulation and R_2 be the thermal resistance of urethane. Hence,

$$\frac{R_2}{R_1 + R_2} = \frac{210}{660}$$

Hence

$$R_2 = \left(\frac{210}{660}\right) \times 22 = 7$$

and

$$R_1 = 15.$$

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Mink-2000 has an average R value of $\frac{1}{0.22}$ between 600°F and 200°F. Hence,

Mink-2000 thickness = 15 x 0.22 = 3.3 inches.

Thus, the insulation sandwich is made up of 14.1 ft^2 of 3.3-inch thick Mink-2000 insulation and 1.0-inch thick urethane.

Cost of Mink-2000 insulation (3.3-inch thick)	= $37/$ft^2$
Cost of urethane (1-inch thick)	= $0.31/$ft^2$
Cost of insulation sandwich	= $37.31/$ft^2$
Total cost of insulation	= 37.31 x 14.1 = $526
Weight of Mink-2000	$= \frac{14.1 \times 3.3 \times 20}{12} = 77.5$ lb
Weight of urethane	$= \frac{14.1 \times 1}{12} \times 1.9 = 2.23$ lb
Total insulation weight	= 77.5 + 2.23 = 80 lb.

Hence, the total weight of the system is:

Weight of the steel	= 129 lb
Weight of the insulation	= 80 lb
Weight of the fluid	= 158 lb
Miscellaneous	= 33 lb
Total	= 400 lb

Overall system dimensions are:

Length = 36 inches (3 ft)

Diameter = 28 inches (2 1/3 ft)

The total cost is:

Cost of steel at $0.30/lb	= $ 38.7
Cost of insulation	= 526.0
Cost of fluid at $13/gallon	= 336.0
Miscellaneous	= 29.3
Total Material Cost	= $930.0

Cost of labor with burden at $1.50/lb of hardware	= $\underline{\$ 360}$
Grand total cost	= $1290

Allowing a profit of 20%, the retail price = 1.2 x 1290 = $1550.

E.8.1 Range Impact

Weight contribution = 0.00026 x 400 = 0.104

Parasitic power = 365 watts

Hence, effective range = 0.85.

E.8.2 Performance Impact

$$\text{Performance Impact} = \frac{3320}{3320 + 400} = 0.894.$$

Scores for the other criteria are given in Table E.4 of Section E.10.

E.9 WATER THERMAL STORAGE FOR HEATING ONLY

If water is to be used only for heating, a heater will be installed in the wate- storage tank on the car and no garage equipment will be required. The heater . rned on at night by using wall-plug electricity in the garage. The system p...ameters are easily derived from Section E.4, Water Thermal Storage for Both Heating and Cooling. The results are:

- First Cost - $300
- Weight of the system on car - 333 lb
- Range impact - 0.86
- Energy efficiency - 0.33
- Storage period - 10 hours
- Performance impact - 0.91.

The values of the other parameters are given in Table E.4 of Sectic E.10.

E.10 SUMMARY OF RESULTS OF RANKING CALCULATIONS

The outcome of the calculations given in this appendix is presented in Table E.4, Marks Assigned to Different Systems for the Various Criteria. Table E.5 presents the same data normalized with respect to the reference system, the gasoline-engine-driven heat pump. Table E.6 presents the data modified by multiplying the marks with weighting factors for the different criteria. These weighting factors are discussed in Appendix D, Table 1. Table E.7 shows the total score of the systems.

TABLE E.4

MARKS ASSIGNED TO DIFFERENT SYSTEMS FOR VARIOUS CRITERIA

Criteria	Gasoline-Engine-Driven Heat Pump	MTI Heat-Activated Heat Pump	Aqua-Ammonia Split Heat Pump	Thermal Storage with Water	Thermal Storage with K_2CO_3-Na_2CO_3-Li_2CO_3	Thermal Storage with Li_2CO_3	Thermal Storage with LiOH	Thermal Storage with Water for Heating Only	Thermal Storage with Oils	LiBr-Water Split Heat Pump
Described on Appendix C, Page:	2-21	2-23	2-50	3-1	3-20	3-20	3-7	3-1	3-28	2-43
First Cost (Dollars)	750	820	726	650	827	827	1645	306	1550	180
System Life (Hours)	>3000	>3000	>3000	>3000	1500	1500	1500	>3000	3000	3000
Range Impact (Effective Range)	0.92	0.89	0.88	0.86	0.90	0.90	0.86	0.86	0.85	0.92
Energy Efficiency	0.5	1.0	0.73	0.42	0.33	0.33	0.33	0.33	0.33	0.26
Storage Period (Days)	60	60	60	1	1	1	1	1	1	10
Maintenance Cost ($/Year)	35	10	15	5	7	7	7	2	7	20
Performance Impact (Per Unit Acceleration)	0.95	0.95	0.92	0.91	0.95	0.95	0.91	0.91	0.89	0.95
Consumer-Perceived Risk	2	4	4	2	3	3	3	2	3	3
Noise	10	5	2	2	2	2	2	2	2	2
Environmental Impact	10	5	1	1	1	1	1	1	1	20
Packaging and Volume	[illegible]	6	6	6	8	8	10	6	8	5
Development S[illegible]	2.5	6.0	5.0	3.0	5.0	5.0	5	1	5	5

TABLE E.5

NORMALIZED MARKS FOR VARIOUS CRITERIA FOR DIFFERENT SYSTEMS

Criteria	Gasoline-Engine-Driven Heat Pump	MTI Heat-Activated Heat Pump	Aqua-Ammonia Split Heat Pump	Thermal Storage with Water	Thermal Storage with $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$	Thermal Storage with Li_2CO_3	Thermal Storage with LiOH	Thermal Storage with Water for Heating Only	Thermal Storage with Organic	LiBr-Water Split Heat Pump
Described on Appendix C, Page:	2-21	2-23	2-50	3-1	3-20	3-7	3-7	3-1	3-28	2-43
First Cost	1	0.92	1.03	1.15	0.92	0.92	0.46	2.5	0.48	4.16
System Life	1	1	1	1.0	0.50	0.50	0.50	1.0	1.00	1.00
Range Impact	1	0.97	0.96	0.94	0.98	0.98	0.93	0.94	0.93	1.00
Energy Efficiency	1	2.00	1.46	0.84	0.66	0.66	0.66	0.66	0.66	0.52
Storage Period	1	1.00	1.00	0.02	0.02	0.02	0.02	0.02	0.02	0.17
Maintenance Cost	1	3.50	2.34	7.00	5.00	5.00	5.00	17.5	5.00	1.75
Performance Impact	1	1.00	0.97	0.96	1.0	1.00	0.96	0.96	0.89	1.00
Consumer-Perceived Risk	1	0.50	0.50	1.00	0.67	0.67	0.67	1.0	0.67	0.67
Noise	1	2.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00
Environmental Impact	1	2.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	0.50
Packing and Volume	1	0.83	0.83	0.83	0.62	0.62	0.50	0.83	0.62	1.0
Development Status	1	0.42	0.50	0.84	0.50	0.50	0.50	0.50	0.50	0.50

TABLE E.6

WEIGHTED MARKS FOR VARIOUS CRITERIA FOR DIFFERENT SYSTEMS

Criteria	Gasoline-Engine-Driven Heat Pump	MTI Heat-Activated Heat Pump	Aqua-Ammonia Split Heat Pump	Thermal Storage with Water	Thermal Storage with $K_2CO_3 \cdot Na_2CO_3 \cdot Li_2CO_3$	Thermal Storage with Li_2CO_3	Thermal Storage with LiOH	Thermal Storage with Water for Heating Only	Thermal Storage with Organix Oil	LiBr-Water Split Heat Pump
Described on Appendix C, Page:	2-21	2-23	2-50	3-1	3-20	3-7	3-7	3-17	3-28	2-43
First Cost (Dollars)	25	23	26	28.8	23	23	11.5	62.5	12	104
System Life (Hours)	5	5	5	5	2.5	2.5	2.5	5.0	5	5
Range Impact (Effective Range)	10	9.7	9.6	9.4	9.8	9.8	9.3	9.4	9.3	10
Energy Efficiency	5	10	7.3	4.2	3.3	3.3	3.3	3.3	3.3	2.6
Storage Period (Days)	5	5	5	0.1	0.1	0.1	0.1	0.1	0.1	0.9
Maintenance Cost ($/Year)	10	35	23.4	70	50	50.0	50.0	175.0	50.0	17.5
Performance Impact (Per Unit Acceleration)	10	10	9.7	9.6	10.0	10.0	9.6	9.6	8.9	10.0
Consumer-Perceived Risk	5	2.5	2.5	5.0	3.3	3.3	3.3	5.0	3.3	3.3
Noise	10	20	50	50	50	50	50	50	50	50
Environmental Impact	5	10	50	50	50	50	50	50	50	2.5
Packaging and Volume	5	4.1	4.1	4.1	3.1	3.1	2.5	4.1	3.1	5.0
Development Status	5	2.1	2.5	4.2	2.5	2.5	2.5	2.5	2.5	2.5
TOTAL	100	136.4	195.1	240.4	207.6	207.6	194.6	376.5	197.5	213.3

TABLE E.7

RESULTS OF RANKING SYSTEM

System Type	System	Score
Both Heating and Cooling	Thermal Storage with Water	240
	Aqua-Ammonia Split Heat Pump System	195
	MTI Heat-Activated Heat Pump	136
	Gasoline-Engine-Driven Heat Pump	100
Heating Only	Thermal Storage with Water	377
	Thermal Storage with K_2CO_3-Na_2CO_3-Li_2CO_3	208
	Thermal Storage with Li_2CO_3	208
	Thermal Storage with Organic Oil	198
	Thermal Storage with LiOH	195
Cooling Only	LiBr-Water Split Heat Pump	213

I. Contract Requirement and Execution

(A) **Since excessive electrolyte temperature may reduce cycle life of the battery pack, estimate the relative vehicle driving range penalty ($\Delta R/Ro$) if the ECS were required to provide a cooling capability to prevent battery electrolyte temperature from exceeding 100°F when the ambient temperature is a constant 120°F.**

(B) Describe the required ECS modifications and/or additions needed to implement the cooling capability alluded to in paragraph (a)(8)(A) above.

In the execution of this task it was deemed desirable to study battery heating together with battery cooling. The study encompassed meeting the battery temperature control algorithm:

a) At the end of a charging cycle, the battery pack is 120°F.

b) The battery temperature controller maintains the battery pack at 120°F ± 10°F except during periods of non-use exceeding 3 days.

The requirement for controlling battery electrolyte temperature to 100°F and less with a constant 120°F ambient temperature was given less attention. Perusal of a climatic atlas of the U.S. indicates that a 120°F ambient is rare indeed and that constant (continous) 120°F ambient is non-existent. It is recognized that precautions are necessary to realize the true ambient as a heat sink; actual temperatures in a parked car can be significantly higher than ambient unless suitable precautions are taken.

II. Technical Approach

The means for temperature conditioning of the battery were categorized. The categories of active and passive were selected as encompassing all possible means applicable to effecting a change of battery temperature.

Categories:

A. Passive

Storing or rejection of heat during charge and/or during operation.

Active

1. The use of on-board energy dedicated to heat or cool the batteries.

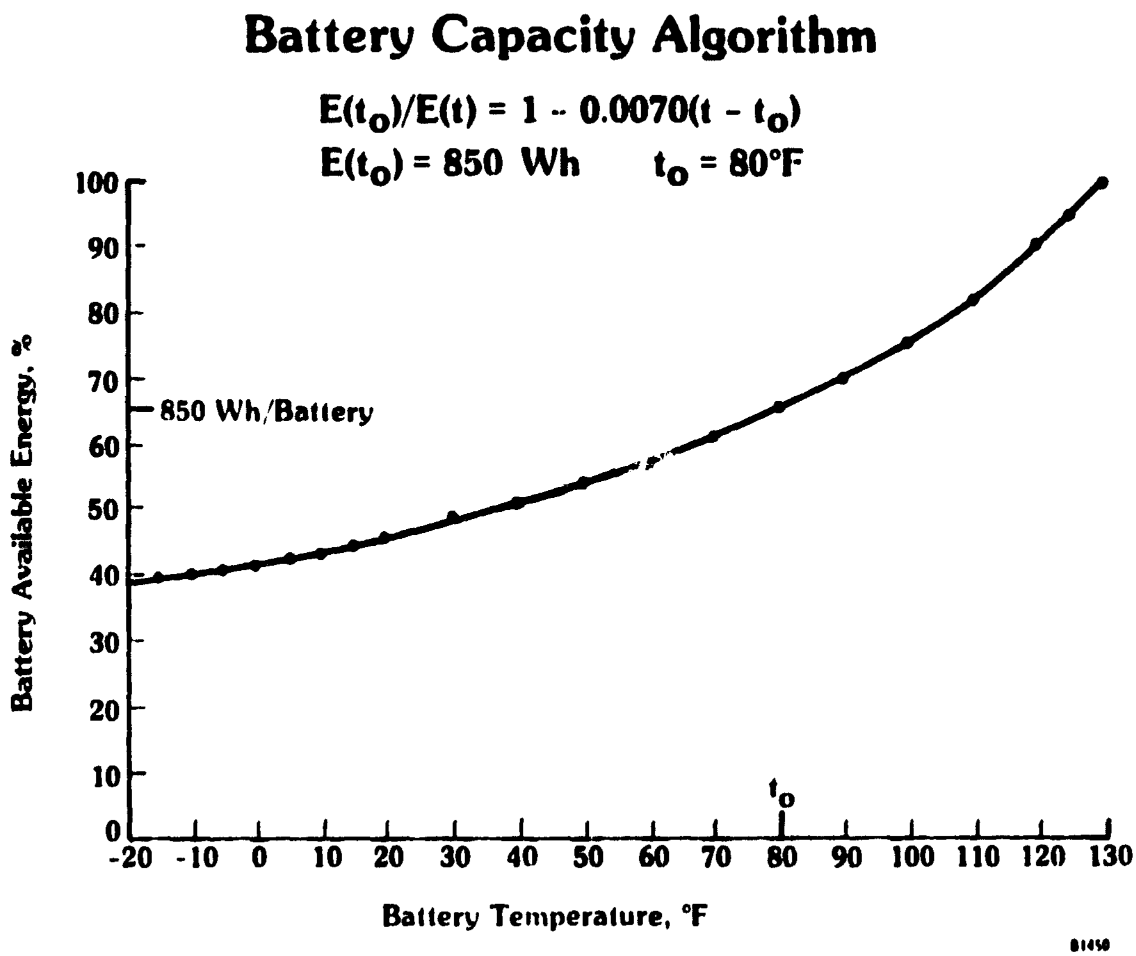
2. The use of off-vehicle energy to heat or cool the batteries.

In review of these categories and consideration of various configurations in a qualitative assessment it became apparent that:

a) It was desirable to keep the batteries at the highest possible safe temperature to achieve the benefit of the greatest possible capacity (Refer to Figure 1).

b) Useful quantities of heat were available, as a "by-product", from the charging process and some heat was also available from discharge.

Therefore, specific configuration were quantitatively examined that used insulation to trap the "by-product" heat available from the batteries. The various



Battery Capacity Algorithm
E(to)/E(t) = 1 - 0.0070(t - to)
E(to) = 850 Wh   to = 80°F
850 Wh/Battery
Battery Available Energy, %
Battery Temperature, °F
to
0
10
20
30
40
50
60
70
80
90
100
-20
-10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
81450

Figure 1.

configurations were selected to seek an optimum for volume devoted to insulation versus heat loss. Only well established, state of the art materials were considered.

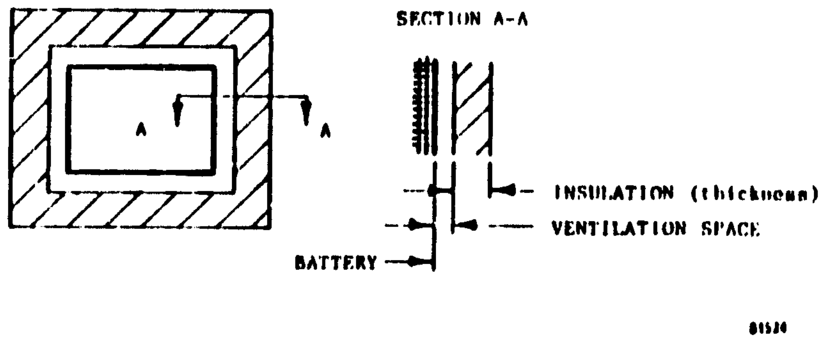
The requirement for rejection of heat from the batteries appears to be straightforward. In the calculations the batteries are permitted to reach 130°F and it is assumed that cooling air at 100°F is available. With that temperature differential available the dissipation from lead acid batteries is within the capability of a small capacity blower.

III. Assessment of Specific Concepts

Four different configurations were considered and competitive worst case results were determined.

A. Insulated Configuration

1. Schematic Diagram:



2. Considerations:

   a) with a starting point of 120°F ± 10°F after battery charging (130°F was elected for these calculations), what is the trade-off in battery capacity with insulation thickness?

   b) in a practical vehicle the battery compartment size limits the acceptable insulation thickness, two inches was used for this analysis.

   c) the insulation of choice for these calculations was polyurethane foam at "R" = 7 per inch of thickness.

3. The results are described in Figure 2 and the following Table.

TABLE

Effect of Insulating Thickness (@ R = 7 per inch) on Vehicle Range (R) and on Final Electrolyte Temperature (Tb-final)

with ambient temperature = 20°F
$T_{battery}$ = 130°F @ time = 0

Elapsed Time	No Insulation		1-1/2 inch insulation		2 inch insulation		2-1/2 inch insulation	
	$\frac{\Delta R}{R}$	T_b-final	$\frac{\Delta R}{R}$	T_b-final	$\frac{\Delta R}{R}$	T_b-final	$\frac{\Delta R}{R}$	T_b-final
10	0	- 7	0.7	100	0.7	107	0.8	111
12	0	- 12	0.7	95	0.8	103	0.8	108
24	0	- 20	0.7	68	0.8	80	0.9	88
36	0	- 20	0.6	48	0.7	62	0.8	72
48	0	- 20	0.5	32	0.6	47	0.7	58

Example:

The temperature dropped from 130°C to 62°F in 36 hours with insulation of R=14 and to -20°F with no insulation. The battery capacity at the time averaged temperature for the interval was used to calculate (ΔR/R), the range effect due to the insulation.

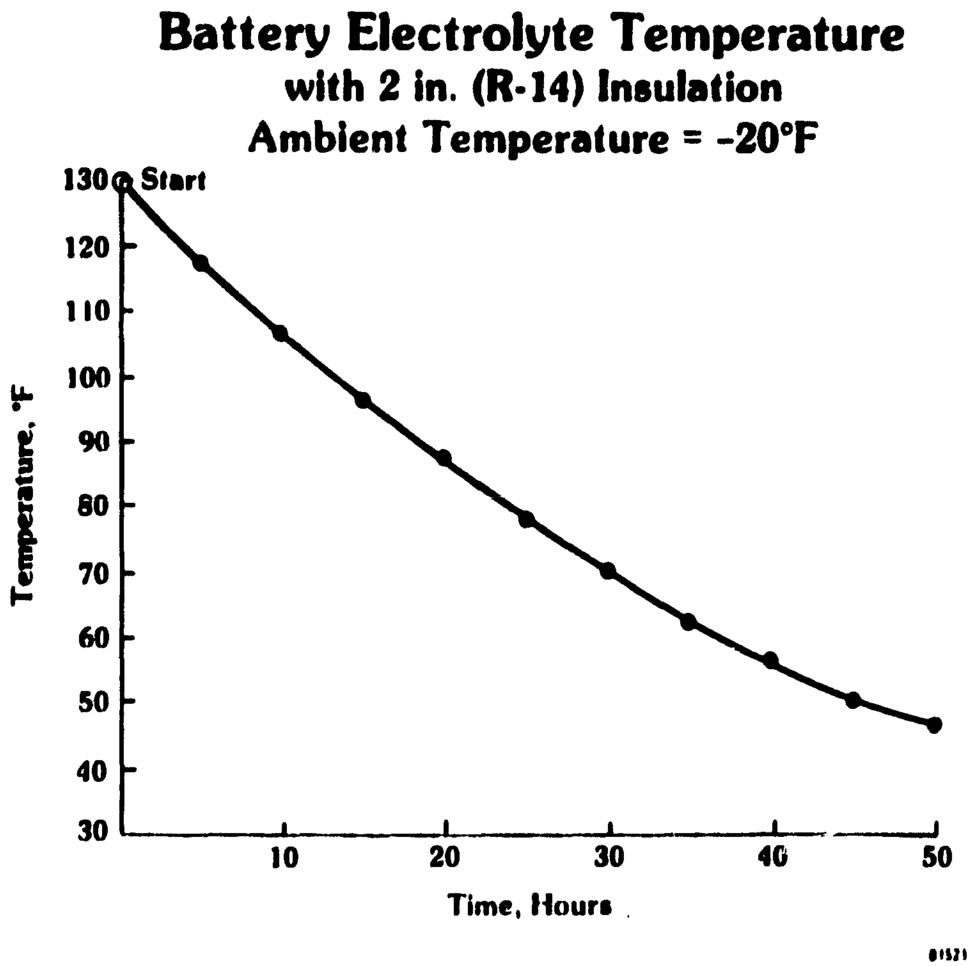
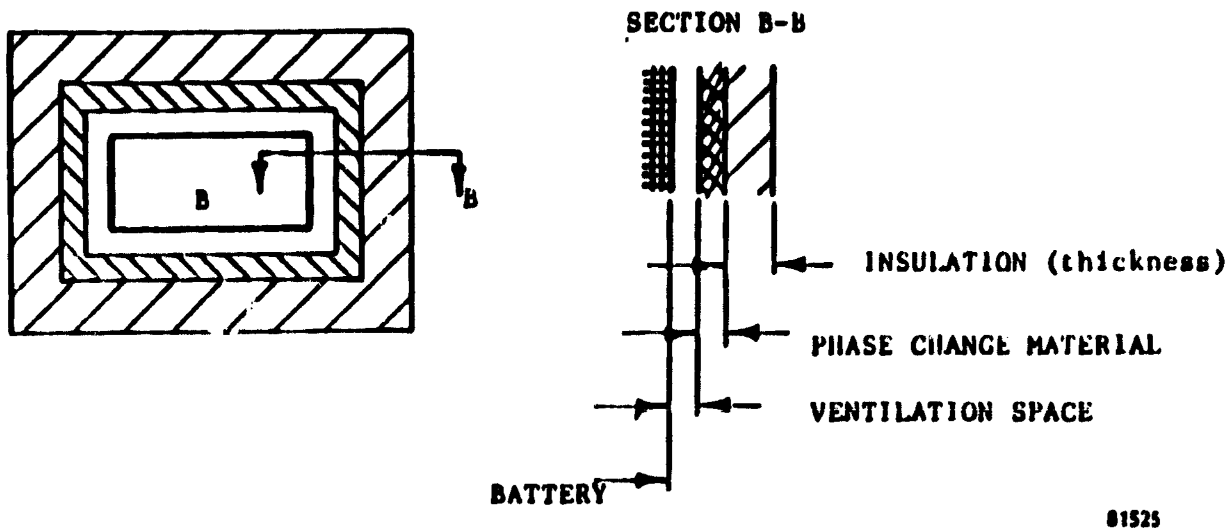

Battery Electrolyte Temperature
with 2 in. (R-14) Insulation
Ambient Temperature = -20°F
Start
Temperature, °F
130
120
110
100
90
80
70
60
50
40
30
10
20
30
40
50
Time, Hours

Figure 2.

B. Insulation with "Thermal Inertia" Configuration

1. Schematic Diagram



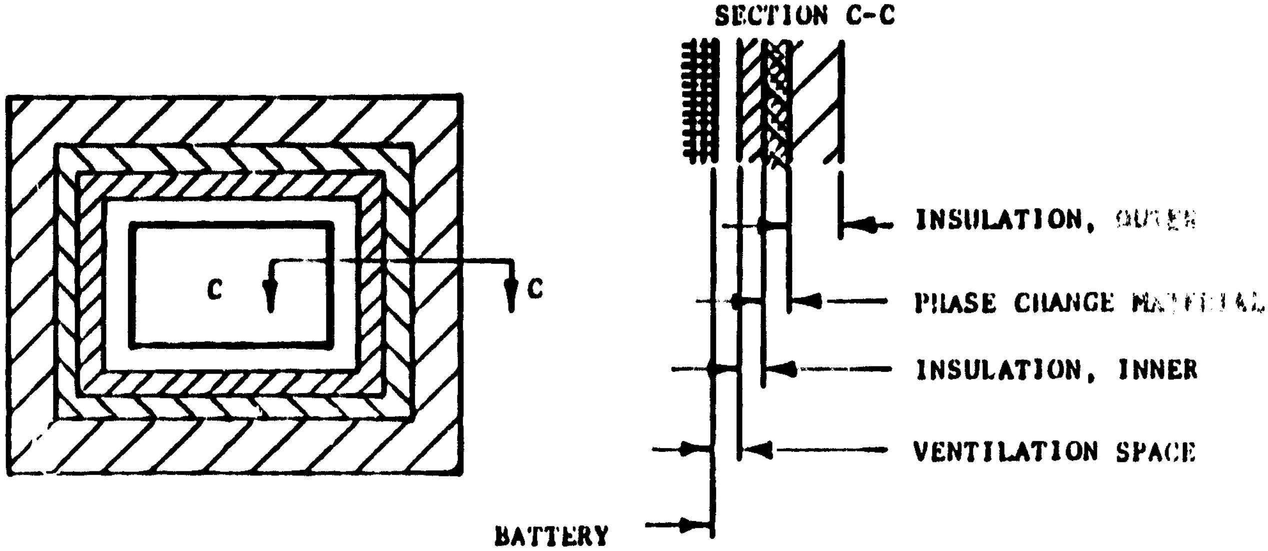
2. Considerations:

With a starting point of 130°F what is the trade-off in battery capacity with insulation thickness, phase change material properties and thickness to obtain the best 10 hour performance with minimum thickness.

3. Cursory analysis indicated that an insulation/phase change material sandwich provided optimal results. This was evaluated as Configuration C.

C. Insulation with Thermal Inertia Sandwich Configuration

1. Schematic Diagram



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2. Considerations:

The placement of the phase change material, within the insulation, results in a thinner total layer around the battery with the same 10 hour effectiveness. This construction was optimized for two phase change fluids applied within 2 inches of R = 7 per inch insulation.

The use of water and also Eicosane was considered as the "thermal inertia material".

3. **Results**

a) **Optimized Placement and "thermal inertia material" applied within 2 inches of R = 7 per inch insulation.**

Material and Quantity of "thermal inertia material"	125 lb. Water	6 lb. Eicosane
Inside Insulation	1.0 inch	1.7 inch
Outside Insulation	1.0 inch	0.3 inch
Total Thickness	2.3 inches	2.1 inches

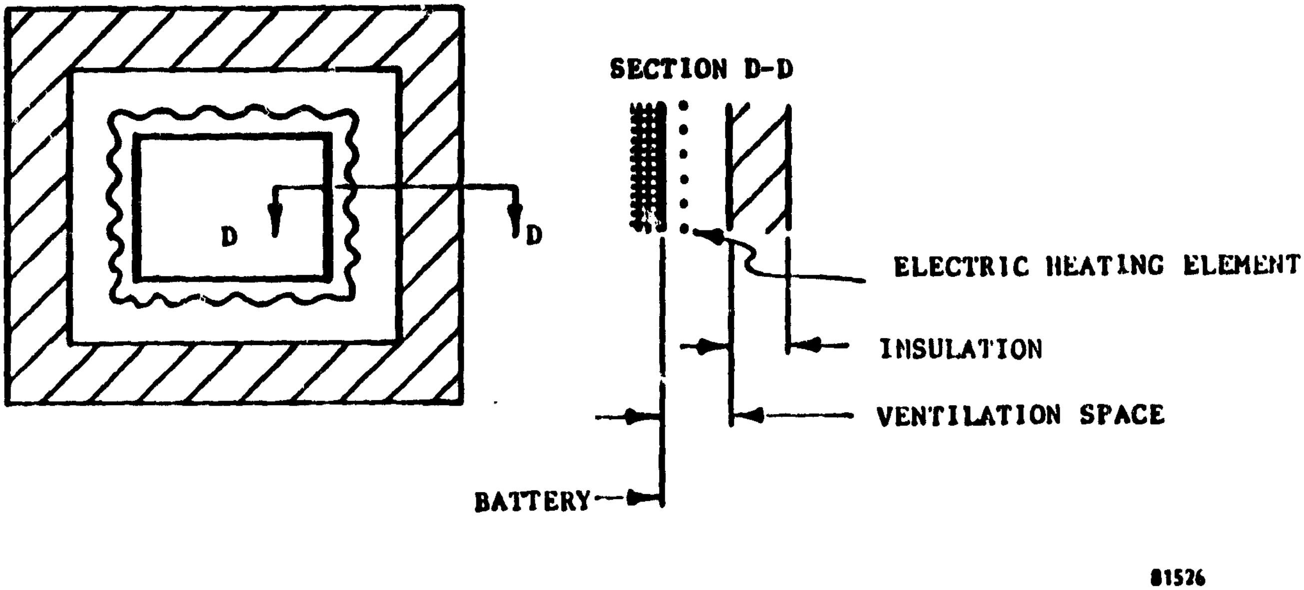
b) **Effect on vehicle range (R) and final battery temperature after 10 hours when starting with battery electrolyte at 130°F, ambient of -20°F and no heat dissipation from the battery during the 10 hour period.**

Material	Water	Eicosane	None	None
Insulation	2 inches	2 inches	2 inches	None
$\frac{\Delta R^*}{R}$	+ 0.76	+0.84	+0.76	0
Final $T_{battery}$	119°F	124°F	107°F	- 7°F

*considers weight effect, which is small, i.e. 0.1

D. Insulated and Electrically Heated Configuration

1. Schematic Diagram



2. Considerations

What are the trade-offs when adding heat derived from electric battery power to keep the battery temperature at a predetermined set value?

The variables considered include:

Insulation
Ambient Temperature
Temperature Set Point
Time for Initiation
State of Charge, at Initiation

3. <u>**Initial Assessment**</u> **(Refer to Table below)**

With 2 inches total insulation of R = 7 perinch

Tambient of -20°F

At time = 0, battery is fully charged, electrotype is at 130°F

Condition	With T_{set} 110°F	With T_{set} 100°F	With T_{set} 90°F	With T_{set} 80°F
Initial Period	$\frac{\Delta R}{R} = 0.74$ for 8 hours	$\frac{\Delta R}{R} = 0.79$ for 13 hours	$\frac{\Delta R}{R} = 0.80$ for 19 hours	$\frac{\Delta R}{R} = 0.80$ for 24 hours
Loss per Hour After Initial Period	185 W	171 W	157 W	143W
Range Effect After Initial Period with a) Heater On per hour	$\frac{\Delta R}{R} = 0.\bar{0}096$	$0.\bar{0}097$	$0.\bar{0}10$	$0.\bar{0}12$
b) Heater Off to 36 hours; Avg. per hour	$\frac{\Delta R}{R} = 0.\bar{0}001$	$0.\bar{0}001$	$0.\bar{0}05$	$0.\bar{0}07$

IV. Conclusion

A. Calculation Methodology

Assessing the range benefit of different concepts for battery temperature conditioning implies that probabilistic effects be considered. These effects include -

periods of non-use

state of charge at non-use.

To avoid complex computations that simulate these effects, the following simplifications have been made in all of the initial assessments.

1. Electrolyte temperature at the initiation of the non-use period is 130°F.

2. State of charge during non-use is 'fully charged'.

3. Optimization is done on an average temperature basis for the time interval analyzed.

4. Initial Assessment

   With Tambient of -20°F

   $T_{battery}$ equal to 130°F at time = 0

   $\frac{\Delta R}{R}$ for the average temperature over the period evaluated.

B. Recommendation

The preferred method is to use simple insulation with a small thermostatically operated blower using ambient air for cooling. Attention will be needed to secure the lowest available temperature, avoiding stagnant under hood heated air.

An option of heating the batteries, using garage power when available and battery power when under way may be useful for the geographic areas of most severe cold.

The benefits of this approach appear significant. The trade-off to providing additional volume to accomodating the insulation has important implication to vehicle design and battery maintenance. No significant improvement was found on this simple configuration.

G.1 Development Effort

The effort to develop EHV Environmental Control Systems has been estimated for three configurations:

- Engine Driven Heat Pump
- Thermal Storage
- Split Heat Pump

The development effort has been estimated to establish the absolute level of effort and also to provide a comparison between the requirements of the three configurations that were studied. In each configuration the estimate comprises one iteration of design through the laboratory evaluation using available components to the greatest possible extent without jeopardizing function and performance. Similar schedules were considered in the three estimates of level of effort. This typical Development Schedule is shown in Figure G-1.

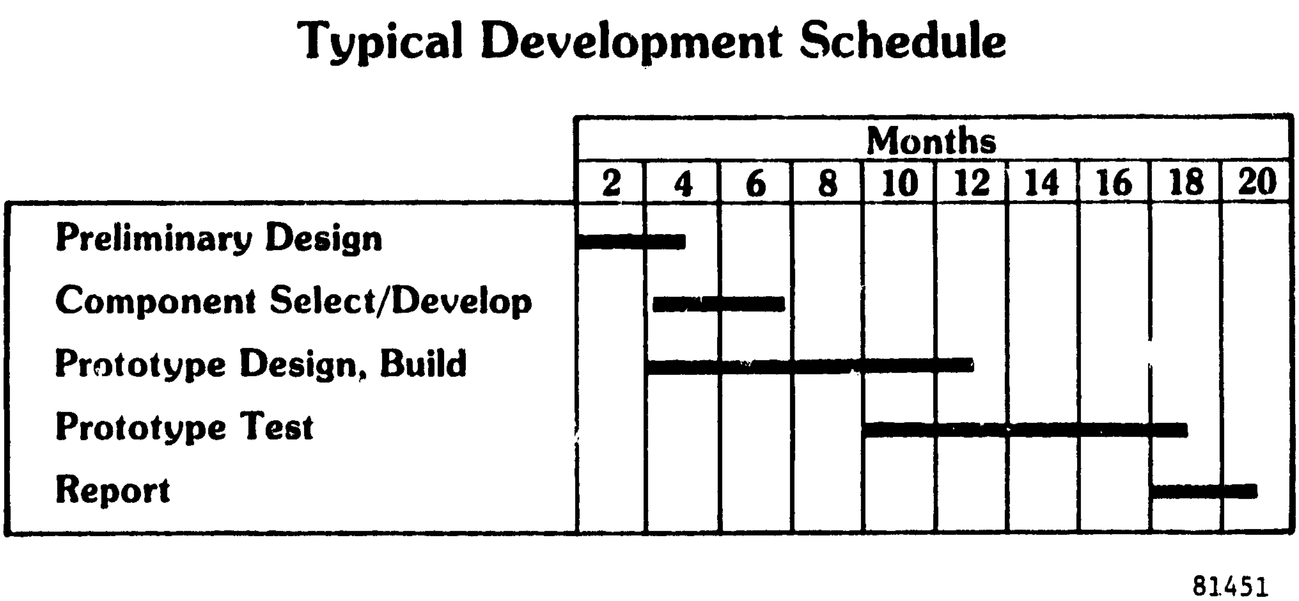


Figure G-1

G.2 Scope of Tasks

The scope of work to be performed in the development was defined for each of the three configurations. The task breakdown was:

a) Preliminary Design which includes:

1. listing and definition of design parameters
2. layout of heating and cooling system
3. layout of charging system (if applicable)
4. definition of control strategy
5. design considerations for safety
6. definition of test procedures and data requirements

b) Component Select/Develop which includes:

1. optimize components for vehicle unit
2. optimize components for recharge unit(s), (if applicable)
3. select best commercially available components

c) Prototype Design and Build which includes:

1. iterate the preliminary design to make use of commercially available components
2. integrate ECS design with a generalized vehicle.
3. design the test equipment
4. complete prototype design, including controls, safety considerations and vehicle integration
5. review design and interaction of test equipment and vehicle
6. detail design prototype parts
7. obtain quotations of vendor parts, conduct vendor discussions
8. order all materials, fabricate parts

d) Prototype Test which includes:

1. assemble prototype
2. provide data acquisition equipment
3. debug prototype, test equipment and data acquisition
4. perform tests in accord with the test procedure
5. collect and log data

e) <u>Report</u> which includes:
   1. analyze data
   2. review results
   3. document all work
   4. deliver draft report
   5. deliver final report

The evaluation for any of the candidate ECS's requires a laboratory test capability that can perform as a Psychrometric Test Chamber. The principle features of such a chamber are identified, refer to Figure G-2.

Psychrometric Chamber Test System

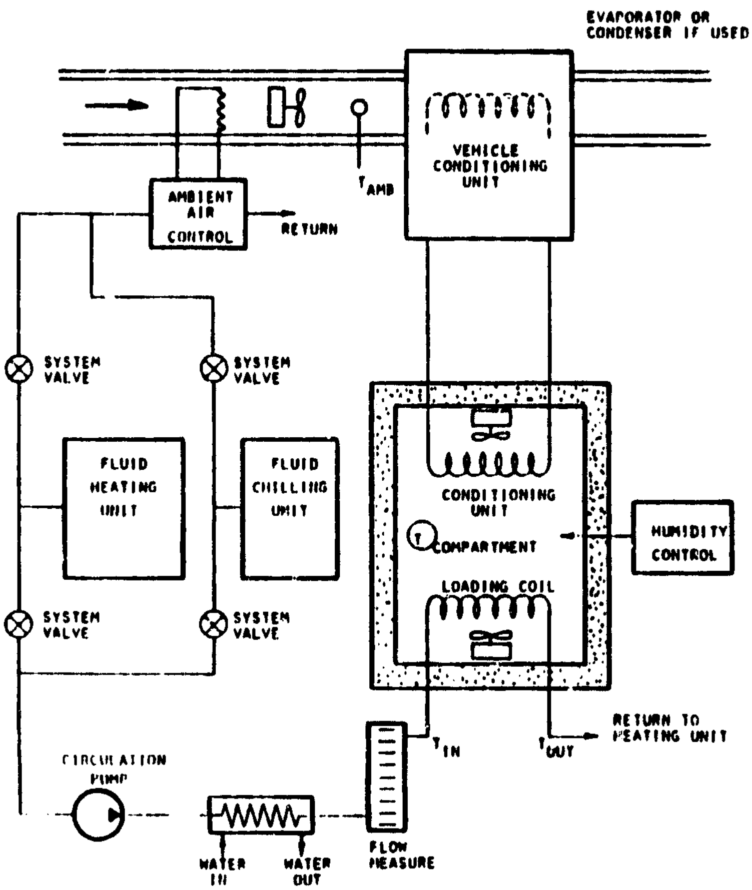


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Figure G-2

G.3 Configurations

The selection of the three configurations recommended for furhter consideration is described in Appendix E of this report. The three configurations are identified again here, as representative of the starting point for the development effort estimate.

a) Engine-Driven Heat Pump
Figure G-3

b) Thermal Storage System
Figure G-4

c) Split Heat Pump System
Figure G-5

Engine-Driven Heat Pump

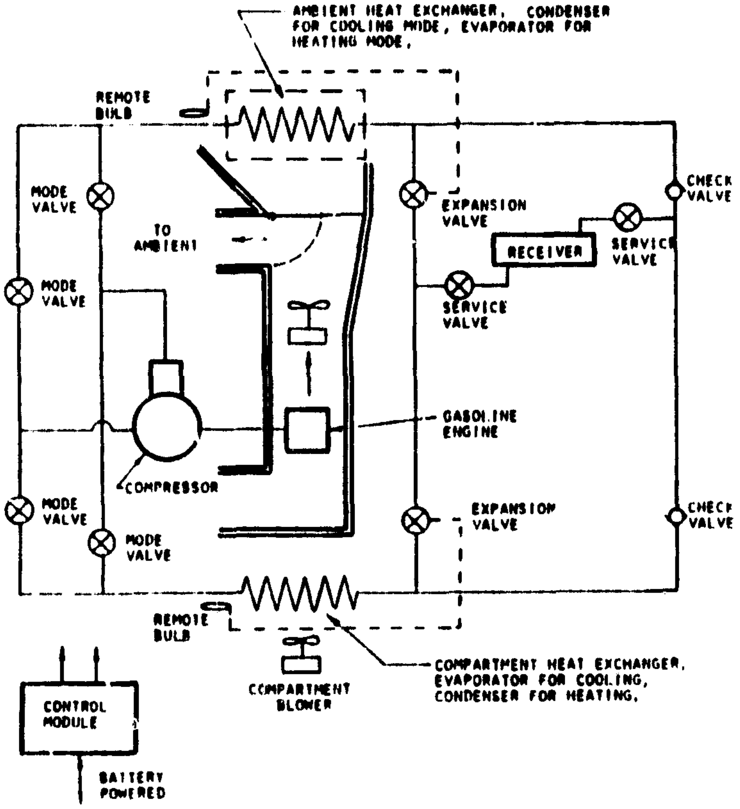

Figure G-3

Thermal Storage System

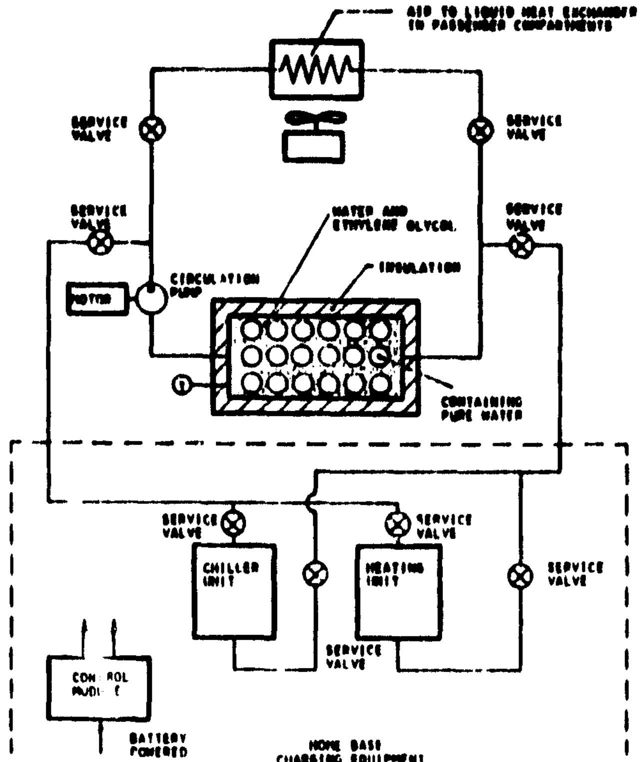

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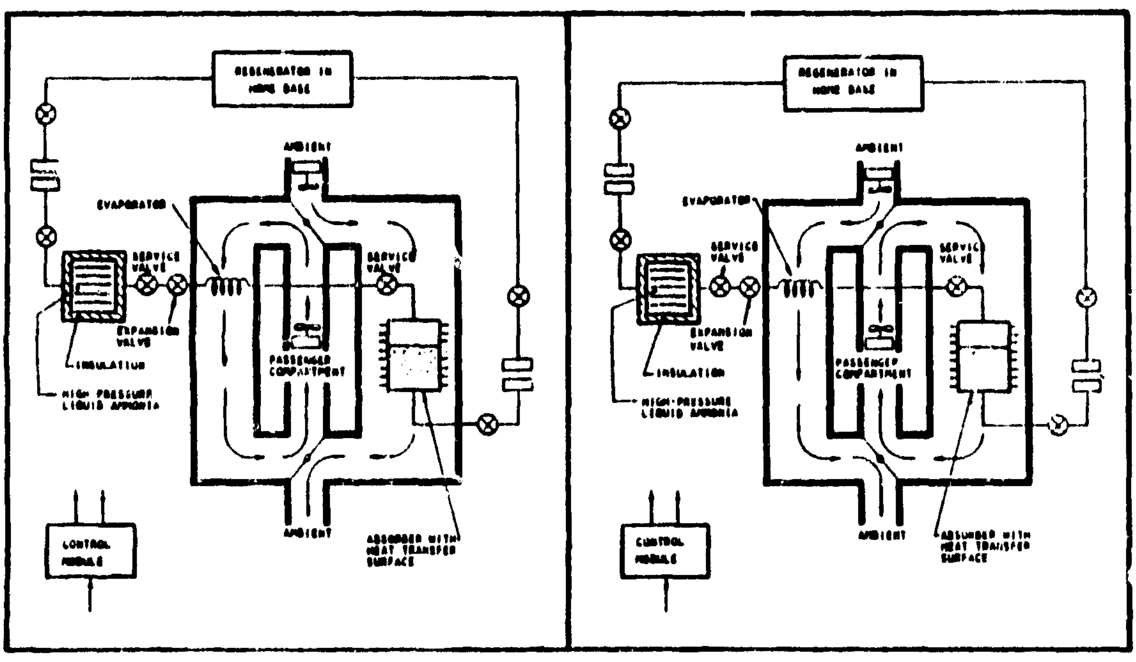
Figure G-4

Split Heat Pump System



Cooling Mode **Heating Mode** 81352

Figure G-5

G.4 Level of Effort

The effort in terms of manhours and material cost is described by task for each of the three configurations in Figure G-6.

EHV Environmental Control System Development

Task	Engine-Driven Heat Pump		Thermal Storage		Split Heat Pump	
	Manhours	Materials $K	Manhours	Materials $K	Manhours	Materials $K
Preliminary Design	840	1	850	1	1,150	2
Component Select/ Develop	920	4	1,490	6	1,360	86
Prototype Design and Build	3,180	63	4,660	65	5,160	35
Prototype Test	3,210	35	3,620	36	3,650	36
Report	650	2	780	2	780	2
Total	8,800	105	11,400	110	12,100	161

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Figure G-6

G.5 Risk Assessment

An evaluation was made of the confidence in the availability of applicable technology for each of the configurations. From this assessment, the risk incurred in achieving a successful development within the estimated budget and schedule can be made. Please note that the code used is:

SSS = State of the Art

RR = Technology exists but may require modification for vehicular application

D = Extensive development is required

a) Engine-Driven Heat Pump using a gasoline engine:

1. Heat Pump Technology = SSS
2. System Efficiency - RR
3. Gasoline Engine Performance - SSS
4. Engine Waste Heat Utilization - RR
5. Heat Exchanger Technology - SSS
6. Passenger Compartment - SSS
   Heat Exchange Equipment

7. Unit Packaging - RR
8. Compartment Ducting - SSS
9. Compartment Lossess - SSS
10. Equipment Reliability - SSS
11. Control Strategy - RR
12. Test Equipment - SSS

b) Thermal Storage System using ethelyne glycol as the working fluid:

1. Ethylene Glycol Storage - SSS
2. System Efficiency - RR
3. Storage Vessels - RR
4. Liquid Storage Vessels - RR
5. Passenger Compartment
   Heat Exchange Equipment - SSS
6. Unit Packaging - D
7. Compartment Losses - SSS
8. Compartment Ducting - SSS
9. Equipment Reliability - D
10. Control Strategy - RR
11. Recharge Equipment
    Technology - SSS
    Packaging - RR
12. Test Equipment - SSS

c) Split Heat Pump System

1. Absorbtion Technology - RR
2. System Efficiency - RR
3. Liquid Ammonia Storage - D
4. Evaporator - RR
5. Absorber/Heat Exchanger - D
6. Passenger Compartment - RR
   Heat Exchange Equipment
7. Unit Packaging - RR
8. Compartment Cucting - RR
9. Compartment Losses - SSS
10. Equipment Reliability - D

11. Control Strategy - D

12. Recharge Equipment
    Technology - RR
    Packaging - D

13. Test Equipment
    Vehicle Unit - SSS
    Recharge Unit - RR