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heat pump technology

A Survey Of Technical Development Market Prospects

June 1978

Prepared For

U.S. DEPARTMENT OF ENERGY
Assistant Secretary For
Conservation And Solar Applications
Division Of Buildings And Community Systems

Under Contract No. EX-76-C-01-2121

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Prepared By

Gordian Associates, Inc.

For The

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Assistant Secretary For
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Washington, D.C. 20545

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GORDIAN ASSOCIATES INC.

HEAT PUMP TECHNOLOGY

A Survey of Technical Developments,
Market Prospects and Research Needs

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SUMMARY

This study evaluated the on-site energy consumption, energy and life cycle cost, market prospects, institutional factors and primary or resource energy efficiency of heat pumps for residential and commercial applications. Only unitary equipment was considered; however, the investigation extended from conventional electric heat pumps now being marketed to experimental gas thermal engine or absorption cycle machines now under development. Other heat pumps were also considered as were heat pump combinations with solar energy systems.

Calculations of the on-site and primary energy effectiveness and cost of conventional and advanced heat pumps in comparison to electric and combustion furnaces and baseboard heat (all with appropriate cooling systems) were made by means of two hour-by-hour computer simulation programs - one for a residential building, one for a commercial building. These programs modelled in significant detail the various mechanisms of building heat loss or gain based on inputs of weather data, data on insolation, building geometry, structural characteristics, occupancy schedules and the like. The heating and cooling energy consumption of conventional and advanced heat pumps and other systems could then be modelled and calculated in considerable detail. As test buildings, a two-story frame single family residence (1850 sq. ft. of conditioned area) and a two-story masonry office building (5580 sq. ft. of conditioned area) were selected. These buildings were hypothetically moved about to nine different locations (cities) selected as representative of the various climatic regions of the continental United States.

Energy costs, installed costs and life-cycle owning and operating costs were then determined for the residential and commercial HVAC systems compared as of the First or Second Quarter of 1977. Energy costs were based on locally obtaining utility rates, accounting for base load usage and demand charges, where appropriate. Installed costs were based on a detailed estimate from a New Jersey area contractor and include all costs of labor, materials, shipping and handling and local taxes. Life cycle costs were also calculated assuming an 8% annual inflation in energy costs and a lower rate of inflation in maintenance costs.

Models of the residential and non-residential building market were used to project market penetrations of electric and gas heat pumps on a regional basis. The models account for replacements and conversions as well as new construction and permit adjustment for factors such as air conditioning demand and gas availability. The market projections were then combined with primary fuels and energy consumption results to arrive at estimates of energy savings potentially obtainable with introduction of advanced electric and gas heat pumps. The study also considered institutional factors such as consumer attitudes, building codes, gas availability and other pertinent factors. The findings of the study were as follows:

As expected, electric heat pumps were found to be more efficient than any of the all-electric HVAC systems examined. High efficiency residential electric heat pumps had performance factors in the 2.56-1.51 (0.78-0.43 on primary energy basis) range for heating and the 1.91-2.56 (0.61-0.82 on primary energy basis) range for cooling, constituting an average 12% improvement over the standard heat pump representing 1975

technology. In primary energy terms, electric heat pumps were comparable to or slightly less efficient than the combustion heating systems combined with electric cooling in regions with a significant heating load. In very cold regions, the combustion systems were more efficient. However, combustion system efficiencies were calculated based on insulated supply and return air ductwork. This is the observed practice with electric but not with combustion heating systems in general. Calculations showed that the performance of the combustion furnaces was considerably improved with duct insulation. With uninsulated ducts, electric heat pumps would probably have a slightly higher efficiency than the combustion systems.

Gas heat pumps now under development have a higher efficiency in a primary energy sense than electric heat pumps, combustion furnaces or resistance heating systems. However, if used to displace electric heat pumps in new construction, gas heat pumps would accelerate the depletion of the scarcest fuel--natural gas--while conserving the more abundant energy resources.

Of the residential gas heat pumps examined, the gas-fired free-piston and the V-type single cylinder Stirling-Rankine heat pump were found to have performance factors in the 1.16-0.91 range for heating and in the 0.68-0.66 range for cooling. The corresponding results for the organic fluid absorption heat pump were lower: 0.95-0.87 for heating and 0.48-0.46 for cooling. The comparatively poor cooling performance of the absorption machine has the result that, in areas with a significant cooling requirement, a marked increase in natural gas consumption would result if this heat pump were substituted for a conventional gas furnace with high efficiency electric air conditioning. However, other forms of primary energy would still be conserved everywhere.

For commercial applications, the Rankine-Rankine and Brayton-Rankine heat pumps have a higher primary energy efficiency (performance factors 0.76-1.43 heating; 0.72-0.91 cooling) than electric heat pumps or roof top air conditioners with gas or electric resistance heat. Again, however, increases in natural gas consumption would result unless used to displace existing gas equipment.

Life cycle cost studies showed that conventional residential space conditioning systems are, in general, still more cost-effective than most electric heat pumps, including the high efficiency models. When inflationary effects on energy and maintenance costs are included in life cycle costs, however, the electric heat pump systems appear competitive with most of the conventional systems, with the exception of the gas warm air furnace and central electric air conditioner. Because of their high efficiency and the low price of gas compared to other fuels, gas heat pumps would have lower energy costs than those of electric heat pumps or other HVAC systems. Because of their increased complexity, the first costs of gas heat pumps for residential application would probably be higher than those of electric heat pumps. Thus whether gas heat pump life-cycle costs would ultimately be competitive with available alternatives is open to question. It is true that, in the one case where a cost estimate was available for a residential gas heat pump (the organic fluid aborption machine), it compared favorably with electric heat pumps. However, since the system is in an early prototype stage, it is likely that the cost estimate is rather optimistic.

Life cycle cost analysis of commercial-size equipment indicates that, except in cold climates where they compare favorably with roof-top air conditioners with resistance heat, electric heat pumps are in general costlier than available alternatives. The gas heat pumps, on the other hand, would be cheaper to own and operate than any of the conventional electric or fossil fuel systems. Since the gas thermal engine heat pumps are more complex than electric heat pumps and because these results are based on first cost estimates for equipment still in the "breadboard" stage, some skepticism is in order. If high-efficiency advanced gas furnaces in combination with electric cooling were successfully introduced, the cost-effectiveness and marketability of gas heat pumps would be impaired.

The general conclusion of this study is that no residential or commercial HVAC system is universally to be preferred to any other if energy effectiveness, life-cycle cost, building application and other pertinent factors are considered. Therefore, legislative or regulatory activity which favors one system over another would be counterproductive from an energy conservation standpoint if freedom of choice is to remain with the consumer. However, in a number of areas identified in the study, Federal Government sponsorship of heat pump research could assist in the evolution and market penetration of this energy-conserving technology.

LIST OF ABBREVIATIONS AND ACRONYMS

AEIC	Association of Electric Illuminating Companies
AGA	American Gas Association
ARI	Air Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
ASME	American Society of Mechanical Engineers
BOCA	Building Officials and Code Administrators
COP	Coefficient of Performance
DOE	Department of Energy
EEA	Electric Energy Association, now merged with EEI
EEI	Edison Electric Institute
EER	Energy Effectiveness Ratio
EPRI	Electric Power Research Institute
ERDA	Energy Research and Development Administration, now DOE
FEA	Federal Energy Administration, now DOE
FERC	Federal Energy Regulatory Commission
FHA	Federal Housing Authority
FPC	Federal Power Commission, now FERC
GAMA	Gas Appliance Manufacturers Association
HUD	Housing and Urban Development
HVAC	Heating, Ventilating and Air Conditioning
ICBO	International Conference of Building Officials
IGT	Institute of Gas Technology

LNG	Liquified Natural Gas
LPG	Liquified Petroleum Gas
MPS	Minimum Property Standard
NBS	National Bureau of Standards
NEC	National Electric Code
NEMA	National Electrical Manufacturers Association
NESCA	National Environmental Systems Contractors Association
NOAA	National Oceanic and Atmospheric Administration
ORNL	Oak Ridge National Laboratory
OSU	Ohio State University
SNG	Synthetic Natural Gas
SPF	Seasonal Performance Factor

1. INTRODUCTION

1.1 Objective and Scope of this Study

The purpose of this study was to evaluate the performance, cost and market prospects of heat pumps - both conventional and advanced - and to assess research and development needs and make recommendations concerning Federal Government policies and support of this type of space conditioning technology. The study covered electric as well as experimental gas and other types of heat pumps and examined equipment for residential and commercial applications. Concerning the latter, unitary equipment of the type applicable to light commercial, institutional or industrial construction was investigated in preference to large built-up or applied systems, since all but one of the experimental heat pump concepts encountered in the course of the study were of the unitary (i.e. self-contained single package or split-system type).*

It is probably impossible to do a meaningful study of advanced heat pump technology without investigating or at least touching on specific designs, "hardware embodiments" or individual experimental heat pump development projects. However, a distinction, albeit fine, must be drawn between technology assessment in a broad sense such as is attempted here and proposal or project evaluation in a narrow sense. The latter would be beyond the scope, time and funding limitations of our study. There is another reason for this distinction also: the availability of sufficient information to support a specific judgement on the merits of a particular project vs. a more general assessment of the technical concepts represented. In some cases the technical or market information made available to us was quite substantial; in other cases, presumably proprietary reasons

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* See Section 1.4.

prevented us from getting any but the most summary or cursory description of the configuration involved or the basis of claims or statements made about the feasibility or performance of these experimental heat pumps. The rest of the details had to be pieced together as well as possible from information available elsewhere. In all cases, but with obviously varied success, the authors attempted to determine the factual basis and degree of credence to be placed in performance and other data presented as well as to conduct a very detailed analysis of the energy consumption and economics of these systems in typical building applications. This is the subject of the chapters that follow.

1.2 Background and Survey of Previous Work

In the last decade and the last several years especially, a considerable literature has grown on the subject of heat pumps, although, of course, a number of articles on the subject date back to much earlier. In view of this, a discussion of selected technical literature describing prior work should be helpful and one is provided. The review makes no attempt to be comprehensive; rather, its purpose is to give the reader some background and additional source material on the topic of heat pump performance improvement.

A survey of the technical literature on heat pumps should properly begin with the well-known monograph by Ambrose (1966). The author discusses the thermodynamics of the Carnot and Rankine refrigeration cycle, heat sources and sinks for the heat pump, typical configurations and applications, energy estimation methods and residential and commercial diversified demand characteristics. Additional discussions of heat pump

fundamentals may be found in the ASHRAE Handbook of Fundamentals (1972) and the "Equipment" and "Systems" volumes of the ASHRAE Handbook and Product Directory (1975, 1976). Early efforts at electric heat pump efficiency and reliability improvement are described in two ASHRAE symposium publications (ASHRAE 1970, 1972) and in Edison Electric Institute's final report on the Heat Pump Improvement Research Project (EEI, 1971).

A number of recent reports contain energy consumption and cost data on heat pumps and other heating and cooling systems. Blacklaw and Johnson (1973) conducted a study of heat pump performance, cost and reliability in the Seattle area on the basis of approximate heating and cooling load and energy calculations. A number of case histories based on interviews with homeowners of heat-pump-conditioned homes were included in their study. Delene (1974) of Oak Ridge National Laboratory calculated energy requirements and life cycle costs in thirteen U.S. cities for a 1800 ft^2 residential building heated with a heat pump, electric furnace, gas furnace, oil furnace, electric ceiling or baseboard heat with air conditioning, where appropriate. The Delene study utilized results of energy calculations which assume steady state efficiency values of 60% for gas and 55% for oil. One conclusion of the Delene study was that "no single residential heating system is superior to all others in all regions of the country," a conclusion that reflects the results obtained by the authors of this study as well. A later study by Hise (1975), also of Oak Ridge National Laboratory, discusses seasonal performance in detail and presents a critical review of some of the literature data on comparative energy consumption. Reliability is discussed in an appendix to the Hise report. Additional more recent comparative energy studies based on theoretical calculations have been published by Dunning et al. (1974), and Dunning and Geary (1974) of Westinghouse.

A precursor to this study, in certain respects, is the report entitled Evaluation of the Air-to-Air Heat Pump for Residential Space Conditioning by Gordian Associates (1976). That study, performed under contract to the FEA, analyzed the reliability, market acceptance and energy effectiveness of conventional air source electric heat pumps in comparison to electric resistance and combustion heating and electric air conditioning systems. The study was based in part on data gathered from heat pump manufacturers, electric utilities, HVAC contractors and other sources and presented energy consumption and annualized owning and operating cost data from an hour-by-hour computer simulation study of the Kray residence, one of the thoroughly instrumented, actually occupied single family houses in Columbus, Ohio used in ongoing research by Charles Sepsy of Ohio State University.*

A study comparing zoned electric vs. central electric (including the heat pump), gas and oil systems was conducted by Energy Utilization Systems (1977?) showing the energy and cost savings obtainable with zoned (i.e. individual-room-controlled) vs. a central space conditioning system. With the current emphasis on system efficiency, this means of minimizing energy usage is sometimes obscured.

Heat pumps have been the subject of a number of technical conferences recently. The Oklahoma State University has organized two conferences on heat pump technology held in 1975 and 1976 at Stillwater, Oklahoma. There have also been two Purdue University conferences on improving the efficiency of residential and commercial HVAC equipment (Goldschmidt and

* The on-site energy consumption calculations in the Gordian study for the FEA (Gordian, 1976) were conducted by C. Sepsy of Ohio State University while under subcontract to Gordian. Sepsy's basic model for residential on-site energy analysis, subsequently substantially modified by Gordian, was used in one part of the residential energy use phase of this study as well.

Didion, 1974 and 1976), as well as the conferences on compressor technology held in 1972, 1974 and 1976. In 1975, the Pennsylvania State University organized a conference on solar energy heat pump systems for heating and cooling of buildings (Gilman, 1975). Proceedings of the above conferences have been assembled or published.

In recent years a number of more fundamental studies of the heat pump cycle have appeared in the literature. Hiller and Glicksman (1976) developed a computer model to calculate the performance of various components of the air-to-air heat pump refrigeration loop from a combination of first principles and empirical correlations, where the latter are necessary. The model employs steady state fluid flow and heat transfer theory but goes into considerable depth, such as calculating two phase fluid flow pressure drops in refrigerant lines, etc. The model was validated by comparing calculated capacity and power consumption results against the manufacturers' experimental data on three different heat pumps and appears to be in good agreement with performance data measured by the manufacturers. The authors applied the model to evaluate an experimental early compressor suction valve closing device as an approach to capacity modulation. To our knowledge, this is the most detailed heat pump refrigerant loop simulation model described in the literature, although some equally detailed proprietary design-type models are known to be in use by several heat pump manufacturers and others.

One interesting approach to heat pump system modelling has been undertaken by Carrier, as described in a paper by Groff and Bullock (1976). Compressor capacity is simulated by a first-order transfer function with different values for the start-up and shut-down time constants; compressor power demand, on the other hand, is assumed to

reach its steady-state value instantaneously, i.e. is modelled as a step-function. Thermostat lag, defrost and other components of the refrigeration loop are also modelled by transfer functions, the model then being used to calculate demand profiles and energy consumption. The utility of this method is, of course, dependent on available experimental data to evaluate gains and time constants of the various system components. A parallel field study is, indeed, underway in cooperation with the Niagara Mohawk Power Corporation and should yield interesting results.

The transient in contrast to the usually measured steady-state performance of electric heat pumps has been the subject of ongoing experimental study at the National Bureau of Standards. Two papers on these studies have been published (Kelly and Bean, 1976; Parken et al., 1977) and will be discussed in subsequent chapters of this report.

A study of electric heat pump performance in the New Mexico area was recently completed by Wildin et al. (1977). The study was partly based on laboratory measurements of heat pump performance, partly on calculations, and accounts for defrost and other heat pump energy consumption aspects.

An analytical study of an optimum heat pump design for colder climates has just been completed by Westinghouse under EPRI sponsorship (Kirschbaum and Veyo, 1977 a,b,c). The study developed data on the effect of component modifications and the combination of heat pumps with solar collectors and thermal storage. It was apparently concluded that only a few significant changes could be made in electric heat pump design which would yield efficiency improvements at an acceptable first

cost and without detracting from equipment reliability. The Westinghouse study considered heat pumps in conjunction with solar energy collection and thermal storage systems as well. Chapter 3 of this report makes use of some of Kirschbaum and Veyo's results.

A number of potentially interesting field investigations of electric heat pumps are now underway. One such study is that of the load and use characteristics of heat pumps in single family residences being conducted by Westinghouse in cooperation with 12 participating electric utilities (under EPRI and the AEIC Load research Committee sponsorship). This study, involving a total of 120 heat pumps of recent make, is designed to accumulate experimental data not only on energy use, but also on demand characteristics and diversity.

The investigations cited above deal mainly with electric heat pumps, although other heating and cooling systems have been considered in some of them. In contrast, comparatively little information has been published on gas heat pumps now under development. An AGA research project final report on gas and electric heat pump concepts (Wurm, 1974) gives a good basic review of the subject. The papers of Wurm and Rush (1975) and Wurm et al. (1976) present theoretical and to some extent idealized analyses of thermal engine heat pumps. Colosimo (1976) reviews the status of the various AGA-sponsored gas heat pump development projects, a subject also covered in a recent ASHRAE Journal article by Sarkes et al. (1977). Coverage of the literature of thermal engines or absorption cooling is beyond the scope of this review.

A simulation study of gas and electric residential heating and cooling equipment, including the experimental free-piston Stirling-Rankine and organic fluid absorption gas heat pumps and a pulse-combustion

furnace, was conducted by AGA (Lewis, 1976). The study, based on modeling the performance of gas and electric test homes in Canton, Ohio, found operating costs of conventional gas furnace-electric air conditioning combinations considerably less expensive than the electric heat pump, and those of the advanced gas systems even lower. Life cycle costs were not considered. As far as the gas heat pumps were concerned, savings in gas consumption in comparison to furnaces were apparently offset to some degree by higher parasitic electrical requirements. Nevertheless, the gas heat pumps could theoretically heat and cool the test home with less gas than a conventional gas furnace would consume for heating alone.

This concludes the brief survey of some pertinent background material to this study. The next section presents a classification of the various heat pump concepts to be described in this report.

1.3 An Ad Hoc Classification of Heat Pumps

A "heat pump" is a device which extracts heat from a source at a low temperature and, by the input of mechanical or electrical work, rejects it to a sink at a higher temperature. Although only one type of heat pumping device, the electric-motor-driven Rankine cycle heat pump, is presently manufactured for residential and light commercial applications, a number of other heat pump cycles or concepts can be and have been devised.

For the purpose of this study, the following ad hoc classification of heat pumps will be used:

- Electric heat pumps,
- Thermal engine heat pumps,

- Absorption cycle heat pumps,
- Other types of heat pumps.

While this classification has obvious defects (at least to the purist), it is convenient for the purpose of the discussion that follows. The reader is forewarned that some of the heat pumps discussed may fit into more than one category and that, therefore, their classification is somewhat arbitrary. A more satisfying classification would have been according to the type of heat pump cycle or heat pumping process (e.g. Rankine, Brayton, Ericsson, absorption, hydration-vaporization, etc.), but that would have made the report organization too lopsided since the majority of the heat pumps under consideration are Rankine cycle machines.

1.4 Editorial Note

This document is a final report of an investigation conducted in the period of August 1976 - July 1977 under contract to the Energy Research and Development Administration. The first complete draft of this final report was issued in July 1977 and reviewed in October-November of that year. In early 1978, an extension of this investigation was commissioned by the sponsor (now reorganized as the Department of Energy) mainly for the purpose of revising the heat pump dynamic performance calculations in the report to incorporate the results of new data on electric heat pump dynamic operation based on ongoing research at the National Bureau of Standards. The results of these revisions are incorporated in this final report. However, while this was in progress a number of changes occurred in the status of the

technology described in this report. For example one manufacturer has announced plans to introduce a two-speed electric heat pump for residential applications; several others have introduced or are testing dual-source solar assisted heat pumps with thermal storage. Changes have occurred in the several gas-fired heat pump development projects as well. For example, the steam turbine-Rankine cycle heat pump is now being developed as a 100 ton built-up system; the organic absorption cycle heat pump field tests have been delayed, according to information available, and the Brayton-Rankine system reportedly encountered problems with the rotating group. Work on the free-piston Stirling-Rankine heat pump appears to be progressing on schedule. Since the field is evolving so rapidly, it would have been impossible to incorporate these and other changes in the text of the report. However, clarifying footnotes are added where appropriate to update the descriptive material.

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2. METHODOLOGY

2.1 General Approach and Study Plan

As already stated in Chapter 1, the purpose of this study was to evaluate advanced heat pump technology and to develop recommendations concerning research needs in this area. The general approach adopted in the investigation consisted of (a) a technical data-gathering phase including visits to or other contacts with heat pump manufacturers, institutions or organizations conducting heat pump research and development, utilities, trade and industry organizations and many others; (b) a computer simulation study of on-site energy consumption of various advanced heat pump and other HVAC systems in a single-family residence and a small office building; (c) an energy and life cycle owning and operating cost analysis; (d) an analysis of market and institutional factors and (e) a study of the primary resource energy impact and effect of heat pumps on utility load characteristics. The methods used in each of the above phases of the study are described further in the following sections.

2.2 Technical Developments Study

The analysis of technical developments presented in Chapters 3-7 of this report was based on a program of visits to selected developers of experimental heat pumps or other pertinent sources of data, augmented by a survey of the technical literature and other published information on the subject.

The purpose of this phase of the investigation was, of course, to establish the exact status of development of the various heat pump

concepts or technologies and to provide a base of data on their configuration, capacity and coefficient of performance for energy effectiveness simulation and cost studies. In connection with this investigation, visits were made to the following:

- Allied Chemical Corp.
- American Gas Association
- Carrier Corp.
- Columbia Gas System Service Corp.
- Consolidated Natural Gas Service Co.
- Dunham-Bush Inc.
- Garrett-Airesearch Manufacturing Co. of California
- General Electric Co., Corporate Research and Development
- General Electric Co., Space Division
- Lennox Industries Inc.
- Mechanical Technology Inc.
- Oak Ridge National Laboratory
- Sunpower Inc.

The field investigation also included attendance at several technical conferences where heat pumps were discussed. Other sources contacted included Copeland Corp., Energy Research and Generation Inc., Fedders Corp., General Electric Co. (Louisville, Ky. and Tyler, Tex.), Institute of Gas Technology, Janitrol (then Division of Andro Corp.), Phillips Engineering, Rovac Corporation, Siemens Corp., Sundstrand Corp., Tectonics Research Inc., Westinghouse Electric Corp., (Pittsburgh, Pa. and Norman, Okla.), York Division of Borg-Warner Co., as well as a number of individual heat pump developers or other sources of information on heat pumps.

Still applicable, too, was some of the information gathered for the FEA heat pump study by the same authors (Gordian Associates, 1976). An extensive literature search was also undertaken for new heat pump concepts and to update information already available.

Whereas unitary electric heat pumps are being manufactured and marketed for residential and commercial applications and their performance parameters are reasonably well established through laboratory and field studies, only one gas heat pump has reached the prototype stage at the present time. The others are in various stages of proof-of-concept and feasibility evaluation, design analysis, component fabrication, testing, performance mapping and have in most cases not operated as a complete heat pump assembly. Data on the capacity and COP of the gas and other experimental heat pumps are still, to a large extent, tentative and subject to verification by measurements on prototype (and, ultimately, production-type equipment). Whenever possible, the authors attempted to verify the exact developmental stage of the system in question and the basis for the performance data made available by the developers. This was all the more important for the systems selected for later simulation studies since the level of effort prescribed by the contract did not permit modelling of each experimental heat pump from first principles, and, therefore, the seasonal on-site energy consumption calculations (to be described in Section 2.3) had to be based on heating and cooling capacity, fuel and electricity consumption or coefficient of performance data provided by the manufacturer or developer.

The amount of information made available to us varied considerably, as already mentioned in Section 1.1. In some cases detailed equipment

and component specifications, drawings and plans, sample experimental or calculated performance points and other information was made available; in others, only capacity and COP data were released. In some cases, equipment or its components were exhibited or demonstrated; in others, not even the system configuration was revealed due to proprietary reasons. Thus the information presented herein had to be pieced together from various data sources and represents our best information as of the date of the investigation.

2.3 Energy Consumption Simulation Study

The purpose of the energy consumption simulation study was to compare on an equal basis the electric and/or fossil fuel energy required by the various heat pumps and other systems under consideration in meeting the same seasonal heating or cooling load in identical selected test buildings. For the residential-size HVAC equipment, the simulation determined the hourly heating and cooling energy requirements of a single-family, two-story frame dwelling having a heated area of 1850 square feet. For the evaluation of commercial-size HVAC equipment, a two-story concrete masonry office building with a heated area of 5580 square feet was utilized. The calculations were performed for the test buildings as if located in nine different cities representative of various geographical and climatological regions of the continental United States.

Two different computer programs were used in the simulation studies, each capable of accepting actual NOAA weather data and data on building and HVAC system characteristics and calculating therefrom hour-by-hour

energy consumptions for the building and system in question. The residential building and systems simulations were carried out by the use of a transient simulation model originally developed by C. Sepsy of Ohio State University under subcontract to Gordian Associates and subsequently substantially improved and modified for the present study. The building thermal load calculation part of that model is a modified version of the U.S. Postal Service load program developed by the General American Transportation Corp. and is a version of the one being developed under EPRI sponsorship for residential building energy use simulation. The simulation model used for the commercial building is the Edison Electric Institute's AXCESS model modified to reflect system part load and dynamic efficiency performance.

Simulation of specific HVAC equipment operating parameters was computerized using manufacturers' performance data or prototype data for advanced heat pumps as supplied by developers. The input energy to each system required to maintain comfort conditions (i.e. satisfy heating and cooling loads) in each building was thus calculated. In broad outline, the computer simulation procedure is similar to that used in an earlier heat pump study for the FEA (Gordian Associates, 1976). However, considerable refinements of the methods used in the earlier study were incorporated in the present work. Although most of the details of the computer methods will be reported in appropriate parts of this report, a general description of some important features is offered below:

- Both the residential and commercial building used represent real existing structures for which detailed architectural information was available. Important characteristics of the buildings, such as the composite construction of

walls and roofs, window areas, and foundation construction, were thus readily available for the accurate estimation of hourly heating and cooling loads by the computerized load programs.

- Both the residential structure load calculation program and AXCESS for the commercial building use accepted ASHRAE load calculation algorithms. However, these have been modified by the latest information currently available. In addition, both load programs and system simulation methods have been validated in previous investigations by comparing computer results with measured data for existing buildings.

A number of important factors, generally neglected in modelling HVAC systems, but having important effects on energy consumption, were included in the present study:

- Supplemental resistance heat required to temper cold air admitted to a residential structure during the defrosting of the outdoor heat exchanger or coil during winter heating was included in the simulation of conventional electrical heat pumps. Heat pump manufacturers' performance data generally include compressor energy used during the defrost period, when the heat pump cycle is reversed and hot condensing refrigerant is used to melt ice on the outdoor coil instead of being delivered to the indoor coil for heating. The heat pump acts as an air conditioner during defrost periods and air circulated through the residence is cooled. The amount of supplemental energy used to partially heat the cooled circulating air to the comfort level is dependent upon too many variables to be included in manufacturers' system performance data, but the effect was modelled for each conventional electric heat pump used in this study.

- At outdoor dry bulb temperatures below about 40°F and at sufficiently high humidities, some residual frost is always present on the outdoor heat transfer coil. An ice layer of sufficient thickness will eventually reduce the heating capacity of a heat pump due to reduced heat transfer rates in the outdoor refrigerant evaporator. The effect of outdoor coil ice film on heat pump operating efficiency was modelled with data reported in the literature.

- The on-off cyclic operation of heating and cooling systems has the effect of reducing operating efficiency below the steady-state performance reported by equipment manufacturers for heating and cooling. Recently published studies

quantifying dynamic efficiency losses for conventional electric heat pumps, natural gas and oil furnaces were used to model cyclic effects. The gas and oil furnace efficiency data also include stack combustion and infiltration losses.

- Current installation practices for HVAC systems generally require insulating basement supply and return air ductwork for electric furnaces and electric heat pumps. No ductwork insulation is generally required for fossil fuel fired furnaces even if installed in combination with an electric air conditioner, except in isolated cases to prevent condensation inside supply ducts. The effect of heat losses for uninsulated ductwork was examined and found to be sufficiently high for the residential building to significantly affect seasonal performance for heating and cooling. A uniform modelling procedure including ductwork insulation for all systems was adopted so that the energy efficiency of all systems could be compared on the same basis.

- Performance data for advanced heat pump systems were generally available from developers for only a single size machine currently under development, usually a 3 ton cooling capacity heat pump for residential service and a 7.5 or 10 ton heat pump for commercial application. Performance data for other size heat pumps required for some cities were scaled from prototype data by a uniform procedure confirmed in discussions with several manufacturers. In only one case did a heat pump developer undertake the task of scaling performance data for a prototype machine to other sizes.

The residential load and system energy use calculation was run for 8760 hours for the test weather year, a year representative of typical weather conditions in each city. To reduce computation expense for the commercial load and system energy simulation, an hour-by-hour calculation was run for every fifth day of each of four heating and cooling months: January, April, July, October. The results were then extrapolated to totals for the entire heating and cooling season. For both residential and commercial buildings, the results consist of monthly, seasonal and annual heating and cooling loads, on-site electric energy and fuels consumed and seasonal performance factors (defined as net heating or cooling loads divided by the sum of all forms of energy consumed by a

system). The results of the analysis are then used to compare on-site energy efficiency of the various HVAC systems examined and are presented in Chapter 8. These results also serve as the basis for determining primary fuel resources consumed by the various HVAC systems as described in Section 2.6.

2.4 Energy and Life Cycle Cost Studies

This section deals with the economic viability of heat pumps, conventional and advanced, in the residential and light commercial building applications. In order to ascertain the potential market acceptance of the heat pump, detailed installed costs and life cycle costs were calculated. Through life cycle cost analyses (with and without inflation) the effects of increased energy efficiency could be compared with the general trend of higher installed costs of a heat pump system.

Energy costs were determined for all systems for which a meaningful hour-by-hour computer simulation of energy consumption could be performed. Nine cities, representing the climatic regions of the United States, were locations chosen for simulation. In this manner, the findings could be used broadly throughout the country.

Cost information was obtained for equipment currently available, either residential or light commercial. Most of the advanced heat pumps are still in early stage of development; hence, equipment costs for these are not generally available at the present time. However, the developers of the organic fluid absorption gas heat pump (which reached the prototype stage at the time of this writing) have made projections of their estimated wholesale price. This, together with costs of labor and materials for installation, was used in the life cycle cost analyses.

Cost data reported herein are in general current as of the First Quarter of 1977 or, in the case of the commercial systems, the Second Quarter of 1977. Since it was our intent to have the cost analyses of conventional heat pumps and other HVAC equipment represent current commercial practice, regional representatives of the manufacturers whose equipment is featured in this study were consulted to verify compatibility of components and equipment sizing procedures. Dealers or the manufacturers themselves were the source of equipment and installation cost data used. The exception was the cost of room air conditioners which was obtained from the Sears and Roebuck Co. Summer 1977 Catalogue. The installed cost obtained consisted of the following major elements:

- Material component prices and installation man-hours required for HVAC equipment;
- Ductwork (insulated and non-insulated), material price and installation man-hours;
- Chimney, where required;
- Oil tank, where required;
- Electrical service change, where required;
- Labor costs for installation;
- Freight costs;
- Applicable state and local sales and/or use taxes;
- Installing contractor's fee.

Labor costs (sheetmetal workers and other HVAC laborers) were taken as representative of the Philadelphia area, since most of the installed costs were obtained from a central New Jersey contractor. For other cities, appropriate wage rates were taken from data published by the Bureau of Labor Statistics for Winter 1976. Regional variations in

first costs thus reflect differences in labor rates, shipping costs and local taxes as well as equipment size differences, where appropriate.

Maintenance cost data were provided by the National Environmental Systems Contractors Association (NESCA) and represent conditions as of Third Quarter of 1976. They represent the average retail price of a full service contract to the consumer. A maintenance fee for the baseboard heaters, as recommended by NESCA, has also been included. The room air conditioner maintenance fee was obtained from Sears. Maintenance costs were assumed not to vary regionally for purposes of these calculations. An estimate of the cost of a full service maintenance contract was obtained from the New Jersey HVAC contractor previously mentioned (Tri-Boro Environmental Systems), and represents prices as of the Second Quarter of 1977.

For purposes of life-cycle calculations, depreciation periods for the various components of each system (e.g. the HVAC equipment, ductwork, chimney, etc.) were taken from the ASHRAE 1975 Systems Handbook and Product Directory, or represent Internal Revenue Service recommended practices.

Current state and local taxes were obtained from the sales tax departments of the individual states in which the facilities were located. A contractor's fee was also applied to determine an installed cost. Where possible, equipment costs and contractor's fees were verified by discussions with other contractors, with manufacturers and by comparisons with data reported in the Means 1977 Building Construction Cost Data Book.

The cost data reported in the study features equipment manufactured by General Electric Co., York Division-Borg Warner Corp., Carrier Corp.,

Lennox Industries, Inc. and Janitol Division-Tappan Company. These companies provided cost information, equipment design specifications, and recommendations concerning installation procedures.

Current rate information, used in determining all energy costs was obtained by contacting the electric and gas utilities servicing each city involved in the simulation study. Oil prices were obtained from usually two private oil companies in the city. The results of the life cycle owning and operating cost analyses appear in Chapter 9.

2.5 Market and Institutional Factors Study

The objective of this segment of the study was to develop an understanding of the space conditioning markets and products, to project the potential market and the expected market share or growth rate for the advanced heat pumps, and to estimate the approximate time schedule of advanced heat pump product introduction. A further goal was to determine the influence of variables such as consumer income level, traditional financing methods, building codes, consumers' and local officials' attitudes as influencing heat pump market penetration, and to identify both accelerating and retarding factors affecting the market prospects for advanced heat pumps.

The approach utilized in the determination of the market prospects for advanced heat pumps was to develop a model of the residential and nonresidential building market for HVAC equipment.

The market models (there are two, because of differences in available input statistics on residential and nonresidential construction) accept input data on new building starts, and calculate projected electric or

gas heat pump installations in the new construction, conversion and replacement sectors of the total market. The conversions and replacements, it should be noted, are calculated as a fraction of the new construction market based on historical data on electric heat pump and gas furnace conversions and replacements. New building construction information is available in the form of number of housing starts or dollar-value put in place for the residential and nonresidential market sectors, respectively. Market-limiting factors such as regional availability of gas as a heating fuel and the demand for air conditioning were accounted for. Estimated penetration rates for the heat pump products in question required by the model were determined on the basis of historical penetration rates (as a function of years from first market introduction) for nearest analogous products. Thus, a high-growth-rate and low-growth-rate projection is calculated. Estimates are also made of the expected regional distribution of the products involved by four major U.S. census regions.

Determination of the institutional factor impact included collection and evaluation of data on lending institutions, energy availability and costs, consumer attitudes on heat pumps and energy conservation, building trends, consumer income and demographic factors.

Market and institutional factor-related data were obtained from a number of sources.

Equipment manufacturers and developers contacted were General Electric Co., Carrier Corp., Consolidated Natural Gas Service Corp., Garrett-Airesearch Corp., and Allied Chemical Co., who provided projections concerning the space conditioning market for conventional and advanced systems as well as data or explanations of their market analysis for

advanced heat pumps.

Electrical utilities contacted were Arkansas Power & Light Co., who provided information on heat pump consumer attitudes and demographics, Mississippi Power & Light Co., who gave information on their heat pump marketing program and on key buying influences, and Columbia Gas System Service Corp., who gave information of the market acceptance of gas air conditioners (significant as to possible impact on the gas heat pump).

Trade association data sources included American Gas Association, Edison Electric Institute, Gas Appliance Manufacturers Association, Air-Conditioning and Refrigeration Institute and National Environmental Systems Contractors Association, who gave information on space conditioning equipment shipments, new, replacement and conversion installations, and equipment maintenance. The National Association of Home Builders, Manufactured Housing Institute and the National Lumber Building Dealers Association provided data and information on residential and nonresidential building new construction.

Additional information on historical data and projections of residential and nonresidential buildings and space conditioning systems was obtained from the U.S. Department of Commerce - Census Bureau, Smith Barney, Harris Upham, Ryan Homes and Lowe's Companies.

Information on natural gas concerning present and future supply, curtailments, prices and allocation policy was obtained from American Gas Association, Federal Power Commission, Bureau of Mines and U.S. Geologic Survey.

Literature and trade publications used as information and data sources were Air Conditioning, Heating and Refrigeration News, Electrical

World, Heating/Piping/Air Conditioning and Professional Builder.

In addition, discussions were held with a number of heating and air conditioning contractors, home mortgage bankers and realtors, code officials and building code organizations, state energy conservation offices, and home builders and home insulation suppliers. Other pertinent contacts are cited in the body of this study. The results of the investigations into the market prospects for advanced heat pumps are presented in Chapter 10. Institutional and other factors are discussed in Chapter 11.

2.6 Primary Energy Resource Impact Study

The objective of this phase in the evaluation of heat pumps and other HVAC systems was the assessment of their impact on primary fuel resources - and particularly scarce fuel resources such as oil and natural gas - due to penetration of these systems into residential and commercial space conditioning markets. In order to assess these consequences, estimates of primary fuel usage were first made in two ways: on the basis of a single house or commercial building in each of the nine test cities and on the basis of fuels and energy consumed per square foot of floor space for the test buildings in each of the nine U.S. census regions. By comparing primary energy resources consumed by the different HVAC systems, specific HVAC systems capable of conserving scarce primary fuels can be identified and an estimate made of the total quantities of energy saved.

Primary energy consumption by space conditioning systems in each of the nine cities was determined from calculated onsite fuels and electric

energy consumption, taking into account such factors as electric generating plant heat rates, and transmission and distribution losses. A single energy value is produced, representing total equivalent primary energy consumption. Conversion of onsite energy consumption to primary energy consumption, by type of onsite fuel or electric energy used, was calculated as follows:

- Electric energy: onsite consumption was increased by 9% to account for transmission losses and then converted to primary energy consumption using average generating plant heat rates in the region in which the city is located;
- Natural gas: onsite consumption was increased by 8% to account for distribution losses;
- Oil: no distribution losses were assumed.

Primary fuels and electric energy consumed on a census region basis were also determined. The method used first correlated onsite energy consumed by each HVAC system examined in each test city with a function combining design load for the building, heating or cooling degree days in each city and the difference between design temperature and comfort temperature. A similar function is recommended by ASHRAE (1976) to estimate seasonal fuel consumption for heating systems. This correlation permits an estimate to be made of energy consumed by each HVAC system for the residential and commercial building in each state, based upon estimates of annual heating and cooling degree days for each state. Weighted average heating and cooling degree days for each state were calculated using the distribution of detached single family residences and commercial buildings within each state as the weighting factor. The estimated energy consumption by the different HVAC systems examined in each state was then combined to obtain weighted average energy consumptions for each of nine census regions, the weighting factor being the total

number of detached single family residences or commercial buildings in each state comprising a census region. Primary fuel and electric energy consumption for each HVAC system in each census region was determined by accounting for generation, transmission and distribution losses by energy type as described above. However, the quantities of electric energy produced at the generating station were converted into primary fuels using published data on average fuel heating values and the seasonal mix of primary fuel resources used by generating plants in each of the nine census regions.

The final result of this process is a list consisting of the quantities of each of three primary fuel resources (pounds of coal, gallons of oil, cubic feet of natural gas) and two primary electric energy resources (kWh of nuclear and hydroelectric energy) consumed by each HVAC system examined in each of the nine census regions. These results are used in conjunction with assumed market penetrations of each HVAC system for the time period 1980-2000 to produce estimated quantities of primary resource consumption.

2.7 Other Studies

In addition to the major phases of the study described in the previous sections, other questions examined were, for example, the role of heat pumps in solar energy building heating and cooling systems, the effect of heat pumps on electric and gas utility load characteristics, etc. Solar energy technology in building heating and cooling is probably one of the most intensively investigated fields at the present time and only an overview of some of the aspects pertinent to heat pumps is provided. On the other hand, comprehensive and sufficiently penetrating

studies of the effect of heat pumps on utility load characteristics have not yet appeared in the literature. Unfortunately, although the tools were available for a more thorough and quantitative investigation of this issue, only a qualitative study fell within the scope of this project.

References to Chapter 2

American Society of Heating, Refrigerating and Air Conditioning Engineers, Handbook and Product Directory (New York, ASHRAE, 1976).

Gordian Associates, Inc. Evaluation of the Air-to-Air Heat Pump for Residential Space Conditioning, (Report prepared for FEA, NTIS No. PB-255-652, April, 1976)

3. ELECTRIC HEAT PUMPS

3.1 Introduction

This chapter is concerned with the approaches to efficiency and reliability improvement of electric heat pumps - the only type of heat pump being manufactured and applied today in residential and light commercial buildings. Emphasis in this chapter is on unitary equipment; i.e., equipment manufactured in one or more complete assemblies designed to be operated as a unit. Applied or "built-up" heat pump systems are not considered in this report. However, the reader should note that because these heat pumps are assembled from components at the building site and because of the size (i.e. refrigeration tonnage) and the large capital investment usually involved in single installations, large built-up heat pump systems have frequently offered more opportunities for innovative engineering and efficiency improvement than unitary equipment. Indeed, many of the technical approaches that have been suggested for improved efficiency of unitary heat pumps are already being practiced in built up heat pump systems at the present time, as a recent study has noted (Crane et al., 1976). On the other hand, unitary equipment offers a more difficult challenge to the industry to improve equipment performance and reliability at a cost which is competitive with market alternatives.

The electric heat pump was commercially introduced in the early 1950's, although reports of construction and installation of experimental units date back to the 1930's. At first conceived as little more than air conditioner with a capability reversing the direction of refrigerant flow in part of the refrigerant loop, the heat pump began its initial

market penetration in the South where post-World-War-II affluence brought about an increasing demand for comfort cooling. Since these areas also have a small-to-moderate winter heating requirement, the ability to offer electric cooling and heating in the same package seemed an ideal way for electric utilities to improve or at least counter the deterioration of their annual load factor and compete with then readily available cheap gas heat.

The rest of the early history of the heat pump and the heat pump industry is well known and needs no repetition here. What is less known is that some manufacturers have made considerable progress in equipment performance and reliability since the early days and, while still considerable variation in product quality exists among manufacturers, efficient and reliable heat pump equipment is now available (Gordian Associates, 1976). Especially notable have been the advances in energy efficiency in the last several years, as this study will document.

3.2 Aspects of Electric Heat Pump Design and Operation

Before discussing the possibility of heat pump performance improvement, it is necessary to review certain fundamental aspects of current heat pump design and operation in order to understand why performance improvement might be desirable in the first place. Since many sources of information on fundamental principles of heat pumps are available (e.g. the ASHRAE Handbook of Fundamentals or the several volumes of the Handbook and Product Directory, this section will dispense with a discussion of ideal thermodynamic cycles and proceed, instead, directly to the question of capacity and efficiency variation with outdoor temperature.

3.2.1 Capacity and Efficiency

It is well known that the heating and cooling capacity and coefficient of performance of the electrically driven Rankine cycle heat pump vary as functions of the outdoor temperature. Exhibit 3-1 shows the temperature dependence of heat pump heating and cooling capacity, normalized by the total rated cooling capacity at 95°F outdoor dry bulb temperature, for air source equipment of three different manufacturers*. It is apparent that the heating capacity of the heat pump decreases with a decrease in the ambient temperature and the cooling capacity decreases with an increase in the ambient temperature. This is just the reverse of the temperature variation of the heating and cooling demand of most buildings. Exhibit 3-2 shows the variation with ambient temperature of the heating and cooling COP. Again, the heat pump is least efficient when the heating or cooling demand is the greatest. A brief summary of the reasons for this behavior is given below:

The thermodynamic cycle parameters of interest in determining the performance of a Rankine cycle heat pump are (a) the evaporator refrigeration effect H_{re}, (b) the work of compression (adiabatic head) W and (c) the condenser heat rejection H_c. The refrigeration effect is the specific enthalpy difference between the compressor suction vapor and the evaporator inlet fluid; the condenser heat rejection (which may include subcooling) is the specific enthalpy difference between compressor discharge vapor and condenser (or subcooler) effluent. Since these quantities are

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* The data represent a General Electric 10 ton single package heat pump, a Carrier 4 ton split system heat pump and a York 2 ton split system heat pump.

defined in terms of a function of state, they may be calculated from measured values of refrigerant temperature and pressure (and quality, if appropriate). As such they will be the same for an ideal or a real cycle operating between the identical state points. The specific work of compression, on the other hand, is path-dependent. If calculated on the basis of an isentropic path, the appropriate compression η_c and mechanical efficiency η_m must be applied; if from a polytropic path, the appropriate index or indices must be known as well.

Since the above quantities are intensive, they must be multiplied by the mass flow w of refrigerant circulating in order to get absolute capacity and energy requirements. The actual mass flow of refrigerant is given by $w = \rho_1 D \eta_v$. where ρ_1 is the mass density at compressor inlet conditions, D is the displacement rate (e.g. in cubic feet per hour) and η_v is the volumetric efficiency defined as the ratio of the actual to the ideal mass flow rate. The displacement rate for a reciprocating compressor is, of course, a function of the cylinder bore and piston stroke, the number of cylinders and the cycle frequency. The volumetric efficiency decreases with increasing fractional clearance volume c and the 1/k power of the internal compression ratio r (k is the ratio of the isobaric and isosteric specific heats of the gas in question.

In terms of the above quantities, then, the heat pump heating capacity Q_h is given approximately by

$$Q_h = wH_c$$

whereas the cooling capacity is

$$Q_c = wH_{re}$$

In proper units, compressor horsepower is

$$W_{act} = wW/(\eta_c \eta_m)$$

to which must be added indoor and outdoor fan energy consumption in order to get the total energy requirements (exclusive of supplemental heat).

As a first approximation, it may be assumed that the condenser heat rejection and evaporator refrigeration effect do not vary significantly with temperature as follows:

- The specific work of compression (adiabatic head) increases with compression ratio (raised to a fractional power) and inlet gas temperature. Total power requirements increase of course, with the mass flow rate of fluid being compresses;
- Fluid friction losses (in terms of power expended) increase with gas density and the cube of the flow rate;
- Mechanical losses (e.g. piston and bearing friction) increase with compression ratio.

In the heating mode, a decrease in compressor power requirements is observed as the ambient temperature drops. This is the result of decreases fluid and mechanical friction due to a lower fluid mass flow rate as well as a decreased volumetric efficiency. In the cooling mode, the same factors operate, although the temperature dependence is less severe.

Valve, valve porting, internal piping and muffling pressure drops are another consideration. The compressor "pumping" energy to overcome

internal fluid friction losses varies as the product of the mass flow of refrigerant fluid and the pressure drop (the latter increasing as the square of fluid velocity - or flow rate, if the flow area remains constant).

In the above discussion, clearly the dependence on ambient temperature of motor and other losses and thus the capacity and power requirements have been neglected. The winding temperature of a hermetic motor cooled by suction gas (as heat pump motors are) must clearly increase as the ambient temperature drops since less mass flow of refrigerant is available to cool the motor. A hotter motor means hotter suction gas entering the compression cylinder and an increase in the specific work of compression.

The reason for the decrease of heat pump heating capacity and COP with ambient temperature may be explained as follows: the lower the ambient temperature of the air entering the outdoor coil (evaporator) (Exhibit 3-3), the lower the evaporator temperature has to be to maintain a sufficient temperature difference for heat transfer to the vaporizing fluid. As the evaporator temperature drops, the evaporator equilibrium pressure decreases but the compression ratio increases. The effect on the refrigerant mass flow is thus twofold: (a) the inlet gas density decreases due to a lower pressure (at roughly constant superheat) and (b) the volumetric efficiency decreases due to a higher compression ratio. The result is a lower mass flow of refrigerant and an increase in the specific work of compression.

In the cooling mode (Exhibit 3-4), cooling capacity and COP decrease as the outdoor temperature rises. The outdoor coil, now the condenser, must be maintained at a temperature 10-15 degrees F higher than the

entering ambient air for adequate heat transfer. As the condensing temperature rises with higher ambient temperature, the compression ratio increases increasing the work of compression and decreasing the volumetric efficiency. The attendant decrease in cooling capacity and COP is compensated somewhat by the higher suction gas density.

The latent heat removal or dehumification capacity of the heat pump also varies with temperature. When the evaporator (or indoor coil) temperature is low (typically, say, 45-50°F), there exists a sufficient vapor pressure difference between the warmer, more humid entering air and the cooler stagnant vapor film over the condensed liquid water at the coil surface to cause rapid mass transfer, the heat of water vapor condensation being absorbed by the evaporating refrigerant along with the sensible heat of the air being cooled. With increasing outdoor coil temperature and increasing evaporator temperature indoors, the bulk air to surface film water vapor pressure differential at the indoor coil decreases and the dehumidification rate drops. That is, the latent heat removal or cooling capacity of the heat pump decreases with increasing outdoor temperature.* When the coil temperature approaches entering air wet bulb temperature, moisture removal capability ceases. This fact is of concern in assuring cooling comfort and may impose an upper limit on the indoor heat exchanger size (and thus a limit on one means of efficiency and capacity improvement).

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* On the other hand, at given outdoor entering air temperature, the total capacity is higher the higher the indoor wet bulb temperature due to increased condensation rather than sensible heat transfer. See the total and sensible cooling capacity curves in Exhibit 3-1.

3.2.2 Controls

Unitary heat pumps employ a number of controls for operational mode activation and termination, motor-compressor protection, refrigerant flow regulation, defrost initiation and termination, indoor and outdoor temperature sensing and, in recent models, user's and servicer's diagnostics. Developments in solid-state sensing and switching devices have substantially increased the speed of response and reliability of motor protection devices. Microprocessor technology, too, appears to be gaining a place in small HVAC system control schemes, enabling the monitoring, analysis and regulation of a larger number of parameters than before. For example, the York CHAMPION heat pump (see York, 1975) uses a solid state logic module which monitors the following:

- Air temperature entering outdoor coil (thermistor);
- Compressor discharge line temperature (thermistor);
- Liquid line temperature (thermistor);
- Pressure differential across outdoor coil (differential pressure switch).

Other features provided includes an adjustable balance point control, with settings in the 23-43°F range, which prevents supplemental resistant heat operation above the balance point (in lieu of a field-installed outdoor thermostat). Plug-in contacts also allow interconnection with a servicer's diagnostic device to facilitate corrective maintenance.

In addition to motor-protection and other devices having an obvious effect on equipment reliability, several control elements (valves) influence heat pump capacity. Therefore, some characteristics of capillaries, thermostatic expansion valves and subooling valves will be discussed below.

A capillary tube (a small diameter coiled tube in the liquid line leading to the coil serving as an evaporator) produces a pressure (and corresponding temperature) drop in the entering fluid. Because the capillary is basically a device which presents a fixed constriction to refrigerant flow, it is unable to significantly adjust mass flow to changing upstream or downstream pressure conditions in a manner desirable. According to Copeland Corp. (1975), if a capillary on the outdoor unit is sized to give 10-15 degrees F superheat at moderate evaporator temperature, it will allow just enough liquid to pass at low evaporator temperature to improve compressor cooling and oil return.*

A thermostatic expansion valve, on the other hand, varies the flow constriction (pressure drop) to maintain a constant amount of superheat in the evaporator effluent vapor. While thermostatic expansion valves were in many cases earlier available only in the larger split-system heat pump sizes, several manufacturers now use thermostatic expansion valves for both the outdoor and indoor coil or for the outdoor coil only in the smaller residential size equipment as well. Thermostatic expansion valves are thought to provide better evaporator flow control, but at low mass flows and low evaporator temperatures unduly restrict refrigerant flow if set for constant superheat. The result is that the excess liquid refrigerant not admitted into the evaporator is stored in the condenser restricting the coil area available for condensation and increasing condenser pressure.

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* Note that the bulletin in question advocates use of a slight amount of liquid flooding at low temperature to insure adequate cooling of the motor-compressor.

Thermostatic expansion valves use a metal diaphragm and a sensing fluid charge, which may be a source of failure.

An additional refrigerant control sometimes incorporated in the heat pump design is a pressure activated subcooling control valve. A pressurized bulb senses the condenser effluent temperature from its set point of 10-15°F from saturation temperature, hence the extent of liquid subcooling.

3.2.3 Coil Freezing and Defrost

In the heating mode, the heat pump evaporator at certain conditions of ambient dry and wet bulb temperature experiences coil freezing. Data from General Electric, quoted by Wildin *et al.* (1977), show that the highest number of defrost cycles seem to occur at air temperatures between 32° and 40°F. Coil freezing seems not to occur, on the other hand, if the relative humidity is below about 60%. Coil freezing can be especially severe if sleet or rain blows onto the evaporator coil. However, little is known quantitatively about conditions causing frost formation, rates of frost layer and density increase and its effect on heat transfer rates, although there has been some research on the subject (Gates *et al.*, 1967; Huffman and Sepsy, 1967; Parish and Sepsy, 1972).

While a moderate* amount of frost on the evaporator may, indeed, enhance heat transfer through increased roughness-induced turbulence and increased surface area, too much frost or ice can block or at least

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* How much freezing can be tolerated is a matter of contention. One controls manufacturer claims that up to about 75% coil air flow blockage can be permitted as due to low mass flow rates on heating there is excess coil area anyway (West, 1976).

significantly reduce the air flow over the coil, allowing suction gas temperatures to rise. At this point in the operation, a defrost cycle has to be initiated to remove the ice.

The most common method of defrosting the outdoor coil is by cycle reversal; that is, the reversing valve is actuated and the heat pump is switched from a heating to a cooling cycle, while resistance heating elements are energized to temper the cold air being input to the conditioned space. Defrosting is accomplished by the high temperature vapor from the discharge side of the compressor entering and condensing in the iced-over outdoor coil and requires on the average 3-5 minutes (but sometimes longer) depending on the method of defrost control, heat pump design and other factors.

Another method of defrost is also possible (and is actually employed on the recently introduced Janitrol heat-only heat pump): injection of hot discharge gas directly into the evaporator, bypassing the condenser (Janitrol, 1976). This method has the advantage of not requiring cycle reversal with the attendant reversal of high and low-pressure coils of the system and the sudden surging of liquid into the accumulator and and compressor. On the other hand, due to reduced heat transfer, longer defrost cycles may result with a corresponding loss of efficiency.

3.2.4 Transient Operation

Although, in the past, theoretical performance analyses of heat pump operation were usually carried out under the assumption of steady state operation, actually much of the operation of the heat pump occurs in the transient mode. This may be attributable to several factors, most important among them being:

- On-off cycling under conditions where heating or cooling capacity exceeds demand;
- Cycle-reversal in the course of coil defrosting;
- Cycling induced by the thermostat dead-band (i.e. control limits), inadequate heat delivery due to dirty air filters.etc.

Recent experiments conducted by the National Bureau of Standards (Kelly and Bean, 1976; Parken et al., 1977) and elsewhere show that the manufacturer's rated steady-state COP is achieved in actual operation only when the heating or cooling load is high and the equipment is operating near its maximum capacity. At low loads, the necessity for overcoming thermal and mechanical inertia in the system apparently consumes some of the energy otherwise available for heating or cooling. Indeed, according to laboratory measurements by Parken and others, as much as a 30% loss in actual over steady state heating or cooling COP occurred at loads of 15-20% of the rated heating or cooling capacity. At loads of the order of 80%, the difference was only 4-5%. A source of efficiency loss on heating is, of course, coil freezing and periodic defrost.

While Parken's measurements were obtained on a single 3-ton heat pump and Kelly's on a 5 ton heat pump and thus may not accurately reflect the performance of equipment of other manufacturers, they, nevertheless point out an important source of performance loss not widely recognized, since analyses of heat pump instantaneous performance under part-load conditions (e.g. at temperatures other than at which rated capacity is achieved) assume that steady-state conditions are attained as far as the equipment and its immediate surroundings are concerned.

3.3 Approaches to Efficiency Improvement

It is apparent (cf. Section 3.2) that, while overall increases in efficiency are not to be scoffed at, the improvement of low temperature capacity as well as the COP of the heat pump is a must for more widespread applications in colder regions.

The sources of thermodynamic inefficiency in a Rankine cycle heat pump refrigeration loop have been well documented (see Spofford, 1959; Dubberley, 1961; Ambrose, 1974) as have the general technical approaches to overcoming them. In broad outline, they are as follows:

- Increase efficiency of refrigeration loop components, decrease heat leaks and fluid pressure losses (e.g. improve compressor and fan motor efficiency, increase heat exchanger effectiveness, etc.);
- Improve thermodynamic "quality" of heat sources and (e.g. evaporator-side thermal storage, air conditioner condenser evaporative pre-cooling, etc.);
- Improve capacity modulation of heat pump to better conform to building heating and cooling load.

Specific component or other improvements proposed fall into one or more of the above categories.

In most but not all cases, the technology for accomplishing the above improvements is either available or could be readily developed provided a perception of sufficient market demand existed to support the research and development required and that there was reasonable confidence that the presumably higher first cost of the more efficient equipment would not be a deterrent to purchase. In a limited number of cases, improvements must await the development of some critical device or system (e.g. a variable speed mechanical gear system for a hermetic reciprocating compressor).

In recent months, a rather comprehensive analytical study of heat pump efficiency and reliability improvement was completed by Westinghouse (a major heat pump manufacturer) under EPRI sponsorship (see Kirschbaum and Veyo, 1977). The study goes into considerable detail on the possible "hardware" changes required for an optimal (in a life cycle cost sense) electric heat pump for Albany, New York - Boston, Mass. -Denver, Colo. climate (although some possible improvements are not described due to "patent interest"). Given the detail of the study, there would seem to be little purpose in our providing another "laundry list" of improvements.* Instead, the sections following will simply summarize the available technical literature on this subject and comment on aspects of special interest.

3.3.1 Compressor Modifications

Hermetic compressor development appears to be directed toward (a) improved reliability, (b) improved COP at all load conditions and (c) improved low-temperature heating capacity. The paragraphs below will attempt to summarize the various concepts that have been advanced.

Hermetic compressor losses are either mechanical or electrical in nature. According to Massa and Ralph (1974) (see also ASHRAE, 1975), major compressor losses are:

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* A less detailed survey of possible heat pump advances was conducted by Crane *et al.* (1976). However, most of the means for electric heat pump efficiency improvement described appear to have been taken from a paper by Comly *et al.* (1975) and the latter is recommended for details.

- Valve, valve port, suction and discharge tube and muffler pressure drops;
- Internal leakage of discharge gas to suction side of compressor;
- Friction, hermetic motor and compression cylinder conduction heat losses to suction gas;
- Valve inefficiencies (mechanical);
- Oil circulation work loss;
- Clearance volume (compressed gas re-expansion loss);
- Anisentropic compression.

Mechanical friction losses and other inefficiencies can be reduced by valve and valve port redesign, reduction of bearing forces on contact surfaces, etc. It is doubtful that much room for improvement exists in the above areas, given a competent design to begin with. In addition, better matching of motor torque-efficiency characteristics to loads encountered especially in colder-climate heating operation, use of heat shields between the compressor high and low pressure sides, use of ringed instead of plug-type pistons is already being implemented or considered (Massa and Ralph, 1974; Kirschbaum and Veyo, 1977). Minor decreases in clearance volume may possibly be achieved by cylinder and piston redesign. However, it is doubtful that much can be done to improve isentropic efficiency (i.e. a closer approach to a reversible adiabatic process), given the basic reciprocating compressor configuration and materials of construction, since strength, durability, ease of fabrication and cost are also important considerations.

Compressor efficiency may also be increased by elimination of suction gas cooling of the hermetic motor and compressor. Riffe (1977) of Westinghouse has proposed that desuperheated and partially condensed

refrigerant be piped into a heat exchanger coil in the compressor oil sump. Heat absorbed from the oil would be transported to the condenser coil and supplied to the indoor air in the heating mode or rejected to outdoor air in the cooling mode. Since motor cooling is now provided by the oil, the suction gas can be introduced directly into the suction muffler and cylinder head, rather than being directed over the motor windings to provide motor cooling. This allows operation with cooler (or less superheated) suction gas with an attendant decrease in compression ratio and increased volumetric efficiency. A possible problem in implementing the oil cooler concept is that in a heat pump the condenser or "high side" coil is alternately the outdoor or indoor heat exchanger, depending on whether the heat pump is in the cooling or heating mode, respectively. This would obviously require a more complex plumbing and valving system. Nevertheless, the idea may have merit in single-package heat pumps. Application to split system heat pumps (especially with the compressor in a separate package) may not be feasible.

Hermetic Motor Efficiency Improvement

Improved hermetic motor efficiency can also improve system COP and capacity by reducing heat leakage to suction gas and the consequent loss of volumetric efficiency. Motor maximum efficiencies are usually 85% or less for single-phase and 90% or less for three-phase motors normally used in heat pump compressors. Motor efficiency can be improved by increasing the "stack" height, providing more capacitance, using a better (higher magnetic permeability) grade of steel in the stator and

by a number of other modifications (Johnson, 1974). Kirschbaum and Veyo (1977a) discuss the efficiency-torque characteristics of single and 3-phase hermetic motors. The authors imply that it might be possible to get a closer match between the torque at which motor maximum efficiency occurs and the load conditions representing most of the heating cycle operating hours of the heat pump. However, it appears that little opportunity exists for substantial electric motor efficiency improvement without undue additional first costs. If, in addition, cost considerations force manufacturers to substitute aluminum wire for copper in stator windings, lower motor efficiency would result due to the lower electrical conductivity of the former. Three phase motors have higher efficiency than single phase motors; however, their use in equipment for residential applications is impractical for obvious reasons.

One possible method for motor efficiency improvement is use of ceramic-metallic composite core materials of higher magnetic permeability than the iron in use today. However, discussions with one compressor manufacturer indicate that this technology is at least five years from realization.

3.3.2 Capacity Modulation

As was described in Section 3.3.1, only a limited degree of further efficiency improvement appears to be possible with the basic constant-speed hermetic reciprocating compressor design. However, even if these improvements were realized there remains the basic problem of the mismatch between the variation with ambient temperature of typical building heat losses (or gains) and the heating (or cooling) capacity of the vapor

compression cycle heat pump. This mismatch is the cause of basically two types of efficiency losses: the loss due to the necessity for supplementing the compressor at low outdoor temperatures with resistance heat at a COP of 1 and (b) the loss due to equipment cycling* under part-load operation. The former has been recognized for a long time in accounting for energy consumption of buildings heated with heat pumps. Evidence of the latter has only recently been published as a result of measurements being carried out at the National Bureau of Standards (Kelly and Bean, 1976; Parken et al., 1977) and is not ordinarily factored into energy estimation calculations. Hence, one method of heat pump efficiency improvement is capacity modulation to conform with actual heating and cooling load variation with ambient temperature.

Two-Speed Compressor

One method of compressor capacity modulation is by speed-halving, i.e. by switching from a two-pole to a four-pole motor configuration. Lennox Industries in 1973 introduced into their larger-tonnage unitary air-conditioner line (Lennox News, 1973) a two-speed hermetic compressor which generates 15 tons of refrigeration at a synchronous speed of 3600 rpm and 7.5 tons at a synchronous speed of 1800 rpm.** Claimed advantages for this method of capacity control include reduction in mechanical friction and refrigerant gas pressure drop losses, producing more than a 50% decrease in power requirements as capacity is halved (Cawley and

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* Not all of the transient character of the capacity can be attributed to the compressor, of course. Heat exchangers, fans, thermostats all have their transients as well. Another cause of efficiency loss is the effect of frost on heat transfer.

** Rated at 45°F evaporator, 125°F condenser, 65°F return gas and zero subcooling temperature.

Pfarrer, 1974). An additional benefit appears to be increased life of refrigeration oil due to lower average discharge temperatures in two-speed mode operation.

The two-speed compressor in question utilizes a 3-phase motor. Recently, the company developed a single-phase two-speed hermetic motor as well and has introduced a 3½ ton two-speed air conditioner utilizing this technology (Lennox, 1977).

While no manufacturer appears to have as yet introduced a two-speed compressor in a heat pump line*, contacts with several indicate that prototype two-speed heat pumps are now being field tested.

Other methods of capacity modulation have also been employed in commercial-size unitary air conditioners. These include cylinder unloading by suction valve lifting mechanisms, discharge gas bypass to the evaporator and compressor discharge-to-suction side pressure equalization. The second method, it should be noted, bypasses the condenser and produces no power reduction at all (Cheney, 1974). While these methods may reduce the compressor pressure ratio and thus the work of compression, only speed-variation of the methods discussed can reduce mechanical and fluid friction. Thus, the two-speed compressor is reported to improve compressor rated EER by 3.3% whereas a cylinder unloaded compressor sustained a 32% EER loss in going from full to half-capacity operation (Lennox News, 1973).

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* As the final draft of this report was being completed Lennox Industries announced the introduction of a two-speed single-phase heat pump for residential applications.

The Rotary Compressor

Although most of the unitary heat pump manufacturers utilize welded hermetic reciprocating motor compressors in their heat pump equipment, Fedders has introduced a split system product with a welded hermetic rotary motor compressor (Boxer, 1976). Advantages claimed for the rotary as opposed to the reciprocating positive displacement compressor are (a) simplicity of construction (fewer moving parts, more rugged rotor, single discharge valve, etc.), (b) better ability to withstand liquid slugging and (c) denser suction gas as suction gas motor cooling is not used. The greater amount of refrigeration oil claimed as a benefit as far as compressor protection is concerned can serve to reduce refrigeration capacity and efficiency elsewhere in the system by decreasing evaporation rates and inhibiting heat transfer. Other disadvantages of the rotary compressor are (a) the greater possibility of leakage of high pressure gas to the low pressure side past rotor and vane clearances, (b) greater heat losses to ambient air due to a "hot shell" design (in the case in question) and (c) the possibility of clearance volume increases (with attendant volumetric efficiency loss) due to vane and system wear. According to Kirschbaum and Veyo (1977c), the rotary compressor is being considered by other heat pump manufacturers for possible adoption in the heat pump.

3.3.3 Heat Exchanger Effectiveness

Since thermal effectiveness is a measure of the thermodynamic availability loss in a heat exchanger, improved heat exchange is clearly one means for improving overall system performance. Heat exchanger effectiveness for an evaporator or a condenser is an exponential function of the number of heat transfer units UA, where U is a suitably averaged overall air-side based heat transfer coefficient and A is the corresponding heat transfer area; that is $\varepsilon = \varepsilon$ (UA; geometry), cf. Kays and London (1964). Hence, increasing the overall heat transfer coefficient (e.g. by increased air flow - since the air-side heat transfer resistance is liable to be controlling in most cases of practical interest) or the coil area will produce the desired result.

Taking up first the question of increasing the coil area, it should be noted that this, indeed, is what many manufacturers have been doing in the last several years. According to West (1976), outdoor coil areas of some manufacturers' equipment have nearly doubled since 1973. Comparing, for example, General Electric Co.'s new high-efficiency (1977) WEATHERTRON "Executive" heat pump with its earlier (1976) 2-ton heat pump, nearly 60% more coil face area per Btu's of nominal (cooling) capacity is provided with the newer model, as shown below:

Trends in Electric Heat Pump Heat Exchanger Area as Contributor to Higher COP*

	1976-Model	1977-Model
Nominal cooling capacity, Btuh	24,000	26,000
Outdoor coil face area, ft^2	8.5	14.5
Face area/capacity ratio, $(ft^2/Btuh) \times 10^4$	3.5	5.6
Heating COP		
17°F	1.6	1.9
47°F	2.5	2.7
Cooling EER, 95°F	7.0	8.1

Increasing the coil area obviously increases heat exchanger cost. Besides cost, however, there are other limitations to increasing the size of the heat exchangers. Pietsch (1974) has discussed the problem of decreased dehumidification capacity in the cooling mode resulting from a larger indoor coil and an attendant higher coil temperature.

Increasing the outdoor coil (the condenser, in the cooling mode) requires an increased amount of refrigerant charge. According to Pietsch, empirical data on system reliability show loss of reliability beyond a certain charge quantity (due in part to increased probability of liquid slugging). Hence, this approach has its restrictions.

Increased airflow (i.e. coil face velocity) for both the indoor and outdoor coil is also a means of improving heat exchanger effectiveness through its effect on U, the overall heat transfer coefficient. Since the refrigerant side condensing or boiling heat transfer coefficient is much greater than on the air side*, the air side film coefficient is controlling, i.e. $U \sim h_{air}$, the latter varying approximately as the eight-tenths power of the face velocity, $h_{air} \sim u^{0.8}$. Fan power, however,

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* Not all of the COP improvement shown is, of course, attributable solely to increased coil area.

increases as the cube of the air flow and increased noise is also a consideration. If the coil in question is the indoor coil, chill drafts and condensate blow-off in the cooling mode are a possible problem as well as the fact that most heat pump indoor coils already approach the ARI maximum of 37.5 cfm per 1000 Btuh, according to Pietsch.

3.3.4 Fan Efficiency and Power Requirements

Another major source of energy loss* is outdoor and indoor fan power consumption; hence decreasing fan power requirements is bound to improve the COP. Fan "air watts" are a product of heat exchanger pressure drop and volumetric air flow rate and thus vary as the cube of air velocity. Fan power is the product of the "air-watts" times the reciprocal of the motor efficiency. In Section 3.3.3 it was noted that the heat transfer coefficient increases (which is one way of increasing coil effectiveness) must come at the expense of increases air velocity, which means increasing the air side pressure drop and the fan power requirements. On the other hand, it may be possible to maintain the normal capacity with a slower larger-diameter fan at considerable savings in power, as is illustrated in the following comparison of the General Electric 1977 high-efficiency or "Executive" heat pump to its 1976 model in Exhibit 3-5.

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* While obviously necessary to provide air movement in a practical system, fan power represents another source of loss of thermodynamic availability in implementing an ideal Rankine cycle heat pump.

3.4 Other Electric Air-Source Heat Pump Concepts

The previous sections have concentrated on conventional (i.e. resistance heat supplemented heating and cooling mode) heat pump design characteristics and efficiency improvement. This section focuses on some interesting variations, dealing with hybrid heat pumps (i.e. those using combustion sources as supplemental heat), heat-only heat pumps and other types.

3.4.1 Heat Pumps with Combustion Supplemental Heat

Heat pumps using a combustion furnace for supplemental heat (termed "hybrid" heat pumps in this report), introduced a number of years ago, seem to be attracting increased market interest for principally two reasons:

- Hybrid heat pumps are applicable to the refit market as add-ons to existing gas or oil furnaces*;
- Hybrid heat pumps have a better annual load factor (ratio of average power demand to peak demand) than resistance heat supplemented heat pumps.**

With the exception of certain control components designed to regulate compressor and furnace operation, essentially standard split-system heat pump components are utilized: a heat pump outdoor unit with fan and heat exchanger, the compressor (a separate unit in some models) and the conditioning or indoor coil for the air handler. The system is operated in the heat pump mode above the balance point, the furnace being switched

* Hybrid heat pumps are designed to be added on to electric as well as combustion furnaces. However, only the latter applications are of interest here.

* While a combustion-supplemented or hybrid heat pump clearly alleviates the low electric load factor problem of a resistance-heat-supplemented heat pump, it decreases the load factor of the gas furnace (if that is the fuel used) by reducing overall gas consumption without affecting the peak demand. Nevertheless, in a period of diminishing gas (and oil) supplies, allowing the gas utility or fuel oil distributor to "stretch" his full supply among his existing customers (and thus stay in business) may be a potential benefit.

on when supplementary heat or air-tempering is required during the heat pump defrosting. Present generation models switch the compressor off completely below the balance point (Lennox, 1976; Carrier, 1976). However, improved models being developed would permit alternating heat pump compressor and furnace operation (to a low temperature - say $0^{o}F$).

Thus, technology is obviously of especial interest in colder climates, where the traditional means of heating existing buildings is largely natural gas, LPG or fuel oil, but a requirement for comfort air-conditioning (say, as an add-on) arises. Even in the coldest climates, there are a sufficient number of heating days above the balance point of a conventionally-sized heat pump to make this system worthy of consideration.

3.4.2 Heat-Only Heat Pump

Another development of interest is the heat-only heat pump. In Europe, the heat-only heat pump, as an alternative to electric resistance heat, is basically the only type of heat pump under consideration as air-conditioning is not a widespread requirement there (Philips Research, 1977; Fikri, 1975). In the United States, Janitrol (now a division of Tappan Co.) has introduced a single size (37,000 Btuh nominal cooling capacity) heat-only heat pump designed as mainly an add-on to a resistance or combustion furnace
(Janitrol, 1976).

Features of interest in the Janitrol heat-only heat pump, the WATTSAVER, are use of compress or discharge gas for evaporator defrost (instead of hot condensing liquid) and the fact that having to design for heating mode operation only enables the engineer to use certain

appraoches to achieve high heating COP but which might be detrimental to cooling performance in a conventional heat pump. Indeed, the heat only heat pump, compared to conventional ones, shows superior low and high temperature heating performance, as may be seen from the table below:

Comparison of Heat-Only Heat pump with Conventional Heat Pumps

	Nominal Heating Capacity at 47°F Btuh	COP 17°F	COP 47°F
Heat-only	37,000	2.3	3.2
Conventional A	37,500	2.1	2.8
Conventional B	39,000	2.0	2.8
Conventional C	37,500	2.1	3.1

Sources: Carrier Corp., Janitrol, General Electric and York technical bulletins.

One possible difficulty, indeed the major one, to a heat-only heat pump approach is that the first cost, which for a conventional heat pump can be paid off through energy savings over the entire year, must be recovered over heating season operation only. Hence, the cost-effectiveness of this system in comparison to alternatives is open to question, unless subsequent designs can effect substantial cost reductions. However, this question is outside the scope of this chapter.

3.4.3 An Advanced Concept Air-Source Heat Pump

An innovative heat pump design has been developed by Mechanical Technology Inc. Originally destined for mobile home applications, the concept* is now being developed for application to small (i.e. of the 1200 ft^2 heated area) "energy-efficient" site-built residential housing as well (MTI, 1976).

* Because of secrecy agreement with the developers, the authors cannot provide a description of this project herein.

The MTI heat pump represents quite a radical departure from conventional electric heat pump designs. Features of interest are unfortunately proprietary and cannot be described further here, except that unconventional heat exchange and compression concepts are used.

Possible disadvantages of this approach are (a) that the compressor is of the open type (i.e. refrigerant loop is not hermetic) and a dynamic shaft seal is required, (b) the design involves high bearing stresses and starting torque and (c) interior thermal leakage from the cold to the hot side via the metal shell. Nevertheless, further development of this and successive generations of this machine should prove interesting.

3.5 Water and Dual Source Heat Pumps

Although most of the unitary equipment manufactured and installed in this country is of the air-to-air type, water source (e.g. water-to-air) heat pumps are growing in market importance because of their adaptability to water-based solar heating systems, to multizone or multi-unit applications where energy recovery is a possibility and because of certain other inherent technically attractive features*.

First, water source equipment offers higher COP's (and a smaller variation of COP with source temperature), at least in principle, since the source temperature is maintained at 32°F of higher**. The source of the heated water may be lake, river or well water (sources with obvious limitations in terms of availability, equipment cost, etc.), a solar or off-peak thermal storage system or, usually in commercial or

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* Water-to-water heat pumps are common in applied or built-up heat pump installations where internal heat sources or heat or cold reclaim is possible.

** If this has to be accomplished by use of resistance heating of the water, the overall COP will be no greater than one.

multi-unit residential installations, a common water loop for heat and cold reclaiming.

A second desirable feature has already been mentioned: compatibility with a water tank thermal storage system. This allows a water-source heat pump* to be used in solar heating or cooling systems or otherwise utilize thermal storage (say, for load management). Water source heat pump evaporator coils do not experience frost problems (unless ice-making is a design feature of the system), a factor further contributing to their high seasonal efficiency.

In solar heating and cooling system application, water-source heat pumps suffer from a limitation in that condenser heat on the cooling cycle must be rejected to water or in the absence of a water-based heat sink, to air via an air-cooled heat exchanger**. This latter requirement may involve additional fan or pumping power diminishing the COP.

A dual (water and air) source heat pump circumvents these disadvantages in that the heating cycle evaporator heat exchanger can be immersed in water or air. This enables the water loop to be used during at least that part of the heating cycle when the water temperature exceeds that of the ambient air and the air coil to be used in warm heating weather or for cooling operation. Several major manufacturers are believed to be developing dual-source heat pumps; however, no major manufacturer has this equipment on the market at the present time.

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* An example of water-source heat pump application is the Annual Cycle Energy System (ACES), now being demonstrated at the University of Tennessee (see Fischer, 1977). However, the cost-effectiveness of this type of system for residential buildings has yet to be demonstrated.

** In some cases, nocturnal heat rejection by evaporation or thermal radiation to the night sky is a possibility as well. Again however, auxiliary power consumption is usually involved.

3.6 Brayton Heat Pump

The electric heat pumps previously described utilize a fluorocarbon refrigerant-based Rankine cycle as the refrigeration system. However, other refrigeration cycles obviously also exist and are suitable for heat pumping, at least in theory. Some of them have been used in aircraft air conditioning. The Brayton refrigeration cycle, for example, uses a permanent gas (e.g. air) which in the ideal cycle is compressed and expanded isentropically, the heat exchange processes being isobaric (e.g. Wurm, 1974). The Wurm report has analyzed Brayton cycle machines in great detail from both the theoretical and practically achievable performance standpoints. In this section, a rather novel air-cycle machine is described which is under development by Rovac Corporation for automobile air-conditioning applications, but is also under consideration for residential heat pumping.*

The Rovac heat pump consists of an elliptic-cylinder-shaped stator housing, in which rotates a circular rotor. The rotor has a series of radial vanes which are controlled to maintain near contact (to ± 0.001 inch) with the stator wall as the rotor rotates. In the closed system design version, a cooler and heater heat exchanger is employed, as well as a regenerator (Rovac, 1976). The stator-rotor system, called the CIRCULATOR, forms a compressor-expander unit which, togther with the heat exchangers, constitute the closed refrigeration loop. The rotor is shaft-driven by an external power source - an automobile engine in the automobile air conditioning version.

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* Laboratory versions of the Rovac heat pump are driven by an electric motor through an external shaft. For this reason, the system is considered to be an electric heat pump. The original version of the Rovac system was designed as an automobile air-conditioner to be driven by a power take-off from the automobile engine.

A novel feature of the Rovac system is the injection of water in the compression and expansion sections of the CIRCULATOR for added efficiency. Evidently the water vaporizes during the compression stage, keeping its temperature rise down; on entering the expander it condenses (and even freezes). Advantages claimed for this system (other than that fluorocarbons are not used) are high COP's (2.6 at 17°F; 3.4 at 47°F; and EER 11.9) for the electrically driven residential unit and ease of capacity modulation by pressure ratio change.

Possible disadvantages of this approach are (a) the non-hermetic refrigerant loop, (b) the necessity for more heat exchanger surface due to the sensible heat transfer processes of the air cycle refrigeration machine, and (c) the problem of heat leakage, seals and wear of the rotary compressor-expander. The above heat pump COP's are projected; for the automotive air conditioner now being tested the EER was lower.

3.7 Summary

The post-Arab-oil-embargo years have seen rapid development in unitary electric heat pumps. However, a number of exiting possibilities for further development still lie ahead as market demand for electric heat pumps expands and heat pump research and development intensifies under competitive and, possibly, government pressures. Most of this development will no doubt be undertaken by the industry as part of normal product evolution and improvement.

Increased sensitivity of the manufacturers to the problem of low load factor that the resistance heat supplemented electric heat pump presents to the utility (and the projection that increased numbers of electric utility systems in this country will be winter-peaking by 1980 due to growth of electric heating load) appears to be encouraging the search for other means of supplementing the capacity of the Rankine cycle heat pump at low outdoor temperature. A number of manufacturers have developed and introduced heat pumps that may be added on to gas or oil warm air furnaces - the latter then supplying the supplementary heat during peak heating hours. Another possible solution, now being intensively researched, is thermal storage as a means of load management. It is believed that the emerging demand charges for residential electric service may spur this development, although in some cases the goals of demand reduction and energy usage reduction may not be entirely compatible.

Thermal heat or cold storage systems (unless in the conditioned space) would be subject to heat leaks to or from the surroundings and may decrease the efficiency of a heat pump at the expense of improving its load factor.

If the storage is replenished solely via the heat pump cycle, as opposed to reliance on resistance heat, this efficiency decrease may not in fact occur.

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EXHIBIT 3-1

NORMALIZED HEATING AND COOLING CAPACITY OF TYPICAL HEAT PUMPS
(Based on Total Cooling Capacity at 95°F Outdoors)

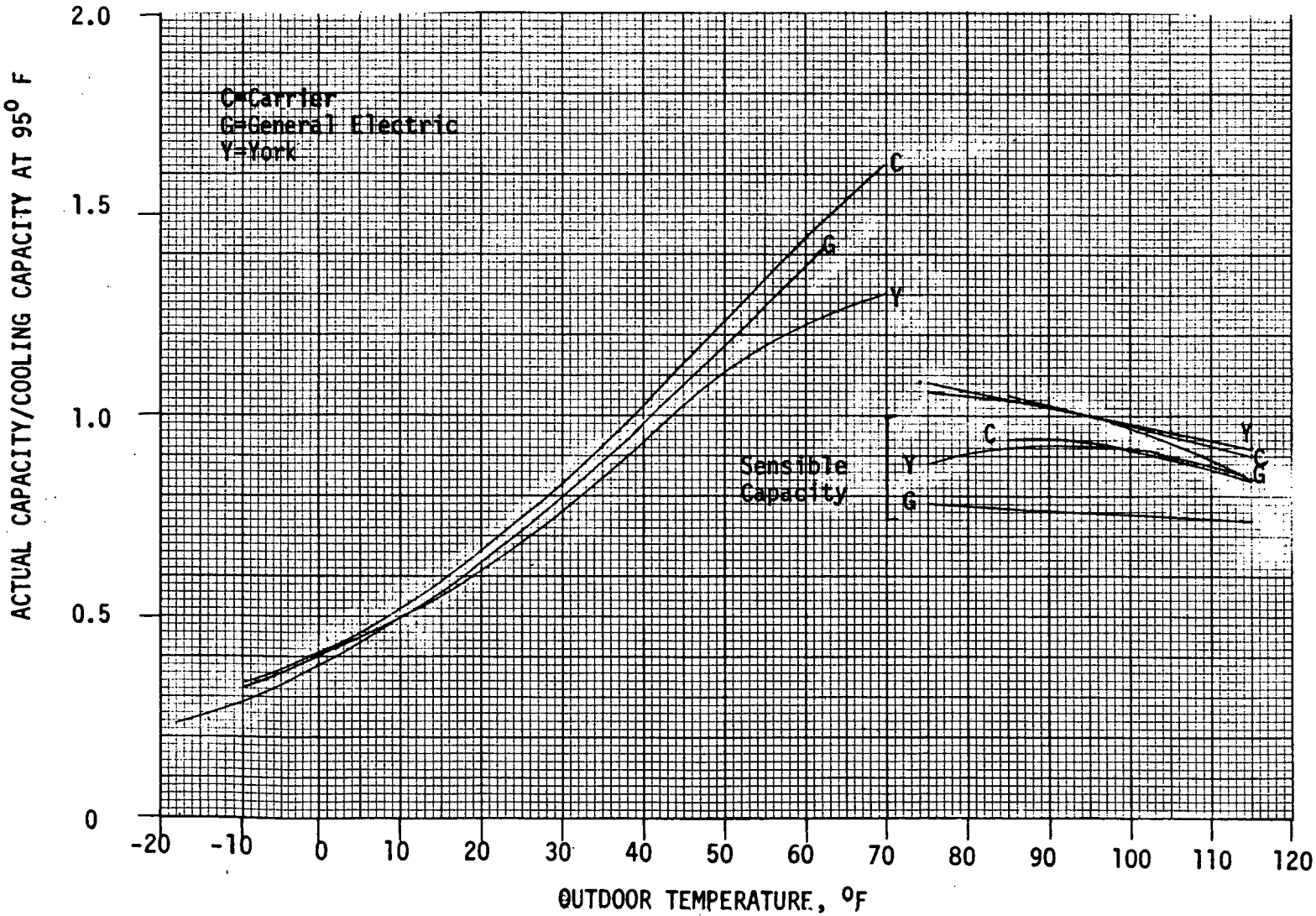


EXHIBIT 3-2

HEAT PUMP HEATING AND COOLING COP

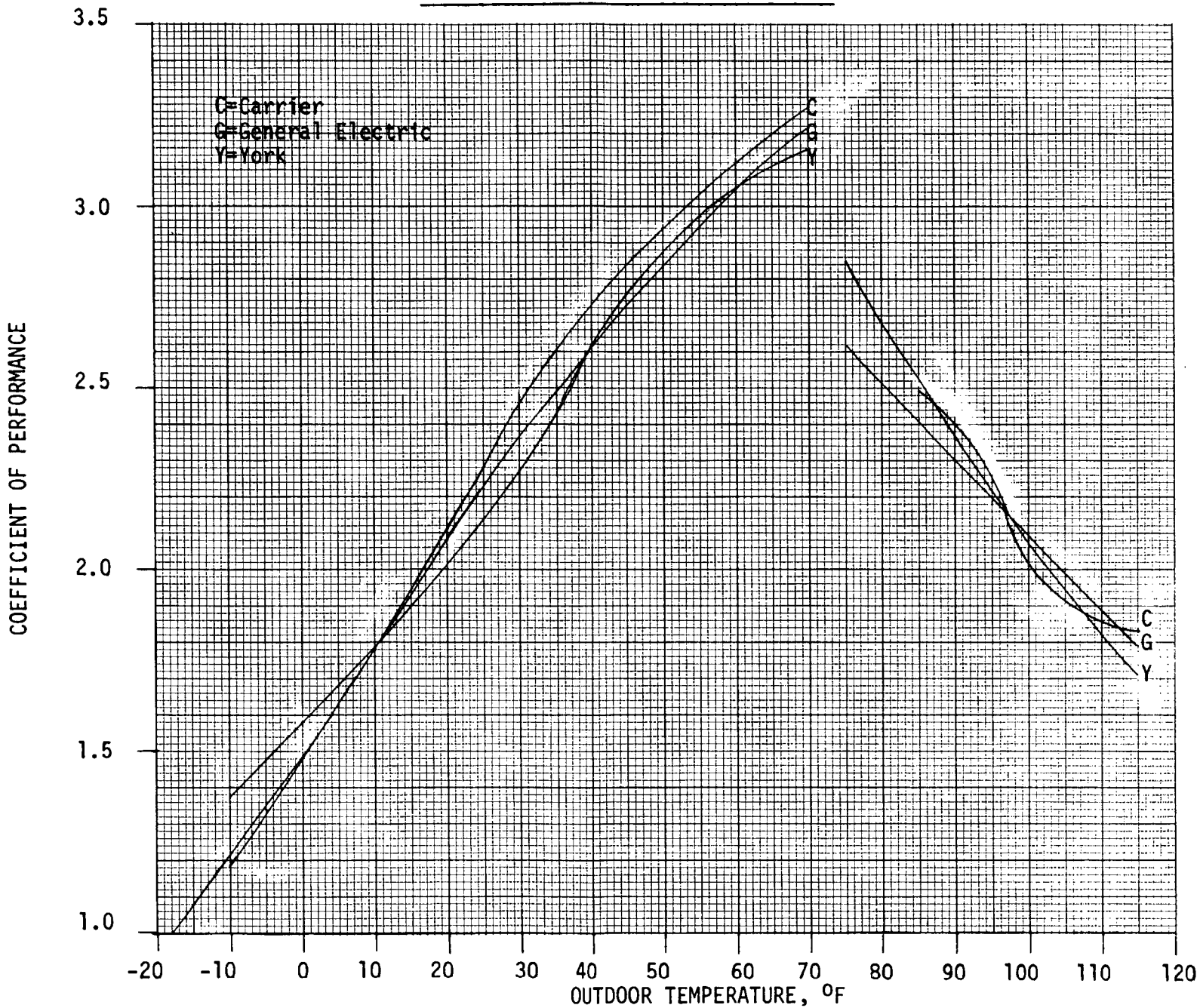
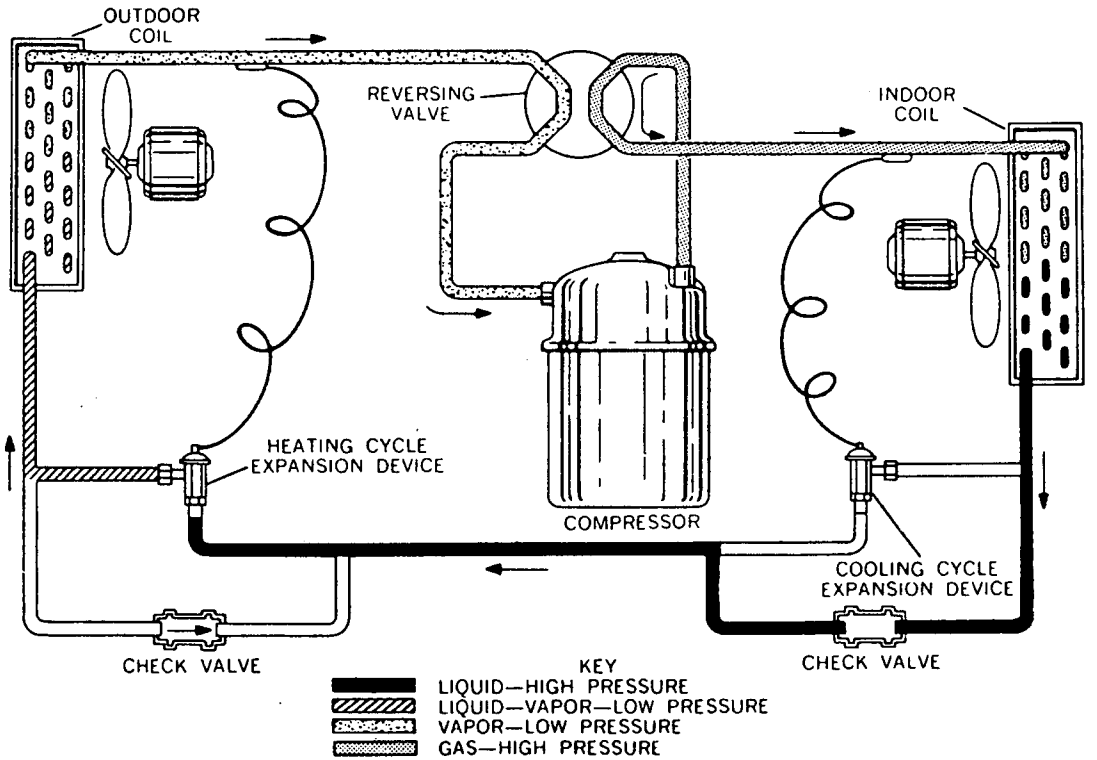


EXHIBIT 3-3

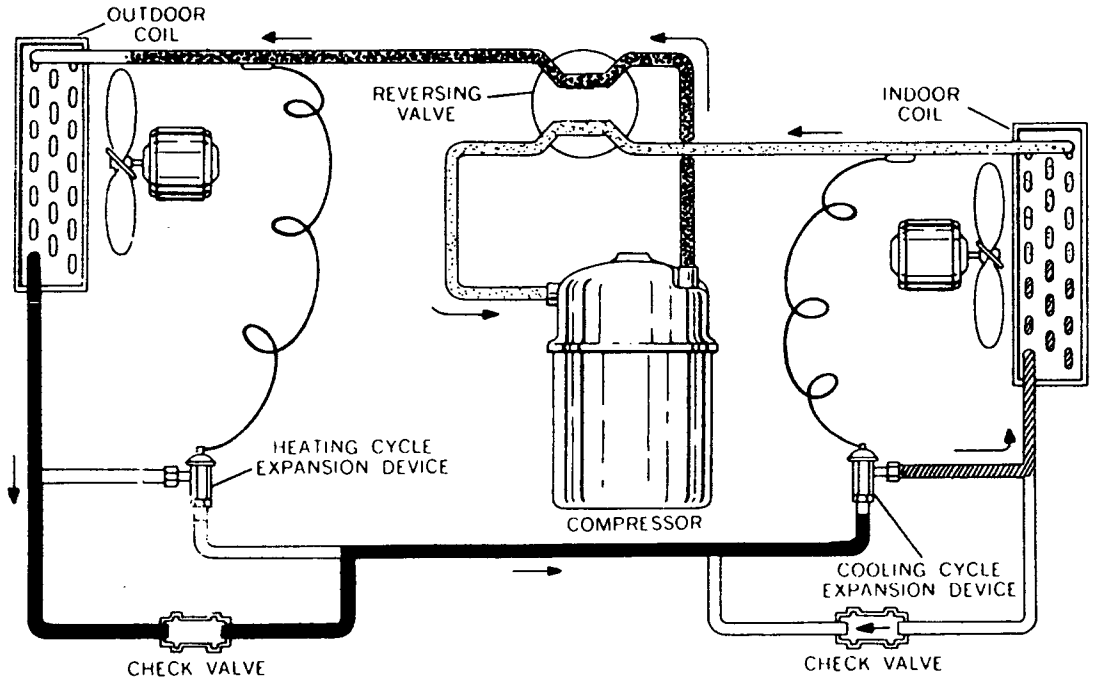
HEAT PUMP REFRIGERANT LOOP SCHEMATIC-HEATING MODE



Source: Westinghouse Electric Corporation (by permission).

EXHIBIT 3-4

HEAT PUMP REFRIGERANT LOOP SCHEMATIC-COOLING MODE



Source: Westinghouse Electric Corporation (by permission).

EXHIBIT 3-5

DECREASED OUTDOOR FAN POWER AS CONTRIBUTOR TO HIGHER COP*

	1976-Model	1977-Model
Nominal cooling capacity, Btuh	24,000	26,000
Outdoor fan (propeller):**		
Diameter, inches	18	22
Motor horsepower	1/3	1/6
Motor speed, rpm	1075	825
Airflow at 0"w.g., cfm	2600	2550
Outdoor coil:		
Face area, ft^2	8.5	14.5
Face velocity, ft/min	301	176
Heating COP		
17°	1.6	1.9
47°	2.5	2.7
Cooling EER, 95^oF	7.0	8.1

* Not all of the COP improvement shown is, of course, attributable to decreased outdoor fan energy.

** No change in indoor fan, which is centrifugal, 9"X9", 1075 rpm and has a 1/4 horsepower motor. Rated air flow 800 cfm.

4. THERMAL ENGINE HEAT PUMPS

4.1 Introduction

Although the electric heat pump employs an electric induction motor hermetically sealed into the refrigerant loop to drive the Rankine 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 chapter 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 Rankine cycle type; in some cases, a cycle operating entirely in the gas phase is treated. It should be noted that none of the heat pumps discussed have been completely assembled and tested at the time of this writing, although in a number of cases a substantial amount of development work has already been accomplished.

4.2 Aspects of Thermal Engine Heat Pump 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 4.3. 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).

4.2.1 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 ones are listed below:

- High overall efficiency (in a primary energy sense) in comparison to state-of-the-art electric heat pumps in heating mode and gas or oil furnaces;
- Ability to utilize prime mover waste heat to supplement refrigeration cycle on heating mode;
- Capability to modulate heat pump capacity by varying fuel input rate to 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, 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 it is possible to design a thermal engine heat pump to "hold its own" on cooling as well.

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. This allows easier modulation of heat pump capacity than with alternating current induction motors presently employed in electric heat pumps.

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 may potentially eliminate what is frequently a major reliability problem with electric heat pumps as well as a cause of some efficiency loss.

4.2.2 Disadvantages

Significant disadvantages applicable in general to thermal engine heat pumps, in contrast to electric heat pumps or gas or oil furnaces, are listed below. As before, these 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, it should be noted that the fact 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 surface, and pump or fan energy is

required to reject engine waste heat. (Obviously, some of that heat could be utilized for service hot water heating, but at the price of increased complexity.)

Experience with small residential and commercial HVAC and refrigeration systems has shown that hermetic systems are preferred from a reliability standpoint. In systems that operate above atmospheric pressure, refrigerant charge leakage is a possibility. In systems that operate at sub-atmospheric pressure, infiltration of air (or possibly other fluids) can occur. Noncondensible gases in condensers, especially, tend to diffuse to and form a film adjacent to the condenser wall thus increasing the inside film resistance to heat transfer (Eckert and Drake, 1972).

With an on-site prime mover, design of a hermetic refrigerant loop becomes difficult although not impossible. One solution is to use a magnetic coupling, as in the Brayton-Rankine case (see Section 4.4). While eliminating a shaft, the mass of magnetic material required to transmit sufficient power at minimum slip may be significant in relation to the rest of the rotating group*, thus increasing the polar moment of inertia of the shaft and lowering the critical rotational speed at the onset of shaft whirl. However, this may not be as serious a constraint as it might appear, as the Brayton-Rankine system, which transmits 13 hp across a samarium-cobalt coupling, is designed to operate at speeds up to 90,000 rpm (1,500 Hz).

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* Because of frequent direction reversals, it is anticipated that a magnetic coupling of a linear engine to a reciprocating load is less practical.

Another interesting solution to this problem is evident in the design of the (linear) free-piston Stirling-Rankine heat pump. This design employs an inertial compressor consisting of a cyclindrical refrigerant compressor housing rigidly connected to the power piston. The cylinder contains a free inertial mass, the compressor piston, which compresses the refrigerant vapor as the housing reciprocates. The refrigerant suction and discharge lines are in the form of tubular helical springs which extend and contract with the motion of the compression cylinder. Some possible problems with this design may be (a) lack of sufficient gas pressure or "spring" in the heating mode to bounce back the inertial piston due to the lower mass density of the refrigerant and (b) suction and discharge tube failure due to large number of flexing cycles encountered over the expected operating life of the equipment.

Rotating high-speed machinery or unbalanced engine-compressors are bound to increase noise and vibration levels over hermetic electric motor compressors or gas furnaces. Much of this noise can be eliminated in a closed-system design, proper vibration damping and sound insulation, but again at some increased cost. On-site fuel combustion may increase CO, NO_x and particulate emission levels unless an existing furnace burning the same fuel is being replaced. Again, these problems are amenable to design solutions.

Another limitation of a combination on-site prime mover and heat pump is the greater system complexity, in most cases, in comparison to an electric heat pump. This is not to say that an elegantly simple design such as the free-piston Stirling engine compressor with only

three moving parts (in the engine compressor) is not possible but, in general, practical thermal engine heat pump designs have to have:

- Two different working fluids, requiring their sealing and isolation from each other;
- Large number of moving machined surfaces requiring machining to close tolerances and lubrication (most heat pump concepts use refrigerant oil, but the engine lubrication may be dry);
- Several thrust and journal bearings;
- Feed pumps or blowers, engine coolant pumps;
- Engine as well as heat pump control devices;
- A separate starting system in some cases;
- Materials to withstand high static or dynamic operating stresses at high temperature.

This increased complexity is likely to result in a high-first-cost equipment and potential problems of system and component reliability, especially in a product that is expected to operate essentially maintenance-free and unattended for a long time.

Any new technology dependent on specific and scarce fuels such as natural gas or oil is liable to increase consumption of and dependence on these fuels in comparison to electrically driven machinery, unless only replacements of existing installations are contemplated. If, in addition, coal is to be the source of gas (or oil), the results of a recent study by Booz Allen (1976), properly interpreted, suggest that some if not all of the efficiency advantage of thermal engine heat pumps may be lost due to inefficiency of coal conversion. This point will be addressed further in Chapter 12.

4.3 Description of Some Engines and Heat Pump Cycles

Having discussed some general advantages and disadvantages of thermal engine heat pumps, some specific embodiments of prime movers and refrigeration machines will be described * for purposes of illustration. The discussion is not intended as comprehensive or exhaustive. Thermal engine heat pumps to be discussed include the following:

- Stirling-Rankine heat pump
- Rankine-Rankine heat pump
- Brayton-Rankine heat pump
- Otto-Rankine heat pump
- Ericsson-Ericsson heat pump

Exhibit 4-1 summarizes some pertinent characteristics of the above prototype systems under development. Information available on the last two of the above is of a very preliminary nature at this time.

4.3.1 Stirling-Rankine Heat Pump

An example of a Stirling cycle engine driving a Rankine cycle heat pump is the system under development by General Electric Company's Space Division.* The prototype is to have a nominal rated cooling capacity of 36,000 Btuh (3 tons) and a heating capacity of 64,000 Btuh (at 47°F).

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* These descriptions are based on best information made available to us at the time of the investigation and are believed to be accurate. However, not all details and data were available for inspection. It is also expected that design concepts may have changed since the time our visits were made or that further development may have changed some aspects of the projects discussed herein.

** Other participants in the project are AGA and ERDA.

The prime mover is a (linear) free piston Stirling engine described in a number of technical articles by the inventor and his coworkers (Beale, et al., 1973). Primarily residential applications are contemplated, but the system appears amenable to scale-up.

A schematic diagram of the Stirling-Rankine heat pump refrigeration loop is shown in Exhibit 4-2, based on a report by the sponsoring organization*, AGA (Colosimo, 1976). The system is shown in heating mode operation. The engine is cooled by a water-glycol mixture which also recovers additional waste heat from the combustor exhaust gas. The coolant then rejects the heat via an additional heat exchanger to the indoor air (as supplementary heat) in the heating mode or to the outdoor air in the cooling mode.

Exhibit 4-3 shows the details of the prototype engine-inertial compressor unit. The engine working fluid is helium heated by a gas combustor in an external combustion chamber. As already mentioned, the refrigerant loop is completely hermetic. The refrigerant is R-22.

The most significant advantage of the Stirling engine as a heat pump prime mover lies in the high theoretical efficiency attainable in principle. Unlike most of the other engines discussed in Chapter 4, the theoretical efficiency of the Stirling engine equals that of the Carnot engine at the same source and sink temperatures (Walker, 1973). Although it is no more possible to build an ideal Stirling machine than a Carnot machine, the former can be more closely approximated in real "hardware." As a result, the design overall engine efficiency of the free piston engine is about 32% in the configuration contemplated for the heat pump

* A review article on this project was recently published by Sarkes et al. (1977), much of it based on the earlier Colosimo paper.

unit. However, according to Beale (1976), efficiencies as high as 48% have been achieved in experimental engines in his laboratory.

As indicated in Exhibit 4-1, the 3 kW engine, which is the size required to power a 2 or 3 ton heat pump, would operate at a reciprocating frequency of about 30 Hertz. According to Beale et al. (1973), the frequency of the engine is proportional to $(kA^2P_0/mV)^{0.5}$ where k is the ratio of the isobaric to the isosteric heat capacity of the charge gas, A is the piston area, P_0 is the charge pressure, m is the mass of the piston and V is the volume of the working gas. Hence, engine power increases approximately as the 1.5 power of the charge pressure. The specfic work per cycle is a function of the operating pressure ratio. To modulate the engine, one would have to vary the charge pressure, keeping the temperature as high as possible for high efficiency. While this might be accomplished by having an external reservoir of charge gas connected to the working space, the complexity of the arrangement and the required controls might make the approach unattractive. According to General Electric, the present approach is to cycle the engine-heat pump on and off, rather than modulate the capacity (at least significantly), however, nothing inherent in the basic engine-compressor design would preclude speed-modulation in principle.

The free piston engine itself has only two moving parts, the compressor, one, making for potentially high reliability. The design appears to facilitate quick disassembly for preventive or corrective maintenance. No oil lubrication system is required for the prime mover since the carbon-graphite bearing surfaces used are self-lubricating. This makes for simpler, less costly system. The refrigeration loop has

lubricating oil, but this may ultimately be eliminated in a commercial product. The free-piston engine in a vertical configuration is self-starting for a horizontal engine, a starting mechanism is required. Since the engine cycle is a closed one (i.e. uses an external combustion system) gas, oil or even solid fuel can be utilized. Indeed, any sufficiently intense source of thermal energy (e.g. solar radiation) capable of producing the required hot end temperatures may be substituted for the combustor. The engine CO emissions (when burning gas) are believed to be comparable to and NO_x emissions are lower than those of a gas furnace. Engine operation is noise free; thus external muffling is not required. (The Beale free-piston engine is not balanced and hence requires appropriate mounting.)

On the other hand, a number of potential problem areas exist requiring further development. For example, the decreased refrigerant pressure during heating mode operation apparently produced insufficient "gas spring" to prevent the inertial compressor piston from banging against the ends of the compressor housing. According to W. Beale of Sunpower Inc., possible solutions to the "gas spring" problem can be found: e.g. an expandable compressor piston, a gas bypass around the piston, etc. Whether this problem has been solved is not known.

Engine hot end materials are required to withstand helium temperatures of 1200°F and pressures as high as 1050 psig. This places stringent requirements on materials of construction. While the engine-compressor has only three moving parts, close tolerances to minimize yaw of piston and displacer and low dead volume are critical if design efficiency is to be achieved. The ability of helical spring compressor

suction and discharge tube joints to withstand an estimated 6-7 billion flexures has to be proven.* It should be noted that the control over engine operation does not appear to be "positive" in the sense of control over internal combustion engines or electric motors. The engine may continue to operate after the fuel supply is cut off apparently until the hot end cools down sufficiently. Experimental engines have been observed to develop abnormal vibrations (shaking) in operation.

The estimated specific power of the projected engine is lower than for Rankine, Otto or Diesel cycle engines. This usually implies a costlier engine.

Many of the problems cited are of course expected in an experimental device and should be amenable to design solution. Needless to say, critical questions that remain to be resolved are, of course, cost and operating reliability of production type equipment.

Although the free-piston Stirling engine seems attractive as a heat pump prime mover from many respects, it is obviously not in commercial production at present. Since there are some small Stirling engines in a more advanced stage of commercial development, it is necessary to consider whether any of those might not serve as a driver as well. As an example, IGT has proposed that a small single cylinder V-type machine developed in Europe** would offer the advantage of (a) being essentially

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* A 30 Hz engine operating for 60,000 full-load hours would subject the compressor tubing to 6.48×10^9 cycles of flexures during its design life.

** IGT has requested that the identity of the manufacturer of the engine not be published at this time.

production-ready and (b) offering a better capability of speed modulation. Since this information became available rather late in our study (Wurm, 1977), it has not been possible to investigate this proposed approach thoroughly, although performance estimates have been published previously (Wurm et al., 1976). It is known that Philips and its licensees have been involved in development of engines for automotive and other applications for a number of years, although little has been published as to the status of their technology. It would be interesting to learn how the problem of engine-heat pump power transmission is to be accomplished in view of the requirement of minimizing refrigerant leakage.

4.3.2 Rankine-Rankine Heat Pump

A Rankine cycle heat pump powered by a steam turbine is being developed by Consolidated Natural Gas Service Company.* The prototype is to have a nominal cooling capacity of 90,000 Btuh (7.5 tons) and a heating capacity of about 150,000 Btuh (47°F)**. The low temperature (17°F) heating capacity of the system is 135,000 Btuh. The heat pump compressor, augmented by turbine waste heat recovery, provides heat down to a temperature of -10°F; below that, direct steam injection into the steam coil in the supply air duct is planned.

Exhibit 4-4 shows the prime mover and heat pump flow schemes in the heating mode; the cooling mode is shown in Exhibit 4-5. The steam

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* Other project participants are Mechanical Technology Inc. and ERDA.

** While the final draft of this report was being completed, the Rankine-Rankine heat pump underwent a change in product concept. The production version is now to be a 100 ton applied or built-up system instead of unitary.

turbine is a radial entry machine driving an alternator and centrifugal R-11 refrigerant compressor via a rigid shaft. The alternator generates all auxiliary power to drive a condensate return pump and combustion air blower for the compact steam boiler and indoor and out-door fans. The turbine is rated at 31 shaft horsepower and operates in the 30,000-60,000 rpm (500 -1000 Hz) range. Present plans call for the turbine to be modulated to about one-fourth load on cooling and to less than one-half load on heating. All turbine waste heat is recovered for maximum efficiency.

The Rankine-Rankine heat pump design exhibits a number of interesting features. First, the power fluid (water) is, of course, stable, non-toxic and non-flammable. While the cost effectiveness of small steam turbines has not yet been universally demonstrated (except where special needs such as for reliability in emergencies or use of waste heat can be justified), the designers claim that the use of steam reduces necessary heat exchanger area for engine heat rejection since its film heat transfer coefficients are higher than for air or refrigerant at the same Reynolds number, due to the higher Prandtl number of steam (Cons. Natural Gas, 1976). A design feature of interest (and on which CNG has filed for patent protection) is the sequencing of the steam and refrigerant heat exchange in the indoor and outdoor coils (Exhibits 4-4, 4-5). Steam and refrigerant are in a crossflow configuration with respect to air but in counterflow with respect to each other. This maximizes the heat transfer effectiveness of the heat exchangers and keeps down the refrigerant compressor pressure ratio.

It should be noted that turbomachinery is potentially highly reliable for long-time unattended operation although there is admittedly little experience with small capacity steam expanders. Other advantages are that the system is self-starting and because the steam turbine cycle is basically a closed loop, with proper design, noise and vibration is not expected to be a problem.

The major drawbacks to this concept appear to be few, but should be mentioned. First, the refrigeration loop is not hermetic. Hence, a dynamic shaft seal is necessary on the common shaft linking steam turbine to the alternator and refrigerant compressor. Engineering a reliable dynamic shaft seal to minimize infiltration of air (both the water and the refrigerant loop have sections where the pressure is sub-atmospheric) and without incurring a significant efficiency penalty is not a trivial problem. Air infiltration can cause corrosion on the steam side and reduce heat exchanger overall heat transfer rates on the refrigerant side and hence the capacity. It should be noted that the present system is designed to use R-11 as the refrigerant. R-11 has been tentatively identified in connection with its possible reactivity with atmospheric ozone.

The Rankine-Rankine heat pump is intended for commercial applications in areas where the dominant requirement is for heating. As such it would primarily compete with the "year-round" air conditioner using gas for heating.

4.3.3 Brayton-Rankine Heat Pump

The gas turbine Rankine heat pump is being developed by Garrett-Airesearch Manufacturing Company of California under sponsorship of AGA. The prototype has a nominal cooling capacity of 120,000 Btuh (10 tons) and a heating capacity of about 180,000 Btuh at maximum turbine speed of 90,000 rpm.

The prototype Brayton-Rankine heat pump configuration is shown in Exhibit 4-6. The system employs a subatmospheric open cycle gas turbine with a centrifugal compressor for compressing the exhaust gas before rejection to the atmosphere. The prime mover shaft is connected via a samarium-cobalt magnetic coupling to the refrigerant compressor which is of the centrifugal type. Thus the refrigerant loop is completely hermetic. The Brayton rotating group employs air-lubricated foil journal and thrust bearings. Other features of the system are engine and exhaust heat recovery and the use of a refrigerant boiler and expander as the start-up system for the engine. Note that the diagram shows an additional turbine running on refrigerant vaporized in a R-12 boiler by gas combustor exhaust heat. This is used as a bottoming cycle to increase efficiency and serve as a start-up mechanism.

Although not particularly so for gas turbomachinery, comparatively high wheel speeds (70,000-90,000 rpm; 1170-1500 Hz) are used, depending on the heating or cooling capacity required by the building load. At low loads, the system will cycle on and off at the lower speed.

The advantages of rotating machinery in terms of inherently higher reliability than reciprocating machinery have already been noted. A gas

turbine is basically a low first-cost prime mover although its efficiency in comparison to other prime movers is not particularly high. Combustors can be designed to accommodate liquid as well as gaseous fuels.

It should be noted that a centrifugal compressor is less susceptible to liquid slugging and contains fewer moving parts than a reciprocating compressor. On the other hand a centrifugal compressor is more susceptible to gas surging at low mass flows than a reciprocating (i.e. positive displacement) compressor.

Some potential problem areas with the Brayton-Rankine concept are as follows: First, clearly the high (90,000 rpm maximum) rotating speeds and turbine wheel temperatures of the order of 1500°F place stringent requirements on rotating group materials. The turbine is not self-starting, requiring a relatively complex vacuum or a refrigerant bottoming start cycle. (The latter cycle is shown in Exhibit 4-6.) The present plan of using reverse cycle defrost (rather than turbine waste heat) will incur an efficiency penalty. Note, also, that the present system design is for R-12 as the refrigerant, a fluid which has been tentatively identified in connection with its reactivity with atmospheric ozone. NO_x emissions should not be a potential problem with this gas turbine since low cycle pressures are employed.

The Brayton-Rankine heat pump is being developed as a unitary gas heat pump for commercial applications. While the design is amenable to scale-up, according to the developers, it is unlikely that a smaller unit than the prototype 10 ton (or perhaps 7 1/2 ton) size could be developed at a reasonable first cost.

4.3.4 Otto-Rankine Heat Pump

Another concept that has been advanced is the use of a spark-ignition* free piston reciprocating gas engine as a heat pump prime mover. The engine was developed as a driver for air compressors by A.T. Braun. Participants in the proposed application of the engine to drive a commercial-scale Rankine cycle heat pump are Tectonics Research Inc. (a Braun subsidiary), Columbia Gas System Service Co. and others (Columbia Gas System, 1976).

The Braun linear engine has been described in some detail in an article by Braun and Schweitzer (1973). Its major advantages over crank-type reciprocating engines are (a) smaller number of moving parts, (b) lower cost due to simpler construction and fewer critical tolerances and (c) lower fuel consumption at equivalent compressed gas (air) output.

The developers claim that this engine compressor is closer to commercialization than any of the other engine driven heat pumps, in that major components of the air-compressor version of the system have been tested for over 45,000 hours of running time and a heat pump-type test engine has been successfully run for 15,000+ hours. The developers also claim that the engine compressor has a higher coefficient of performance than other engine systems, higher part-load efficiency (due to the ability of the prime mover to modulate speed) and is of simpler construction. The compressor can run without refrigerant oil.

The scope of the present study did not permit a thorough investigation of this system and evaluation of the validity of the above performance

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* Development of a compression-ignition engine in this configuration is also possible.

claims. However, this engine driven heat pump has a number of apparent limitations, the first being that, although the refrigerant compressor in the design configuration is hermetic, the membrane* isolating the refrigerant loop from the prime mover is subject to fatigue failure as a result of a large number of flexures over the expected 60,000 hour design-life of production-type equipment. Indeed, although having fewer moving parts than an automotive-type, crank-driven engine, the power transmission from the engine to the refrigerant compressor utilizes a number of mechanical components (which cannot be described here for reasons already stated) that are potentially subject to high wear in heat pump service under possibly very adverse conditions (e.g. cold-soaked start-up).

Also, while the engine itself may have only a few moving parts, carburetion or fuel injection, ignition, a start-up system and other auxiliaries are required.

Finally, the Braun linear engine is an internal combustion engine developing relatively high combustion chamber pressures at high temperature. Thus noise and pollutant emissions are expected to be a problem which, although, amenable to solution, would surely detract from overall system efficiency.

4.3.5 Ericsson - Ericsson Heat Pump

The heat pumps described in the previous sections of this chapter have been two-fluid heat pumps: that is, different fluids were used in

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* Details of the actual design are confidential.

the engine and in the refrigeration loop. This is because higher overall efficiencies are obtained, in practice, if the power fluid and refrigerant are each selected on the basis of offering the most optimal set of thermodynamic, transport, flammability, toxicity, cost and other properties. On the other hand, separate power and refrigeration loops make for a more complex and costly machine. Hence, there are certain advantages in a single-fluid thermal engine heat pump concept in principle. One such concept is a gas-fired Ericsson cycle engine driving an Ericsson cycle refrigerating machine.

The Ericsson cycle is similar to the Stirling cycle except that the latter ideally uses constant volume regenerative heat exchange whereas the former uses constant pressure regenerative processes. That is, in the ideal Ericsson engine, the working gas (e.g. hydrogen, helium, carbon dioxide or air) is compressed isothermally by the power piston, heated at constant pressure to the higher cycle temperature, expanded isothermally producing work and cooled at constant pressure to the lower cycle temperature (Walker, 1973). As with the Stirling cycle, the thermodynamic efficiency of the Ericsson cycle is identical to that of a Carnot machine operating between the same terminal temperatures. Again, however, machines approximating the Ericsson cycle are easier to build than Carnot machines. The Ericsson refrigerator employs the same processes as the Ericsson engine but in a reverse cyclical order (counter-clockwise on a T-s diagram). Thus work has to be input to the refrigerator to produce a net refrigerating effect.

An Ericsson-Ericsson heat pump has been proposed by Energy Research and Generation Inc. (Benson, 1975; ERG, 1977). The concept utilizes a single fluid hermetic free-piston machine with an engine displacer, heat pump displacer and two phaser pistons actuated by working gas pressure. The engine is valveless. Since this machine is still in the proof-of-concept stage (as far as a heat pump development is concerned) the developers' claim of attaining 80% of the efficiency of an ideal Carnot engine-heat pump at the same operating temperature and pressure conditions has yet to be proven.

4.4 Summary

This chapter has surveyed a number of experimental thermal engine heat pump concepts now being developed through participation or sponsorship of the gas utility industry. Thermal engine heat pumps, especially those employing engine cycles of high ideal efficiency, offer theoretically the best promise of overall system energy efficiency of any of the heat pump technologies especially in applications and climates where the major space conditioning requirement is for heating. But an on-site engine-driven refrigeration machine with all its necessary auxiliaries is apt to be more complex and costly than a hermetic electric motor-driven machine of equal capacity. Therefore, whether a practical, cost-competitive, reliable and maintenance-free unitary thermal engine heat pump can be successfully developed, manufactured and marketed is certainly an open question. Of the various heat pump concepts examined, the ones employing rotating machinery and a hermetic refrigeration loop would

appear to be technically the more feasible. It is also more likely that commercial-size rather than residential-size machines will be eventually developed and commercialized. In the former case, first cost would seem to pose less of a constraint to the design solution of the numerous problems of engine cooling, lubrication, waste energy recovery, refrigerant loop integrity (hermeticity), cooling-mode efficiency, evaporator defrost method, engine and refrigeration machine controls, fans, pumps and other auxiliaries, noise, vibration and exhaust pollutant emissions, to name several.

Ultimately, it is more likely that manufacturing and market economics, rather than technological viability, will be the deciding factors on whether thermal engine heat pump technology will emerge successfully from its present development stage.

As this section was being prepared, notice was received (Didion et al., 1977) of experiments carried out at the National Bureau of Standards with operation of an air-source Rankine cycle heat pump by power from a Philips I-98 gas fired, water cooled Stirling engine. The maximum thermal efficiency of the engine was 28% over the range of conditions tested. The experiments found that between 50-60% of the fuel energy input was rejected to engine coolant (in contrast to, say, 15-30% for an internal combustion engine)--energy that is potentially partially reclaimable as supplemental heat for heating mode operation. Another finding was that higher efficiency was obtained at the lower engine speed as the compressor was speed-locked in constant ratio to the engine through a pulley drive. Hence, the authors suggest that speed and capacity modulation is indicated only below low speed heat pump balance point.

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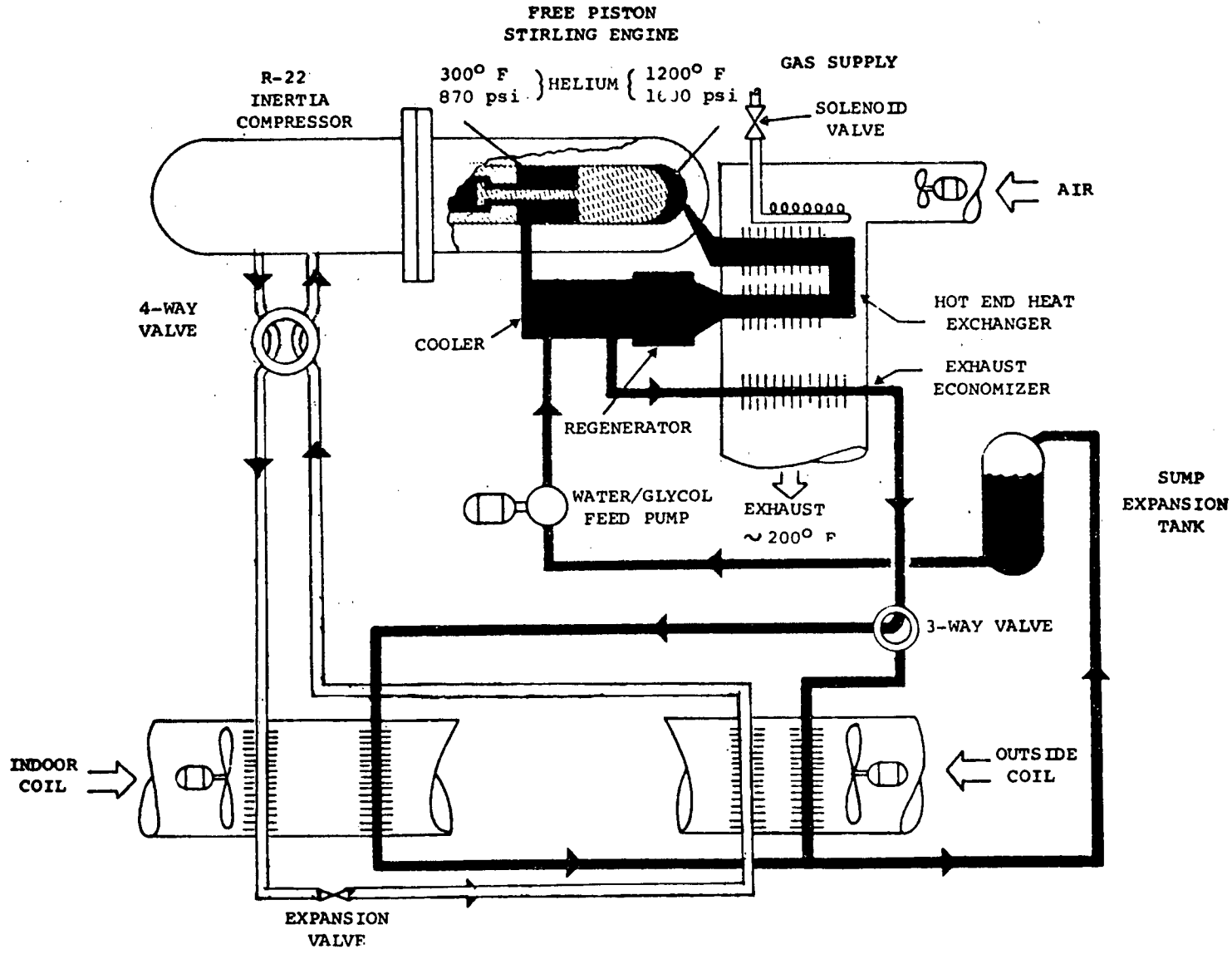
EXHIBIT 4-1

SUMMARY OF SOME THERMAL ENGINE DRIVEN HEAT PUMP CHARACTERISTICS

	Stirling-Rankine	Rankine-Rankine	Brayton-Rankine	Otto-Rankine	Ericsson-Ericsson
Engine Data					
Thermodynamic cycle	Free piston Stirling Engine	Rankine steam turbine	Brayton gas turbine	Braun linear Otto engine	Ericsson
Power	3kW	31 HP (23 KW)	13 HP (9.6 kW)		2 HP (1.5KW)
Power fluid	Helium	Steam	Combustion gas	Combustion gas	Hydrogen
Efficiency(max.)	32-36%	23%	27%	33%	
Frequency	30 Hz	500 Hz	1170-1500 Hz	Variable	
Coolant	Glycol water	Glycol-water			
Fuel (s)	Gas (other fuel, solar)	Gas	Gas (fuel oil)	Gas	Gas (other fuel, solar)
Self-starting	Yes	Yes	No	No	Yes
Compressor Data					
Thermodynamic cycle	Rankine	Rankine	Rankine	Rankine	Ericsson
Refrigerant	R-22	R-11	R-12	R-22?	Hydrogen
Type	Inertial (recip.)	Centrifugal	Centrifugal	Reciprocating	Reciprocating
Nominal cooling	3 tons	7.5 tons	10 tons	20 tons	
Hermetic	Yes	No	Yes	Yes	Yes
Other Data					
Electrical power required	Yes	None (generates 8kVA own power)	Yes	Yes	None

EXHIBIT 4-2

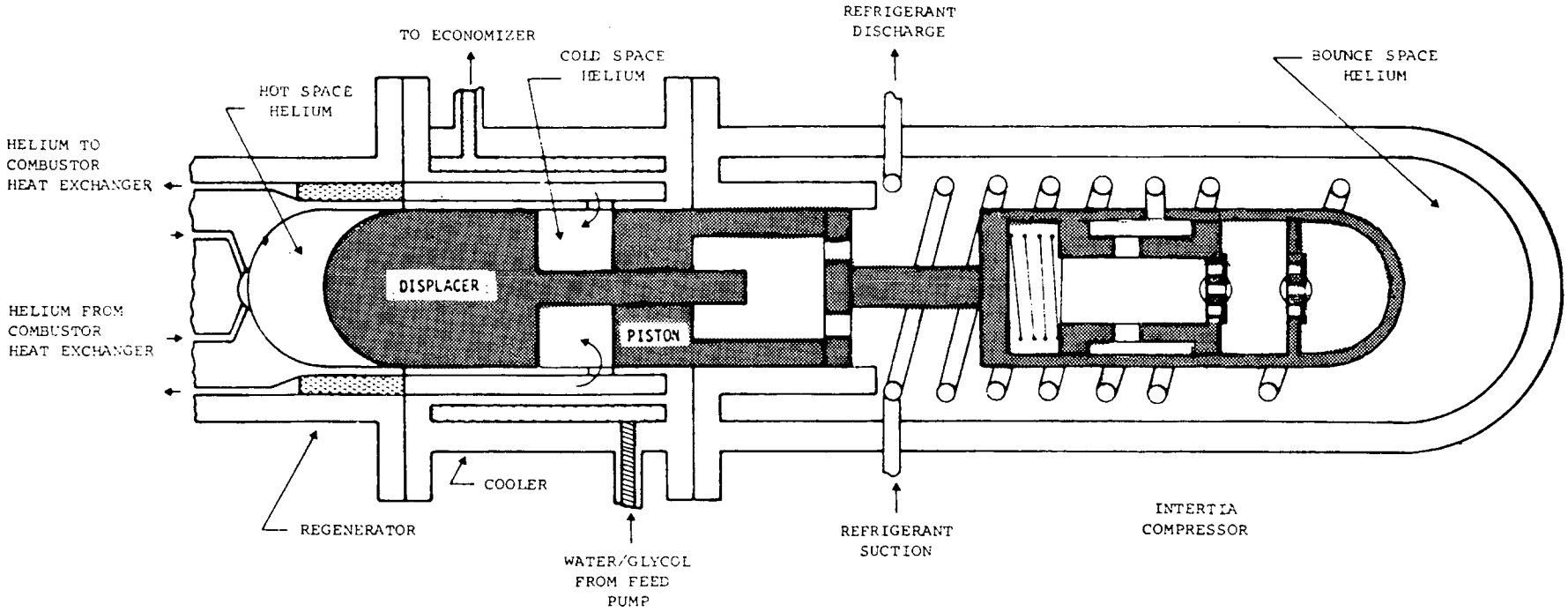
FREE PISTON STIRLING-RANKINE HEAT PUMP-HEATING MODE



Source: Colosimo (1976); by permission.

EXHIBIT 4-3

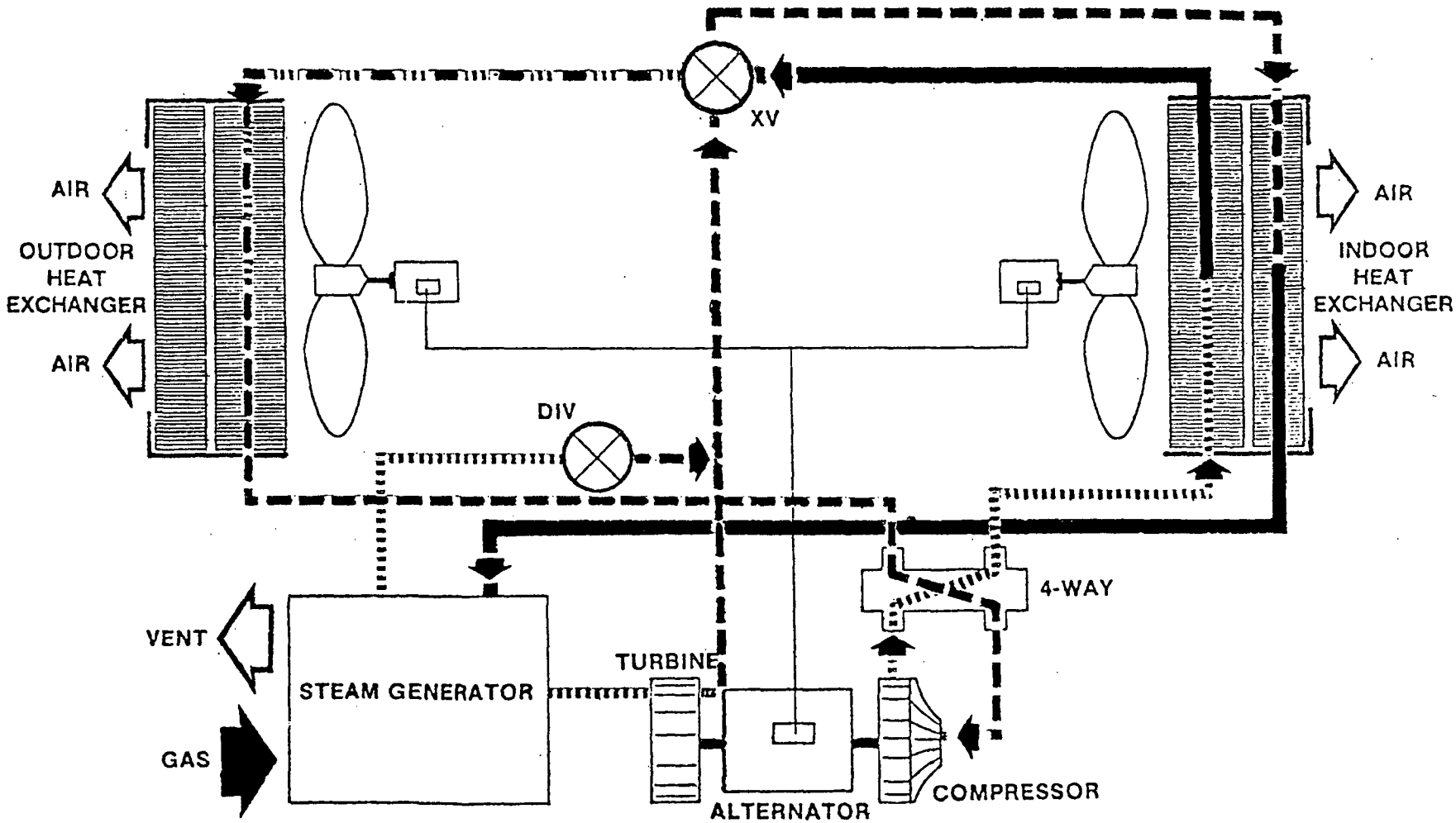
FREE PISTON STIRLING ENGINE WITH INERTIAL COMPRESSOR



Source: Colosimo (1976); by permission.

EXHIBIT 4-4

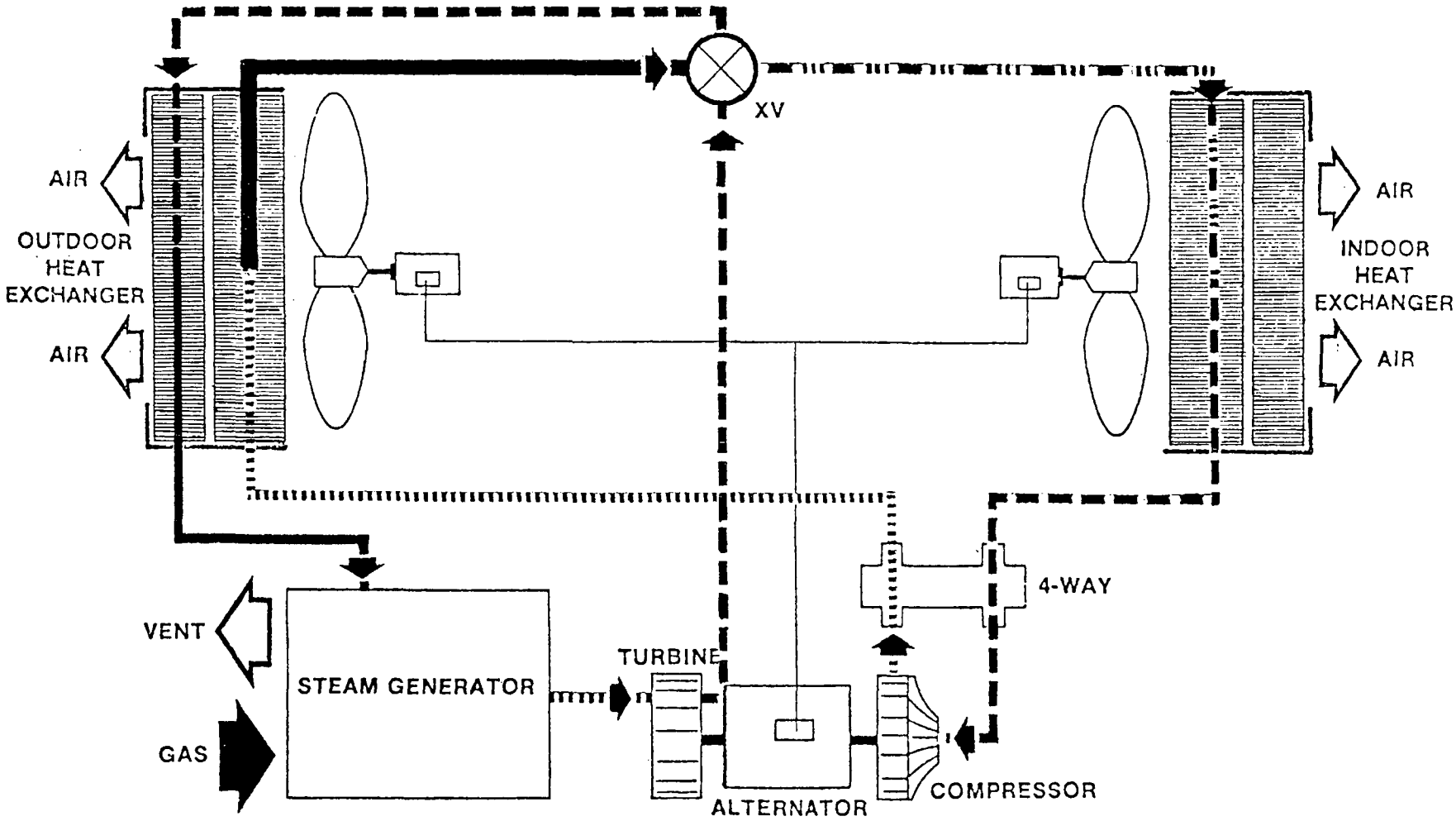
ERDA/CNG/ MTI GAS HEAT PUMP SYSTEM SCHEMATIC - HEATING SERVICE



Source: Consolidated Natural Gas Service Co. (1977); by permission.
Heat exchanger patents pending.

EXHIBIT 4-5

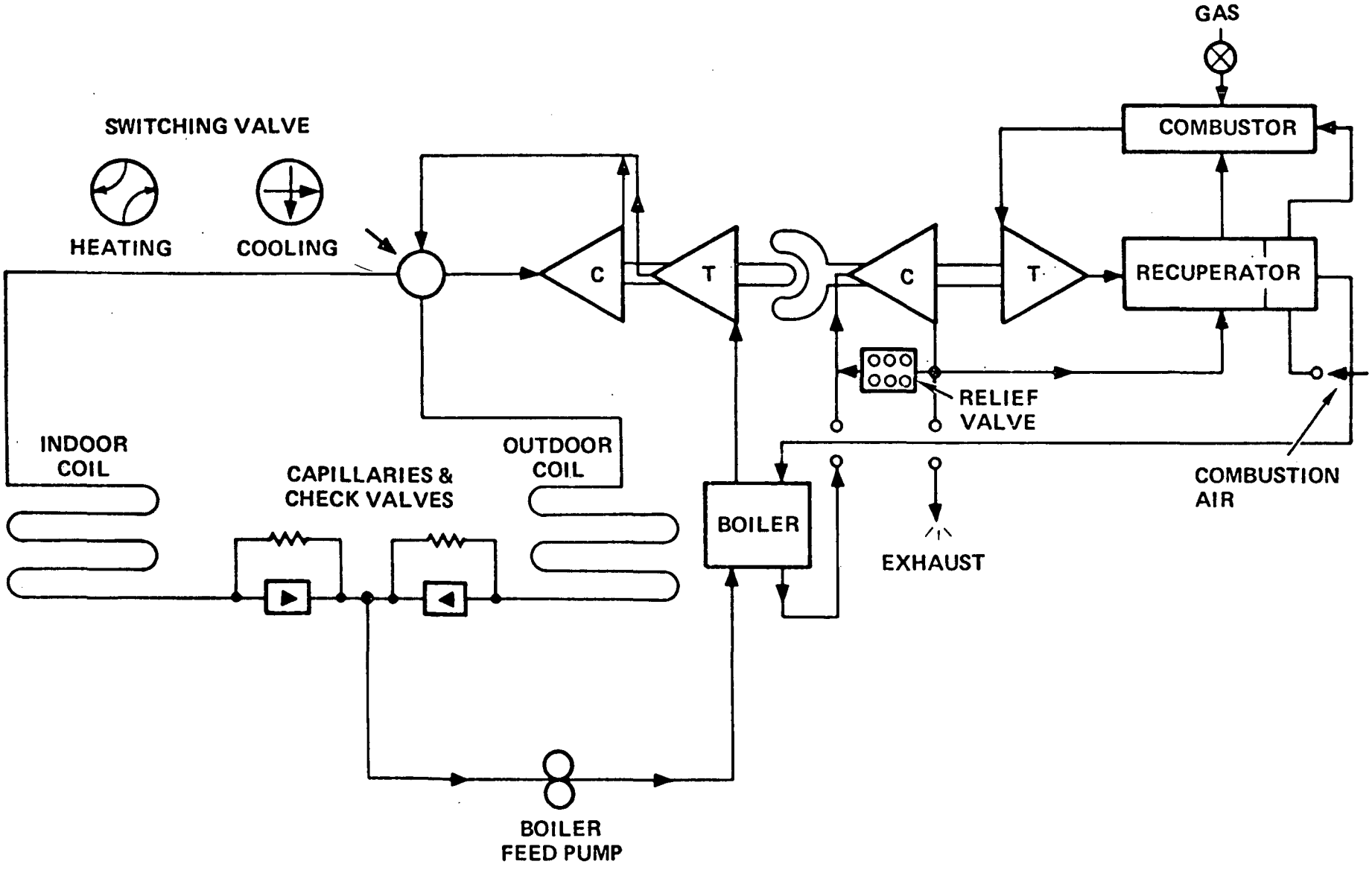
ERDA/CNG/ MTI GAS HEAT PUMP SYSTEM SCHEMATIC - COOLING SERVICE



Source: Consolidated Natural Gas Service Co. (1977); by permission
Heat exchanger patents pending.

EXHIBIT 4-6

BRAYTON RANKINE HEAT PUMP



Source: Garrett Corp. (1976); by permission.

5. ABSORPTION CYCLE HEAT PUMPS

5.1 Introduction

The electric and thermal engine heat pumps treated in Chapters 3 and 4, respectively, have utilized homogeneous fluids (refrigerants) undergoing a Rankine or other refrigeration cycle. This chapter describes another method of heat pumping; namely, by one of several absorption refrigeration cycles.

In many respects an analogy may be said to exist between an absorption refrigeration cycle and a vapor compression or Rankine cycle. In the latter case, a compressor is used to increase the pressure of the refrigerant vapor prior to its condensation; in the former case, the chemical affinity between a refrigerant vapor and an absorbent solution and a solution pump produces a similar effect. Indeed, the expression for the COP of a simple ideal absorption cycle has the form of a product of the efficiency of the Carnot engine cycle and the COP of a Carnot refrigeration cycle operating at the same limiting temperatures (e.g., Curran, 1975). Absorption air-conditioning machines are commercially available today in large-tonnage (100 and up) sizes. On the other hand, some unitary smaller tonnage residential and commercial absorption cycle air conditioners are also being produced by a single manufacturer, but to a steadily shrinking share of the total unitary air-conditioning market. No absorption cycle unitary heat pump is commercially available at this time.

5.2 Aspects of Absorption Cycle Heat Pump Design and Operation

This section will enumerate and discuss some general advantages and disadvantages of absorption cycle heat pumps based on information obtained

from our field investigation and technical literature sources (Wurm, 1974; ASHRAE, 1975; Curran, 1975; Phillips 1976 a,b). It should be stated, however, that our available information on the organic fluid based machine is very limited due to, apparently, proprietary considerations.

5.2.1 Advantages

Absorption cycle heat pumps, depending somewhat on the cycle and the particular design configuration employed, have the following major advantages:

- Inherently simple and potentially highly reliable equipment, in that the solution pump and air moving fans (or water pumps for water-cooled systems) are the only moving parts (no compressor is needed);
- High efficiencies in the heating mode due to possibility of heat recovery from on-site combustion as supplementary heat to that provided by the absorption cycle condenser itself;
- Lower parasitic energy requirements than for thermal engine heat pumps;
- Refrigerant-absorbent fluids can be used which are chemically nonreactive with atmospheric ozone;
- The generator of the absorption cycle heat pump can more easily be energized by a solar thermal source, if of sufficiently high temperature, than using solar energy to power a thermal engine*;
- Availability of on-site waste heat for evaporator defrost.
- Inherent capability of storing energy as high pressure refrigerant vapor.**

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* However, a thermal engine may be preferred from the viewpoint of efficiency and cost (Curran, 1975).

** The authors are indebted to one of the reviewers for bringing this advantage to their attention.

Absorption machine efficiency is very much a function of the fluids used. The gas fired absorption cycle heat pump benefits from having the fuel combustion take place on-site just as a thermal engine heat pump does, but the former is much simpler insofar as lack of reciprocating or rotating machinery is concerned. The on-site waste heat is available for heat pump supplementation and evaporator defrost, this aspect being important not only from the standpoint of efficiency and reliability but because refrigerant flow reversal is not as easily handled as with Rankine cycle machines.

5.2.2 Disadvantages

Absorption cycle heat pumps also have a number of disadvantages:

- Comparatively large heat exchanger areas (at attendant high first cost) required for realization of acceptable COP's;
- Usually lower cooling mode COP's than obtainable with Rankine refrigeration machinery (including electric heat pumps);
- Toxicity of ammonia requires that direct refrigerant/conditioned air contact in aqua-ammonia systems must be avoided (necessitating the use of either a water loop or other intermediate means of heat transfer);
- Corrosiveness of aqua-ammonia systems requires the use of steel rather than aluminum or copper as material of construction; the organic absorption system uses aluminum;
- Diameter limitation of generator pressure vessel requires use of a tall unit for adequate vapor capacity with attendant bulkiness of equipment;

- "Plumbing" of absorption machines, although obviously dependent on the particular design, appears to be more complex than that of a Rankine cycle machine;
- Dependence on availability of scarce fossil fuels (e.g. natural gas).

A problem that needs mention is the fact that initial experience with aqua-ammonia absorption-cycle gas air conditioners has been rather dismal and their market share, since their highest penetration (about 4%) of the residential unitary air conditioning market, has decreased steadily. Cost and equipment reliability problems have been the cause, so that only one of the original three unitary gas air conditioner manufacturers remains in the market today (Columbia Gas, 1976; Phillips Engineering, 1976). As is the thermal engine gas heat pump, the absorption cycle machine is dependent on specific and scarce fuels (i.e. natural gas). In comparison to the available alternative of electrically driven systems, new installations are liable to increase consumption of and dependence on these fuels. As before, (see Section 4.2) the efficiency advantage over an electric system will vanish if coal is to be the source of the fuel gas.

5.3 Description of Some Absorption Cycle Heat Pumps

Having discussed some general advantages and disadvantages of absorption cycle heat pumps, two specific embodiments of these will be described for purposes of illustration. Absorption heat pumps to be discussed are

- Organic fluid absorption heat pump
- Aqua-ammonia absorption heat pump.

5.3.1 Organic Fluid Absorption Heat Pump

A nominal 36,000 Btuh (3 ton) cooling capacity absorption heat pump utilizing a proprietary fluorocarbon refrigerant and aromatic hydrocarbon absorbent is under development by Allied Chemical Corp. under sponsorship of AGA. The system is water cooled and has a nominal rated heating capacity of about 87,000 Btuh. Both residential and commercial applications are contemplated.

In spite of various attempts, we have been unable to obtain enough information about this heat pump from the developers in order to identify potential problem areas or compare the anticipated advantages of this technology to other systems. According to the information available from a recent paper* (Colosimo, 1976), the non-corrosive nature of the fluids enables the hardware to be of aluminum construction (with attendant lower costs in comparison to stainless steel). In addition, it is claimed that the thermodynamic properties of the fluids permit more internal heat exchange and higher heat utilization than is possible with ammonia-water systems.

Exhibit 5-1, taken from the Colosimo paper,* shows a single-effect machine with absorber and generator heat exchangers, a separator heat exchanger and a pre-cooler. Three fluid circulation pumps are also shown.

5.3.2 Aqua-Ammonia Absorption Heat Pump

Another gas direct-fired absorption cycle heat pump is being developed in a 36,000 Btuh (3 ton) nominal cooling capacity** configuration by

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* According to Colosimo, the flow diagram does not correspond exactly to the prototype configuration as it stands now. However, no details are available to update the figure.

** Heating capacity data are not available at this time.

Columbia Gas System Service Corp. The device utilizes a proprietary multi-component aqua-ammonia base absorbent-refrigerant system and is undergoing performance testing and development in commercially available hardware.

The gas fired inorganic fluid absorption heat pump development follows an apparently technically successful but commercially unsuccessful aqua-ammonia gas air conditioner (the COLUMBIA 360) and combination air conditioner modular gas furnace (the COLUMBIA 360H) development program initiated in 1970, now since terminated. The heat pump development program is in the fluid performance testing stage, the testing being done in available aqua-ammonia air conditioner hardware at this stage of the project. No extensive performance mapping has been undertaken, the design cooling coefficient of performance of 0.85 representing a development target rather than an achieved result (at least as of mid-October 1976). However, considerable in-house technology on absorption systems is available from two previous gas air conditioner development programs (one of which we mentioned above).

5.4 Summary

As with thermal engine heat pumps, it is likely that manufacturing and market economics and gas availability, rather than performance, will determine the future of the absorption cycle heat pump. Traditionally, absorption machines have been deficient in cooling mode COP in comparison to electric air conditioners. It is possible to improve the cooling mode performance dramatically by using a multi-effect machine, but at probably prohibitive first cost. Absorption machines are clearly capable of high

heating mode efficiency and high reliability. From the standpoint of system simplicity and potentially maintenance-free operation, they might even be preferred to the generally more efficient thermal engine heat pumps for residential applications. The organic fluid absorption heat pump is certainly rather well along in its development path, as far as available information indicates. A limited field test of the prototype should yield more information on the technical feasibility of the concept.

References to Chapter 5

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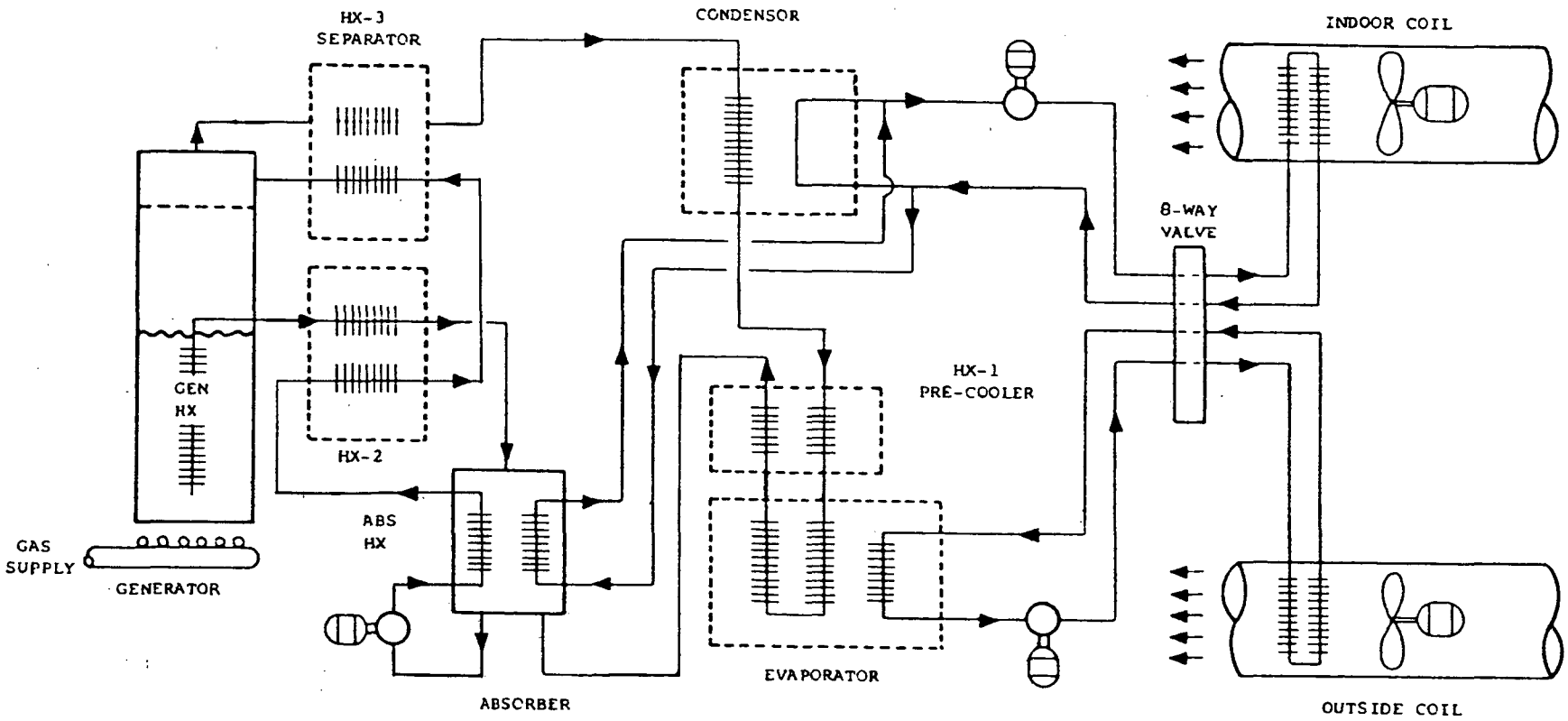
Phillips, B.A. "Absorption Cycles for Air-Cooled Solar Air-Conditioning" Paper presented at ASHRAE meeting, Dallas, Texas, February 1-5, 1976a.

Phillips, B.A., Personal communication, December 15, 1976b.

Wurm, J. "An Assessment of Selected Heat Pump Systems" Project HC-4-20 for the American Gas Association, March 1974.

EXHIBIT 5-1

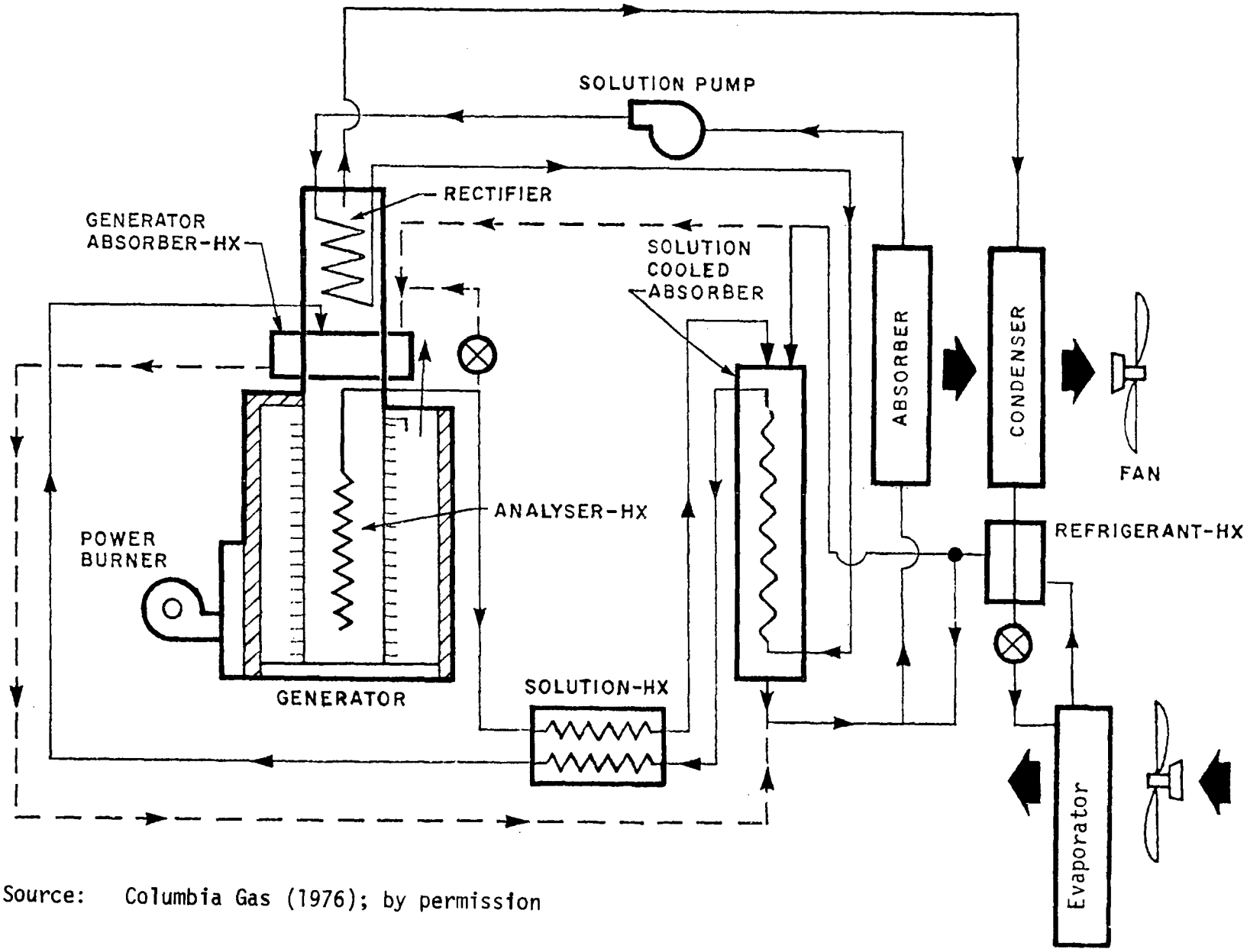
ORGANIC FLUID ABSORPTION CYCLE HEAT PUMP-HEATING MODE



Source: Colosimo (1976); by permission

EXHIBIT 5-2

GENERATOR-ABSORBER CYCLE SCHEMATIC



Source: Columbia Gas (1976); by permission

6. OTHER HEAT PUMPS

6.1 Introduction

Besides the electric, thermal engine and absorption cycle heat pumps described in the previous chapters, other heat pumping devices have been devised or proposed. Since these are either in the very early experimental stage or, while technically feasible, appear to be uneconomical for space conditioning applications, only brief descriptions of selected systems will be presented in this report. Among the systems discussed will be those utilizing the Peltier effect and several processes involving the heat of adsorption or absorption accompanied by a reversible chemical reaction.

6.2 Description of Selected Heat Pump Concepts

6.2.1 Peltier Effect Heat Pumps

The Peltier effect, the reverse of the thermocouple effect, is the arising of a temperature gradient when electric current flows through two dissimilar materials thermally and electrically joined. As far as electric heat pumps are concerned, Peltier heat pumps are inherently the simplest and most reliable of all, consisting entirely of solid materials and requiring no moving parts or working fluids. The theory of the Peltier heat pump or refrigerator has been described in the ASHRAE Handbook of Fundamentals (1972) and will not be discussed here. However, some considerations in developing potentially practical equipment will be summarized below.

First of all, it should be realized that the magnitude of the Peltier effect in most materials is relatively small depending, quantitatively,

on the relative magnitude of the Seebeck coefficient, and the thermal conductivity of the materials involved*. A high Seebeck coefficient implies a large temperature gradient for a given voltage drop; a low thermal conductivity is desirable since thermal conduction from the hot to the cold end opposes the gradient set up by the potential difference. However, it is difficult to achieve desired extreme values of both in real materials, since materials with a high Seebeck coefficient are also good thermal conductors. In this respect, semiconductor materials are preferred over metals.

Although the Peltier effect is used in cooling of electronic circuits, no space conditioning applications of it have been made, as far as we know. It is true that Siemens (in West Germany) several years ago conducted research in thermoelectric air conditioners for potential room use, but this program, according to a representative of the organization (Siemens, 1977), was discontinued due to their projected high first cost. Thermoelectric elements or aggregates are still being manufactured by Siemens and a few other firms in Europe. According to Fikri (1975), a research project in Peltier effect heat pumps is also underway in Sweden; however, no details are available.

From all aspects with the exception of first cost, a Peltier effect heat pump would seem very attractive. But the probability of developing a cost-competitive Peltier system appears rather remote at this time, unless there is a dramatic breakthrough in materials.

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* Additional phenomena arising out of the effect are production of Joule (i^2R) heat and the second order Thompson effect.

6.2.2 Chemical Heat Pumps

Also suitable for heat pumping is any reversible chemical reaction with a sufficiently high latent heat. For example, processes that have been studied include:

- Hydration of magnesium chloride dihydrate to a higher hydrated state in the presence of water vapor. The process is accompanied by the release of heat of chemisorption. Dehydration is accomplished by use of solar energy. (Chemical Energy Specialists, 1976);
- Other metal salt hydration-dehydration reactions;
- Absorption-desorption of water vapor from concentrated sulfuric acid solution (accompanied by a very high temperature change).

6.3 Summary

It is not particularly instructive to catalogue all, or even a representative selection, of the various chemical or physical phenomena theoretically suitable for heat pumping. Clearly any such phenomenon accompanied by a temperature change (increase for heating; decrease for cooling) could be used. However, at the present time it appears that Rankine cycle refrigeration machines are among the most efficient and practical (although there are strong advocates of Brayton and other cycles as well).

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7. HEAT PUMPS AND SOLAR ENERGY

7.1 Introduction

A subject not yet discussed, but one that is getting major attention by the industry and elsewhere is the role of heat pumps in utilization of solar energy for space heating and cooling. Although the scope of this investigation did not permit a comprehensive review of the extensive ongoing work in demonstrating solar heating and cooling systems using heat pumps, developing solar assisted or supplemented heat pumps or even solar engine-driven or absorption cycle heat pumps, a brief treatment of these topics is nonetheless necessary to place our assessment of the future direction of heat pump technology in proper perspective. Solar heat pump technology surveyed herein includes solar supplemented, solar assisted and solar energized systems.

7.2 Solar Supplemented Heat Pumps

Solar supplemented or "boosted" heat pumps are defined for the purpose of this report as "conventional" heat pumps (electrically driven -- but gas-fired configurations are also feasible) which use solar heat for supplementing refrigeration-cycle heat by a solar energy system heat exchanger directly in the conditioned air (or water) circulation loop. This definition implies that solar supplementation of heat pumps is applicable in the heating mode only, since, obviously, no supplemental heat is required (or even desired) in the cooling mode.

A solar-supplemented heat pump system, although probably not yet cost-competitive with a conventional heat pump or other heating system, offers the owner (as well as the manufacturer, it would seem) a number of advantages. First, the heat pump is frequently basically a conventional

air-source heat pump with a solar hydronic heat exchanger in the air handler in place of the resistance heating elements.* This makes the heat pump and solar heating system essentially independent of each other and allows the former to operate on ambient air as a heat source, when that would be more efficient. On the cooling cycle, the arrangement allows the heat pump to reject heat into the atmosphere directly, without going through a water-air heat exchanger system.

The benefit to the manufacturer is obvious: he can sell his conventional electric heat pump equipment with only a few modifications to it (except, perhaps, having to offer additional components such as storage tanks, a water-antifreeze circulation system, solar collectors, etc.) and thus participate in a segment of the HVAC market which is the focus of much public attention in these days.**

7.3 Solar Assisted Heat Pumps

Solar assisted heat pumps may be defined as "conventional" *** heat pumps utilizing solar energy as a heat source. By definition, then, solar assistance of heat pumps is possible only in the heat pump heating mode, since in the cooling mode the conditioned space itself is the heat source.

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* Resistance heat as a backup is usually still provided, however, with the heating elements (immersion heaters) being installed in the water thermal storage tank instead.

** In addition, any tax or other financial incentives offered by the federal or state governments would also accrue to the purchaser of a solar-supplemented heat pump.

*** The distinction between solar "assisted" and solar "supplemented" heat pumps, made in this report, is not usually made in the technical literature. Indeed, heat pump systems using evaporator side storage or a solar heating coil for supplemental heat are both sometimes referred to as solar assisted heat pumps. The distinction, nevertheless, is useful for purposes of clarity.

A number of configurations of heat pumps with solar assistance are, of course, possible, most potentially practical ones usually involving thermal energy storage of one type or another. A common and technically rather attractive concept is evaporator-side storage. However, condenser or "hot-side" storage is possible as well as are configurations where more complex series and parallel circuiting of heat pump and solar loops are encountered. Evaporator-side storage is described in somewhat greater detail below.

7.3.1 Heat Pumps with an Evaporator-Side Heat Source

The concept of combining a solar energy collector and storage system with a heat pump is not new. Jordan and Threlkeld (1954) proposed in the early fifties that using stored solar thermal energy as a heat source for the evaporator would allow the heat pump to operate at higher COP than if on ambient air. At the same time, using a heat pump to "step up" distribution air temperature to comfort levels would enable the solar (usually flat plate) collectors to operate at higher average collection efficiency, since the collector inlet fluid (and thus the collector cover-glass and absorber plate) temperature would be lower. * On the other hand, "draw down" of the thermal storage to an extent where the evaporator is forced to operate in a low temperature environment could severely deterioriate the system COP. Hence, ample storage and a means (e.g. by supplemental resistance heating elements or a fossil fuel fired heater) of maintaining thermal storage system temperature above a minimum is required.

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* Non-circulating solar collector efficiency η as determined from experimental measurements in solar simulators, say, is usually a monotonically decreasing function of the paramerer $(T_{in} - T_{air})/I$ where I is the insolation and the meaning of the other symbols is self-evident. For more details, see for example, Jordan and Liu (1977).

A water-source heat pump would seem to have obvious advantages in solar-assisted applications in that the water coil is readily incorporated with a water-tank solar thermal storage unit. Indeed, a number of solar heating system designs incorporating water source heat pumps utilizing water source equipment from Singer, York, Carrier and several other manufacturers * have been reported in the literature (see, for example, Barkmann et al.,1976) . On the other hand, novel solar assisted air source heat pump configurations have also been described -- some using part of a solar-heated attic space as the source (Bourne, 1976), some using a rock bed storage system. In the latter case, the advantage of temperature stratification is claimed to overcome the lower collection efficiency of an air-type solar collector (Löf, 1977). These advantages may well hold true in a solar air-source heat pump application where the source is a rock bed.

7.4 Solar Energized Heat Pumps

By a solar energized heat pump is meant a heat pump where solar energy is used as the "prime" energy source of the heat pump system. This may be accomplished in one of several ways: (a) solar thermal energy could be used to evaporate or expand the working fluid of a prime mover which in turn is used to drive a heat pump or "refrigeration cycle" machine or (b) solar heat could be used to vaporize the refrigerant in the generator of an absorption cycle machine. Experimental work using both of these concepts is known to be ongoing, although no hardware has as yet reached the prototype stage, as far as is known.

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* Some of the system designs are admittedly of an experimental type or may utilize pre-production equipment.

The gas-fired free-piston Stirling engine developed by Beale (see Chapter 4) is also capable of using solar energy (Beale, 1975). A version is under development where instead of combustion gas passages in the "hot" end of the engine a quartz Fresnel lens would concentrate and transmit solar energy from a concentrating collector onto the outer surface of the "hot end."

A number of experimental solar energized Rankine cycle engines typically utilizing fluorocarbons as a working fluid are known to be operating as demonstration devices or undergoing development. An example is the rotary pivoting vane expander-compressor unit under development at Battelle Columbus Laboratories (Eibling and Frieling, 1975). A relatively recent review of solar Rankine cycle and absorption cooling machines was written by Curran (1975). An excellent article on the potential of air-cooled aqua-ammonia based solar cooling cycles was written by Phillips (1976).

As a note of interest in this connection, Arkla Industries has recently introduced a 25 ton lithium bromide-water absorption chiller able to operate on solar-heated hot water at 195°F. (Arkla Industries, 1977a). If the water temperature drops to 160°F, the capacity is reduced to 7.4 tons. A 3 ton solar air conditioner is also available having the rated capacity at a water inlet temperature of 195°F (Arkla Industries, 1977b). Lithium bromide absorption machines, while capable of "pumping heat" in principle, cannot do so over the whole temperature range which is of practical interest in heating service due to water freezing in the evaporator. However, from a solar system implementation viewpoint a practical cooling machine operable on the output-temperature of an efficient

non-concentrating solar collector is quite significant. (As solar energy may be easily used for heating directly, any means for running a cooling machine on solar thermal energy is an advantage, at least from a technical viewpoint.)

7.5 Summary

Heat pumps, being devices for elevating the temperature of basically low-temperature, low-grade heat sources, add great flexibility to practical solar heating system design, at the same time providing a needed cooling function. While a substantial amount of research is still needed to develop practical systems utilizing solar energy as the principal energy source for heat pumping (i.e. systems we have called "solar energized " heat pumps), solar supplemented and solar assisted heat pumps can utilize current conventional heat pump technology with only minor modifications necessary to accommodate additional components of the solar collection and storage system. Indeed a practical solar system of the next decades is, in our judgement, more likely to be one utilizing basically proven, conventional heat pump technology rather than some of the more exotic chemical and other heat pump cycles or solar-powered engines being researched. The requirement of reliable relatively maintenance-free operation is likely to be a major factor in dictating the allowable complexity of a system for widespread residential and light-commercial use.

An emerging important factor to be considered (especially in view of demand charges for residential customers) in connection with solar heat pump technology is the potential effect on electric (or gas) utility annual load factors. As will be discussed in a later chapter, without

provision for sufficient solar energy storage of some type, solar energy systems are likely to have a significant deteriorating effect on utility load characteristics (solar energized systems on the cooling cycle may improve the summer load factor instead).

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8. HEAT PUMP PERFORMANCE IN RESIDENTIAL AND COMMERCIAL APPLICATIONS

8.1 Introduction

The principal topic of this chapter is a comparison of energy usage by conventional and advanced heating and cooling systems for residential and commercial space conditioning. The comparison of energy consumption is based on computer simulations of 21 HVAC systems, 16 residential and 5 commercial. Heating season, cooling season and total annual energy consumption for each of these systems are determined in terms of the quantities of electric and/or fossil fuel energy consumed onsite, the thermal equivalent of these energies and system performance factors. Because of its importance in interpreting calculated system performance data some attention will be paid here to describing the operating characteristics of the systems modelled as well as to the computer simulation methods used to develop the energy consumption data.

8.2 Heat Pump Performance in Residential Applications

8.2.1 Methodology

The calculation of heating and cooling energy requirements to maintain residential comfort conditions was performed with the aid of two computer programs. The first is a load program which estimates hourly heating and cooling loads for a residence. The second program simulates the performance of heating and cooling systems and determines the energy input to the systems just needed to balance hourly loads on the residence.

In order to properly compare energy requirements for different heating and cooling systems at a specific physical location, it is important that identical loads are satisfied by the different HVAC

systems examined. This can be accomplished most easily by selecting a single residential structure for any specific site and calculating a single set of heating and cooling loads based on actual local weather conditions. The systems simulation program will then produce estimates of energy input to each HVAC system in direct response to identical loads. Thus the relative energy efficiency of different systems can be compared in a straightforward manner. If it is also necessary to determine how HVAC energy requirements change with climate then it is equally important that the same residential structure be used to generate heating and cooling loads at each different climatological location. The method used, therefore, involves the selection of a single residential structure, or test house, and, along with local weather data for a number of different locations in the U.S., generating hourly heating and cooling load data. For HVAC systems of interest in this study, manufacturers' performance data were then used in the systems program and heating and cooling energies calculated for an entire year.

Computerized residential load calculations and systems energy simulations were performed on the test house in nine different cities within the continental U.S. The selection of the nine locations was based on the need to represent as completely as possible the large climatological variations existing in the country and the availability of detailed weather data at the sites selected. Exhibit 8-1 lists the cities finally chosen as representative of U.S. climatological variations as characterized by heating degree days. Seattle was chosen because of its rather anomalous climatic conditions. Although it has a similar number of heating degree days as Philadelphia, its summers are much

milder; these conditions are taken to represent the climatic conditions of the Pacific Northwest.

Local weather data for each of the nine test cities were selected from among extensive meteorological data available from a U.S. weather station maintained in each city. The year of weather data chosen for each city was representative of long term average weather conditions at each site. The method used to select a year of typical weather data for each city was based upon a modified version of the procedure recommended by ASHRAE (1973). The procedure requires that the year of weather data to be chosen is that year of 12 consecutive months having the smallest deviation of heating degree days on a month-by-month basis from long term monthly average heating degree days.

8.2.2 Residential Building Selection and Design Load Calculation

The test house selected for computer simulation of residential heating and cooling loads was a single family detached two story residence. This residence is an existing structure located in the suburbs of Columbus, Ohio. It was chosen for this study principally because it is serving in an ongoing study of residential load and system simulation modelling under the guidance of research staff at the Ohio State University. Research results obtained with this house have directly contributed to the load and systems simulation programs used in this study. In fact, the results have served to validate the modelling procedures used.

The test house is occupied by a family of four, two adults and two children. Detailed data on internally generated loads due to human activity and appliance usage have been measured over a period of several years and incorporated in this study. Further, the Columbus test house

is of relatively new construction and incorporates modern construction features and insulation standards.

Front elevation and floor plans of the test house are shown in Exhibit 8-2 and general construction features tabulated in Exhibit 8-3. Insulation used in the test house thermal load program and heat transfer characteristics of opaque surfaces are shown in Exhibit 8-4 along with a comparison to ASHRAE 90-75 standards. Air infiltration rates around exterior doors, windows and sliding doors are shown in Exhibit 8-5 and compared with ASHRAE 90-75 standards.

Design heating and cooling loads to be used for heating and cooling equipment selection were determined using the load simulation program, with design weather data in lieu of NOAA weather data. Design heating and cooling loads estimated in this manner are shown in Exhibit 8-6.

8.2.3 Heating and Cooling System Selection

A total of 16 heating and cooling systems were selected for the examination of energy usage for residential comfort conditioning. These systems were divided into two broad classifications: conventional heating and cooling systems representing equipment more or less extensively used in residential applications and new technology advanced heating and cooling systems currently being developed.

Conventional Heating and Cooling Sytems

Within the classification of conventional heating and cooling systems, six represent electric heat pumps while four are fossil fuel fired warm air furnaces or electric resistance heaters in combination with electric air conditioners. A more detailed categorization of conventional equipment examined is shown below. Each designation indicates a single representative system unless otherwise indicated.

Conventional Heating and Cooling Systems

- Split system electric heat pumps
  - o Standard electric heat pump (moderate COP/EER)
  - o High efficiency electric heat pumps (3 different systems) (high COP/EER)

- Add-on split system electric heat pump
  - o Hybrid electric heat pump added onto gas or oil fired central warm air furnaces
  - o Heat only electric heat pump added onto a central electric furnace

- Fossil fuel fired warm air furnaces
  - o Gas fired
  - o Oil fired

- Electric resistance heating
  - o Central electric furnace
  - o Baseboard convectors

- Mechanical air conditioners used in conjunction with the heat only heat pump, fossil fuel furnaces and electric resistance heating.

Some operating characteristics of these systems will be found in Exhibit 8-7.

The basic component of the split system electric heat pump is an electrically driven reciprocating Rankine cycle compressor using a fluorocarbon refrigerant as working fluid. The standard for this group is a split system heat pump with a heating COP of 2.3-2.6 (47°F rating) and an EER of 6.8-7.1 for cooling (95°F rating). This heat pump, with minor modifications, has been commercially available from several manufacturers for about two decades. The high efficiency split system heat pumps represent rather new improvements of the standard system (more efficient compressors, larger heat exchangers, lower balance point, new defrost control) and have become available only within the last two or three years. Three different manufacturers' models of high efficiency split

electric heat pump systems were examined to determine if any significant performance differences for these new systems could be discerned. As a group, the high efficiency systems exhibit increased heating COP's to 2.7-3.1 (47°F) and cooling EER's to 7.7-8.4 (95°F) or approximately 15-25% increased efficiency for heating and 15-20% increased efficiency for cooling at rating points compared with the standard electric heat pump.

The hybrid electric heat pump has the basic configuration of the standard split system electric heat pump and provides both heating and cooling capability. In very cold weather, when the load on the residence exceeds the capacity of the heat pump, the heat pump is shut down and an oil or gas fired warm air furnace is used to satisfy the load. Thus, not only supplementary heat is supplied by the fossil fuel fired furnace during very cold weather, but also the baseload. When the heating load for the residence is satisfied the furnace is shut down. The control system is designed so that the heat pump is always started up first, even below the balance point, and furnace heat is called for only when the heat pump cannot meet the load on the residence. The gas or oil furnace is sized so as to be capable of satisfying the design heating load without the support of the heat pump.

The heat-only electric heat pump is an add-on system that can be used in conjunction with fossil fuel fired furnaces or, more frequently, with central electric warm air furnaces. It is identical in operation to the split system electric heat pump with the exception that there is no reversing valve to allow the system to operate in the air conditioning mode. Because the heat-only electric heat pump has no air conditioning capability it uses hot compressor discharge gas for defrost control

rather than hot liquid as do conventional electric heat pumps. In this way there is some capacity for residential heating during defrost. Supplemental heat is provided by the central electric furnace below the balance point. The heat-only heat pump has an oversized compressor, relative to compressor heating capacity for standard electric heat pumps sized to meet an air conditioning design load. Compared to the high efficiency electric heat pumps, the heat-only heat pump is similar in rated heating capacity at 47°F but somewhat higher in heating COP.

With the exception of the heat-only electric heat pump, heat pump equipment was selected based on the design total cooling load for the test house in each of the test cities with proper consideration given to the design latent load. Manufacturers' published performance data were used to select the closest available off-the-shelf equipment for each case. Supplemental electrical resistance heaters were sized for 80% of design heating load in all cases.

Other conventional heating systems are represented by oil and gas fossil fuel fired furnaces and electric resistance heating. In each case, manufacturers' off-the shelf equipment were selected with maximum heating capacities equal to or greater than the design heating load for the test house. In order to properly compare annual energy usage for residential heating and cooling, electric central air conditioning systems were added to the oil, gas and central electric furnaces. Two central air conditioners were examined; a high efficiency unit (COP's of 7.1-7.8 at 95°F) generally selected by individual homeowners and a standard unit (COP's of 6.2-6.8 at 95°F) most usually selected by builders as original equipment for developments. All of the final results of

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* Resistance elements for heating only heat pump are sized for 100% design heating load.

energy usage are reported for the high efficiency central air conditioner although a brief comparison with the standard unit is made later.

For the test house heated with baseboard convectors, room air conditioners were selected as cooling equipment. The entire first floor was assigned two large air conditioners, one in the family room and another in the living room. The two largest bedrooms were assigned two smaller room air conditioners. Capacities of the room air conditioners were estimated by apportioning the design total cooling load among the rooms to be conditioned on the basis of relative floor area. The air conditioner selected were of the high-EER type.

Advanced Heating and Cooling Systems

Six different advanced systems were examined, four by computer simulation and two by scaling calculated energy consumption data for similar systems on a city-by-city basis. A classification of the advanced systems is shown below:

Advanced Heating and Cooling Systems

- Advanced electric heat pumps (2 systems)
- Thermally activated heat pumps
  - o Free piston Stirling-Rankine gas heat pump
  - o V-type single cylinder Stirling-Rankine gas heat pump
  - o Organic liquid absorption gas heat pump
- Pulse combustion gas furnace

A seventh system, an Ericsson-Ericsson cycle gas heat pump, was examined with results reported in Appendix A.

One of the advanced electric heat pumps examined is based on a new proprietary design for which full details cannot be reported here. This

new system is designed for increased heating performance in cold climates (COP of 2.7 at 47°F) with a lower balance point than conventional electric heat pumps, and requires no defrosting or supplemental heating. Cooling performance is reported to be about equal to standard electric heat pumps.

The advanced electric heat pump briefly described above was the only electric system for which adequate performance data were available for simulation. In order to explore the performance anticipated for new technology advanced electric heat pumps, energy consumption data were estimated for a hypothetical advanced electric system. The estimation method will be described later, but this heat pump was assumed to have a high performance capability based upon a number of modifications currently being explored by equipment manufacturers. Examples of these modifications include the use of fossil-fuel-independent energy sources (such as solar energy) and increased low temperature heating capacity brought about by series or parallel compression or continuous capacity modulation.

Three heat activated heat pumps were examined within this group of advanced systems. Two represent Stirling-Rankine cycle gas heat pumps and the third, a gas fired absorption system heat pump. One of the Stirling-Rankine systems employs a free piston Stirling cycle engine driving a Rankine cycle heat pump. In its present configuration the system operates cyclically, starting up and shutting down in response to an indoor thermostat.* The second Stirling-Rankine system consists of a V-type single cylinder Stirling engine driving a reciprocating Rankine cycle heat pump compressor. This prime mover can be modulated however and the capacity of the heat pump can therefore be modulated to respond

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* However, nothing in the free piston Stirling-Rankine heat pump concept would prevent it from being designed as a capacity-modulating system.

to load changes. Little is known about the configuration of the gas fired absorption heat pump except that the working fluid in the heat pump cycle is an organic fluid of proprietary composition. Characteristic of all three heat activated heat pumps is their high heating capacity rating (see Exhibit 8-7) compared with conventional electric heat pumps of identical cooling rating.

One additional heating system included in this group was a pulse combustion furnace. Energy consumption for residential heating with this system was not simulated however, but scaled from conventional gas furnace energy consumption calculation assuming a 92.5% combustion efficiency (Hollowell, 1976).

8.2.4 System Simulation and Computer Modelling

The residential building HVAC system computer simulations were accomplished in two parts: a load simulation and a system simulation. Since these models are described in an appendix to this study only a brief description will be presented here

The load calculation model determines essentially how much heat is required to be added or removed from the control volume (the test building) to maintain indoor dry bulb conditions of 70 or 75°F (winter and summer, respectively) and 50% relative humidity. The program performs a heat balance on the control volume, taking into account such factors as heat transfer through windows and walls, insolation, infiltration of outside air, and internally generated heat from the activity of occupants.

The load program is a modification of the Post Office Program, developed by General American Transportation Corporation for the U.S.

Postal Service. The modifications were made to adapt the program for use with residential structures. Additionally, new calculational routines were included, representing new developments in thermal load modelling. As input, the program requires various parameters, such as building construction details and materials, orientation, geographic location, and local weather conditions. The latter information was obtained from NOAA in the form of a magnetic tape containing hourly data for an entire year, for each of the nine test cities.

The output of the load simulation program consists of hourly sensible and latent load data, which are used in the systems simulation program. In general, the heating or cooling capacity of the systems is dependent on outdoor temperature. System operating parameters were obtained from manufacturers or developers and were fitted to mathematical functions of outdoor temperature. System energy consumption was determined hourly using the load data, weather data and system operating data. Energy consumption data included contributions from all system components (fans, blowers, pumps, etc.). The output from the systems simulation programs consists of a monthly breakdown of energy consumption by fuel type (electric energy, natural gas, fuel oil) for the entire year.

The load program is identical to that used in previous Gordian Associates studies; the systems simulation program has been substantially revised and upgraded, as new developments in space conditioning system performance modelling have become available. Some of these changes are discussed below.

Air Conditioning Simulation

Summer air conditioning involves two simultaneously occurring

processes: the reduction of both air temperature and water content (humidity) in the conditioned space. As warm, humid air passes over the indoor (evaporator) coil, the dry bulb temperature of the air is reduced. If the air is sufficiently humid and the evaporator coil sufficiently cold, some of the air passing over the coil becomes saturated and water vapor condenses on the coil. Hence, outlet air is cooler and drier than inlet air.

Equipment manufacturers list both sensible and total cooling capacities for air conditioners, both as functions of indoor wet bulb and indoor and outdoor dry bulb temperatures. Sensible capacity is a measure of the extent to which the air may be cooled. The difference between total and sensible capacity is a measure of the maximum extent to which entering air can be dehumidified (the latent capacity). If the full latent capacity is not utilized, air may be cooled in excess of the rated sensible capacity.

The cooling subroutine of the systems simulation program first compares the latent load calculated in the load simulation program with the latent capacity of the system. The equipment capacity is determined hourly from manufacturers' data using the outdoor dry bulb temperature for that hour, and set indoor conditions of 75°F dry bulb temperature and 50% relative humidity. If the latent load equals or exceeds the latent capacity of the air conditioner, then the effective sensible capacity is taken to be the manufacturers' minimum rating. If the latent load is less than the latent capacity, the excess latent capacity is added to the minimum sensible capacity. In this way, the total capacity of the system remains constant while the effective sensible capacity is a variable parameter dependent on the latent load imposed on

the residence. In no case, however, is the effective sensible capacity allowed to exceed the total capacity of the air conditioner or be less than the rated sensible capacity. Although recent information (General Electric, 1977) indicates that this procedure is just now being considered and implemented in systems simulation, to our knowledge, this method is generally not used. Instead, the manufacturers' reported sensible capacity is used almost exclusively.

The value of this effective sensible capacity method lies in the fact that in actual operation the air conditioner is responsive to the indoor thermostat which, in turn, responds only to changes in indoor dry bulb temperature (or sensible load). Thus, the length of time the air conditioner operates is strongly dependent on the temperature within the residence and the sensible capacity of the equipment. The fraction of an hour that the air conditioner will operate may be estimated as the ratio of sensible load for an hour and system effective sensible capacity, ignoring heat gains through ductwork. The greater the effective sensible capacity of the system, the shorter the time on to satisfy comfort conditions. Therefore, energy consumption for that hour will also be less.

Manufacturers' published sensible and latent capacity data on air conditioners were used in the sizing procedure. If the maximum latent capacity of the equipment was insufficient to meet the design latent load, a larger size unit was selected, even though the total capacity was sufficient to satisfy the total design load. In this way, equipment was selected to insure that sufficient capacity was available for both sensible cooling and dehumidification.

Insulated Ductwork

It is common practice among builders and installers of electric heating systems (i.e. central electric warm air furnaces and electric heat pumps) to insulate the supply and return ducts attached to the plenum. Gas or oil warm air furnace basement ducts are generally not insulated even when electrically operated central air conditioners are used except in very humid climates to prevent condensation in supply ducts carrying cooled air to the conditioned space.

In order to estimate system energy consumption on a common basis, it was decided to perform all of the system simulations assuming one-inch of fiberglass insulation in all supply and return ducts in the basement. In addition, several of the conventional systems were simulated assuming no duct insulation for the purpose of determining the effect of this insulation on system performance. These results are discussed elsewhere in this chapter.

Defrost Simulation

When frost accumulates on the outdoor evaporator coil, the heating capacity of a heat pump decreases. In order to maintain capacity, the outdoor coil must be defrosted. Coil defrosting is usually accomplished by reversing the heat pump cycle, causing hot gas from the compressor to enter the coil which now acts as a condenser. An alternate method of defrosting, used in the heat only heat pump, for which cycle reversal is not possible, is to bypass hot gas intended for the indoor condenser to the outdoor evaporator. In the cycle reversal method the heat pump acts as an air conditioner and cools the house instead of heating it, and a

supplementary source of heat is used to temper the air to maintain comfort conditions. In the hot gas bypass method, some heat reaches the indoor coil but supplementary heating is still required. Because defrosting represents an additional energy requirement for heat pump heating, it was routinely modelled as part of the systems simulation.

Until recently, many heat pump manufacturers employed a pressure differential switch to initiate the defrost cycle. This control unit operates by sensing a reduced air flow through the outdoor coil as frost buildup clogs air flow passages, and calls for defrost at a preset pressure drop. Because new, more efficient outdoor coil heat exchangers employ a low pressure drop configuration, the pressure method of defrost initiation is not satisfactory. Almost all manufacturers now employ a timed cycle coupled with temperature measurement of the refrigerant in the outdoor coil. In order to initiate a defrost cycle, a preset time period must have elapsed since the last defrost cycle and a refrigerant temperature condition in the outdoor coil must be met. Defrost terminates when the refrigerant temperature rises sufficiently to indicate that ice has been removed from the coil. However, a time limit is placed on the length of the defrost cycle.

The defrost simulation subroutine does not model the heat pump refrigerant circuit in sufficient detail to use the temperature condition of the outdoor coil for defrost initiation and termination. An alternate method is used based principally on a timed cycle procedure. Discussions with heat pump manufacturers indicated that a 90 minute time interval, based on compressor running time, between defrost cycles is generally

employed. Although other time intervals can be preset at the factory or during field installation if local conditions warrant it, the 90 minute cycle is most often used. Further, observations on the duration of heat pump defrost cycles suggest that each cycle lasts, on the average, about 3 1/2 minutes. The defrost simulation was based on these considerations in place of simultaneous timing and temperature control.

The frequency of defrost is a complicated function of local weather conditions (outdoor humidity and dry bulb temperature) and the configuration of the heat exchanger and the air flow over it. Information from heat pump manufacturers (General Electric, 1977; Reedy et al., 1976) indicates, however, that defrost cycles rarely occur when outdoor temperatures are above 47°F and coincident outdoor relative humidity below 60%. Hence, our simulation program accounts for a 3 1/2 minute defrost cycle at the end of each 90 minute compressor running time cycle if the outdoor dry bulb temperature is below 47°F and the relative humidity above 60%.

The defrost simulation method described above is used to account for supplementary heating during defrost periods. For systems with single stage electric resistance supplemental heat, initiating a defrost cycle caused the resistance element to be energized. When defrost is initiated with two stages of supplemental resistance heating, the second stage is energized providing the first stage is already in operation. For the hybrid electric heat pumps, defrost initiation automatically calls for supplementary heat from a gas or oil fired furnace.

When a heat pump operates under ambient conditions conducive to frost formation on the outdoor heat exchanger (evaporator), the frost layer after awhile begins to reduce the rate of heat transfer to the

evaporator. Thus the heating capacity of the heat pump decreases until a defrost cycle is initiated. During the defrost cycle the heat pump actually extracts heat from the return air (i.e. operates at a negative heating capacity), this cooling effect being, of course, countered by the resistance heating elements which are energized during the defrost cycle.

The ASHRAE Standard 37-69 heat pump performance testing procedures used by most manufacturers for performance data acquisition specify that, when frost begins to form on the outdoor coil under certain air temperature and humidity conditions during the test, data for performance rating purposes are to be taken over a two-hour period including at least one complete defrost cycle. Hence, it would seem that manufacturers' "integrated" capacity data under ambient conditions at which frost may be expected to form include the effects of capacity degradation due to coil frosting and the cooling effect of the defrost cycle. However, the experimental work of Kelly and Bean (1976) and Parken et al. (1977) at the National Bureau of Standards, indicates that at least for the two heat pumps tested so far, a loss in capacity and COP is observed in excess of that supposedly "integrated" into manufacturers' performance data for that temperature and humidity region of operation. It should be stated that the observations cited were made on only one manufacturer's heat pump equipment. (Central air conditioners of another manufacturer have been measured as well, with apparently less severe cooling performance degradation results than in the case of the above heat pumps. To what extent this performance degradation phenomenon is applicable to other manufacturer's heat pump models is uncertain and will not be known until

experimental data on these units become available. However, comparison of utility field test data on heat pump performance in actual installations vs. seasonal performance factors computed on the basis of manufacturers' integrated data above (i.e. without accounting for defrost losses and the cycling losses described in the following section) suggests that the calculated seasonal performance factors are higher than those observed and can be brought into better general agreement if defrost losses are accounted for by the NBS data. Although this apparent better agreement does not, of course, constitute adequate evidence of the more "universal" applicability of the early NBS data, better information is unavailable at this time. Therefore, in this study the NBS data were used in all of the electric heat pump dynamic performance calculations to account for coil frosting and defrost. Since seasonal performance factors based on only the manufacturers' integrated data are also provided in the tabulated results, it is hoped that the true values of the seasonal performance factor are at least "bracketed" by the results given in this chapter.

Part-Load Operation Modelling

All HVAC systems simulations performed in this study are based on manufacturers' reported performance data. Such data are generally quite detailed for commercially available heating and cooling systems. Although, in most cases, less detailed information was available from developers of advanced HVAC systems, the data were sufficient for the modelling task.

Manufacturers' and developers' data, however, almost uniformly represent equipment performance on a steady-state or full-load basis, although it has been known for some time that fossil fuel fired warm air

furnaces suffer from losses due to such factors as off-cycle flue losses, outside combustion air infiltration and equipment cycling.

Recent studies of equipment performance have indicated that, in addition to frost buildup on outside coils, equipment cycling due to part-load operation can substantially reduce heat pump and air conditioner efficiencies. In order to present results for HVAC equipment more adequately, transient or part load and efficiency losses have been accounted for in the following manner for conventional heating and cooling systems:

> Heat pumps and central air conditioners: dynamic efficiency performance was modelled using data developed experimentally by Kelly and Bean (1976). Since no dynamic performance data have been published on central air conditioners, dynamic efficiency losses for central air conditioners were assumed to be the same as for heat pumps.
>
> Central gas furnace: experimental part load performance data developed by IGT (Macriss, 1975; Macriss *et al.*, 1977) were used in the modelling of central gas furnace.
>
> Central oil furnace: dynamic efficiency data were obtained from Honeywell, Inc. (Bonne, *et al.*, 1976; Macriss *et al.*, 1977). Data represent oil furnace model simulation calculations based on experimental studies of test furnaces in the Boston area.

The Kelly and Bean data were obtained experimentally for a test house in the Washington, D.C.area space conditioned with 5-ton single phase split system heat pump--a product similar to the heat pump we have termed "standard" in this report.* There is, of course, no evidence that heat pumps of other manufacturers (including the hybrid or heat-only heat pump) would experience identical dynamic or transient losses, although it is reasonable to expect that the losses would be similar.

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* The Parken data are based on environmental chamber tests on a 3-ton electric heat pump.

In the absence of experimental data specific to the other conventional electric heat pumps or air conditioners simulated, the Kelly and Bean heat pump data were used for all single speed electric vapor compression heat pumps or air conditioners investigated in the study.

No dynamic efficiency losses were assumed for the central electric furnace or baseboard convectors. Because of the lack of detailed operating information on advanced heat pumps currently under development, these systems were all modelled without penalty for dynamic efficiency losses. Results presented in the discussion below, however, estimate what new system performance would be if they were subject to the same kinds of losses as conventional electric heat pumps in the heating and cooling modes.

8.2.5 Analysis of Onsite Heat Pump Performance in Residential Applications

The results of the computerized simulation of heating and cooling systems are shown in three different ways in Exhibit 8-8 through Exhibit 8-15. Fuels and electric energy consumed by conventional and advanced HVAC systems to provide comfort conditioning for the test house in nine cities representing U.S. climatic conditions are shown in Exhibits 8-8 through 8-11. Seasonal performance factors, defined as net heating or cooling load divided by the sum of all forms of energy consumed by the system, are shown in Exhibits 8-12 and 8-13. The annual seasonal performance factors for each system shown in these tables are combined heating and cooling loads for an entire year divided by the annual energy consumption of the system. Finally, the total thermal equivalent of the fuels and electric energy consumed by each heating and cooling system is shown in Exhibits 8-14 and 8-15. The data in these last tables were produced by multiplying each form of energy consumed by its gross heating value or, in the case of electric energy, the thermal equivalent of a kilowatthour.*

The information listed in Exhibits 8-14 and 8-15 is presented graphically in Exhibits 8-16 through 8-19, changed, however, to a per square foot of residence basis. The graphical correlations shown are related to a well known method recommended by ASHRAE (1976) for the estimation of seasonal heating energy. The correlating parameter shown as the abcissa of the graphs, if multiplied by the identity 24 hours/day, is the equivalent of seasonal load, for either heating or cooling seasons,

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* Average heating values of fuels were used; 1000 Btu per cubic foot of gas and 139,000 Btu per gallon of fuel oil. For electric energy, 1 kWh equals 3413 Btu.

based upon design conditions and average heating or cooling degree days at a particular geographical location. These correlations are shown in greater detail in Appendix B for each HVAC system examined. The goodness of fit of the energy consumption data generated by the load and system simulation program with these correlating parameters is suggested as indicating the validity of the overall modelling techniques used. Also, the correlations indicate that a general linear relationship of the form

Seasonal energy consumption per = a + b(Seasonal design thermal load /sq.ft. of residence

is an appropriate function for the estimation of energy consumption by HVAC systems. The constants a and b are specific for a particular heating or cooling system but the relationship appears to be quite general otherwise, and can probably be used for any house at any geographical location for which a seasonal design thermal load can be estimated. The graphical relationships shown in Exhibits 8-16 through 8-19 are identical to those shown in greater detail in Appendix B except that on the former the data points have been left out for clarity. Also, a single line is shown for all three high efficiency heat pumps. This can be done because statistically there is no significant difference between individual correlations of these systems, allowing a pooled estimate to be made for the set. A single line is also shown for the central electric furnace and the baseboard convector systems because there is very little difference in heating season energy consumption between them.* Slightly more energy is consumed by the central electric furnace due to small heat

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* This may not be the case if temperature settings of the zones served by baseboard heaters differ from those assumed for the central systems.

losses from insulated basement supply and return ducts and operation of the warm air circulating fan. Also, the hybrid heat pump systems are represented by a single line for both gas and oil furnace cases. Although the quantities of fossil fuels used for the different furnaces are, of course different, there is no distinction in the thermal equivalents of the seasonal quantities consumed of these fuels.

General Findings

As would be expected, the energy consumption shown in Exhibits 8-8, 8-9, and 8-10 reflects the climatic condition in each city; as the number of heating degree days increases, energy consumption for heating increases. The converse is true for cooling systems. It can also be noted in Exhibits 8-12 and 8-13 that seasonal performance factors for heating and cooling have in general an inverse relationship to the number of heating or cooling degree days in each particular city. Of course, this is due to the operating characteristics of the systems simulated, particularly those of heat pumps and air conditioners (fossil fuel fired warm air furnace and electric resistance heating systems are less sensitive to climatic conditions). Heat pump COP for heating decreases with decreasing outdoor temperature and vapor compression air conditioner efficiency decreases with increasing outdoor temperature. Hence, it would be expected that high heat pump efficiencies on heating are experienced in the warm cities (such as Atlanta and Birmingham), while cooler cities (such as Cleveland and Concord) would enjoy high cooling system efficiencies.

The simulations of commercially available heating and cooling systems were performed both taking into account dynamic efficiency

losses and on a full load or steady-state basis. These latter results are shown in parentheses in Exhibits 8-8, 8-9, 8-12, and 8-14. As can be seen in these exhibits, dynamic efficiency losses, due to such factors as equipment cycling and defrost, substantially increase energy requirements for space conditioning, and hence lower both heating and cooling SPF's. Indeed, a reduction in energy consumption of about 6-10% for the heating season and 13-16% for the cooling season was observed when the simulation was performed on the traditional basis of assuming steady state operation. In that sense, then, failure to account for losses due to defrost and equipment dynamics, as has been the practice in the past, results in an overestimate of the SPF and an underestimate of the energy consumption. Unfortunately, however, dynamic efficiency data are not available for the non-modulating advanced heat pumps (for which they are specifically needed). Therefore, the results presented in Exhibits 8-11, 8-13, and 8-15 for advanced heating and cooling systems could only be determined on a full load or steady-state basis (except for those systems indicated) from system developers' data. It would be expected that in actual operation, dynamic efficiency losses would reduce system performance to some extent. If these systems experienced similar dynamic losses to those of conventional electric heat pumps, average increased energy consumptions of 6-10% for heating and 13-16% for cooling would be expected.

In the simulation of currently available electric heat pumps (standard, high efficiency, hybrid and heat-only), the effect on system performance of outside coil defrosting during the heating season was included. A significant portion of the seasonal energy consumption can be assigned to this process, as can be seen in the table below:

City	Heating Degree Days	Number of Defrost Cycles	Resistance Heat During Defrost (% Total Energy)
Houston	1290	118	3.8
Birmingham	2844	397	5.5
Atlanta	2821	370	4.4
Tulsa	3504	657	4.7
Philadelphia	4508	699	5.1
Seattle	4407	1208	7.9
Columbus	5407	1262	7.5
Cleveland	6097	1523	7.6
Concord	7377	1727	7.5

These results are specific for the standard electric heat pump but are similar to those of other electric heat pumps simulated. As can be seen, up to almost 8% of seasonal energy consumption is due to resistance heat used in tempering of air into the conditioned space as the heat pump operates in the defrost mode.

The results from the simulation of heat pump defrost cycle energy consumption may be compared to experimentally measured values presented by Groff and Bullock (1976) for a residence located in Syracuse, New York (6463 heating degree days). A total of 1222 defrost cycles were measured over the entire heating season, which is less than would be expected based on the data in the above table. However, the average length of each defrost cycle was about 3.3 minutes, lending credence to the 3 1/2 minute cycle length assumed in this simulation. Groff and Bullock also present results for a number of residences in Minneapolis (8382 heating degree days). Here, the number of defrost cycles varies from 1653 to 2164, in an inverse relationship to the capacity of the heat pump. Again, these figures would appear to be less than the number of cycles expected by extrapolation of the data in the table above. In

reality, initiation of a defrost cycle is dependent on a number of factors, such as outdoor humidity, temperature, wind speed, heat exchanger design, and is a most difficult effect to estimate. However, the simple approach used in this study reasonably approximates measured defrost occurrences. As an approximation to the defrost energy consumption, the results presented here also appear satisfactory.

As mentioned previously, in order to present results on a comparable basis, system simulations were carried out assuming insulated basement ductwork for all HVAC systems. In order to determine the effect of basement ductwork insulation on HVAC system performance, however, selected systems were also simulated with uninsulated ductwork. These results are tabulated below.

Effect of Basement Ductwork Insulation on HVAC System Performance

	Energy Consumption - Btu x 10^{-6}		
HVAC System	Uninsulated Ductwork	Insulated Ductwork	Energy Savings Due to Insulated Ducts
Gas warm air furnace	117.59	79.94	32%
Central air-conditioner, high efficiency	10.68	9.26	13%
Electric heat pump-standard			
Heating season	34.38	25.98	24%
Cooling season	11.96	10.26	14%

Although these results are specific for the test house in Philadelphia, similar results in other cities were obtained. The actual test house in Columbus, Ohio as originally built had a gas fired warm air furnace with no insulation in the basement ducts. Tests made by Ohio State University

researchers indicated that almost one third of the furnace output was lost to the basement (McBride, 1976).* These results show that the common practice of not insulating basement ductwork for fossil fuel fired systems can result in rather severe energy penalties. Also, the results show the necessity for simulating the system with ductwork insulation if equipment comparisons are intended.

Specific Findings: Conventional Systems

In Exhibit 8-16, heating season energy consumption per square foot of residential building heated area is shown as a function of the heating correlation parameter. (The greater the value of the parameter the colder the location or the greater the house heat loss.) The curve for the central electric furnace and baseboard convectors separates all-electric HVAC systems from systems consuming both electric energy and fossil fuels. Compared with the central electric furnace and baseboard convectors, which have heating season performance factors of 0.98 and 1.0 respectively, electric heat pumps have heating SPF's greater than one; systems firing fossil fuels onsite have SPF's less than one. Thus, gas or oil fired furnaces consume considerably more total energy, about 62% more, than do all-electric resistance heating systems. The hybrid heat pump, which uses fossil fuel only below the residence balance point, consumes less total energy than do systems operating almost exclusively on fossil fuels (and only a relatively small amount of electric energy for circulating fans). The hybrid heat pump saves about 33-64% of the onsite energy consumed by a gas or oil fired furnace, the larger amount representing warmer climates and the smaller percentage

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* In this study, the basement is not considered part of the conditioned space of the house.

saving representing colder climates. The reason for the variation in energy saved is due mainly to the improved performance achieved by an electric heat pump in warmer regions than in colder climates.

Exhibit 8-16 also shows that all three types of all-electric heat pumps considerably reduce heating season energy consumption compared with electric resistance heating and fossil fuel fired heating systems, the greatest savings occurring in warmer climates. The highest energy consuming all-electric heat pump, the standard electric heat pump, uses only about 29-53% of the energy consumed by electric resistance heating and about 56-73% of the energy consumed by fossil fuel fired warm air furnaces. Thus, in general, quite significant savings in total energy consumed on-site for heating can be realized by the use of conventional electric heat pumps.

Upon comparing the different electric heat pumps, the data show that the new high efficiency heat pumps use an average of 10% less energy for heating than the standard electric heat pump. The lowest energy consuming heat pump, the heat-only heat pump, reduces heating season energy consumption by an average of 19% compared with the standard electric heat pump.

It must be pointed out that, while a comparison of total energy consumption at the residential site is quite valid as a measure of the thermal equivalent of all forms of energy consumed by the HVAC system, comparing electric energy consumption with the consumption of fossil fuel burned onsite on this basis does not account for fuels consumed at the electric generating plant. To do so would require the energy balance envelope to be extended to include the utility as well. This will be done in Chapter 12.

Turning now to the conventional electrically operated cooling systems, Exhibit 8-17 graphically displays the energy consumed in the operation of heat pumps in the cooling mode and central air conditioners. The high efficiency electric heat pumps show the lowest seasonal energy consumption for cooling. Compared with the standard electric heat pump, the high efficiency units reduce cooling energy consumption by about 13%. The hybrid heat pump appears to have the largest energy consumption of all the heat pumps, but this should be considered a temporary finding. The heat pump portion of the hybrid system is an early standard model and is less efficient for either heating or cooling (see Exhibit 8-7 where comparative ratings are tabulated) than the new model which has since been developed by the manufacturer.

The standard central air conditioner, usually installed in tract housing developments, consumes the greatest amount of energy for cooling of all the systems shown, about 17% more than the high efficiency central air conditioner. The high efficiency central air conditioner is less energy consuming than the standard electric heat pump in all regions of the country examined and about equal to the average energy consumed by the high efficiency electric heat pump. Cooling season energy consumption for room air conditioners in the test house is quite low. It must be remembered though that the entire house is not cooled in this case and therefore one would expect this method of cooling to be the lowest energy user of all the cooling systems examined.

On an annual basis the combination of baseboard convectors for heating and room air conditioners for cooling consumes the smallest total amount of onsite energy, one-third less total energy than the combination of warm air fossil fuel furnace and high efficiency central air conditioners. The heat only electric heat pump plus high efficiency central air conditioner and the high efficiency electric heat pumps consume about the same amount of annual total energy which is about 36-70% of the annual total energy consumed by the combination of warm air furnace and high efficiency central air conditioners. Total annual energy savings for the standard electric heat pump relative to warm air furnaces with central electric air conditioners is about 26-64% and and is not as great as for the high efficiency electric heat pumps.

Specific Findings: Advanced Systems

Exhibits 8-18 and 8-19 graphically display heating and cooling energy usage per square foot of test house area for advanced heating and cooling systems. Considering heat activated heat pumps first, the data show Stirling-Rankine cycle heat pumps to have smaller energy consumption for heating and cooling than the absorption heat pump, some 25% less seasonal energy for heating and 34% less seasonal energy for cooling.

It would appear from Exhibits 8-18 and 8-19 that the V-type single cylinder Stirling-Rankine heat pump is only marginally better for heating than the free piston Stirling-Rankine heat pump, while for cooling the free piston system is more energy efficient than the V-type system. However, this may have as much to do with the assumption of capacity modulation for the latter system in the determination of the basic COP

data on which the calculations of this report are based as inherent system differences. Hence, for the V-type single cylinder Stirling-Rankine heat pump, dynamic efficiency losses should be relatively small and the data reported here should reasonably approximate energy consumption for the fully assembled field operated machine.* In its current design concept, the free-piston Stirling-Rankine machine will cycle. Based on this assumption, it would experience dynamic efficiency losses due to cycling not accounted for in the steady-state calculation of this report. Both would also, of course, experience additional losses due to defrost. These cannot be specified on the basis of data available.

Without fully and properly measuring performance data for the advanced heat pump equipment there is no completely satisfactory way of determining its energy consumption in actual practice. However, an approximation can be made if it is assumed that the ratio of energy consumed including dynamic efficiency losses to the energy consumed based on manufacturers' data is the same as that for conventional electric heat pumps. In that case the data of Exhibit 8-8 for the standard electric heat pump can be used to predict these ratios. When this is done for heating season, cooling season and annual energy usage averaged for all nine test cities, the following factors are produced:

$$\frac{\text{(Energy consumption including dynamic efficiency losses)}}{\text{(Energy consumption based on manufacturers' steady-state performance)}}$$

Heating season	1.097
Cooling season	1.164
Annual	1.118

* However, a constant heat exchanger effectiveness was assumed in the V-type Stirling-Rankine case. Hence, that system does not fully represent a realistic design configuration.

These factors can now be used to estimate energy consumption for equipment whose performance data are based on steady-state values. It should be understood that these factors provide only crude approximations to reality. It is not quite correct to assume, for example, that the free piston Stirling engine will have the same dynamic efficiency losses as the electric motor driving the reciprocating compressor of the standard electric heat pump. It might be expected that such losses will be collectively greater for the free piston Stirling-Rankine heat pump than for a conventional electric heat pump since in the former case the prime mover will exhibit its own dynamics.

The factors in the above table can be applied to all the advanced heating and cooling systems to produce energy consumption data on the same basis as for conventional systems with the exception of the following systems: advanced electric heat pump II, pulse combustion furnace and the V-type single cylinder Stirling-Rankine heat pump. The first and last are modulated systems for which dynamic efficiency losses should be quite small. The pulsed combustion furnace efficiency used already represents an integrated value over the heating season.

Applying the factors developed above to the energy data for the advanced electric heat pump I indicate that in the heating mode, the system can offer energy savings over commercially available electric heat pump systems. However, in the cooling mode this system appears to perform only slightly better than the electric heat pump standard.

All of the gas heat pumps under consideration in this study consume substantially more energy on-site for both heating and cooling than commercially available HVAC systems, regardless of whether the comparison is made with or without consideration of dynamic efficiency losses.

The advanced electric heat pump II represents a hypothetical system employing realizeable modifications to conventional electric heat pumps but having a significantly better operating performance. Energy consumption data for this hypothetical heat pump not separately calculated, rather scaled on energy consumption data produced in the simulation of the standard electric heat pump. The SPF for this high efficiency heat pump was taken as 2.4 in Columbus, an increase of 71% over an SPF of 1.4 for the standard electric heat pump in the same city. This increase is a reasonable assumption of the higher performance characteristics of a new system employing advanced techniques (Comly et al., 1975). All other SPF's and energy consumed for heating and cooling the test house in each of the nine test cities with this heat pump were then produced by scaling values for the standard electric heat pump using the factor (2.4/1.4).

A similar technique was used for the pulse combustion furnace. Based on information that this type of system would have a seasonal efficiency of about 90-95% (Hollowell, 1976), the gas energies calculated for the conventional gas warm air furnace were adjusted to reflect an overall efficiency of 92.5%. Electric energy to drive the indoor circulating fan was assumed to be slightly higher for the pulse combustion furnace than for the gas warm air furnace. Comparing the pulse combustion furnace at a 92.5% efficiency with a gas warm air furnace at a 60% efficiency suggests that a 32% saving in total energy would result with the more efficient system.

8.3 Heat Pump Performance in Commercial Applications

8.3.1 Methodology

The methodology used to compare the energy effectiveness of advanced heat pump systems with available conventional heating and cooling systems parallels the systems study of energy effectiveness for residential comfort conditioning. Steps in the overall procedure are as follows:

Commercial building selection. A single commercial building was selected and thoroughly characterized with respect to construction details for the purpose of estimating heating and cooling design loads for the structure in each of nine cities located in different climatological regions of the U.S. This characterization included structural features of the building, including insulation details needed for the subsequent calculation of hourly heating and cooling loads and systems energy requirements in each climatological location.

Weather data and design load calculation. Design heating and cooling loads were estimated for the selected building. Weather data representing long term average weather conditions at each of the nine cities were also used for the examination of energy effectiveness of alternative heating and cooling systems for commercial building application.

Systems selection. Conventional heating and cooling systems were selected from among available representative equipment generally used for the commercial structure chosen. Advanced heat pumps were chosen from among those currently being developed for similar commercial building applications. Equipment sizes were then specified for the commercial building in each of the nine cities based upon estimated heating and cooling design loads.

Computer model. A computerized calculation was made to estimate monthly and annual heating and cooling loads and systems energy input. Performance data for the equipment selected were acquired from manufacturers (heating and coooling capacities and operating COP as a function of outdoor temperature, air flows and parasitic energy requirements, special characteristics of each equipment type). Performance data were modeled in a manner capatible with the computerized calculation of the commercial building heating and cooling loads and system energy requirement.

The final computer load and systems energy calculation were worked up in consistent form suitable for comparative analyses. These include a comparison of on-site energy consumption for conventional systems and advanced heat pumps, seasonal and annual performance factors--the subject of this chapter. In addition the data were further developed for a cost analysis and a comparison of primary energy resources required for each system examined.

8.3.2 Commercial Building Selection and Design Load Calculation

The selection of a single commercial building, which is then artificially moved to different climatological regions of the U.S., is purely a device through which a reliable comparison of energy effectiveness for alternative heating and cooling systems may be made. The building is essentially used to determine the heating and cooling loads which must be satisfied by the competing systems. To some extent, then, an almost arbitrary selection of building type could be made. However, the type of advanced technology heat pumps currently under development should obviously be a determinant in building selection. Several market studies commissioned by organizations involved in advanced heat pump development have indicated that light commercial structures (e.g., of banks, community or professional services buildings, light manufacturing and small office buildings) requiring unitary equipment of less than 25 tons cooling capacity represented the largest single potential. Typical are the 10 ton cooling capacity Brayton-Rankine and the 7.5 ton RankineRankine gas heat pumps under current development for light commercial building application.*

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* However, see Section 1.4.

Several small commercial buildings were examined and a small office building currently under construction in Mountainside, New Jersey was selected. The Mountainside building is a two story, cement block structure having 5580 square feet of conditioned space. Architectural plans were made available as well as details of HVAC equipment selected for the building in its New Jersey location. Architectural views of the office building are shown in Exhibit 8-20 and general features outlined in Exhibit 8-21. The insulation and heat transfer properties of the walls, roof, foundation and glass areas of the building are compared with ASHRAE 90-75 thermal design standards for non-residential buildings (ASHRAE Standard 90-75) in Exhibit 8-22. Design loads were calculated for each location using the NESCA Manual J Supplement procedure (NESCA 1975) and are shown in Exhibit 8-23.

8.3.3 Heating and Cooling System Selection

Five different heating and cooling systems were selected for study; three represent conventional "baseline" systems frequently installed in the light commercial building application used here along with two advanced heat activated heat pumps currently being developed for similar installations. The specific systems selected were:

- Electric Heat Pump - Standard
- Electric Resistance Heating/Electric Air Conditioning
- Rooftop Gas Furnace/Electric Air Conditioning
- Brayton-Rankine Gas Heat Pump
- Rankine-Rankine Gas Heat Pump

Because the advanced heat pump systems being developed are in the 7-1/2 10 ton nominal cooling capacity range, and also because conventional

systems are available within this size range, it was desirable to subdivide the building so that multiples of the basic 7-1/2-10 ton sizes could be used. This was readily done by dividing the building into two zones, each zone having half the total floor area and half the design heating and cooling loads shown for the nine cities in Exhibit 8-23. When this is done the nine cities fall into two groups. The first group is represented by "warmer" climatological locations, Houston through Philadelphia, for which 10 tons of cooling capacity are required. Within this group the largest design total cooling load is about 103,000 Btuh (Houston) and the smallest is 94,000 Btuh (Philadelphia) just above the 7-1/2 ton size. The second group consists of "colder" climatological regions, Seattle through Concord, within which the largest design cooling load is about 89,000 Btuh (Columbus) and the smallest is about 63,000 Btuh (Seattle). For cities in this group, 7 1/2 tons of cooling capacity is required. It might possibly have been better to select a lower cooling capacity system for Seattle but this was not done for two reasons. Commercially available rooftop single package gas furnace/electric air conditioners are not made in less than 7 1/2 ton cooling capacities and the Brayton-Rankine gas heat pump is not scheduled for commercialization below 7 1/2 tons of cooling capacity because of technical and cost factors. The system sizes selected for Seattle are therefore all oversized with respect to cooling capacity but represent realistic sizes that would be installed in that region.

A short description of each heating and cooling system selected is found below:

Electric Heat Pump-Standard: Single package; two unit sizes employed based on design cooling load (see text above) with a nominal cooling capacity of 90,000 and 120,000. Lower capacity unit has a 90,000 Btuh cooling capacity at 95°F ODBT* (EER 7.8) and heating capacities of 90,000 Btuh (COP 2.6) and 50,000 Btuh at 17°F ODBT (COP 1.9); air flow is 3,000 cfm for heating and cooling. Higher capacity unit has a 117,000 Btuh cooling capacity at 95°F ODBT (EER 8.2) and heating capacities of 120,000 Btuh at 47°F (COP 2.1); and 67,000 Btuh at 17°F; airflow is 4,000 cfm for heating and cooling. Supplementary heaters sized for full design heating load for each city. Systems are equipped with outdoor temperature controlled economizer dampers. Performance data for both heating and cooling were based on manufacturer's reported steady-state data corrected for dynamic efficiency effects with data of Kelly and Bean (1976).

Electric Resistance Heating/Electric Air Conditioner: Heating provided by electrical resistance heaters sized for full mounted, single package unit with 2-stage modulated capacity (50% cooling capacity for each stage) and outdoor temperature controlled economizer dampers. Two unit sizes were required, 90,000 and 120,000 Btuh nominal cooling capacity, depending on design cooling load. For the 90,000 Btuh air conditioner (45,000 Btuh each stage), first stage EER 6.62 at 95°F ODBT and combined stages EER 7.69, airflow was 3,000 cfm (heating and cooling). For the 90,000 Btuh air conditioner (45,000 Btuh each stage), first stage EER 6.62 at 95°F ODBT and combined stages EER 7.69; airflow was 3,000 cfm (heating and cooling). The 120,000 Btuh air conditioner (60,000 Btuh each stage), first stage of 40,000 cfm (heating and cooling). Both systems are equipped with outdoor temperature controlled economizer dampers. Electric resistance heating was assumed 100% efficient. Performance data for the electric air conditioner were based on manufacurer's reported steady-state data corrected for dynamic efficiency effects with the Data of Kelly and Bean (1976) for electric heat pumps in the cooling mode.

Rooftop Gas Furnace/Electric Air Conditioning: Single package; unit sizes selected on the basis of cooling design load requirements. Each unit has 2 compressors for 2-stage cooling capacity modulation operation. The smaller cooling capacity unit was rated at 90,000 Btuh cooling with 2-stage operation (EER 7.26 at 95°F ODBT) and 45,000 Btuh cooling with 1-stage operation (EER 5.85 at 95°F ODBT). The accompanying 2-stage gas furnace capacity is rated at 180,000 Btuh output for 2-stage and 135,000 Btuh output for 1-stage operation. These outputs are AGA ratings for full load combustion efficiencies of 75% for each stage. The larger cooling capacity unit was rated at 120,000 Btuh cooling with 2-stage operation (EER 7.19 at 95°F ODBT) and 60,000 Btuh with 1-stage operation (EER 5.69 at 95°F

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* ODBT stands for "outdoor dry bulb temperature" and refers to the air temperature at the outdoor unit heat exchanger. Heating and cooling capacity and COP's are directly dependent upon this temperature.

ODBT). Gas furnace capacity for this larger cooling unit was 225,000 Btuh output for 2-stage operation and 168,000 Btuh output for each stage. Air flows for the two units were 3,000 cfm (90,000 Btuh nominal cooling capacity) and 4,000 cfm (120,000 Btuh nominal cooling capacity). Steady-state furnace output capacities were reduced using dynamic performance data for residential gas furnaces (Macriss, 1975). Performance data for the electric air conditioner were based on manufacurer's reported steady-state data corrected for dynamic efficiency effects with data of Kelly and Bean.

Brayton-Rankine Gas Heat Pump: This system is capable of modulated operation over a major portion of its heating and cooling operating range. For small outdoor dry bulb temperature deviation (±5°F) about 65°F, the system ventilates only. As progressively greater temperature deviations occur (and the heating and cooling loads increase) the system first operates in an on/off cycling mode during which heating and cooling capacities are constant (45° to 65°F ODBT for heating and 70° to 75°F ODBT for cooling). At ODBT below 45°F for heating and above 75°F for cooling, the system is fully modulated. Performance data provided by the manufacturer for a nominal 10 ton system showed a cooling capacity of 119,000 Btuh at 95°F ODBT (COP 1.06 and a heating capacity of 100,000 Btuh at 47°F ODBT (COP 1.32). Parasitics (indoor and outdoor coil fans, sink heat exchanger fan and systems control) for the 10 ton system were assumed to be 2.4 kW of electric energy (0.24 kW per ton of nominal cooling capacity). Performance data for a nominal 7-1/2 ton cooling capacity system were scaled from data for the 10 ton system, assuming that COP vs. ODBT remains constant. Estimated performance data for the 7-1/2 ton system at ARI rating points are 89,000 Btuh of cooling capacity at 95°F ODBT (COP 1.6) and a heating capacity of 75,000 Btuh at 47°F ODBT.

Rankine-Rankine Gas Heat Pump: This system is a fully modulated gas fired steam turbine driven heat pump. Parasitic electrical energy requirements are produced by the turbine with a high frequency alternator. In addition to heat pump operation, steam turbine exhaust is used to heat air for space conditioning and at very high heating loads (low outdoor temperatures) supplemental heat is provided by direct injection of generated steam. Data provided by the developer consisted of heating and cooling capacity ratios as percentages of full load cooling capacity and corresponding COP's related to outdoor dry bulb temperatures. For heating service, the heat pump is cut off at 0°F and all heating energy is provided by steam injection below this temperature. Two sizes of heat pumps were modeled for the commercial building application, 120,000 Btuh and 90,000 Btuh of cooling capacity at 95°F ODBT (COP 0.73 for both sizes). On heating service, high temperature capacities (47°F ODBT) of the heat pumps were taken as 120,000 Btuh and 90,000 Btuh, respectively (representing 100% of cooling load capacity) for which both heat pumps have COP's of 1.5. The developer specifically stated that reported operational heating COP's were substantially lower than design COP's due to off-cycle thermal losses, start-up losses at low loads and modulation losses at intermediate loads. Computed system energy requirements are therefore considered to include dynamic efficiency effects.

8.3.4 Systems Simulation and Computer Modeling

The computer program used for the determination of energy effectiveness of commercial building heating and cooling systems was AXCESS (Alternate Cost Comparison for Energy System Simulation), a program developed for the Edison Electric Institute (EEI) by several contractors largely for the purpose of comparing energy requirements for alternative heating and cooling systems applied to a single structure. AXCESS has a broad flexibility with respect to the buildings and HVAC equipment it can examine, as well as a number of internal variations that can be programmed. Although other building energy analysis programs are available (including some which can simultaneously perform an economic analysis of systems examined) AXCESS was selected because of the known accessibility of backup and support in its use by EEI as well as its proven ability to model systems and structures similar to those used in the present study (Reeves et al., 1976). A full description of AXCESS is not possible here, nor necessary, and only a brief account will be given of the program and its data requirements and output capabilities for the specific systems examined in this study.

Building input data for AXCESS can be brief or detailed depending on the user's knowledge of the building being modeled. Simple information such as window areas, net wall area and floor areas are required along with detailed information of wall and roof construction features. Heat transfer response factors for envelope surfaces are supplied internally or can be input by the user. AXCESS performs an internal calculation of heating and cooling loads on an hourly basis and simultaneously calculates

and outputs systems energy to satisfy these loads. Part of the flexibility of the program results from the ability to subdivide a building into areas, each area served by its own HVAC system, and into zones, each zone having a specific response. A further subdivision into meters stores data on specific types of energy used within the building. For example, meters can be programmed to record lighting and office equipment electric energy requirements as well as to record energy in various forms (cubic feet of gas, gallons of oil, kWh of electric energy) for HVAC equipment in each area. In addition, separate sub-meters can be used to tally energy usage for equipment components, for example compressor energy for a heat pump, supplemental strip heat required below the balance point, and the energy used to operate fans and other parasitic devices.

Although AXCESS calculates heating and cooling loads and then uses input data on the HVAC systems programmed to calculate energy needed to satisfy these loads, the program does not directly output calculated loads. It was important, though, for this study that the loads corresponding to calculated energy used be known so that performance factors for the equipment examined could be estimated. This shortcoming was surmounted by running this program with a "synthetic" HVAC system, one which had a virtually infinite capacity for heating and cooling and an energy input exactly equal to its output. In this way the printed output was exactly equal to the load on the building.

Input data for the program also consisted of local weather data for each city and a detailed occupancy schedule in terms of percent of base load energy for lights and office machinery as a function of the hour of day, so that internal energy gains could be realistically estimated. Night time hours were each assumed to require 5 percent of the base load demand of 14kWh. In addition, each heating system was programmed to operate so that the full heating load was satisfied during working hours (70°F comfort level) and Monday to Friday nonworking hours (10°F setback to 60°F). The cooling system was set to satisfy a comfort level of 75°F during working hours only. During weekends and holidays, no heating or cooling was supplied.

Operating performance data for simulation of the five space conditioning systems considered were derived from information by manufacturers or developers. Version "6" of the AXCESS model was used which allows part load behavior and dynamic efficiency losses to be reflected in system energy consumption, where appropriate. In case of the electric heat pump, this was done on the basis of the 5-ton heat pump data of Kelly and Bean (1976). It is recognized that these data were developed for a residential system and that the 10-ton commercial heat pump, say, may utilize two 5-ton compressors. The dynamic efficiency data used included the effect of reduced heating-capacity due to the formation of frost on the outdoor heat exchanger coil and outside dry bulb temperature below 40°F. Energy consumption during defrosting cycles is included in the electric heat pump input data but there is no provision in the AXCESS program for energy consumed to temper cold air transferred to the building during defrost. However, the Kelly and Bean model of heat pump performance in

the defrost region included the first stage of supplemental heating for tempering air during the defrost cycle. This was one of the reasons for choosing this model over that of Parken and Kelly (1977). Also, it will be recalled that Parken and Kelly model was applied to residential heat pump performance in a defrost region defined as outdoor dry bulb temperature less than about 47°F and coincident relative humidity in excess of 60%. Because AXCESS has no data input capability for ambient relative humidity, the single definition of the defrost region inherent in the Kelly and Bean data was used. No information about defrost energy requirements for the advanced heat activated heat pumps was available. Most probably there is sufficient heat rejection from the combustion furnaces for these heat pumps to provide energy for defrosting outdoor heat exchanger surfaces. With the exception of the electric heat pumps operating in Concord, representing the coldest weather conditions, no supplemental heat was required for any of the others. (Supplemental heat is incorporated as part of the operating scheme of the Rankine-Rankine heat pump.)

Thus, for heating and cooling modes, the energies calculated by the AXCESS program reflect system response to the building load under dynamic conditions wherever possible. Dynamic efficiency effects included for the systems examined in this study were mentioned in the system descriptions above, along with the source of the data used.*

Output data from the AXCESS program consist, in part, of energy consumed at each active meter for all usages specified. This output

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* Part load performance data were available for all commercial building HVAC systems except the Brayton-Rankine Gas Heat Pump. Energy consumption data presented here for the Brayton-Rankine Gas Heat Pump are probably slightly underestimated by the steady-state performance simulation which had to be resorted to.

could be reported on an hour-by-hour basis or accumulated over longer intervals. In addition to energy consumed, the program stores hourly electric energy demands and outputs the maximum demand at each electric meter for the time period examined.

The method of calculation employed was to run AXCESS for each city for a full 24 hours every fifth day for four months; January, April, July and October. When instructed to run every fifth day in any month, the computer internally sums the results for the six days computed, multiplies by five, and prints a monthly estimate of energies for each meter. This method of calculating monthly energy consumption is entirely adequate for estimating annual energy usage and was the method used to validate the program.The validation study (Reeves et al., 1976) shows quite good agreement between energy consumption estimated by calculating every fifth day for a full year and measured energy consumption for an existing building. Energy consumption data for the entire year were determined by a simple multiplication of the four-month-data by three. Prior to commencing full scale runs for the present study using every fifth day for four months a year, it was demonstrated that a reliable estimate of annual energy consumption could be produced with these limited data by calculating heating and cooling energy consumption and performance factors for an entire year for two of the cities. These results were then compared to the results obtained by calculating energies and performance factors for four months alone (and multiplying by 3). Differences in the yearly energy consumption computed by the two methods were of the order of 10-20% at the most. Differences in HVAC system performance factors calculated by the two methods were of the order of 1-2%.

The final outcome of the computer modelling with AXCESS for the five heating and cooling systems examined is a set of monthly heating and cooling loads for the commercial building in nine climatologically different regions of the country, monthly on-site energy inputs to each space conditioning system and one hour peak electric energy demands. These results are shown in Exhibits 8-24 to 8-26; in terms of annual fuels and energy consumed on-site, Exhibit 8-24; system performance factors, Exhibit 8-25; thermal equivalent of annual onsite energy consumption, Exhibit 8-26.

8.3.5 Analysis of Heat Pump Performance

For the two all-electric systems, the electric heat pump and the electric resistance heater/electric air conditioner, the electric heat pump consistently uses less electric energy in all nine cities. Savings in annual electric energy by the heat pump range from 11% in Houston to 47% in Seattle and are generally larger for colder climate cities than for cities in warmer climates. This is reflected also by a superior heating performance factor for the heat pump (2.0 to 2.7 for all cities) compared with a uniform heating performance factor less than one for electric resistance heating as is shown in Exhibit 8-25. In contrast, the cooling performance factors are approximately the same for the electric heat pump and the rooftop air conditioner. Therefore, in Houston, where the greatest energy use is for cooling, there is less of an advantage for the heat pump than in Concord, where the major energy requirement is for heating.

As expected in energy use comparisons on an on-site basis, the rooftop combination of gas furnace/electric air conditioner uses more annual energy than either of the conventional all-electric systems, since electric generation transmission and distribution losses are not included in this comparison. However, a part of the increased energy consumption of the gas furnace/electric air conditioner is also due to a lower efficiency air conditioning operation than the all-electric system. The rated EER for the electric air conditioner used with the gas furnace is 7.2 (120,000 Btuh capacity, 2 stages) and 7.3 (90,000 Btuh capacity 2 stages). These are lower, for example, than the EER for the standard electric heat pump (8.2 for the 120,000 Btuh cooling capacity unit and 7.8 for the 90,000 Btuh unit). The higher EER electric heat pumps are clearly the reason for the better cooling season performance factors for the heat pump as shown in Exhibit 8-25.

Gas furnace heating season performance factors for residential service were previously shown to be about 0.62-0.65 whereas the rooftop gas furnace for light commercial building application shows heating performance factors of 0.36-0.52, both results including dynamic losses. The reason for the lower efficiency gas furnace performance in commercial buildings is partly due to a lower full load (steady state) efficiency for the large rooftop units (75%) compared with residential furnaces (80%), both certified ratings by the AGA. In addition, the rooftop units are grossly oversized compared with design heating loads. All rooftop gas furnaces are rated at 180,000 Btuh output heating capacity (1st stage) compared with a maximum heating design load of 83,000 Btuh for Concord and 43,000 Btuh for Houston. As a consequence of oversizing

the furnaces cycle more frequently, each combustion cycle being of shorter duration compared with a gas furnace whose rated output is more nearly matched to design load. Dynamic efficiency losses due to greater cycling will considerably reduce heating performance factors.

It is difficult to compare the advanced gas heat pumps with the conventional systems because of the different forms of energy utilized on-site. It is obvious that, since the on-site efficiency of the electric HVAC systems is higher, the all-electric conventional systems use more on-site energy than the advanced systems. Both advanced systems show high heating performance factors (greater than one) compared with a heating season performance factor of less than 0.5 for the gas furnace. Thus, with respect to the same energy source, the advanced heat pumps show superior performance.

Upon comparing the two advanced gas heat pumps, it appears that the Rankine-Rankine system uses more total on-site energy in all cities except Concord than does the Brayton-Rankine system. However, the latter system uses purchased electricity for its auxiliary equipment whereas the Rankine-Rankine machine generates all of its own electricity. Hence, better comparisons of these and other systems are provided in Chapter 9 and 12.

8.4 Summary

This chapter has compared the on-site electrical and fossil-fuel energy consumption and seasonal performance factors for a number of conventional and advanced HVAC systems in both residential and commercial applications. The comparisons represent results of computer simulations

of the energy use of a single-family residence and a small two-story office building in nine climatologically and geographically different locations in the continental U.S. Since some of the systems involved use electricity alone or in combination with gas or fuel oil, comparisons of on-site performance are not particularly meaningful. However, comparisons of some all-electric systems are of interest. For example, high efficiency conventional electric heat pumps which have recently become commercially available use about 11% less energy annually than standard electric heat pumps which have been on the market for some time. Electric resistance systems, represented by the central electric furnace and baseboard convectors, are less energy-efficient than the electric heat pumps, as expected.

The high efficiency electric heat pumps use the least amount of energy for residential cooling of all currently available vapor compression air conditioning systems with the exception of zone demand cooling with room air conditioners. High efficiency air conditioners use about 10% more energy than standard electric heat pumps in the cooling mode, but are more efficient than standard air conditioners generally used by builders.

Advanced electric heat pumps use less energy than the most efficient currently available electric heat pumps in residential heating. A hypothetical electric heat pump system representing anticipated improvements in system performance would reduce energy consumption for heating by about 40%. The advanced electric heat pump I, currently under development, appears to consume about as much energy for space cooling as currently available high efficiency electric heat pumps. In contrast, the hypothetical

continuously modulated advanced electric heat pump II should offer substantial energy savings in comparison to current cooling technology.

It is significant that in moderate-to-cold locations gas heat pumps consume less gas for heating and cooling, than the conventional gas furnace does for heating alone. In Tulsa, for example, the free-piston Stirling-Rankine heat pump will use 28% less gas over the entire year than the gas furnace will consume for heating only. In Concord, almost 52% less. However, these results do not take into account possible part load operation losses on the part of the gas heat pumps, so that actual savings may be somewhat less dramatic. (Parasitic electric power requirements for heating by the same heat pump are almost three to four times higher than by gas furnace, however. Additional electric energy is consumed by the air conditioner as well.) The organic fluid absorption cycle heat pump is considerably less efficient on cooling in comparison to the engine-driven heat pump and, therefore, uses less gas annually than the conventional gas furnace only in cities with a low cooling load (Seattle, plus the three coldest cities). Against an advanced gas furnace (such as the pulse combustion system), the gas heat pumps begin to look attractive in terms of on-site gas savings also primarily in cold cities.

Among commercial-sized presently available units examined, the standard electric heat pump consumes the least amount of onsite energy for year round comfort conditioning. The least efficient all-electric unit in terms of on-site energy consumption is single package electric air conditioning with electric resistance heating.

Among the advanced systems considered, both the Brayton-Rankine gas heat pump and the Rankine-Rankine gas heat pump consume in cold cities less gas on-site than the rooftop gas furnace/electric air conditioning system. In Concord, for example, the savings are of the order of 26%. The systems are about comparable in Philadelphia. It should be noted, however, that part load losses in the case of the Brayton-Rankine heat pump have not been accounted for.

References to Chapter 8

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EXHIBIT 8-1

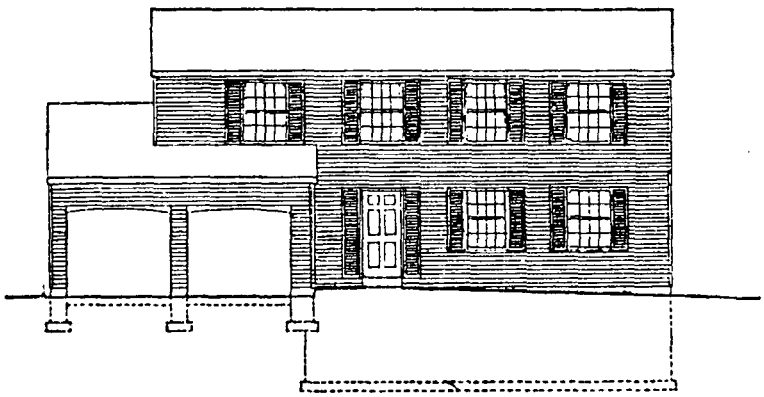
WEATHER DATA FOR TEST CITIES

City	ASHRAE[a] Heating Degree Days	Representative Heating Degree Day Range	WEATHER DATA USED IN THIS STUDY			
			Heating Degree Days	Cooling Degree Days	Year of Weather Data	Deviation Level
Houston	1278	500 - 2000	1290	2339	1955	3%
Birmingham	2551	2000 - 3000	2844	1928	1955	7%
Atlanta	2961	2500 - 3500	2821	1589	1959	6%
Tulsa	3680	3000 - 4000	3504	1949	1950	6%
Philadelphia	4486	4000 - 5000	4508	1104	1951	8%
Seattle	4424	--	4407	183	1960	2%
Columbus	5560	5000 - 6000	5467	809	1964	10%
Cleveland	6351	6000 - 7000	6097	613	1964	10%
Concord	7383	7000 - 8000	7377	349	1964	10%

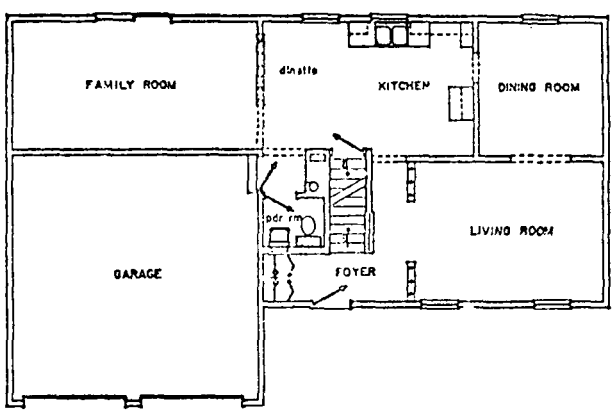
(a) Heating degree days from ASHRAE 1973 Systems Handbook, Chapter 43.

EXHIBIT 8-2

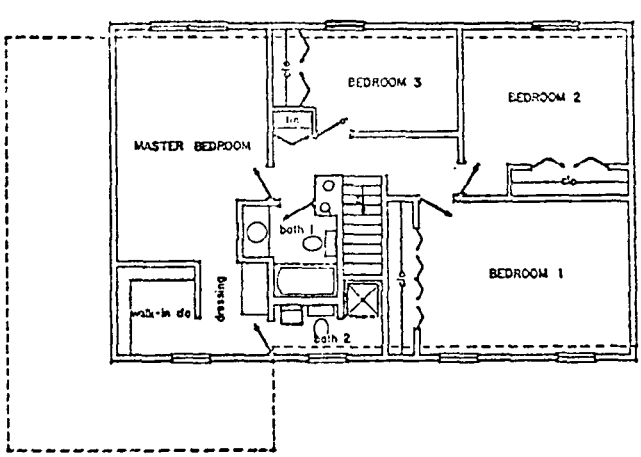
RESIDENTIAL TEST HOUSE: FRONT ELEVATION AND FLOOR PLANS



front elevation



first floor plan



second floor plan

Scale in feet

0 4 8

EXHIBIT 8-3

TEST HOUSE GENERAL CHARACTERISTICS

House Style	Two stories, wood frame, 2 car garage
Living Area	1850 sq. ft. (excluding basement).
Room Complement	
First Floor	Foyer, living room, dining room, family room, kitchen, powder room.
Second Floor	Master bedroom, large bedroom, 2 small bedrooms, 2 baths.
Exposure	Front of house faces north.
Exterior Surface	Aluminum siding.
Doors	One - wood panel construction.
Roof	Black asphalt shingles over building paper and 5/8" plywood deck.
Glass Areas	16 glass areas, including patio sliding door. (For total glass area see bldg. plans.) All windows are aluminum cased, double hung, weather stripped, double pave insulating glass. Basement windows (2) are single glazed, 1/8" sheet glass.
Basement	
Walls	11 course, 8 inch concrete block
Floor	4 inch concrete slab over 4 inch gravel bed, 662 sq. ft.
Crawl Space	4 inch gravel bed over plastic vapor barrier.

EXHIBIT 8-4

TEST HOUSE INSULATION USED AND COMPARISON OF HEAT TRANSFER COEFFICIENTS WITH ASHRAE 90-75 STANDARDS

	EXTERIOR WALLS			CEILINGS			FLOORS OVER UNHEATED SPACES		
	Insulation Specified	Overall Thermal Transmittance (BTU/Hr-Ft^2°F) ASHRAE 90-75	Test House	Insulation Specified	Overall Thermal Transmittance (BTU/Hr-Ft^2°F) ASHRAE 90-75	Test House	Insulation Specified	Overall Thermal Transmittance (BTU/Hr-Ft^2°F) ASHRAE 90-75	Test House
Houston	R-9	0.29	0.132	R-19	0.05	0.048	R-7	0.305	0.086
Birmingham	R-9	0.27	0.132	R-19	0.05	0.048	R-7	0.220	0.086
Atlanta	R-9	0.27	0.132	R-19	0.05	0.048	R-7	0.200	0.086
Tulsa	R-9	0.25	0.132	R-19	0.05	0.048	R-7	0.135	0.086
Philadelphia	R-9	0.24	0.132	R-19	0.05	0.048	R-9	0.080	0.073
Seattle	R-9	0.24	0.132	R-19	0.05	0.048	R-9	0.080	0.073
Columbus	R-9	0.23	0.132	R-19	0.05	0.048	R-9	0.080	0.073
Cleveland	R-9	0.22	0.132	R-19	0.05	0.048	R-9	0.080	0.073
Concord	R-9	0.20	0.132	R-19	0.05	0.048	R-9	0.080	0.073

EXHIBIT 8-5

COMPARISON OF TEST HOUSE AIR INFILTRATION RATES WITH ASHRAE 90-75 STANDARDS [a]

City	Windows (ft^3/min per ft of sash crack)		Sliding Doors (ft^3/min per ft^2 of door area)		Entrance Doors (ft^3/min per ft^2 of door area)	
	ASHRAE 90-75	Test House	ASHRAE 90-75	Test House	ASHRAE 90-75	Test House
Houston	.5	.25	.5	.25	1.25	1.25
Brimingham	.5	.25	.5	.25	1.25	1.25
Atlanta	.5	.25	.5	.25	1.25	1.25
Tulsa	.5	.25	.5	.25	1.25	1.25
Philadelphia	.5	.25	.5	.25	1.25	1.25
Seattle	.5	.25	.5	.25	1.25	1.25
Columbus	.5	.25	.5	.25	1.25	1.25
Cleveland	.5	.25	.5	.25	1.25	1.25
Concord	.5	.25	.5	.25	1.25	1.25

(a) Air infiltration data for windows and sliding doors from "Technical Data - Computing Heat Loss for Anderson Windows and Sliding Doors", Anderson Corporation, Bayport, Minnesota (Dec. 1971)

EXHIBIT 8-6

DESIGN TEMPERATURES AND HEAT LOAD FOR HEATING AND COOLING EQUIPMENT

	HEATING		COOLING			
	97-1/2% DESIGN DRY BULB TEMPERATURE	DESIGN HEAT LOAD (BTUH)	2-1/2% DESIGN DRY BULB TEMPERATURE	2-1/2% DESIGN WET BULB TEMPERATURE	DESIGN SENSIBLE COOLING LOAD (BTUH)	DESIGN TOTAL COOLING LOAD (BTUH)
Houston	32	27,030	94	80	24,069	36,844
Birmingham	22	33,915	94	78	25,335	35,803
Atlanta	23	33,288	92	77	23,126	32,994
Tulsa	16	37,671	99	78	29,383	38,351
Philadelphia	15	38,242	90	77	24,880	35,302
Seattle	32	27,621	79	65	18,757	20,365
Columbus	7	42,913	88	76	23,468	33,090
Cleveland	7	42,913	89	75	24,454	32,845
Concord	-7	52,088	88	73	23,837	30,420

EXHIBIT 8-7

GENERAL CHARACTERISTICS OF HVAC SYSTEMS (f)

	Heating COP (a) 17°F	Heating COP (a) 47°F	Cooling EER[a] 95°F	Heating Capacity (Btuh)(a) 17°F	Heating Capacity (Btuh)(a) 47°F	Cooling Capacity (Btuh)(a) 95°F	Auxiliaries (w/ton)
ELECTRIC HEAT PUMPS							
Electric Heat Pump - Standard	1.8	2.6	6.9	22,400	39,000	36,000	277
Electric Heat Pump - High Efficiency I	2.1	3.1	7.9	22,600	38,900	34,100	267
- High Efficiency II	2.1	2.8	7.8	21,000	37,500	33,500	-
- High Efficiency III	2.0	2.8	8.2	22,000	39,000	36,000	275
Hybrid Electric Heat Pump	1.4	2.1	6.4	22,000	37,000	36,000	314
Electric Heat Pump - Heat Only	2.3	3.2	-	21,500	36,500	-	-
Advanced Electric Heat Pump I	2.2	2.7	7.5	38,100	36,600	39,500	0
THERMAL ENGINE AND ABSORPTION CYCLE HEAT PUMPS			(COP)				
Free Piston Stirling-Rankine Gas Heat Pump	1.4	1.7	(0.9)[g]	55,600	68,100	35,000	517
V-Type Single Cylinder Stirling-Rankine Gas Heat Pump	1.3	1.5	(1.1)	41,000	58.000	36,000	-
Organic Fluid Absorption Gas Heat Pump	1.1	1.2	(0.5)[g]	76,000	87,000	36,000	268
OTHER SYSTEMS							
Gas Warm Air Furnace	80%(b)	80%(b)	-	64,000(c)	64,000(c)	-	-
Oil Warm Air Furnace	85%(b)	85%(b)	-	85,000(c)	85,000(c)	-	-
Central Electric Furnace	100%(b)	100%(b)	-	(d)	(d)	-	-
Baseboard Convectors	100%(b)	100%(b)	-	(e)	(e)	-	-
Central Air Conditioner - High Efficiency	-	-	7.7	-	-	35,500	238
- Standard	-	-	6.5	-	-	35,000	280

Notes to Exhibit 8-7.

(a) Rating conditions are 47/43-70 for high temperature heating, 17/15-70 for low temperature heating and 80/67-95 for cooling. (b) Manufacturer's full load system efficiency. (c) Constant capacity at all outdoor temperatures. (d) Total heating capacity varied by location from 9.6 to 15.36 kW. (e) Total heating capacity varied by location from 9.75 to 16.0 kW. (f) Characteristics of heat pumps are for 3 ton (cooling) models of each type. (g) COP for compressor only; auxiliaries not included.

EXHIBIT 8-8

FUELS AND ELECTRIC ENERGY CONSUMED FOR SPACE CONDITIONING BY THE TEST HOUSE IN SELECTED CITIES
CONVENTIONAL ELECTRIC HEAT PUMPS

	Length of Season (months)	Degree Days or Cooling Hours	Electric Heat Pump-Standard Elec. (kWh)	Electric Heat Pump-High Efficiency I Elec. (kWh)	Electric Heat Pump-High Efficiency II Elec. (kWh)	Electric Heat Pump-High Efficiency III Elec. (kWh)
Houston						
Total Heating Season	4	1,290	1,402 (1,261)*	1,173 (1,056)	1,254 (1,125)	1,331 (1,196)
Total Cooling Season	8	2,022	7,215 (6,273)	5,930 (5,178)	6,536 (5,607)	6,286 (5,478)
Total Annual			8,617 (7,534)	7,103 (6,234)	7,790 (6,732)	7,617 (6,674)
Birmingham						
Total Heating Season	6	2,844	3,787 (3,414)	3,250 (2,940)	3,379 (3,053)	3,603 (3,256)
Total Cooling Season	6	1,480	5,010 (4,301)	4,236 (3,657)	4,561 (3,868)	4,369 (3,754)
Total Annual			8,797 (7,715)	7,486 (6,597)	7,940 (6,921)	7,972 (7,010)
Atlanta						
Total Heating Season	5	2,821	3,789 (3,437)	3,235 (2,935)	3,562 (3,231)	3,612 (3,284)
Total Cooling Season	7	1,393	4,700 (4,001)	3,980 (3,406)	4,381 (3,778)	4,100 (3,497)
Total Annual			8,489 (7,438	7,215 (6,341)	7,943 (7,009)	7,712 (6,781)
Tulsa						
Total Heating Season	6	3,504	7,878 (7,252)	6,855 (6,304)	6,741 (6,198)	7,687 (7,069)
Total Cooling Season	6	876	3,873 (3,261)	3,158 (2,669)	3,449 (2,931)	3,889 (3,277)
Total Annual			11,751 (10,513)	10,013 (8,973)	10,190 (9,129)	11,576 (10,346)
Philadelphia						
Total Heating Season	7	4,508	7,613 (6,958)	6,615 (6,048)	6,798 (6,215)	7,283 (6,663)
Total Cooling Season	5	917	3,006 (2,549)	2,527 (2,155)	2,725 (2,283)	2,622 (2,227)
Total Annual			10,619 (9,507)	9,142 (8,203)	9,523 (8,498)	9,905 (8,890)
Seattle						
Total Heating Season	9	4,407	6,861 (6,101)	6,238 (5,499)	5,982 (5,333)	6,101 (5,431)
Total Cooling Season	3	532	981 (875)	801 (719)	905 (795)	837 (739)
Total Annual			7,842 (6,976)	7,039 (6,218)	6,887 (6,128)	6,938 (6,170)
Columbus						
Total Heating Season	7	5,407	10,338 (9,488)	9,212 (8,434)	9,283 (8,513)	9,968 (9,146)
Total Cooling Season	5	861	2,801 (2,396)	2,351 (2,023)	2,543 (2,148)	2,441 (2,092)
Total Annual			13,139 (11,884)	11,563 (10,457)	11,826 (10,661)	12,409 (11,238)
Cleveland						
Total Heating Season	7	6,097	12,364 (11,362)	11,077 (10,145)	11,112 (10,205)	11,928 (10,963)
Total Cooling Season	5	670	2,153 (1,831)	1,792 (1,532)	1,945 (1,634)	1,878 (1,600)
Total Annual			14,517 (13,193)	12,869 (11,677)	13,057 (11,839)	13,806 (12,563)
Concord						
Total Heating Season	9	7,377	16,796 (15,743)	15,613 (14,611)	14,225 (13,306)	14,891 (13,943)
Total Cooling Season	3	435	1,366 (1,184)	1,128 (978)	1,263 (1,077)	1,098 (942)
Total Annual			18,162 (16,927)	16,741 (15,589)	15,488 (14,383)	15,989 (14,885)

* Numbers in parentheses are on-site fuels and electric energy consumption calculated on steady state or full load performance basis, i.e., no dynamic efficiency losses are included.

EXHIBIT 8-9

FUELS AND ELECTRIC ENERGY CONSUMED FOR SPACE CONDITIONING BY THE TEST HOUSE IN SELECTED CITIES - ADD-ON HEAT PUMPS

	Electric Heat Pump-Heat Only With Central Air Conditioner High Efficiency	Hybrid Electric Heat Pump - Gas Furnace		Hybrid Electric Heat Pump - Oil Furnace	
	Elec. (kWh)	Gas (cft)	Elec. (kWh)	Oil (Gal)	Elec. (kWh)
Houston					
Total Heating Season	1,172 (1,049)*	943 (847)	1,572 (1,416)	7 (7)	1,572 (1,416)
Total Cooling Season	6,510 (5,664)	--	7,570 (6,506)	--	7,570 (6,506)
Total Annual	7,682 (6,713)	943 (847)	9,142 (7,922)	7 (7)	9,142 (7,922)
Birmingham					
Total Heating Season	3,138 (2,831)	6,141 (5,157)	3,719 (3,450)	45 (38)	3,719 (3,450)
Total Cooling Season	4,524 (3,887)	--	5,273 (4,476)	--	5,273 (4,476)
Total Annual	7,662 (6,718)	6,141 (5,157)	8,992 (7,926)	45 (38)	8,992 (7,926)
Atlanta					
Total Heating Season	3,185 (2,899)	5,903 (4,701)	3,777 (3,520)	44 (35)	3,777 (3,520)
Total Cooling Season	4,242 (3,614)	--	4,959 (4,178)	--	4,959 (4,178)
Total Annual	7,427 (6,513)	5,903 (4,701)	8,736 (7,698)	44 (35)	8,736 (7,698)
Tulsa					
Total Heating Season	6,201 (5,694)	15,802 (14,008)	5,742 (5,373)	114 (101)	5,743 (5,374)
Total Cooling Season	3,788 (3,194)	--	4,024 (3,329)	--	4,024 (3,329)
Total Annual	9,989 (8,888)	15,802 (14,008)	9,766 (8,702)	114 (101)	9,767 (8,703)
Philadelphia					
Total Heating Season	6,176 (5,645)	17,377 (15,173)	6,738 (6,313)	127 (111)	6,740 (6,314)
Total Cooling Season	2,711 (2,300)	--	3,171 (2,658)	--	3,171 (2,658)
Total Annual	8,887 (7,945)	17,377 (15,173)	9,909 (8,971)	127 (111)	9,911 (8,972)
Seattle					
Total Heating Season	5,445 (4,855)	20,249 (11,424)	5,792 (6,270)	147 (85)	5,794 (6,271)
Total Cooling Season	889 (786)	--	1,048 (914)	--	1,048 (914)
Total Annual	6,334 (5,641)	20,249 (11,424)	6,840 (7,184)	147 (85)	6,842 (7,185)
Columbus					
Total Heating Season	8,509 (7,775)	36,344 (30,892)	6,784 (6,698)	262 (223)	6,786 (6,700)
Total Cooling Season	2,527 (2,163)	--	2,957 (2,500)	--	2,957 (2,500)
Total Annual	11,036 (9,938)	36,344 (30,892)	9,741 (9,198)	262 (223)	9,743 (9,200)
Cleveland					
Total Heating Season	10,221 (9,354)	47,207 (38,902)	7,568 (7,728)	340 (281)	7,572 (7,732)
Total Cooling Season	1,940 (1,652)	--	2,278 (1,916)	--	2,278 (1,916)
Total Annual	12,161 (11,006)	47,207 (38,902)	9,846 (9,644)	340 (281)	9,850 (9,648)
Concord					
Total Heating Season	13,028 (12,191)	58,538 (52,890)	8,179 (8,018)	420 (380)	8,186 (8,024)
Total Cooling Season	1,184 (1,017)	--	1,431 (1,191)	--	1,431 (1,191)
Total Annual	14,212 (13,208	58,538 (52,890)	9,610 (9,209)	420 (380)	9,617 (9,215)

* Numbers in parentheses are on-site fuels and electric energy consumptions calculated on steady state or full load performance basis, i.e. no dynamic efficiency losses are included.

EXHIBIT 8-10

FUELS AND ELECTRIC ENERGY CONSUMED FOR SPACE CONDITIONING BY THE TEST HOUSE IN SELECTED CITIES - OTHER CONVENTIONAL HEATING AND COOLING SYSTEMS *

	Gas Warm Air Furnace With Central Air Conditioner-High Efficiency		Oil Warm Air Furnace With Central Air Conditioner-High Efficiency		Central Electric Furnace With Central Air Conditioner-High Efficiency	Baseboard Convectors With Window Air Conditioners
	Gas (cft)	Elec. (kWh)	Oil (Gal)	Elec.(kWh)	Elec.(kWh)	Elec.(kWh)
Houston						
Total Heating Season	17,112	98	119	60	3,039	2,974
Total Cooling Season	--	6,510	--	6,510	6,510	3,402
Total Annual	17,112	6,608	119	6,570	9,549	6,376
Birmingham						
Total Heating Season	41,176	236	289	161	7,450	7,295
Total Cooling Season	--	4,524	--	4,524	4,524	2,386
Total Annual	41,176	4,760	289	4,685	11,974	9,681
Atlanta						
Total Heating Season	42,078	241	296	163	7,603	7,441
Total Cooling Season	--	4,242	--	4,242	4,242	2,289
Total Annual	42,078	4,483	296	4,405	11,845	9,730
Tulsa						
Total Heating Season	73,451	397	510	286	13,342	13,142
Total Cooling Season	--	3,788	--	3,788	3,788	1,909
Total Annual	73,451	4,185	510	4,074	17,130	15,051
Philadelphia						
Total Heating Season	78,406	448	554	325	14,454	14,243
Total Cooling Season	--	2,711	--	2,711	2,711	1,578
Total Annual	78,406	3,159	554	3,036	17,165	15,821
Seattle						
Total Heating Season	69,235	348	492	190	12,906	12,629
Total Cooling Season	--	889	--	889	889	768
Total Annual	69,235	1,237	492	1,079	13,795	13,397
Columbus						
Total Heating Season	94,396	540	668	407	17,677	17,418
Total Cooling Season	--	2,527	--	2,527	2,527	1,507
Total Annual	94,396	3,067	668	2,934	20,204	18,925
Cleveland						
Total Heating Season	110,649	633	785	487	21,015	20,708
Total Cooling Season	--	1,940	--	1,940	1,940	1,247
Total Annual	110,649	2,573	785	2,427	22,955	21,955
Concord						
Total Heating Season	127,258	603	900	483	24,013	23,622
Total Cooling Season	--	1,184	--	1,184	1,184	804
Total Annual	127,258	1,787	900	1,667	25,197	24,426

* All fuels and electric energy reported represents part load results, i.e. dynamic efficiency losses are included.

EXHIBIT 8-11

FUELS AND ELECTRIC ENERGY CONSUMED BY TEST HOUSE IN SELECTED CITIES: ADVANCED HEATING AND COOLING SYSTEMS*

	Advanced Electric Heat Pump I	Advanced Electric Heat Pump II	Free Piston Stirling - Rankine Gas Heat Pump		V-Type Single Cylinder Stirling-Rankine Gas HP		Pulse Combustion Gas Furnace with Central Air Conditioner-High Efficiency		Organic Fluid Absorption Gas Heat Pump	
	Electric (kWh)	Electric (kWh)	Gas (cft)	Elec (kWh)	Gas (cft)	Elec (kWh)	Gas (cft)	Elec (kWh)	Gas (cft)	Elec (kWh)
Houston										
Total Heating Season	**	722	6,109	242	5,604	247	11040	104	8,477	150
Total Cooling Season	**	3,622	45,369	1,587	51,835	1,750	-	5664	76,446	1,546
Total Annual	**	4,344	51,478	1,829	57,439	1,997	11040	5768	84,923	1,696
Birmingham										
Total Heating Season	2,977	1,994	15,373	618	14,264	639	26447	250	21,226	378
Total Cooling Season	4,034	2,510	33,204	1,092	38,608	1,261	-	3887	55,927	1,066
Total Annual	7,011	4,504	48,577	1,710	52,872	1,900	26447	4137	77,153	1,444
Atlanta										
Total Heating Season	3,015	2,001	15,625	626	14,499	640	27058	256	21,587	384
Total Cooling Season	3,760	2,335	30,944	1,020	34,979	1,196	-	3614	52,096	998
Total Annual	6,775	4,336	46,569	1,646	49,478	1,836	27058	3870	73,683	1,382
Tulsa										
Total Heating Season	5,489	4,458	28,366	1,130	27,498	1,317	47433	423	38,967	689
Total Cooling Season	3,082	1,903	24,299	834	29,829	1,173	-	3194	40,893	817
Total Annual	8,571	6,341	52,665	1,964	57,327	2,490	47433	3617	79,860	1,506
Philadelphia										
Total Heating Season	5,896	4,110	30,321	1,225	28,301	1,248	50920	481	41,774	745
Total Cooling Season	2,402	1,483	19,968	654	22,125	781	-	2300	33,590	641
Total Annual	8,298	5,593	50,289	1,879	50,426	2,029	50920	2781	75,364	1,386
Seattle										
Total Heating Season	**	3,441	26,093	1,062	17,266	1,900	45721	379	36,180	651
Total Cooling Season	**	511	7,134	216	7,449	298	-	786	11,950	216
Total Annual	**	3,952	33,227	1,278	24,715	2,198	45721	1165	48,130	867
Columbus										
Total Heating Season	7,391	5,515	37,899	1,543	35,618	1,549	61645	582	51,956	930
Total Cooling Season	2,254	1,398	18,662	612	21,272	713	-	2163	31,394	600
Total Annual	9,645	6,913	56,561	2,155	56,890	2,262	61645	2745	83,350	1,530
Cleveland										
Total Heating Season	8,787	6,598	45,012	1,833	42,400	1,822	72867	688	61,713	1,104
Total Cooling Season	1,729	1,069	14,259	472	15,683	556	-	1652	23,954	464
Total Annual	10,516	7,667	59,271	2,305	58,083	2,378	72367	2340	85,667	1,568
Concord										
Total Heating Season	**	9,116	52,361	2,236	49,749	2,823	83325	656	71,851	1,317
Total Cooling Season	**	691	9,186	294	9,208	467	-	1017	15,432	288
Total Annual	**	9,807	61,547	2,530	58,957	3,290	83325	1673	87,283	1,605

* The energies presented may be underestimated for the Advanced Electric Heat Pump I, free Piston Stirling-Rankine gas heat pump and the Organic Fluid Absorption Gas Heat Pump due to possible underestimation of cycling and defrost losses. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, then adjusted estimates of energies which include these effects can be obtained by multiplying the values in this table by a factor of 1.097 for the heating season, 1.164 for the cooling season and 1.118 for annual energy values.

** System is not sized for application in these cities.

EXHIBIT 8-12

CONVENTIONAL ELECTRIC HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF SEASONAL PERFORMANCE FACTORS ON AN ON-SITE BASIS

Heating	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	2.14 (2.38)*	1.93 (2.14)	1.97 (2.17)	1.68 (1.82)	1.87 (2.05)	1.84 (2.07)	1.69 (1.84)	1.68 (1.82)	1.41 (1.50)
Electric Heat Pump - High Efficiency I	2.56 (2.84)	2.25 (2.49)	2.31 (2.55)	1.93 (2.10)	2.16 (2.36)	2.03 (2.30)	1.89 (2.07)	1.87 (2.04)	1.51 (1.62)
Electric Heat Pump - High Efficiency II	2.39 (2.67)	2.17 (2.40)	2.10 (2.31)	1.96 (2.13)	2.10 (2.30)	2.12 (2.37)	1.88 (2.05)	1.86 (2.03)	1.66 (1.78)
Electric Heat Pump - High Efficiency III	2.26 (2.51)	2.03 (2.25)	2.07 (2.27)	1.72 (1.87)	1.96 (2.14)	2.07 (2.33)	1.75 (1.91)	1.74 (1.89)	1.59 (1.70)
Hybrid Electric Heat Pump - Gas Furnace	1.62 (1.80)	1.33 (1.47)	1.36 (1.53)	1.27 (1.38)	1.21 (1.33)	1.03 (1.32)	1.00 (1.11)	0.97 (1.08)	0.93 (1.01)
Hybrid Electric Heat Pump - Oil Furnace	1.61 (1.78)	1.32 (1.46)	1.34 (1.51)	1.27 (1.39)	1.20 (1.32)	1.08 (1.30)	1.00 (1.11)	0.97 (1.08)	0.93 (1.01)
Electric Heat Pump - Heat Only	2.56 (2.86)	2.33 (2.58)	2.35 (2.58)	2.13 (2.32)	2.31 (2.53)	2.32 (2.61)	2.05 (2.24)	2.03 (2.21)	1.81 (1.94)
Gas Warm Air Furnace	0.59	0.59	0.59	0.60	0.61	0.61	0.62	0.63	0.62
Oil Warm Air Furnace	0.61	0.61	0.61	0.63	0.62	0.62	0.63	0.64	0.64
Central Electric Furnace	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
Baseboard Convectors	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Cooling									
Electric Heat Pump - Standard	1.97 (2.26)	1.94 (2.26)	1.95 (2.29)	1.92 (2.28)	1.97 (2.32)	2.09 (2.34)	1.96 (2.30)	1.99 (2.34)	1.98 (2.28)
Electric Heat Pump - High Efficiency I	2.31 (2.64)	2.28 (2.64)	2.30 (2.68)	2.30 (2.72)	2.32 (2.73)	2.56 (2.85)	2.32 (2.69)	2.37 (2.77)	2.39 (2.75)
Electric Heat Pump - High Efficiency II	2.14 (2.50)	2.12 (2.50)	2.10 (2.44)	2.17 (2.55)	2.16 (2.57)	2.27 (2.59)	2.15 (2.54)	2.19 (2.61)	2.13 (2.50)
Electric Heat Pump - High Efficiency III	2.26 (2.59)	2.23 (2.59)	2.24 (2.62)	1.91 (2.27)	2.26 (2.66)	2.46 (2.79)	2.25 (2.63)	2.29 (2.69)	2.46 (2.86)
Hybrid Electric Heat Pump	1.85 (2.15)	1.83 (2.16)	1.83 (2.18)	1.82 (2.20)	1.84 (2.20)	1.94 (2.22)	1.84 (2.18)	1.87 (2.22)	1.84 (2.22)
Central Air Conditioner - High Efficiency	2.18 (2.50)	2.15 (2.51)	2.16 (2.54)	1.96 (2.33)	2.18 (2.57)	2.32 (2.62)	2.18 (2.54)	2.21 (2.60)	2.23 (2.59)
Central Air Conditioner - Standard	1.82 (2.09)	1.80 (2.09)	1.81 (2.12)	1.87 (2.21)	1.82 (2.15)	1.97 (2.20)	1.82 (2.12)	1.85 (2.17)	1.97 (2.28)
Window Air Conditioners	2.19 (2.52)	2.32 (2.69)	2.30 (2.69)	2.28 (2.69)	2.28 (2.69)	2.10 (2.34)	2.31 (2.69)	2.29 (2.69)	2.32 (2.69)
Annual Heating and Cooling									
Electric Heat Pump - Standard	2.00 (2.28)	1.94 (2.21)	1.94 (2.23)	1.76 (1.96)	1.90 (2.12)	1.87 (2.10)	1.75 (1.96)	1.73 (1.89)	1.45 (1.55)
Electric Heat Pump - High Efficiency I	2.35 (2.67)	2.26 (2.56)	2.31 (2.61)	2.05 (2.29)	2.30 (2.46)	2.09 (2.36)	1.48 (2.19)	1.94 (2.14)	1.57 (1.69)
Electric Heat Pump - High Efficiency II	2.18 (2.53)	2.15 (2.44)	2.10 (2.37)	2.03 (2.26)	2.14 (2.37)	2.14 (2.40)	1.44 (2.15)	1.91 (2.11)	1.70 (1.83)
Electric Heat Pump - High Efficiency III	2.26 (2.58)	2.10 (2.41)	2.15 (2.45)	1.78 (2.02)	2.04 (2.27)	2.12 (2.39)	1.85 (2.04)	1.81 (1.99)	1.65 (1.77)
Hybrid Electric Heat Pump - Gas Furnace	1.81 (2.08)	1.57 (1.79)	1.58 (1.82)	1.42 (1.60)	1.34 (1.50)	1.15 (1.38)	1.12 (1.25)	1.05 (1.19)	0.98 (1.06)
Hybrid Electric Heat Pump - Oil Furnace	1.80 (2.08)	1.57 (1.79)	1.58 (1.82)	1.42 (1.60)	1.34 (1.49)	1.15 (1.38)	1.12 (1.25)	1.05 (1.18)	0.98 (1.06)
Gas Warm Air Furnace/Central Air Conditioner - High Efficiency	1.48	1.01	0.99	0.80	0.77	0.68	0.75	0.72	0.67
Oil Warm Air Furnace/Central Air Conditioner - High Efficiency	1.50	1.03	1.01	0.83	0.79	0.69	0.76	0.73	0.69
Central Electric Furnace/Central Air Conditioner - High Efficiency	1.79	1.42	1.40	1.20	1.17	1.07	1.13	1.08	1.04
Baseboard Convectors/Window Air Conditioners	1.63	1.33	1.31	1.16	1.13	1.06	1.10	1.07	1.04
Electric Heat Pump - Heat Only/Central Air Conditioner - High Efficiency	2.24 (2.56)	2.22 (2.54)	2.24 (2.56)	2.07 (2.28)	2.27 (2.54)	2.32 (2.61)	2.08 (2.31)	2.06 (2.27)	1.84 (1.99)

* Numbers in parentheses represent total on-site seasonal performance factors assuming steady state or full load performance, dynamic efficiency losses are not included.

EXHIBIT 8-13

ADVANCED HEATING AND COOLING SYSTEMS: COMPARISON OF SEASONAL PERFORMANCE FACTORS ON AN ON-SITE BASIS

HEATING	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Advanced Electric Heat Pump I	**	2.46	2.48	2.37	2.42	**	2.35	2.35	**
Advanced Electric Heat Pump II	3.43	2.81	2.93	2.28	2.64	2.98	2.40	2.38	2.02
Free Piston Stirling Rankine Gas Heat Pump	1.48	1.43	1.44	1.40	1.41	1.45	1.38	1.38	1.35
V-Type Single Cylinder Stirling Rankine Gas HP	1.59	1.52	1.53	1.41	1.50	1.82	1.45	1.45	1.34
Organic Fluid Absorption Gas Heat Pump	1.14	1.11	1.11	1.09	1.10	1.13	1.08	1.08	1.06
Pulse Combustion Furnace	0.90	0.91	0.91	0.93	0.93	0.93	0.94	0.94	0.94
COOLING									
Advanced Electric Heat Pump I	**	2.41	2.43	2.44	2.45	**	2.43	2.47	**
Advanced Electric Heat Pump II	3.29	3.29	3.29	3.22	3.33	3.57	3.33	3.33	3.36
Free Piston Stirling Rankine Gas Heat Pump	0.95	0.90	0.91	0.93	0.91	0.90	0.90	0.92	0.90
V-Type Single Cylinder Stirling Rankine Gas HP	1.06	1.04	1.07	0.98	1.10	1.16	1.07	1.12	1.17
Organic Fluid Absorption Gas Heat Pump	0.59	0.56	0.56	0.58	0.56	0.56	0.56	0.57	0.56
ANNUAL HEATING AND COOLING									
Advanced Electric Heat Pump I	**	2.43	2.45	2.40	2.43	**	2.37	2.37	**
Advanced Electric Heat Pump II	3.31	3.08	3.12	2.56	2.82	3.06	2.59	2.52	2.11
Free Piston Stirling Rankine Gas Heat Pump	1.01	1.07	1.09	1.19	1.21	1.33	1.22	1.27	1.28
V-Type Single Cylinder Stirling Rankine Gas HP	1.11	1.17	1.21	1.19	1.33	1.65	1.31	1.36	1.31
Organic Fluid Absorption Gas Heat Pump	0.64	0.71	0.72	0.83	0.86	0.99	0.88	0.94	0.97
Pulse Combustion Furnace/Central Air Conditioner-High Efficiency	1.91	1.43	1.41	1.19	1.14	1.02	1.11	1.06	1.01

The performance factors presented may be underestimated for the advanced electric heat pump I, free piston Stirling-Rankine gas heat pump and organic fluid absorption gas heat pump due to possible underestimation of cycling and defrost losses. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, then adjusted estimates of performance factors which include these effects can be obtained by multiplying the values in this table by a factor of 1.097 for the heating season, 1.164 for the cooling season and 1.118 for annual performance factor values.

** System is not sized for application in these cities

EXHIBIT 8-14

CONVENTIONAL HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF TOTAL FUEL AND ELECTRIC ENERGY CONSUMPTION ON-SITE

(ALL FIGURES ARE BTU X 10^{-6})

Heating	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	4.79 (4.30)*	12.92 (11.65)	12.93 (11.73)	26.88 (24.75)	25.98 (23.74)	23.42 (20.82)	35.28 (32.38)	42.20 (38.78)	57.33 (53.73)
Electric Heat Pump - High Efficiency I	4.00 (3.60)	11.09 (10.04)	11.04 (10.02)	23.40 (21.52)	22.58 (20.64)	21.29 (18.77)	31.44 (28.79)	37.81 (34.63)	53.29 (49.87)
Electric Heat Pump - High Efficiency II	4.28 (3.84)	11.53 (10.42)	12.16 (11.03)	23.01 (21.16)	23.20 (21.21)	20.42 (18.20)	31.68 (29.06)	37.93 (34.83)	48.55 (45.41)
Electric Heat Pump - High Efficiency III	4.54 (4.08)	12.30 (11.12)	12.33 (11.21)	26.24 (24.13)	24.86 (22.74)	20.82 (18.54)	34.02 (31.22)	40.71 (37.42)	50.82 (47.59)
Hybrid Electric Heat Pump - Gas Furnace	6.31 (5.68)	18.84 (16.93)	18.79 (16.72)	35.40 (32.35)	40.38 (36.72)	40.02 (32.82)	59.50 (53.75)	73.04 (65.28)	86.45 (80.25)
Hybrid Electric Heat Pump - Oil Furnace	6.37 (5.74)	18.98 (17.07)	18.96 (16.87)	35.45 (32.42)	40.59 (36.93)	40.16 (33.28)	59.54 (53.86)	73.11 (65.49)	86.38 (80.23)
Electric Heat Pump - Heat Only	4.00 (3.58)	10.71 (9.66)	10.87 (9.89)	21.17 (19.43)	21.08 (19.27)	18.58 (16.57)	29.04 (26.53)	34.89 (31.93)	44.47 (41.61)
Gas Warm Air Furnace	17.45	41.98	42.90	74.81	79.94	70.42	96.24	112.81	129.32
Oil Warm Air Furnace	16.81	40.73	41.68	71.82	78.10	69.15	94.27	110.78	126.86
Central Electric Furnace	10.37	25.43	25.95	45.54	49.33	44.05	60.33	71.72	81.96
Baseboard Convectors	10.15	24.90	25.40	44.85	48.61	43.10	59.45	70.68	80.62
Cooling									
Electric Heat Pump - Standard	24.63 (21.41)	17.10 (14.68)	16.04 (13.66)	13.22 (11.13)	10.26 (8.70)	3.35 (2.99)	9.56 (8.18)	7.35 (6.25)	4.66 (4.04)
Electric Heat Pump - High Efficiency I	20.24 (17.67)	14.46 (12.48)	13.58 (11.63)	10.78 (9.11)	8.63 (7.36)	2.74 (2.45)	8.03 (6.90)	6.12 (5.23)	3.85 (3.34)
Electric Heat Pump - High Efficiency II	22.31 (19.14)	15.57 (13.20)	14.95 (12.89)	11.77 (10.00)	9.30 (7.79)	3.09 (2.71)	8.68 (7.33)	6.64 (5.57)	4.31 (3.68)
Electric Heat Pump - High Efficiency III	21.46 (18.70)	14.91 (12.83)	13.99 (11.94)	13.28 (11.19)	8.95 (7.60)	2.86 (2.52)	8.33 (7.14)	6.41 (5.46)	3.75 (3.21)
Hybrid Electric Heat Pump	25.84 (22.01)	18.00 (15.27)	16.92 (14.26)	13.73 (11.36)	10.82 (9.07)	3.58 (3.12)	10.09 (8.53)	7.78 (6.54)	4.89 (4.06)
Central Air Conditioner - High Efficiency	22.22 (19.33)	15.44 (13.27)	14.48 (12.34)	12.93 (10.90)	9.26 (7.85)	3.04 (2.68)	8.62 (7.38)	6.62 (5.64)	4.04 (3.47)
Central Air Conditioner - Standard	26.50 (23.10)	18.45 (15.88)	17.31 (14.77)	13.60 (11.47)	11.07 (9.41)	3.54 (3.18)	10.31 (8.84)	7.92 (6.76)	4.68 (4.04)
Window Air Conditioners	11.61	8.14	7.81	6.52	5.39	2.62	5.14	4.25	2.74
Annual Heating and Cooling									
Electric Heat Pump - Standard	29.41 (25.71)	30.02 (26.33)	28.97 (25.38)	40.11 (35.88)	36.24 (32.45)	26.76 (23.81)	44.84 (40.56)	49.55 (45.03)	61.99 (57.78)
Electric Heat Pump - High Efficiency I	24.24 (21.28)	25.55 (22.52)	24.62 (21.64)	34.18 (30.62)	31.20 (28.00)	24.03 (21.22)	39.47 (35.70)	43.92 (39.86)	57.14 (53.21)
Electric Heat Pump - High Efficiency II	26.59 (22.98)	27.10 (23.62)	27.11 (23.92)	34.78 (31.16)	32.51 (29.00)	23.51 (20.92)	40.36 (36.39)	44.57 (40.40)	52.86 (49.09)
Electric Heat Pump - High Efficiency III	26.00 (22.78)	27.21 (23.95)	26.32 (23.15)	39.51 (35.31)	33.80 (30.35)	23.68 (21.06)	42.36 (38.36)	47.12 (42.88)	54.57 (50.80)
Hybrid Electric Heat Pump - Gas Furnace	32.15 (27.89)	36.83 (32.21)	35.72 (30.98)	49.14 (43.71)	51.20 (45.79)	43.59 (35.94)	69.59 (62.29)	80.81 (71.82)	91.34 (84.32)
Hybrid Electric Heat Pump - Oil Furnace	32.22 (27.95)	36.97 (32.35)	35.89 (31.13)	49.19 (43.78)	51.41 (46.00)	43.74 (36.40)	69.63 (62.40)	80.89 (72.02)	91.24 (84.30)
Gas Warm Air Furnace/Central Air Conditioner - High Efficiency	39.64	57.42	57.38	87.74	89.30	73.46	104.86	119.43	133.36
Oil Warm Air Furnace/Central Air Conditioner - High Efficiency	39.03	56.17	56.16	84.74	87.36	72.18	102.89	117.42	130.91
Central Electric Furnace/Central Air Conditioner - High Efficiency	32.59	40.87	40.43	58.47	58.59	47.09	68.95	78.34	86.00
Baseboard Convectors/Window Air Conditioners	21.76	33.04	33.21	51.37	54.00	45.72	64.59	74.93	83.36
Electric Heat Pump - Heat Only/Central Air Conditioner - High Efficiency	26.22 (22.91)	26.15 (22.93)	25.35 (22.23)	34.10 (30.33)	30.34 (27.12)	21.62 (19.25)	37.66 (33.91)	41.51 (37.57)	48.51 (45.08)

* Numbers in parentheses represents total on-site energy consumption assuming steady state or full-load performance dynamic efficiency losses are not included.

EXHIBIT 8-15

ADVANCED HEATING AND COOLING SYSTEMS: COMPARISON OF TOTAL FUEL AND ELECTRIC ENERGY CONSUMPTION ON-SITE

(All figures are Btu x 10^{-6})

	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Heating									
Advanced Electric Heat Pump I	**	10.16	10.29	19.07	20.12	**	25.35	30.12	**
Advanced Electric Heat Pump II	2.46	6.81	6.83	15.52	14.03	11.74	18.82	22.52	31.11
Free Piston Stirling-Rankine Gas Heat Pump	6.93	17.48	17.76	32.22	34.50	29.72	43.17	51.27	60.00
V-Type Single Cylinder Stirling-Rankine Gas HP	6.45	16.44	16.68	31.99	32.56	23.75	40.90	48.62	60.06
Organic Fluid Absorption Gas Heat Pump	8.99	22.52	22.90	41.32	44.32	38.40	55.13	65.48	76.35
Pulse Combustion Furnace	11.39	27.30	27.93	48.88	52.56	47.01	63.63	75.22	86.06
Cooling									
Advanced Electric Heat Pump I	**	13.77	12.83	10.52	8.20	**	7.69	5.90	**
Advanced Electric Heat Pump II	12.50	8.57	7.97	6.49	5.06	1.74	4.77	3.65	2.36
Free Piston Stirling-Rankine Gas Heat Pump	50.79	36.94	34.43	27.15	22.20	7.87	20.75	15.87	10.19
V-Type Single Cylinder Stirling-Rankine Gas HP	57.81	42.91	39.06	33.83	24.79	8.47	23.71	17.58	10.80
Organic Fluid Absorption Gas Heat Pump	81.72	59.57	55.50	43.68	35.78	12.68	33.44	25.54	16.41
Annual Heating and Cooling									
Advanced Electric Heat Pump I	**	23.93	23.12	29.59	28.32	**	33.04	36.02	**
Advanced Electric Heat Pump II	14.96	15.37	14.80	22.02	19.09	13.49	23.59	26.17	33.47
Free Piston Stirling-Rankine Gas Heat Pump	57.72	54.42	52.19	59.37	56.70	37.59	63.91	67.14	70.19
V-Type Single Cylinder Stirling-Rankine Gas HP	64.25	59.36	55.74	65.83	57.35	32.22	64.61	66.20	70.86
Organic Fluid Absorption Gas Heat Pump	90.71	82.08	78.40	85.00	80.10	51.09	88.57	91.02	92.76
Pulse Combustion Furnace/Central Air Conditioner - High Effic ency	30.83	40.63	40.33	59.83	60.45	49.71	71.05	80.88	89.65

* All data in this table are based on steady state or full-load performance operation, dynamic efficiency losses are not included. If dynamic losses of the advanced heat pumps were similar to those of the conventional electric heat pumps, then multiplying the numbers in the table above by the following factors, 1.097 for heating season values, 1.164 for cooling season values, and 1.118 for annual values will result in approximate total energy consumption corrected for dynamic efficiency effects. Exceptions include the modulated V-type Single Cylinder Stirling-Rankine Gas Heat Pump for which the tabular data already include dynamic effects and the Pulse Combustion Furnace for which data was scaled from Gas Warm Air Furnace values.

** System is not sized for application in these cities.

EXHIBIT 8-16

CORRELATION OF RESIDENTIAL HEATING ENERGY FOR CONVENTIONAL HEATING AND COOLING SYSTEMS

(Data points not included for clarity)

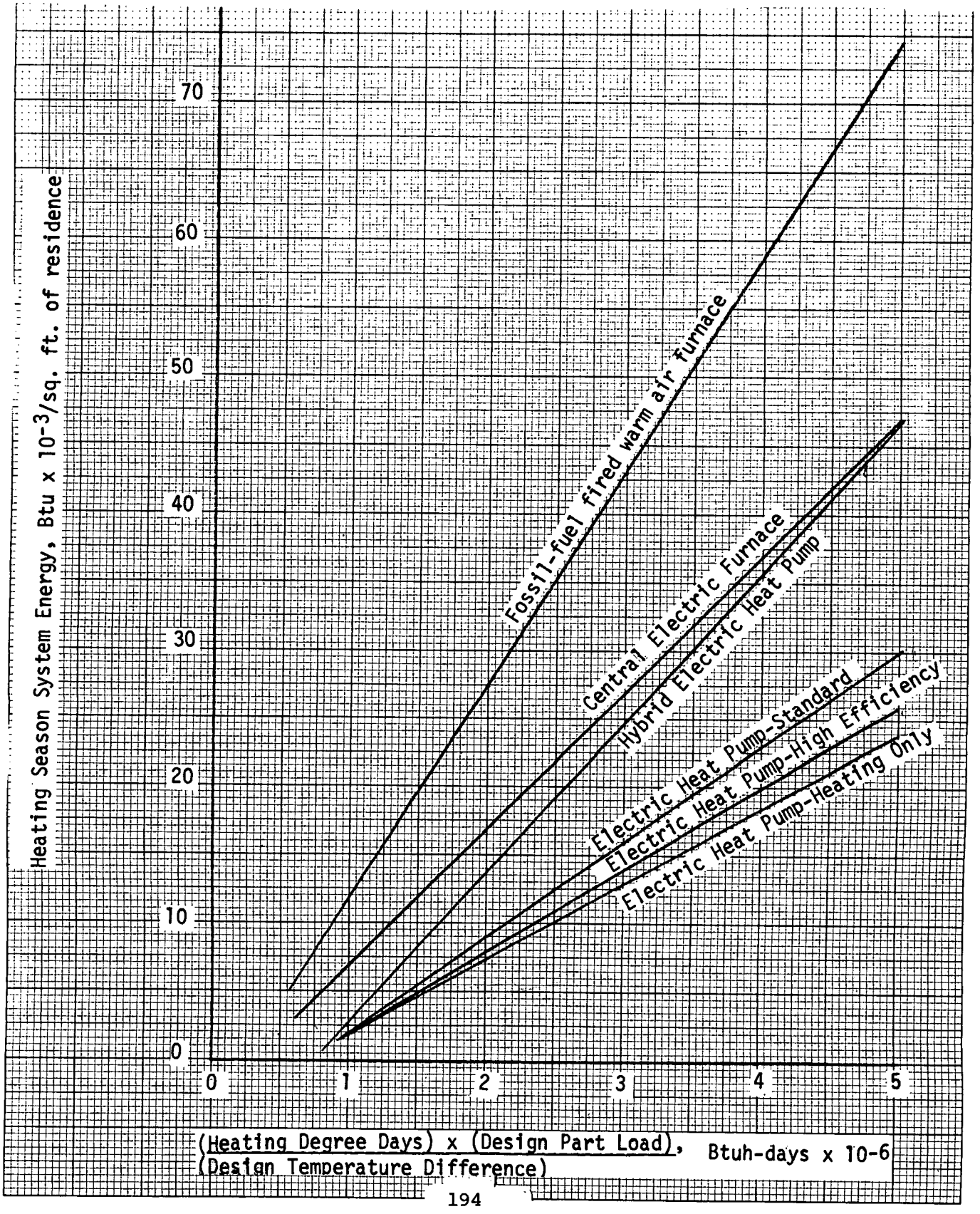


EXHIBIT 8-17

CORRELATION OF RESIDENTIAL COOLING ENERGY FOR CONVENTIONAL HEATING AND COOLING SYSTEMS

(Data points not included for clarity)

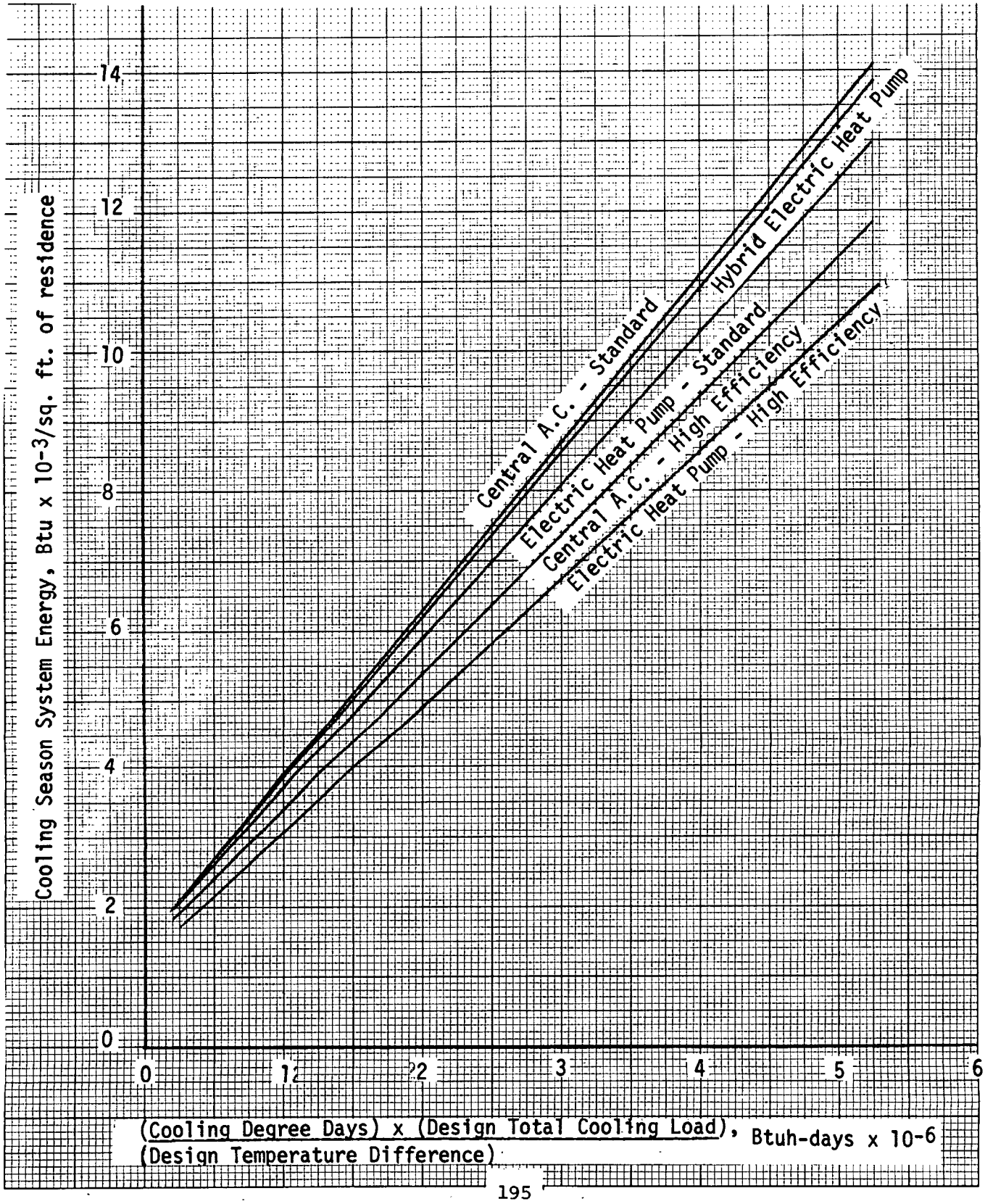


EXHIBIT 8-18

CORRELATION OF RESIDENTIAL HEATING ENERGY FOR ADVANCED HEATING AND COOLING SYSTEMS

(Data points not included for clarity)

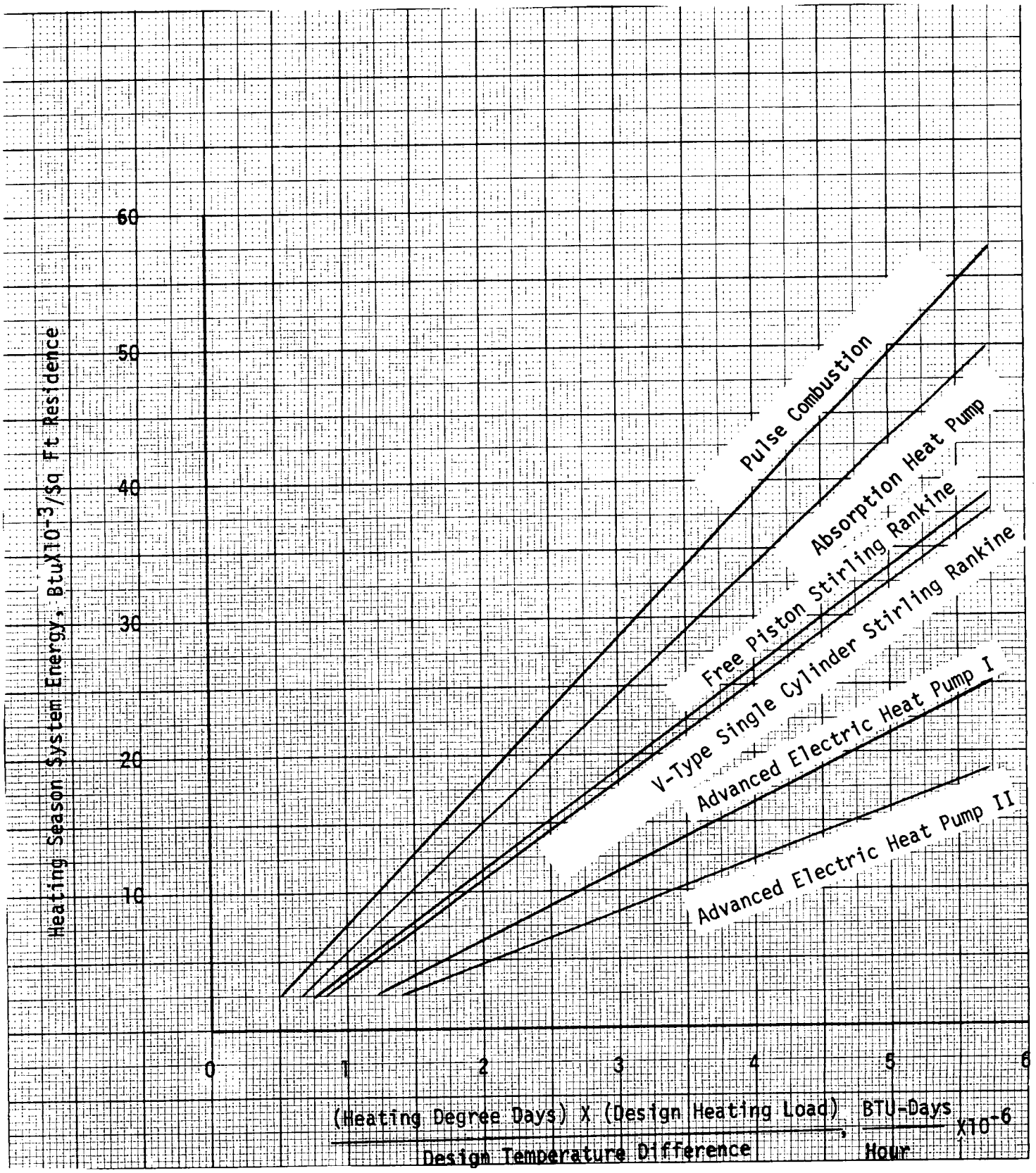


EXHIBIT 8-19

CORRELATION OF RESIDENTIAL COOLING ENERGY FOR ADVANCED HEATING AND COOLING SYSTEMS

(Data points not included for clarity)

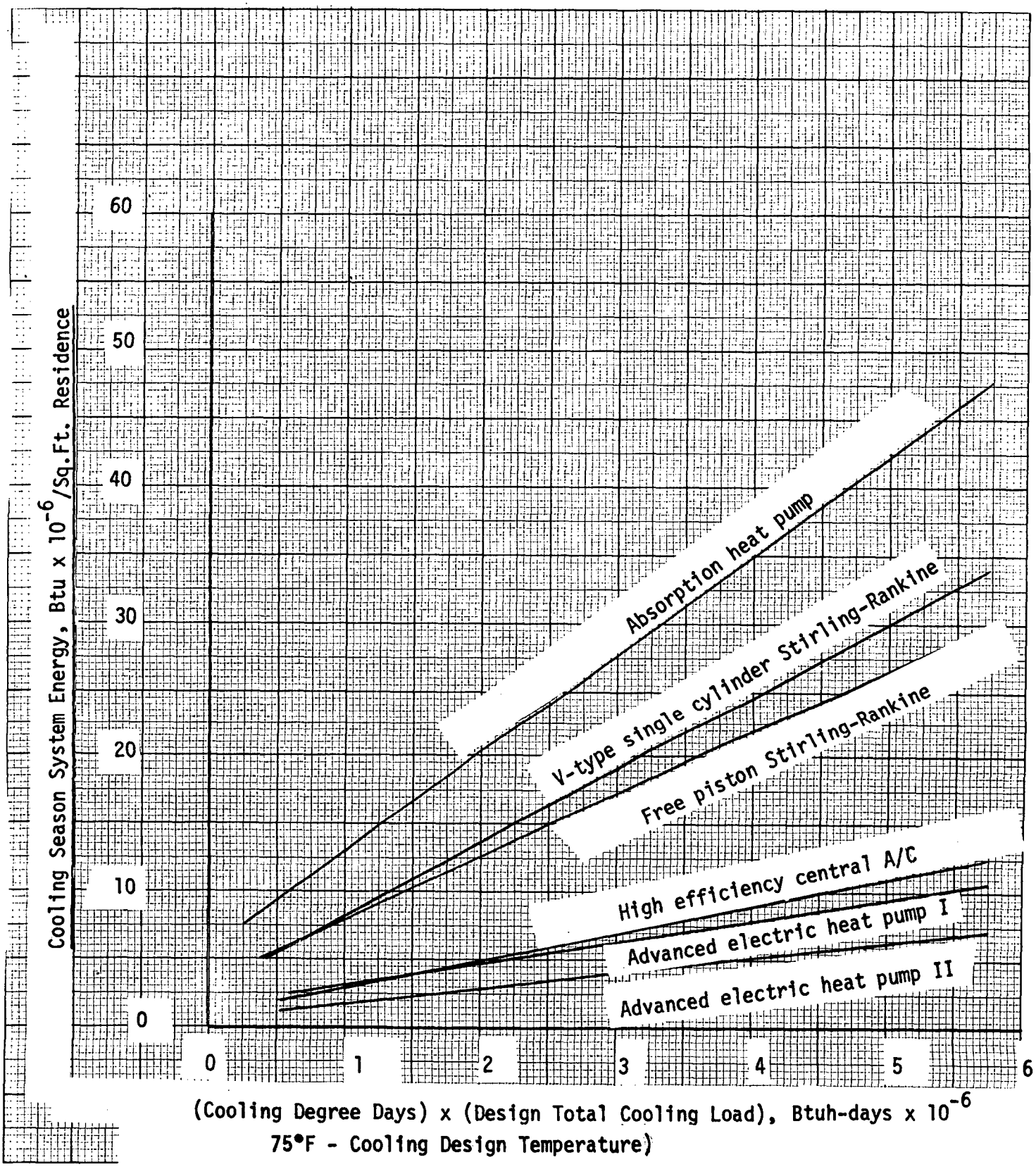
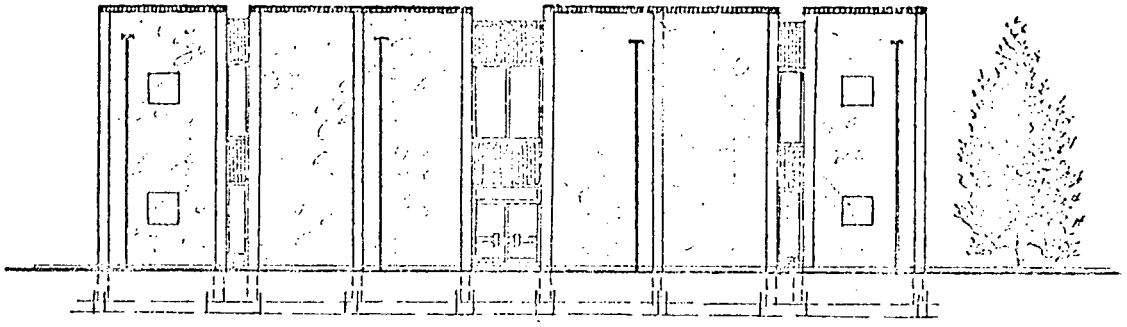
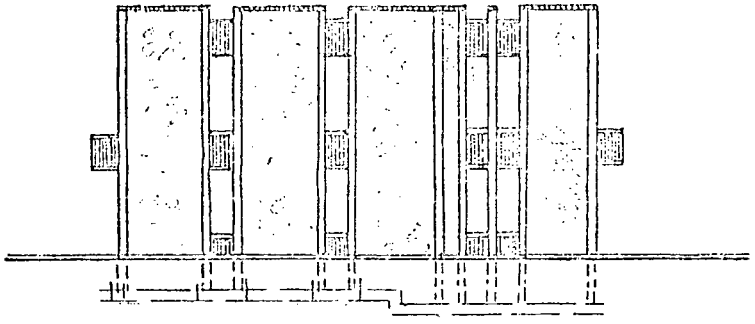


EXHIBIT 8-20

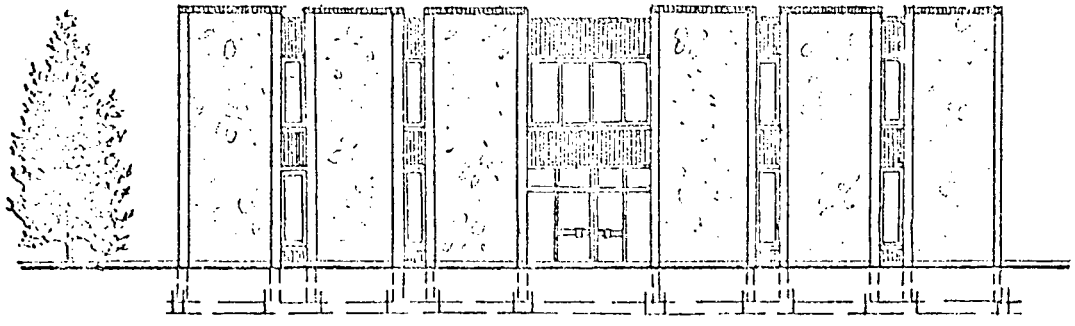
COMMERCIAL BUILDING: ARCHITECTURAL SKETCHES



Rear Elevation



Right Side Elevation



Front Elevation

EXHIBIT 8-21

MOUNTAINSIDE COMMERCIAL BUILDING GENERAL CHARACTERISTICS

Building Style	Two stories, masonry construction
Gross Floor Area	5580 square feet
Exposure	Front of building faces north
Occupancy	40 persons (maximum)
Lighting and Equipment Baseload	14 KW (maximum)
Construction Walls	Stucco surface over 12" concrete blocks (38# light, filled), 1-1/4" furring (16" on centers) and 1/2" styrofoam. Inside surface is 1/2" gypsum board. Net wall area: 4724 square feet.
Roof	3/8" asphalt built up roof over 1/2" plyscore sheathing, 2" x 10" rafters (16" on centers), fiberglass insulation (6") above 1/2" acoustical tile. Roof area: 2790 square feet.
Windows and Doors	Safety glass, 1/2" double glazed. Total glass area: 732 square feet.
Floor Slab	Concrete slab, on grade, 24" insulation
Ceilings	Hung ceiling between floors carries ducts and electric and water services. Ceiling height is 11 feet for each floor
Stairwell	Center staircase located just inside front entrance, double back design with turnaround landing.

EXHIBIT 8-22

MOUNTAINSIDE COMMERCIAL BUILDING: THERMAL INSULATION PROPERTIES COMPARED WITH ASHRAE STANDARD 90-75

City	Heating Degree Days	Construction Wall Overall Heat Transfer Coeffient ($Btu/Hr/ft^2/^oF$)		Roof Overall Heat Transfer Coeffient ($Btu/Hr/ft^2/^oF$)		Slab on Grade Heat Transfer Resistance Factor ($^oF\text{-}Hr\text{-}ft^2/Btu$)		Overall Thermal Transfer Factor ($Btu/Hr/ft^2$)	
		ASHRAE	Building	ASHRAE	Building	ASHRAE	Building	ASHRAE	Building
Houston	1278	0.366	0.121	0.100	0.034	1.672	5.263	30.498	12.091
Birmingham	2551	0.341	0.121	0.100	0.034	2.527	5.263	31.589	12.316
Atlanta	2961	0.333	0.121	0.100	0.034	2.802	5.263	31.593	12.214
Tulsa	3360	0.316	0.121	0.094	0.034	3.406	5.263	32.405	12.646
Philadelphia	4486	0.304	0.121	0.089	0.034	3.826	5.263	33.248	12.338
Seattle	4424	0.305	0.121	0.090	0.034	3.785	5.263	35.443	12.453
Columbus	5660	0.282	0.121	0.080	0.034	4.615	5.263	33.500	12.308
Cleveland	6351	0.268	0.121	0.074	0.034	5.079	5.263	33.784	12.437
Concord	7303	0.249	0.121	0.066	0.034	5.771	5.263	34.330	12.533

(a) ASHRAE refers to heat transfer coefficients and thermal resistance factors recommended for commercial buildings in ASHRAE Standard 90-75.

(b) Building refers to heat transfer coeffient and thermal resistance factors calculated for the Mountainside commercial office building.

EXHIBIT 8-23

MOUNTAINSIDE COMMERCIAL BUILDING HEATING AND COOLING DESIGN LOADS

	Heating		Cooling			
	97½% Design Dry Bulb Temperature (°F)	Design Heat Load (Btuh)	2½% Design Dry Bulb Temperature (°F)	2½% Design Wet Bulb Temperature (°F)	Design Sensible Cooling Load (Btuh)	Design Total Cooling Load (Btuh)
Houston	32	86,151	94	80	143,447	206,119
Birmingham	22	104,613	94	78	143,447	195,667
Atlanta	23	106,664	92	77	139,380	189,992
Tulsa	16	118,971	99	78	155,656	203,852
Philadelphia	15	121,022	90	77	135,315	187,535
Seattle	32	86,152	79	65	112,937	124,957
Columbus	7	139,484	88	76	131,244	178,640
Cleveland	7	139,484	89	75	133,277	176,653
Concord	-7	166,150	88	73	131,244	164,972

EXHIBIT 8-24

ELECTRIC HEAT PUMP, CONVENTIONAL SYSTEMS, AND ADVANCED HEAT PUMPS FOR LIGHT COMMERCIAL BUILDING APPLICATION: ANNUAL ON-SITE FUELS AND ELECTRIC ENERGY CONSUMPTION FOR COMBINED HEATING AND COOLING

Electric Heat Pump and Conventional Systems		Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	Elec (kWh)	32,364	27,036	24,768	26,382	19,635	15,435	21,594	21,156	24,108
Electric Resistance Heating/ Electric Air Conditioning	Elec (kWh)	36,357	36,528	34,107	39,549	34,317	29,286	36,885	37,881	42,138
Rooftop Gas Furnace/Electric Air Conditioning	Gas (cft)	53,817	103,479	108,930	141,489	139,422	164,826	183,675	200,079	219,330
	Elec (kWh)	33,024	23,658	21,006	20,010	13,590	6,489	10,047	8,394	8,940
Advanced Heat Pump Systems										
Brayton-Rankine Gas Heat Pump	Gas (cft)	228,438	193,050	177,384	190,635	135,753	104,769	147,654	142,914	163,317
	Elec (kWh)	5,097	4,305	3,954	4,266	3,168	2,676	3,363	3,288	4,530
Rankine-Rankine Gas Heat Pump	Gas (cft)	253,917	199,497	181,443	184,425	137,460	96,048	136,401	128,985	142,839

EXHIBIT 8-25

ELECTRIC HEAT PUMP, CONVENTIONAL SYSTEMS, AND ADVANCED HEAT PUMPS FOR LIGHT COMMERCIAL BUILDING APPLICATION: COMPARISON OF AVERAGE HEATING, COOLING AND ANNUAL PERFORMANCE FACTORS - ON-SITE BASIS

Electric Heat Pump and Conventional Systems	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard									
Heating	2.67	2.48	2.40	2.35	2.44	2.33	2.14	2.13	2.04
Cooling	2.14	2.13	2.13	2.13	2.11	2.18	2.19	2.20	2.18
Annual	2.17	2.20	2.20	2.20	2.24	2.28	2.16	2.16	2.09
Heating/Electric Air Conditioning									
Heating	0.95	0.95	0.94	0.95	0.95	0.97	0.98	0.97	0.98
Cooling	2.13	2.13	2.13	2.13	1.78	2.21	2.20	2.22	2.13
Annual	1.93	1.63	1.60	1.47	1.28	1.20	1.26	1.20	1.19
Rooftop Gas Furnace/Electric Air Conditioning									
Heating	0.36	0.47	0.44	0.49	0.46	0.47	0.51	0.51	0.52
Cooling	1.97	1.97	1.97	1.97	1.95	2.06	2.05	2.07	2.06
Annual	1.44	1.10	1.03	0.95	0.81	0.64	0.73	0.68	0.69
Advanced Heat Pump Systems									
Brayton-Rankine Gas Heat Pump									
Heating	1.22	1.16	1.10	1.09	1.15	1.17	1.05	1.05	1.00
Cooling	0.96	0.93	0.93	0.91	0.95	0.88	0.94	0.94	0.89
Annual	0.98	0.98	0.97	0.97	1.03	1.05	1.00	1.01	0.96
Rankine-Rankine Gas Heat Pump									
Heating	1.54	1.46	1.42	1.41	1.45	1.46	1.38	1.38	1.38
Cooling	0.91	0.93	0.93	0.95	0.92	0.98	0.95	0.98	0.99
Annual	0.95	1.02	1.02	1.08	1.09	1.24	1.17	1.21	1.20

EXHIBIT 8-26

ELECTRIC HEAT PUMP, CONVENTIONAL SYSTEMS, AND ADVANCED HEAT PUMPS FOR LIGHT COMMERCIAL BUILDING APPLICATION: TOTAL ON-SITE ENERGY CONSUMPTION (MILLION OF BTU'S FOR COMBINED HEATING AND COOLING)

Electric Heat Pump and Conventional Systems	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	110.46	92.27	84.53	90.04	67.01	52.68	73.70	72.21	82.28
Electric Resistance Heating/Electric Air Conditioning	124.09	124.67	116.41	134.98	117.12	99.95	125.89	129.29	143.82
Rooftop Gas Furnace/Electric Air Conditioning	166.53	184.22	180.62	209.78	185.80	186.97	217.97	228.73	249.84
Advanced Heat Pump Systems									
Brayton-Rankine Gas Heat Pump	245.83	207.74	190.87	205.19	146.57	113.90	159.13	154.14	178.78
Rankine-Rankine Gas Heat Pump	253.92	199.50	181.44	184.43	137.46	96.05	136.40	128.99	142.84

9. HEAT PUMP OWNING AND OPERATING COSTS

9.1 Introduction

The focus of this chapter is on heat pump owning and operating costs. In order to place these costs in the proper perspective, owning and operating costs were also determined for the conventional residential and light commercial space conditioning systems.

The system owning and operating costs (or life-cycle costs) were determined for both residential and commercial applications by first calculating installed costs, maintenance costs, and energy costs. Life-cycle costs were then calculated with and without inflationary effects on energy and maintenance charges. The discussion of the above costs is organized into three parts: Section 9.2 discusses the methodology and data sources utilized, expanding upon those in Chapter 2. Section 9.3 presents the results of the cost analysis of residential heat pumps and other heating and cooling systems; Section 9.4 presents the results of the cost analysis of commercial building space conditioning systems.

The analysis presented here will form an integral part in determining the market prospects for advanced heat pumps as discussed in Chapter 10.

9.2 Methodology and Data Sources

This section complements Chapter 2, Methodology. A brief description of those data needed for an understanding of the life cycle calculations is included in this section. The basic calculational approach was identical for both residential and commercial HVAC systems. Where circumstances required variations, these are discussed in the appropriate place in the text.

9.2.1 Data Sources

Cost data reported in the study are in general current as of the First Quarter of 1977 or in the case of the commercial systems, the Second Quarter of 1977. Regional representatives of the manufacturers whose equipment is featured in this study were consulted to verify compatibility of components and equipment sizing procedures. Installing contractors or the manufacturers themselves were the source of equipment and installation cost data used. In addition, the installing contractors provided information relating to freight costs, required electrical service charges, profit and overhead charges. The installed cost obtained, in addition to those previously mentioned, consisted of the following elements: material component prices and installation man-hours; insulated and non-insulated ductwork material costs and installation man-hours; installation labor costs, state and local sales/use taxes; and, as required, the cost of a chimney and oil tank. Additional costs peculiar to a commercial installation included those for a rigger and operator to lift the units to the roof for mounting, an equipment site delivery charge, locking thermostat covers and additional ductwork. The cost of room air conditioners was obtained from the Sears and Roebuck Co. Summer 1977 Catalogue. Where possible, equipment costs and contractor's fees were verified by discussions with other contractors, with manufacturers and by comparisons with data reported in the Means 1977 Building Construction Cost Data Book (Means, 1977).

Labor costs (sheetmetal workers and other HVAC laborers) were taken as representative of the Philadelphia area, since most of the installed

costs were obtained from a central New Jersey contractor. For other cities, appropriate wage rates were taken from data published by the Bureau of Labor Statistics for Winter 1976.

Maintenance cost data for residential installations were provided by the National Environmental Systems Contractors Association (NESCA, 1976) and represent conditions as of Third Quarter of 1976. They represent the average retail price of a full service contract to the consumer. The room air conditioner maintenance fee was obtained from Sears. Maintenance costs were assumed not to vary regionally for purposes of these calculations. An estimate of the cost of a full service maintenance contract for commercial installations was obtained from the New Jersey HVAC contractor previously mentioned (Tri-Boro, 1977) and represents prices as of the Second Quarter of 1977. Maintenance costs for commercial and residential systems are included as Exhibit 9-1.

Depreciation periods for the various components of each system (e.g. the HVAC equipment, ductwork, chimney, etc.) were taken from the ASHRAE Systems Handbook and Product Directory (ASHRAE, 1975) or represent current Internal Revenue Service recommended practices. Depreciation periods for commercial and residential systems are shown in Exhibit 9-2.

Current state and local taxes were obtained from the sales tax departments of the individual states in which the facilities were located.

Current rate information, used in determining all energy costs, was obtained by contacting the electric and gas utilities serving each city involved in the simulation study. Oil prices were obtained from usually two private oil companies in the city.

The cost of money was taken as 9 percent, the current cost of a home mortgage loan. (The portion of the purchase price of the system which is paid in the down payment on the house is assumed to have the same opportunity value to the consumer as that paid over the span on the mortgage. Hence the full installed cost of each system was amortized.)

9.2.2 Calculational Approach

Since the various heating and cooling systems had unequal service lives, it was necessary to compare life-cycle costs on a "perpetual replacement" basis. Two basic approaches were possible: the systems may be compared in terms of their capitalized costs or their annualized costs. We chose the latter method, since the result represents the yearly cash outflow resulting from the cost of operation and maintenance of each system (residential or commercial) as well as the annual payments required to equal the present value of the initial outlay for the installation. Where practical, costs associated with particular pieces of equipment, e.g. ductwork for the heat pump, were consigned to that equipment to allow the effect of differing depreciation periods to be fully illustrated. The following formula (simplified for illustration purposes here) was used for the calculations:

Annualized Cost = (Installed Cost) (Capital Recovery Factor) + (Annual Energy Use) (Unit Price of Energy) + Annual Maintenance Cost

Although this procedure would indeed give the yearly cost outlay necessary for a system, inflationary increases in energy and maintenance charges could not be represented. Therefore, the effects of inflation on energy and maintenance fees on annualized costs were calculated by a second method described in the next paragraph.

In order to determine inflationary effects on annualized costs, a method of analysis suggested in a paper by F.R.S. Dressler (1974) was utilized which computes the effective annual cost of owning and operating the HVAC equipment over its expected life cycle allowing for inflation in operating costs. Again, the cost of money was taken as 9 percent per year. The economic life of the system was considered to be the smallest depreciation period of any major component of the system. An overall energy inflation of 8 percent per year was used in the calculation. This represents current thinking as to the forecast electricity price increase in the case of total decontrol of natural gas and oil over a nine year heat pump depreciated life. Maintenance costs, composed of both labor and material fees were escalated at 6 percent per year, which follows relevant consumer price index trends.

9.2.3 Basis for the Analysis

The determination of life cycle costs for the alternative residential and commercial heating and cooling systems utilized the techniques outlined in the previous sections. Systems were treated in a number of different ways in order to represent either prevailing conditions or most likely near future conditions. The cost of insulation of ductwork was considered. All systems are shown with insulated ducts which, according to our investigation, is the practice with electric heating systems but not with combustion systems. In addition, costs for combustion systems with uninsulated ducts were also determined, this representing current practice. For comparative purposes, all life-cycle costs were calculated using insulated duct systems.

In a number of cases, equipment was priced as either an add-on (replacement) installation or as a new installation. The basis of a

new installation involves placing an entire system into a home being built. This may involve a central electric furnace, central air conditioner, and ductwork. An add-on installation cost would assume that all auxiliary equipment, for example, insulated ducts or air handler, is already in place and, therefore, only the cost of the HVAC equipment itself would be considered. Substantial first cost reductions are, therefore, realized in add-on installations.

All five commercial HVAC systems were considered as new installations and were priced out including insulated ductwork, wiring, plumbing, electrical service change, etc. The single package electric heat pump was priced including an economizer package. The economizer package includes additional dampers, controls, and the economizer. The economizer package gives greater flexibility to the heat pump system since outside air may be used for makeup requirements. This is also an energy saving device, in that, on a cooler day, the outside air (which is cooler than the inside room air) can reduce the air conditioning requirements.

Depreciation periods of commercial and residential installation were considered identical where appropriate. This was especially true in the case of commercial or residential heat pumps with service lives of approximately 9 years. However, in the commercial installation of a rooftop gas heater/electric air conditioner package, the service life was taken as 10 years. This figure was based on discussions with the aforementioned New Jersey HVAC contractor.

Charges for taxes and shipping ($4.50 per 100 pounds) were identical for the commercial and residential systems. Contractor overhead and

profit margins followed the actual practice of higher rates for residential installations, lower for commercial. Generally, the procedure calls for a lower overhead and profit margin where the labor to material ratio is low, since tighter controls could be placed on costs. Smaller installations, characteristic of the residential market, are also not as influenced as commercial systems are by competing bids from other contractors.

Seasonal and annual energy costs for conventional and advanced heat pumps and other HVAC systems were calculated on the basis of the actual system fossil fuel or electric energy consumption as determined from the simulation study described in Chapter 8. The price for electricity, natural gas and No. 2 fuel oil was determined by contacting the electric or gas utility in each of the nine cities included in the study. In addition to applicable current utility rates for residential service, cost adjustment factors for purchased gas or electric power, fuel adjustment charges, cost of service rider, customer service charge, and state and local sales and utility taxes were also obtained, and are accounted for in the calculations. Fuel oil prices used are based on an actual price quotation from fuel oil distributors in eight of the nine cities in the study for No. 2 fuel oil delivered to a residential location in that city. No price quotation was obtained for Houston since apparently there is little or no residential market demand for No. 2 fuel oil in that city. Instead, the price of No. 2 fuel oil in Tulsa was used as a representative price for Houston for the purpose of cost comparisons.

The costs presented account for all of heating and cooling system energy consumption. For example, the gas furnace energy cost includes both the cost of gas and the cost of electricity for indoor blower operation. In determining the residential energy cost, the test building

was assumed to have a certain base load of lighting and electric or gas appliances, based on the usage reported in the Department of Commerce 1970 Census of Housing. Energy use was simulated in the commercial building by assuming an occupancy schedule and a certain base load (including lighting, office machines). The base load was the actual maximum design load demand of 14kW adjusted by occupancy schedule. If the electric utility offered an "all-electric" rate, a household with electric heating and cooling was assumed to have all electric appliances as well. A house with a gas or oil-fired heating system was assumed to have appliances typical of that area (as per census data).

The energy costs are presented on a seasonal basis and yearly basis for residential systems. Such seasonal distinctions were not possible in the commercial systems since both heating and cooling may be necessary during any month. Therefore, only yearly energy costs could be computed for the commercial systems.

9.3 Analysis of Residential Heat Pump System Costs

This section presents a brief analysis of residential heat pump system installed cost (first cost), energy cost and life-cycle cost. The latter, of course, is dependent on the first two measures, and permits comparison of projects or elements of projects with unequal economic lives.

The basis for developing these cost measures was discussed in the preceeding section, and will not be discussed further here, except where such observation would further elucidate the analysis.

9.3.1 Installed Cost Analysis

A comparison of installed cost of conventional electric heat pumps and other systems is shown in Exhibit 9-4. The lowest cost system for any of the locations simulated is the baseboard convector with window air conditioners. This minimum cost is achieved, however, through avoidance of installing a central air distribution system and by use of window air conditioners which do not condition the total building space as is the case with systems that centrally distribute the comfort cooling. Since installation of a central air distribution system in an existing building has been stated to be prohibitively expensive (Hittman, 1976), it should be recognized that installation of a baseboard convector/window air conditioner system would effectively prevent future upgrading with a more energy efficient system that would require central air distribution. Since all of the advanced systems studied in this report use central distribution, the absence of a central distribution system would tend to restrict the owner of such a building from enjoying any possible benefits from future HVAC energy efficiency improvements.

Of the central air distribution systems studied, the gas warm air furnace with high efficiency electric air conditioner was found to be the lowest first-cost system for all locations considered. Use of uninsulated ducts lowers the system first-cost still further to where the gas furnace system has a lower cost in Concord than the baseboard convector system. However, use of uninsulated ducts is false economy. Based on unreported simulations run during this study, the added investment for insulated ducts will be recovered in about two years or less. (In Philadelphia, for example, the annual energy cost with uninsulated ducts is about 1/3 greater for a gas warm air furnace with high efficiency electric air conditioner system).

The first-cost for electric heat pumps not unexpectedly shows a trend toward higher prices for the high efficiency models. However, there are differences among the models priced such that the high efficiency III is only about 5 percent more expensive than the standard heat pump, while high efficiency I and II models are 10 to 20 percent more expensive than the standard heat pump.

The hybrid electric heat pump with oil furnace has a much higher first-cost than the other conventional heat pump systems because of higher (oil) furnace (and auxiliary) equipment cost. The gas furnace-hybrid heat pump, while more expensive to install than the standard electric heat pump, is competitive in first-cost with the high efficiency heat pumps.

For the advanced systems, only the developer of the organic fluid absorption gas heat pump was able to provide an estimate of the equipment cost. This information was used as the basis for developing a projected equipment installed cost for a new installation and for an add-on installation as shown in Exhibit 9-5.

When viewed as an alternative to a conventional high efficiency heat pump, the first cost for a new organic fluid absorption gas heat pump installation is not much greater. However, the product most likely to be the competitive option to the organic fluid absorption gas heat pump is expected to be the gas warm air furnace with a high efficiency central air conditioner. The sizable first-cost differences in favor of the latter system clearly puts the organic fluid absorption gas heat

pump at a cost disadvantage. Although the organic fluid absorption gas heat pump equipment costs used to develop the installed costs are not firm and are therefore subject to change, it is not expected that the gas heat pump would be competitive on a first-cost basis to a gas furnace with central air conditioner system.

The organic fluid absorption gas heat pump add-on system costs shown in Exhibit 9-5 represent the installed cost for a conversion of a gas warm air system to the organic fluid gas heat pump. The first costs for an add-on system are much lower than for a new installation, which would offer some competitive advantage should the consumer wish to upgrade a warm air system to include comfort cooling. The system's developer in fact expects the retrofit market to be greater than the new installation market for their gas heat pump.

9.3.2 Energy Cost Analysis

A comparison of total annual energy costs for conventional and advanced heat pumps is presented in Exhibits 9-6 and 9-7, respectively. Exhibits 9-8 and 9-9 present cost comparisons for currently available conventional and advanced systems on a seasonal basis.

On an annual energy cost basis, the gas warm air furnace with high efficiency air conditioner has a lower energy cost than the standard or even the high efficiency electric heat pumps in a number of locations. This, of course, is not entirely related to energy efficiency but is also an artifact of the rate structure. (The variations in rates for the locations used in the simulation studies may be seen in Exhibit 9-3, which presents effective unit energy prices for residential heating and cooling, and for commercial comfort conditioning on an annual basis.)

Not unexpectedly, the heat pumps show the best advantage in the warmer climates, while the gas and oil furnace systems continue to be competitive in the cooler locations. Seattle, with its low electric rates, represents an anomaly in this regard. In general, the increasing cost of gas and oil have improved the operating economics of the heat pump over those results reported in the previous Gordian study (1976).

Energy costs of the highly efficient advanced systems are, of course, lower than those of conventional systems. However, it should be noted that dynamic efficiency losses due to cycling and defrost may be understated for the advanced heat pump I, free piston Stirling-Rankine gas heat pump and the organic fluid absorption gas heat pump. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, the annual energy costs shown would be about 12 percent higher. If this were the case, part of the advanced system's advantage would be negated. Since the advanced systems will most likely be higher in first-cost than most of the conventional systems, if the advanced systems are unable to show a compensating advantage in lower energy costs, their market prospects would likely be adversely affected. This aspect is considered in the following discussion on life-cycle analysis of residential systems.

9.3.3 Life-Cycle Cost Analysis

The rationale of life-cycle analysis is that by use of appropriate capital recovery factors, the effective owning and operating cost for each system can be determined over the life of that system, thereby putting each system into a directly comparable position.

Life-cycle costs of the residential space conditioning systems simulated (without and with the effects of inflation on energy and maintenance costs) are shown in Exhibits 9-10 and 9-11, respectively. The results show that the electric heat pump we have termed "standard" (which represents 1975-1976 heat pump technology in terms of COP and EER ratings) is slightly more cost effective in general than the more recent high-efficiency models under the conditions of our analysis. An exception to this generalization appears to be the electric heat pump-high efficiency III, which provides a slight annual cost advantage compared to the standard electric heat pump in all locations but Seattle, where the difference is fairly minor. For the other high-efficiency models, only at an assumed 8 percent annual rate of inflation in energy cost (and 6 percent in maintenance cost) are the high-efficiency models comparable in a life cycle owning and operating cost sense to the standard model.

On the other hand, gas and oil combustion furnaces with electric air conditioning still represent a low life-cycle cost system especially in colder regions. However, the basis of the cost comparison is the furnace with insulated ducts and a minimum of oversizing. This is not quite the current practice. With uninsulated ducts, an oil furnace is attractive only in colder climates in comparison to the electric heat pump--even less so with inflation in energy costs.

As noted previously, the only advanced system for which installed costs could be developed is the organic fluid absorption gas heat pump. The life-cycle cost for this system, both new and as an add-on, and with

and without energy inflation, is presented in Exhibit 9-12.

Reflecting its higher efficiency in the cooler climates, the organic gas heat pump life-cycle cost compares favorably with the high efficiency electric heat pump life-cycle costs for the cooler locations simulated beginning with Tulsa. However, when compared to what may well be its strongest competitor, the gas warm air furnace with central air conditioner, only the Concord location appears to show a slight cost advantage toward the gas heat pump. The effect of energy inflation does not result in a reversal of this cost comparison situation. Also, as previously noted, the energy costs for the organic absorption gas heat pump are possibly understated by as much as 12 percent, which, if corrected by that quantity, would place the gas heat pump at a still lesser advantage against either the gas furnace or the electric heat pump on a life-cycle cost basis. The organic fluid absorption gas heat pump system does appear to have a life-cycle cost advantage compared to the hybrid heat pump with gas furnace, which could also be viewed as competing for the same gas heating system customer.

When considered as an add-on system, the organic absorption system offers a life-cycle cost in line with the conventional systems. This tends to confirm the developer's belief that the organic absorption gas heat pump will be more readily able to penetrate the retrofit market.

9.4 Analysis of Commercial Heat Pump System Costs

The previous section presented installed cost, energy cost and life-cycle cost data for residential building HVAC systems. In this section, a similar discussion is presented of light commercial HVAC systems for use in nonresidential buildings.

9.4.1 Installed Cost Analysis

A comparison of installed cost for conventional and advanced commercial systems is presented in Exhibit 9-13. Because of its heating system simplicity, the single package, two stage central air conditioner with strip heating (i.e. rooftop warm air electric furnace) has the lowest first-cost of the systems simulated. The rooftop gas furnace with electric air conditioner system is next lowest with an installed cost 5 to 10 percent greater than the electric strip heat system. The third conventional system costed, the standard electric heat pump, is about 15 to 20 percent more expensive in first-cost compared to the electric strip heat systems. In general, system installed costs are higher in the locations Houston through Philadelphia and lower in Seattle through Concord due primarily to differences in the equipment sizes. (A system capacity of 20 tons of cooling was used for the Houston through Philadelphia locations, while a capacity of 15 ton cooling was used for the other locations). On a cost per ton installed cooling capacity basis, the 20 ton installations actually have a lower unit cost than the 15 ton installations.

This same cost per ton effect is also seen with the two light commercial gas heat pumps studied (i.e. Brayton-Rankine and Rankine-Rankine rooftop gas heat pumps). The installed costs developed for the two advanced systems show them to be cost-competitive with the standard electric heat pump in most locations. However, they are 10 to 15 percent more costly than a gas or electric furnace with air conditioner system. Also, since the advanced systems cost are based on estimates of the developers and are not yet commercially available, it is possible that some variation from these estimates may be found when they are put into commercial production.

9.4.2 Energy Cost Analysis

The total annual energy costs for the light commercial systems simulated are presented in Exhibit 9-14. Inspection of the data indicates that the gas heat pumps, while offering lower energy costs for most locations, are more effective in the cooler climates. The exceptions for Houston and Seattle are due to the low electric rates available in these cities, along with gas rates that are higher than in the other cities studied in this report.

A further observation is that the Rankine-Rankine gas heat pump offers the best advantage in the colder locations, while the Brayton-Rankine gas heat pump performs better (relative to the Rankine-Rankine gas heat pump) in the warmer climates.

The net energy-cost-saving advantage in most locations (excluding Houston and Seattle for the reasons stated above) between a gas heat pump and the gas warm air furnace with air conditioner (a likely competitive product) is $400 to $600. While this appears insufficient to offer a two-year payback of the added investment needed to install a gas heat pump (as compared to a gas warm air furnace with air conditioner), a five-year payout appears feasible in several locations (as discussed in Chapter 11, the nonresidential HVAC market will readily accept a two-year payback of added investment, and will consider up to five years payback if the builder intends to own the building for that length of time [Levin and Associates, 1976]).

9.4.3 Life-Cycle Cost Analysis

The life-cycle costs for the light commercial HVAC systems are presented in Exhibit 9-15 without inflation of operating costs, and

in Exhibit 9-16 with inflation of the energy and maintenance costs. Generally, the gas heat pumps are more cost-effective in the cooler climates.

However, since the gas rates per unit energy are generally lower even in those areas where its efficiency is less (i.e. warmer climates), the gas heat pump presents itself as a viable alternative for all locations except Houston with its high cooling load, and Seattle, with its low electric rates.

When operating costs are inflated over the life of the project (8 percent for energy, 6 percent for maintenance) the life-cycle cost comparison offers still further advantage to the gas heat pumps because of their greater energy efficiency. (Again Houston and Seattle are outliers). An increase in the gas inflation rate to 14 percent was also tested for the Rankine-Rankine gas heat pump, with results similar to those found under the 8 percent gas inflation rate calculation.

9.5 Summary

Although the advanced residential HVAC systems offer lower energy costs, the added system first-cost may not be justifiable. For example, the organic fluid absorption heat pump does not appear at this time to be life-cycle cost competitive with most of the conventional systems and, in particular, the gas warm air furnace with central air conditioner system which is viewed as the logical competitive product. This would be expected to negatively influence the market prospects of the absorption machine for new installations. The refit market appears to offer a better potential for the organic absorption gas heat pump at this time and even more under conditions of 8 percent inflation.

The lowest cost residential space conditioning system (with the exception of baseboard convector/window air conditioners) continues to be the gas warm air furnace with high efficiency air conditioner. High efficiency heat pumps, because of their 5-20 percent higher first cost over the standard heat pump, do not appear to offer any life-cycle cost advantage over the standard heat pump at present energy prices. In general, the higher cost for greater energy efficiency does not appear to pay. However, the more energy efficient systems will be the beneficiaries of future higher energy costs, which will of course improve their economics.

A potential new development, the pulse gas furnace with high efficiency central air conditioner, offers system energy costs equal to the advanced heat pumps (and much better than for the conventional systems studied). Since the pulse gas furnace is expected to be marketed at a price near the present gas warm air furnace, it would appear to make this system an obvious minimum cost choice, assuming the gas hook-up would be permitted.

The hybrid electric heat pumps (with gas or oil furnace) are substantially more expensive to own and operate than any of the other heat pumps studied.

The organic fluid absorption gas heat pump is not cost-effective compared to the gas warm air furnace with central electric air conditioner. Additionally, transient losses not accounted for the energy modelling of this system may result in energy costs of up to 12 percent higher.

The more attractive gas heat pump market possibility at this time is the one for light commercial service. Under present cost assumptions,

the light commercial gas heat pumps are cost-effective compared to the electric heat pump for most all locations, and to the electric and gas furnace with air conditioner in the cooler climates. When operating costs are inflated, the gas heat pumps are cost-effective for all locations used in the simulation study with the exception of Houston and Seattle, both of which have low electric rates available for commercial service.

References to Chapter 9

American Society of Heating, Refrigerating and Air Conditioning Engineers Handbook and Product Directory (New York, ASHRAE, 1975).

Dressler, F.R.S. "Life-Cycle Evaluation of Small/Medium HVAC Systems to Promote Energy Conservation". Proceedings of the Conference on Improving Efficiency in HVAC Equipment and Components (Purdue Research Foundation, 1974).

Gordian Associates Inc. Evaluation of the Air-to-Air Heat Pump for Residential Space Conditioning (Report prepared for FEA, NTIS No. PB-255-652, April 1976).

Hittman Associates, Inc; "Barriers Connected with Certifying or Listing of Energy Conserving Products Used in Buildings (Report to ERDA, November 1976).

Levin and Associates, Philadelphia, Pa. Personal communication, 1976.

National Environmental Systems Contractors Association (NESCA), Personal communication, 1976.

Tri-Boro Environmental Systems, Bound Brook, N.J. Personal communication 1977.

EXHIBIT 9-1

ANNUAL MAINTENANCE COST

SYSTEM	Residential Systems [a] ANNUAL MAINTENANCE COST DOLLARS
Heat Pump	149.60
Gas Furnace and Central Air Conditioner	104.50
Electric Furnace and Central Air Conditioner	109.70
Oil Furnace and Central Air Conditioner	127.90
Electric Baseboard Resistance Heaters and Room Air Conditioner Units	110.00
Window Air Conditioner [b]	21.00

SYSTEM	Commercial Systems ANNUAL MAINTENANCE COST[c] DOLLARS
All 15 ton Heat Pump Systems	840.00
All 20 ton Heat Pump Systems	1120.00

Sources:
a. National Environmental System Contractors Association (NESCA) Mid-1976 Prices.
b. Sears-Roebuck Co. average price per air conditioner unit.
c. Tri-Boro Environmental Systems.

EXHIBIT 9-2

DEPRECIATION PERIOD OF HEATING AND COOLING SYSTEMS

Component	Capital Recovery Factor (at 9% Interest)	Depreciation Period (Years)
Residential Systems		
Heat Pump	0.1668	9[a]
Central Air Conditioner	0.1560	10[b]
Gas Furnace	0.1203	16[b]
Oil Furnace	0.1560	10[b]
Oil Tank	0.1095	20[b]
Electric Warm Air Furnace	0.1240	15[b]
Electric Baseboard Heaters	0.0929	40[b]
Window Air Conditioners	0.1560	10[b]
Ductwork, Chimney	0.0929	40[c]
Commercial Systems		
Rooftop Gas Heating/Electric Cooling Package	0.1560	10[d]
Single Package, Two Stage Central Air Conditioning with Strip Heating	0.1560	10[d]
Single Package Standard Electric Heat Pump	0.1668	9[d]
Brayton-Rankine Gas Heat Pump	0.1668	9[d]
Rankine-Rankine Gas Heat Pump	0.1668	9[d]

Sources:

a. Industry spokesmen give the life as 8-10 years.

b. ASHRAE Handbook and Product Directory, 1975 Systems

c. Internal Revenue Service

d. Tri-Boro Environmental Systems

EXHIBIT 9-3

EFFECTIVE UNIT ENERGY PRICE FOR HEATING AND COOLING

	Houston		Birmingham		Atlanta		Tulsa		Philadelphia		Seattle		Columbus		Cleveland		Concord	
Residential Rates	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu	$/unit	$/$10^6$Btu
Electricity Heating (kWh)	0.0271	7.94	0.0409	11.98	0.0311	9.12	0.0251	7.35	0.0479	14.04	0.0092	2.70	0.0402	11.78	0.0469	13.75	0.0387	11.34
Electricity Heating, All-Electric (kWh)	0.0217	6.36	0.0356	10.43	0.0269	7.88	0.0239	7.00	0.0285	8.35	0.0091	2.67	0.0282	8.26	0.0353	10.34	0.0387	11.34
Electricity Cooling (kWh)	0.0264	7.74	0.0409	11.98	0.0358	10.49	0.0348	10.20	0.0558	16.35	0.0092	2.70	0.0402	11.78	0.0472	13.83	0.0387	11.34
Electricity Cooling, All-Electric (kWh)	0.0264	7.74	0.0408	11.95	0.0357	10.46	0.0348	10.20	0.0542	15.88	0.0090	2.64	0.0398	11.66	0.0464	13.60	0.0387	11.34
Natural Gas Heating (10^3 cft)	2.83	2.83	1.69	1.69	1.79	1.79	1.79	1.79	2.51	2.51	2.89	2.89	1.92	1.92	1.96	1.96	3.25	3.25
Natural Gas Cooling (10^3 cft)	2.43	2.43	1.47	1.47	1.74	1.74	1.67	1.67	2.52	2.52	2.93	2.93	1.92	1.92	1.98	1.98	3.93	3.93
No. 2 Fuel Oil (gal*)	**	**	0.422	3.03	0.440	3.16	0.435	3.13	0.469	3.37	0.459	3.30	0.419	3.01	0.451	3.24	0.469	3.37
Commercial Rates																		
Electricity (kWh)	0.0182	5.34	0.0672	19.70	0.0529	15.51	0.0409	12.00	0.0436	12.78	0.0132	3.87	0.0633	18.55	0.0681	19.97	0.0348	10.20
Electricity All-Electric (kWh)	0.0182	5.34	0.0617	18.08	0.0484	14.19	0.399	11.70	0.0313	9.18	0.0135	3.96	0.0463	13.57	0.0479	14.04	0.0399	11.69
Natural Gas (10^3 cft)	3.55	3.55	2.57	2.57	2.13	2.13	1.93	1.93	2.93	2.93	3.72	3.72	1.97	1.97	2.11	2.11	3.20	3.20

* Price quotations for first quarter, 1977.
** Not sold in Houston, assume Tulsa prices.

SOURCE: Gordian Associates Inc.

EXHIBIT 9-4

CONVENTIONAL ELECTRIC HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF INSTALLED COSTS
(First Quarter 1977 Dollars)

	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump-Standard	3255.24	3119.14	3079.57	3397.40	3304.03	2671.29	3234.22	3406.22	2749.34
Electric Heat Pump-High Efficiency I	3598.18	3458.84	3417.67	3913.75	3646.99	3025.20	3634.96	3816.80	3165.80
Electric Heat Pump-High Efficiency II	4170.41	4043.84	3660.18	4120.73	4232.01	3257.34	4144.46	4329.58	3558.78
Electric Heat Pump-High Efficiency III	3369.13	3234.12	3197.19	3515.02	3422.28	2853.96	3347.02	3520.43	2932.41
Hybrid Electric Heat Pump-Gas Furnace	3645.17	3491.91	3453.25	4051.71	3719.87	3260.69	3628.84	3841.92	3200.87
Hybrid Electric Heat Pump-Oil Furnace	4489.03	4367.51	4292.81	5069.55	4564.95	4034.35	4468.40	4685.62	4039.59
Gas Warm Air Furnace/Central Air Conditioner-High Efficiency	2745.00	2602.43	2552.58	2947.48	2668.18	2345.43	2717.19	2881.25	2365.95
Oil Warm Air Furnace/Central Air Conditioner-High Efficiency	3614.87	3560.62	3437.51	3832.08	3545.02	3148.33	3580.08	3754.58	3211.11
Central Electric Furnace/Central Air Conditioner-High Efficiency	2882.93	2746.14	2717.75	3090.18	2850.39	2429.62	2899.34	3060.90	2508.79
Baseboard Convectors/Window Air Conditioners	2378.14	2264.37	2165.29	2283.81	2452.54	2251.78	2367.93	2562.84	2305.43
System with Uninsulated Ducts*									
Gas Warm Air Furnace/Central Air Conditioner-High Efficiency	2558.83	2546.26	2496.41	2918.82	2612.01	2234.26	2634.26	2825.08	2282.18
Oil Warm Air Furnace/Central Air Conditioner-High Efficiency	3458.70	3504.45	3381.34	3803.42	3488.85	3037.16	3496.93	3698.41	3127.34

* Current Practice

Source: Gordian Associates, Inc.

EXHIBIT 9-5

ORGANIC FLUID ABSORPTION GAS HEAT PUMP: INSTALLED
(Dollars)

	New**	Add-on**
Houston	4044.09	2665.85
Birmingham	3910.89	2662.68
Atlanta	3857.44	2616.05
Tulsa	4350.97	2972.24
Philadelphia	4099.04	2745.90
Seattle	3170.65	2040.93
Columbus	4007.06	2670.30
Cleveland	4200.78	2785.15
Concord	3360.73	2226.67

* First Quarter 1977 costs.
** Cost based on developer's wholesale equipment cost estimate for a 3 ton unit, scaled for certain locations as follows: Seattle, 2 ton; Concord, 2½ ton; Tulsa, 3½ ton.
*** Assumes air handler and ducts in-place.

Source: Allied Chemical Co. (1976);
Gordian Associates Inc.

EXHIBIT 9-6

CONVENTIONAL ELECTRIC HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF TOTAL ANNUAL ENERGY COST*

(Dollars)

System	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump-Standard	221.13(193.39)**	339.46(299.24)	270.09(237.42)	323.36(287.25)	380.00(336.69)	71.27(63.15)	403.47(365.18)	536.35(490.92)	703.72(655.88)
Electric Heat Pump-High Efficiency I	182.36(160.14)	291.51(257.93)	231.69(204.62)	273.90(243.79)	325.59(289.37)	63.72(56.23)	355.98(322.26)	480.17(439.14)	648.63(603.99)
Electric Heat Pump-High Efficiency II	200.14(173.00)	308.81(270.16)	253.54(224.70)	281.45(245.80)	341.60(301.09)	62.33(55.42)	365.86(329.66)	487.51(444.84)	600.07(557.24)
Electric Heat Pump-High Efficiency III	195.14(171.06)	307.99(272.20)	244.82(216.10)	319.34(283.43)	349.80(310.87)	62.79(55.81)	379.53(344.02)	511.25(468.62)	620.62(576.71)
Hybrid Electric Heat Pump-Gas Furnace	236.80	360.23	290.95	305.88	408.39	121.37	390.96	480.55	580.53
Hybrid Electric Heat Pump-Oil Furnace	237.20	367.23	298.14	327.20	423.46	129.24	430.78	540.83	569.70
Electric Heat Pump-Heat Only/Central Air Conditioner-High Efficiency	197.67(172.90)	299.73(263.82)	239.77(211.08)	280.28(247.63)	323.09(285.84)	57.30(51.01)	344.53(310.56)	458.39(418.66)	550.64(511.75)
Gas Warm Air Furnace/Central Air Conditioner-High Efficiency	223.21	264.31	236.78	273.09	369.75	211.31	304.65	338.24	482.54
Oil Warm Air Furnace/Central Air Conditioner-High Efficiency	225.70	313.63	287.16	360.40	426.74	236.12	397.79	468.49	487.50
Central Electric Furnace/Central Air Conditioner-High Efficiency	236.97	428.82	343.92	451.16	559.10	136.18	584.93	809.13	976.26
Baseboard Convectors/Window Air Conditioners	153.17	341.39	270.16	379.50	490.12	131.91	537.40	770.05	946.35

* First quarter 1977 costs

** Numbers in parenthesis are energy costs calculated on steady state or full-load performance basis, dynamic efficiency losses are not included.

Source: Gordian Associates Inc.

EXHIBIT 9-7

ADVANCED HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF TOTAL ANNUAL ENERGY COST**
(Dynamic efficiency not accounted for*)
(Dollars)

System	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Advanced Electric Heat Pump I	***	274.41	218.95	241.16	298.29	***	305.10	403.71	***
Advanced Electric Heat Pump II	132.93	211.61	167.24	219.05	250.01	43.39	268.04	361.26	482.31
Free Piston Stirling Rankine Gas HP	180.25	149.25	144.10	149.36	222.57	109.57	194.16	220.85	332.67
V-Type Single Cylinder Stirling Rankine HP	186.81	162.11	155.00	172.44	231.13	93.05	199.04	227.78	353.48
Organic Fluid Absorption Gas Heat Pump	255.87	178.77	179.77	183.22	261.66	148.67	221.66	244.25	374.48
Pulse Gas Furnace/Central Air Conditioner-High Efficiency	206.56	238.79	210.20	227.74	303.60	145.92	244.55	268.19	347.00

* The energy costs presented may be understated for the advanced electric heat pump I, free piston Stirling Rankine gas heat pump and the organic fluid absorption gas heat pump due to possible understating of cycling and defrost losses. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, the energy costs above should be multiplied by the correction factor of 1.12 for annual energy values.

** First quarter 1977 costs

** System is not sized for application in these cities

EXHIBIT 9-8

CONVENTIONAL ELECTRIC HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF HEATING AND COOLING SEASON ENERGY COST*
(Dollars)

HEATING	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump-Standard	30.44(27.46)**	134.88(123.28)	102.08(93.81)	188.54(173.56)	217.05(198.33)	62.43(55.26)	292.02(269.83)	436.35(403.80)	650.77(609.98)
Electric Heat Pump-High Efficiency I	25.61(23.14)	118.19(108.30)	89.03(81.93)	164.06(150.86)	188.59(172.42)	56.51(49.79)	262.39(241.73)	394.53(364.28)	604.93(566.10)
Electric Heat Pump-High Efficiency II	27.33(24.61)	122.23(111.89)	96.73(88.93)	161.34(148.34)	193.83(177.19)	54.18(48.28)	264.66(244.14)	395.80(365.66)	551.16(515.52)
Electric Heat Pump-High Efficiency III	28.95(26.10)	129.21(118.38)	97.93(90.23)	183.96(169.18)	207.66(189.98)	55.26(49.17)	282.37(260.72)	422.22(390.83)	578.10(540.22)
Hybrid Electric Heat Pump-Gas Furnace	36.79	144.66	113.83	165.74	236.42	111.92	273.31	375.58	525.08
Hybrid Electric Heat Pump-Oil Furnace	37.19	152.06	121.02	187.06	251.48	119.79	313.13	435.86	514.25
Electric Heat Pump-Heat Only	25.58(23.02)	114.67(104.80)	87.86(81.09)	148.41(136.25)	176.09(160.97)	49.29(43.95)	243.98(224.47)	366.83(338.67)	504.76(472.33)
Gas Warm Air Furnace	51.12	79.25	84.87	141.22	218.33	203.13	203.20	246.69	436.66
Oil Warm Air Furnace	53.61	128.57	135.25	228.53	275.32	227.94	296.34	376.94	441.62
Central Electric Furnace	64.88	248.56	192.01	319.29	412.10	128.17	484.38	717.58	930.38
Baseboard Convectors	63.54	243.76	188.21	314.28	406.07	125.01	477.61	707.61	915.21
COOLING									
Electric Heat Pump-Standard	190.69(165.93)	204.58(175.96)	168.01(143.61)	134.82(113.69)	162.95(138.31)	8.84(7.88)	111.45(95.35)	100.00(87.12)	52.95(45.90)
Electric Heat Pump-High Efficiency I	156.75(137.00)	173.32(149.63)	142.66(122.69)	109.84(92.93)	137.00(116.95)	7.21(6.46)	93.59(80.53)	85.64(74.86)	43.70(37.89)
Electric Heat Pump-High Efficiency II	172.81(148.39)	186.58(158.27)	156.81(135.77)	120.11(97.46)	147.77(123.90)	8.15(7.14)	101.20(85.82)	91.71(79.18)	48.91(41.72)
Electric Heat Pump-High Efficiency III	166.19(144.96)	178.78(153.82)	146.89(125.87)	135.38(114.25)	142.14(120.89)	7.53(6.64)	97.16(83.30)	89.03(77.79)	42.52(36.49)
Hybrid Electric Heat Pump	200.01	215.58	177.12	140.14	171.98	9.45	117.65	104.97	55.54
Central Air Conditioner-High Efficiency	172.09(149.88)	185.06(159.02)	151.91(129.98)	131.87(111.38)	151.42(128.52)	8.18(7.23)	101.45(86.90)	91.55(79.99)	45.88(39.42)
Central Air Conditioner-High Efficiency (All electric rate)	172.09(149.88)	185.06(159.02)	151.91(129.98)	131.87(111.38)	147.00(124.87)	8.01(7.06)	100.55(86.09)	91.65(79.99)	45.88(39.42)
Central Air Conditioner-Standard	205.12(178.98)	220.43(190.34)	181.11(155.19)	138.73(117.21)	181.11(154.01)	9.55(8.57)	121.06(104.34)	106.69(93.01)	53.17(45.85)
Central Air Conditioner-Standard (all Electric Rate)	205.12(178.98)	220.43(190.34)	181.11(155.19)	138.73(117.21)	175.82(149.62)	9.36(8.39)	120.15(103.43)	106.69(93.01)	53.17(45.85)
Window Air Conditioners	89.63 (78.13)	97.63 (84.10)	81.95 (69.98)	65.22 (55.18)	84.05 (71.17)	6.90(6.17)	59.79 (51.32)	62.44(53.29)	31.14(26.91)

* First quarter 1977 costs.

** Numbers in parenthesis are energy costs calculated on steady state or full-load performance basis, dynamic efficiency losses are not included.

Source: Gordian Associates Inc.

EXHIBIT 9-9

ADVANCED HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF HEATING AND COOLING SEASON ENERGY COST**

(Dynamic efficiency not accounted for*)

HEATING	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Advanced Electric Heat Pump I	***	109.39	83.90	133.68	168.04	***	216.38	320.66	***
Advanced Electric Heat Pump II	18.98	90.62	67.71	138.72	153.60	38.19	202.16	302.24	451.06
Free Piston Stirling Rankine Gas HP	23.96	53.10	49.78	78.20	135.42	86.56	133.65	168.48	281.29
V-Type Single Cylinder Stirling Rankine HP	22.58	52.02	48.19	80.81	131.48	68.39	129.53	168.36	295.33
Organic Fluid Absorption Gas Heat Pump	28.71	52.90	52.90	86.31	141.13	111.71	137.23	173.33	302.71
Pulse Gas Furnace	34.47	53.73	58.29	95.87	152.18	137.74	143.10	176.64	301.12
COOLING									
Advanced Electric Heat Pump I	***	165.01	135.05	107.48	130.25	***	89.72	83.05	***
Advanced Electric Heat Pump II	113.95	120.99	99.53	80.33	96.41	5.20	65.88	59.02	31.25
Free Piston Stirling Rankine Gas HP	156.28	96.16	94.31	71.15	87.15	23.02	60.52	52.37	51.37
V-Type Single Cylinder Stirling Rankine HP	164.22	110.10	106.82	91.63	99.64	24.67	69.51	59.42	58.15
Organic Fluid Absorption Gas Heat Pump	227.17	125.86	126.87	96.91	120.53	36.96	84.43	70.92	71.77

* The energy costs presented may be understated for the advanced electric heat pump I, free piston Stirling Rankine gas heat pump and the organic fluid absorption gas heat pump due to possible understating of cycling and defrost losses. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, the energy costs above should be multiplied by a correction factor of 1.09 for heating season values and by 1.18 for cooling season values.

** First quarter 1977 costs

*** System is not sized for application in these cities

EXHIBIT 9-10

LIFE CYCLE COSTS FOR ALTERNATIVE RESIDENTIAL SPACE CONDITIONING SYSTEMS - NO INFLATION
(First Quarter, 1977 Dollars)

System as New Installation	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump-Standard	826.01	931.48	855.46	954.97	995.10	590.29	1007.59	1163.75	1236.63
Electric Heat Pump-High Efficiency I	844.39	940.19	873.46	991.63	997.90	641.77	1027.61	1176.05	1251.01
Electric Heat Pump-High Efficiency II	957.62	1055.07	935.76	1033.71	1111.49	679.10	1121.81	1268.92	1268.00
Electric Heat Pump-High Efficiency III	819.02	919.19	849.81	954.28	984.63	612.28	1002.46	1157.70	1184.07
Hybrid Electric Heat Pump-Gas Furnace	934.85	1043.42	968.04	1074.72	1119.39	755.07	1088.57	1202.94	1217.91
Hybrid Electric Heat Pump-Oil Furnace	1077.08	1194.78	1115.49	1262.58	1277.33	908.12	1269.75	1412.43	1346.72
Gas Warm Air Furnace/Central Air Conditioner-High Efficiency	671.30	699.03	664.72	753.25	806.39	605.53	750.33	803.07	882.75
Oil Warm Air Furnace/Central Air Conditioner-High Efficiency	820.34	906.55	862.76	989.05	1011.84	771.73	989.16	1081.80	1029.14
Central Electric Furnace/Central Air Conditioner-High Efficiency	705.94	885.00	795.88	952.98	1022.42	544.97	1057.11	1300.58	1450.78
Baseboard Convector/Window Air Conditioners	561.41	730.26	648.55	768.90	896.47	508.88	934.62	1186.33	1335.21
Electric Heat Pump-Heat Only	700.40	762.28	727.11	795.27	853.21	691.32	908.34	1055.34	1128.77
System as Add-On Installation									
Electric Heat Pump-Heat Only	533.65	605.74	573.43	636.75	681.04	554.38	738.60	877.61	983.82

Source: Gordian Associates Inc.

EXHIBIT 9-11

LIFE CYCLE COSTS FOR ALTERNATIVE RESIDENTIAL SPACE CONDITIONING SYSTEMS-

EFFECTS OF INFLATION ON ENERGY AND MAINTENANCE COSTS INCLUDED*

System as New Installation	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump-Standard	968.52	1125.43	1019.25	1141.92	1206.67	667.67	1229.36	1443.27	1588.90
Electric Heat Pump-High Efficiency I	970.05	1113.29	1020.56	1157.08	1185.82	688.17	1228.74	1431.15	1579.33
Electric Heat Pump-High Efficiency II	1091.01	1235.69	1092.36	1202.44	1306.37	752.59	1327.23	1527.21	1575.22
Electric Heat Pump-High Efficiency III	950.24	1099.46	1002.62	1139.48	1183.07	685.97	1213.82	1426.31	1500.22
Hybrid Electric Heat Pump-Gas Furnace	1102.79	1265.01	1159.52	1272.69	1361.92	882.84	1323.52	1479.83	1535.26
Hybrid Electric Heat Pump-Oil Furnace	1251.41	1425.63	1316.41	1476.03	1532.62	1035.52	1528.22	1715.73	1665.57
Gas Warm Air Furnace/Central Air Conditioner-High Efficiency	815.13	862.76	815.12	921.23	1021.16	743.60	933.59	1002.59	1152.21
Oil Warm Air Furnace/Central Air Conditioner-High Efficiency	973.49	1102.17	1129.80	1207.31	1262.21	929.82	1225.52	1352.39	1308.93
Central Electric Furnace/Central Air Conditioner-High Efficiency	858.21	1130.15	999.93	1208.95	1330.64	648.45	1377.84	1729.85	1908.08
Backboard Convectors/Window Air Conditioners	673.21	933.18	816.99	990.27	1171.39	610.39	1232.43	1596.77	1830.99
Electric Heat Pump-Heat Only	757.92	858.52	811.70	906.18	976.15	759.14	1060.79	1261.18	1394.56
System as Add-On Installation									
Electric Heat Pump-Heat Only	591.17	701.98	658.02	747.66	803.98	622.20	891.05	1083.45	1249.61

* Annual inflation: energy cost, 8 percent; maintenance cost, 6 percent.

Source: Gordian Associates Inc.

EXHIBIT 9-12

ORGANIC FLUID ABSORPTION GAS HEAT PUMP: LIFE-CYCLE COST*

(No dynamic efficiency losses accounted for)
(Dollars)

	ANNUALIZED COST		EFFECTIVE ANNUALIZED COST WITH INFLATION**	
	New	Add-on***	New	Add-on***
Houston	1007.06	850.14	1164.35	1007.43
Birmingham	917.63	772.51	1041.46	896.34
Atlanta	909.67	763.73	1033.93	887.99
Tulsa	987.66	827.59	1114.42	954.35
Philadelphia	1024.15	869.28	1184.28	1029.41
Seattle	765.77	638.70	876.54	749.47
Columbus	969.47	816.67	1111.92	959.12
Cleveland	1018.96	858.41	1171.21	1010.66
Concord	1024.15	895.49	1232.92	1104.26

* First Quarter 1977 costs.
** Calculation on following assumptions: energy inflation - 8 percent, maintenance cost inflation - 6 percent, cost of money - 9 percent, project life - 9 years.
*** Assumes air handler and ducts in-place.

Source: Gordian Associates Inc.

EXHIBIT 9-13

LIGHT COMMERCIAL HEATING AND COOLING SYSTEMS: COMPARISON OF INSTALLED COSTS

(Second Quarter 1977 Dollars)

SYSTEM	HOUSTON	BIRMINGHAM	ATLANTA	TULSA	PHILADELPHIA	SEATTLE	COLUMBUS	CLEVELAND	CONCORD
Rooftop Gas Heating/Electric Cooling Package	19009.66	18811.27	18600.28	18790.57	19769.50	16877.56	16303.05	17346.40	15650.77
Single Package, Two Stage Central Air Conditioning with Strip Heating	17427.14	17427.20	17032.83	17223.11	18167.92	15791.99	15442.20	16473.13	14823.03
Single Package Standard Electric Heat Pump	20517.16	20528.98	20077.64	20278.64	21322.66	18583.57	18172.88	19292.55	17437.17
Brayton-Rankine Gas Heat Pump	21242.50	21298.75	20792.80	20983.08	22038.66	17812.72	17409.50	18497.66	16637.74
Rankine-Rankine Gas Heat Pump	21317.50	21374.47	20867.09	21057.37	22114.37	17888.00	17483.79	18573.01	16709.17

Source: Gordian Associates, Inc.

EXHIBIT 9-14

LIGHT COMMERCIAL HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF TOTAL ANNUAL ENERGY COSTS
(First Quarter 1977 Dollars)

System	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Gas Warm Air Furnace/ Central Air Conditioner-High Efficiency	790.74	1856.01	1342.49	1091.23	1002.03	699.61	998.00	993.07	1012.23
Central Electric Furnace Central Air Conditioner-High Efficiency	660.24	2010.78	1528.44	1513.91	1059.50	397.12	1557.39	1706.17	1727.70
Electric Heat Pump-Standard	587.73	1668.23	1198.56	1052.31	614.73	208.21	1000.78	1012.98	962.67
Brayton-Rankine Gas Heat Pump	797.34	692.16	422.60	533.05	495.18	450.37	450.00	462.73	736.62
Rankine-Rankine Gas Heat Pump	774.88	421.12	347.76	345.42	420.25	383.43	272.38	282.40	479.39

SOURCE: Gordian Associates Inc.

EXHIBIT 9-15

LIFE CYCLE COSTS FOR LIGHT COMMERCIAL HEATING AND COOLING SYSTEMS
(Second Quarter 1977 Dollars)

System	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Rooftop Gas Heating/Electric Cooling Package	4458.20	5505.61	4958.93	4729.24	4760.41	3771.10	3999.12	4118.06	3924.64
Single Package, Two Stage Central Air Conditioning with Strip Heating	4080.83	5444.47	4900.36	4907.39	4568.03	3299.26	4424.21	4618.77	4510.78
Single Package Standard Electric Heat Pump	4640.40	5738.19	5192.95	5070.72	4769.40	3677.83	4424.45	4577.86	4278.67
Brayton-Rankine Gas Heat Pump	5118.79	5038.32	4684.08	4816.76	4917.08	3939.21	3894.14	4042.83	4070.07
Rankine-Rankine Gas Heat Pump	5108.84	4779.91	4621.63	4641.52	4854.78	3884.83	3728.91	3875.07	3821.76

SOURCE: Gordian Associates Inc.

EXHIBIT 9-16

LIFE CYCLE COSTS LIGHT COMMERCIAL HEATING AND COOLING SYSTEMS INCLUDING INFLATIONARY EFFECTS ON ENERGY AND MAINTENANCE COSTS
(Second Quarter 1977 Dollars)

System	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Rooftop Gas Heating/Electric Cooling Package	5224.39	6787.51	5992.23	5640.90	5628.89	4397.33	4769.80	4886.36	4702.01
Single Package Two Stage Central Air Conditioning with Strip Heating	4783.84	6801.29	6008.41	6023.67	5464.33	3779.05	5465.69	5732.28	5634.71
Single Package Standard Electric Heat Pump	5243.29	6810.73	6030.87	5875.54	5384.03	4028.90	5120.02	5278.73	4957.68
Brayton-Rankine Gas Heat Pump	5812.79	5686.61	5215.21	5395.89	5479.76	4395.54	4350.31	4504.54	4650.82
Rankine-Rankine Gas Heat Pump	5793.08	5310.39	5120.22	5139.10	5384.88	4312.07	3717.11	4258.40	4290.71

SOURCE: Gordian Associates Inc.

Calculation based on the following assumptions: energy inflation - 8 percent, maintenance cost inflation - 6 percent, money - 9 percent.

10. MARKET PROSPECTS OF ADVANCED HEAT PUMPS

10.1 Introduction

The market prospects of advanced heat pumps are influenced by a number of factors such as first-cost, life-cycle cost, consumer acceptance--and the potential market for space conditioning. Economic factors such as capital and operating costs were discussed in Chapter 9. Institutional factors, consumer attitudes about heat pumps and the consumer's willingness to pay more for energy efficient space conditioning is covered in Chapter 11. This chapter treats primarily the question of the market prospects for the advanced systems from the HVAC market potential point of view.

A brief analysis of the building market as it pertains to space conditioning is given in Section 10.2. In Section 10.3, an analysis of present market trends in heating and cooling equipment is presented. Projections for the market prospects for advanced heat pumps are provided in Section 10.4. The impact of various economic and institutional factors on market prospects for advanced heat pumps is discussed in Section 10.5. The findings of this chapter are summarized in Section 10.6.

10.2 Building Market Analysis

The building market analysis presented in this report is divided into the two primary building categories of residential and nonresidential buildings, a distinction based on their major differences in composition and space conditioning requirements. For each building category, opportunity for installation of a heat pump would include new construction, replacements

and conversions. A replacement unit is defined as a new unit of the same general type put in place of an existing unit which has become unserviceable or obsolete. A conversion is understood to be the installation of a heat pump in a building previously space conditioned by a system other than a heat pump.

10.2.1 Residential Building Market

The residential building market may be characterized as being fairly homogeneous as to building type within three basic structure classes--single family, multifamily and mobile homes. In the latest reported survey by the Census Bureau, the total U.S. housing stock in 1974 consisted of 51.2 million single family buildings, 20.9 million multifamily buildings and 3.7 million mobile homes (Census Bureau, 1976).

In assessing the residential building market within the context of space conditioning by either gas or electric heat pumps, the Census Bureau's major region grouping (Northeast, North Central, South and West) was used because of the wide regional variation in space conditioning characteristics such as natural gas use in heating or central air conditioning saturation. Regionally, the South has the major share of new residential construction with 45 percent of the total, the West and North Central each have 21 percent, while the Northeast has only 13 percent. Installation of natural gas heating in new residential construction predominates in the West (71 percent) and North Central (65 percent) while the South and Northeast have installation rates of 38 and 35 percent respectively. Not unexpectedly, the South leads in central air conditioning installed in new construction with 62 percent, followed by the West at 39 percent, North Central at 32 percent and Northeast at 13 percent (Census Bureau, 1975).

The annual conventional housing production for the 1977 to year 2000 period is estimated to be between 1.3 and 2.0 million units, with an average of 1.75 million units per year (Smith Barney, 1976). Moderating influences on housing growth during this 1977 to 2000 period include declining population growth rate and an increase in the number of people, 65 and older, who are not prime home buying candidates (Census Bureau, 1977).

10.2.2 Nonresidential Building Market

The nonresidential building market may be characterized as fairly heterogeneous, with a wide diversity of buildings including hospitals, churches, stores, office buildings, warehouses and manufacturing facilities. Unlike residential building data, there is much less information available both as to the existing building stock as well as new construction. Overall new nonresidential construction of all types is estimated to be 1895 million ft^2 by 1981 (General Electric, 1976). Regional nonresidential new construction indicates a greater concentration of new construction in the South and West (60 percent) and away from the Northeast and North Central regions (40 percent) (Commerce Department, 1976).

This regional trend in new construction appears to hold for both office and store buildings. An exception is institutional building new construction which is much lower in the West and about equal in the other three regions. Building type is significant because this study is concerned with nonresidential buildings that have an air conditioning requirement which could be satisfied by light commercial service space conditioning equipment (i.e. up to 25 ton cooling capacity). Hence high rise buildings which have large built-up systems and buildings that are not air conditioned would not be heat pump candidate buildings within the scope of this study.

10.3 Space Heating and Cooling Equipment Market Trends

The previous section viewed the space conditioning market without considering the specific HVAC equipment involved. In this section, consideration is given to the trends in the types of space conditioning equipment supplied to the market. Space heating equipment market trends are discussed in Subsection 10.3.1, and space cooling equipment market trends are covered in Subsection 10.3.2. Heat pumps are considered in both heating and cooling applications, since most heat pumps provide both comfort heating and comfort cooling, and are in fact advertised in this manner.

10.3.1 Heating Equipment Market Trends

The trend in the years 1967 to 1975 in shipments of central heating equipment (gas, oil, electric furnaces and electric heat pumps) is illustrated in Exhibit 10-1. As shown here, gas warm air systems dominated the market until recently since gas has been a preferred fuel from both an economic and a consumer utilization viewpoint. Lack of gas availability (discussed in Chapter 11) and the emergence of electric heating has cut into this dominance. Shipments for both gas and electric warm air furnaces declined significantly in 1975, due at least in part to the overall downturn in the construction industry. Generally, heating equipment shipments follow the same trend as new residential construction, with the moderating influence of a strong equipment after-market in replacements, conversions or inventory.

To further illustrate this point, heating equipment shipment data were compared to new single family residential completions and to other installations (including nonresidential) or inventory. The comparisons

given in the table below shows that between 52 and 60 percent of central heating equipment units shipped are for use in other than new single family construction. This other use, of course, represents a sizeable market and would account for the heating equipment market showing less of a response to the vagaries of new residential building construction completions.

Central Heating Equipment Shipment Analysis

Year	Total Shipments Units X 10^{-3}	Units Installed in New Single Family Home Completions Units X 10^{-3}	Percent	All Other Installations or Inventory Units X 10^{-3}	Percent
1971	2,470	1,014	41.1	1,456	58.9
1972	2,778	1,143	41.1	1,635	58.9
1973	2,556	1,174	45.9	1,382	54.1
1974	2,301	932	40.5	1,369	59.5
1975	1,805	866	48.0	939	52.0

Source: National Association of Homebuilders, Census Bureau.

As discussed in Chapter 11, costs of all forms of energy have increased rapidly since the Arab oil embargo of 1973. One response to higher electricity cost has been the use of a heat pump in place of an electric furnace. Heat pump shipments--which had been under 100,000 units per year until 1973--have been steadily on the increase since then. According to the Air-conditioning and Refrigeration Institute, 1976 heat pump shipments totalled 312,000 units: a figure which is more than double that of the previous year. When compared to all central heating units shipped, heat pumps have increased their share of the total market to 8.3 percent in 1975 from 4.7 percent in 1973.

The distribution of heat pump installations within the residential building market is given in Exhibit 10-2 for the years 1971-1976. New installations comprise about two-thirds of the total units installed in residential service. The reduction in the percent of multifamily installations would be related to the observed trend away from new multifamily building construction.

Commercial service heat pumps--generally 3 tons or larger--comprised about 25 percent of all heat pumps shipped in 1974 and 1975. Of this, heat pumps 5 tons or larger accounted for 12.3 percent of all shipments in 1974 and 10.1 percent in 1975. The proportion of heat pumps shipped 7 1/2 tons or larger was 2.9 percent in 1974, which declined to 2.4 percent of total shipments in 1975 (Air-Conditioning and Refrigeration Institute, 1976).

One of the possible influences on the small proportion of 7 1/2 tons and larger size units could be related to the substantial use of built-up systems for larger installations. About half the commercial building installed heat pump tonnage is estimated to be of the built-up type (Edison Electric Institute, 1976).

10.3.2 Cooling Equipment Market Trends

Manufacturers' shipment data of unitary central air conditioning equipment for the years 1967-1975 is illustrated in Exhibit 10-3. Included are data for the electric air conditioner, the electric heat pump and the gas air conditioner.

Comparison of unitary air conditioner shipment data with total completed housing units adjusted by the central air conditioning installation rate for new housing indicates that up to about one-third

of the units shipped are installed in new residential construction, the remainder going into existing buildings, new nonresidential construction, or inventory. This comparison is shown in the table below.

Central Air Conditioning Shipment Analysis

Year	Total Shipments Units X 10^{-3}	Units Installed In New Single Family Home Completions Units X 10^{-3}	Percent	All Other Installations or Inventory Units X 10^{-3}	Percent
1971	1,877	614	32.7	1,263	67.3
1972	2,403	848	35.3	1,555	64.7
1973	2,839	988	34.8	1,851	65.2
1974	2,489	811	32.6	1,678	67.4
1975	1,623	635	39.1	988	60.9

Source: Air-Conditioning and Refrigeration Institute, National Association of Homebuilders, Census Bureau.

A further indication of the growth in central air conditioning can be seen from the Annual Housing Survey (Census Bureau, 1976) report for 1970 and 1974. In 1970, 7.3 million housing units or 10.7 percent of the total housing stock had central air conditioning. By 1974, this penetration had about doubled to 14.4 million units, or 18.9 percent of all housing units in place. Based on the air conditioning saturation data for new housing completions during the 1971-1974 period, about 3.3 million of the increase in units with central air conditioning came from new residential

construction, while 3.8 million units with central air conditioning would have been retrofit installations in existing houses.

In the previous discussion of heating equipment market trends for commercial installations, it was reported that heat pump shipments in sizes 5 tons or larger were 10 to 12 percent of all heat pump shipments for the years 1974 and 1975. By comparison, industry-reported shipments of year-round air conditioners 5 tons or more were over three times greater than the number of heat pumps shipped in 1975. The ratio for year-round air conditioners 7½ tons and over for the same year was about seven times greater than the reported shipments of heat pumps of a comparable size (Air-Conditioning and Refrigeration Institute, 1976).

The large difference in year-round air conditioner versus heat pump shipments suggests that a substantial potential exists for heat pumps in the larger sizes. Moreover, the year-round air conditioner is generally a roof top unit with heating supplied by natural gas or oil fired equipment. This would be one of the target markets for the light commercial heat pump. To the extent that the market is not now served by the electric heat pump, a good market potential could develop for a gas heat pump, particularly if natural gas is the present heating fuel.

One aspect of special interest to possible use of a gas heat pump is the prevalence of gas air conditioning. As shown in Exhibit 10-3, gas air conditioning shipments peaked in 1969 with under 70,000 units shipped, and have continued to decline to 20,000 units in 1975. Possible explanations for this decline are:

- Increased cost and lack of gas availability (discussed in Chapter 11);
- Higher equipment cost. In 1973 the cost of an installed gas air conditioner was $200 to $300 more than a comparable electric air conditioner;

- Former poor reliability. As gas air conditioning became more heavily promoted in the late 1960's, there was pressure to cut equipment costs to expand the market. This apparently resulted in equipment of lower reliability being offered. The reliability problems, however, are not inherent in the technology of gas absorption air conditioning. It is believed that the reliability of present gas air conditioning equipment is good.

Only one manufacturer of gas air conditioners remains in the market, two others having ceased production of this equipment (Columbia Gas Co., 1977).

10.4 Market Potential for Advanced Heat Pumps

This section presents projections of the total potential heat pump market and estimates of the market prospects of electric and gas heat pumps. The products considered are defined in Subsection 10.4.1. The building model used for estimating the residential and nonresidential space conditioning market is discussed in Subsection 10.4.2. Subsection 10.4.3 gives the projection of the total heat pump market for electric and gas heat pumps, while 10.4.4 presents the market size estimates for a conventional electric heat pump, an advanced electric heat pump, and a gas heat pump.

10.4.1 Definition of Products Considered

In the course of this study, over 20 different heat pump concepts or projects under development were investigated. Because of their varying stages of development, it is not possible to identify a distinct market for each product. Instead, the approach taken was to consider a generalized class of advanced heat pump products, without specifying which particular product would be commercialized or the extent of its market success. A definition for each type of heat pump product is given below.

- Standard Electric Heat Pump. The standard electric heat pump envisioned is basically a single speed hermetic electric motor driven unitary appliance in the nominal 1-25 ton cooling capacity range. For this report, moderate efficiency improvements are assumed (i.e. from the present on-site 1.6 SPF to an SPF of 1.8). *

- Advanced Heat Pump. The advanced electric heat pump considered here is expected to initially have a two-speed motor, and later, to be continuously modulated, staged or otherwise be substantially modified over the standard electric heat pump. It is expected to be offered in the nominal 1-25 ton cooling capacity range as a high efficiency model (i.e. on-site SPF of 3)* in parallel with the standard electric heat pump.

- Gas Heat Pump. The gas heat pump considered in this report is a thermal engine driven Rankine cycle or absorption cycle product now under development. It is expected to be offered in the nominal 1-25 ton capacity range as a high efficiency--on-site SPF of 1.4 * --gas-fired product.

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* The on-site seasonal performance factors (SPF's) shown are for a particular location (Philadelphia), and represent an integrated value for the steady state coefficient of performance (COP) and energy efficiency rating (EER) characteristics of the equipment as follows:

	Standard Electric Heat Pump	High Efficiency Electric Heat Pump	Gas Heat Pump
COP, 17°F	1.8	2.1	1.5
COP, 47°F	2.2	2.9	1.7
EER, 95°F	7.1	7.8	(0.7 COP)
SPF, on-site	1.6	1.8	1.4

An SPF of 3 is projected for the advanced electric heat pump to indicate a substantial improvement over any currently produced heat pump. It is not possible, however, to estimate the expected COP for the advanced electric heat pump at this time.

The gas heat pump COP is on an on-site basis and is not directly comparable to the electric heat pump COP's given because of the energy losses of electricity generation and in the transmission and distribution of both forms of energy.

10.4.2 Model Description

To project the market prospects for advanced electric and gas heat pumps, an estimate was developed for the total market potentially available for that particular type of heat pump. The general equation for total market is given as

Total Market = (New Building Construction + Net Conversions + Replacements) x (Adjustment Factor)

For the purpose of this study, total market is understood to mean the total domestic market. The adjustment factors in the above equation include corrections for central air conditioning saturation, the penetration of warm air heating systems, the incidence or availability of gas heating fuel and, in the case of nonresidential buildings, the proportion of new buildings expected to meet the light commercial building criteria (i.e., low rise, about 3 stories or less).

The estimated number of installations of a particular type of heat pump for any particular year can then be determined from the equation:

Installations of Heat Pump Product in year t = (Estimated Market Penetration in year t) x (Total Market in year t)

The market penetration rate for the standard electric heat pump was based on industry leader projections for new residential construction market share and trend extrapolation. Since the standard heat pump is a currently marketed item, it is reasonable to utilize available current market information and projections.

Because neither the advanced electric heat pump or the gas heat pump are currently marketed, the methodology used to estimate market

penetration was to assume certain patterns of development based on analogous space conditioning equipment. Two growth rates were studied: a high growth rate based on the relatively good market acceptance of the unitary electric air conditioner when it was initially introduced, and a low growth rate based on the actual growth of the electric heat pump, which had numerous market acceptance problems when first introduced due, among other things, to poor reliability and inadequate industry servicing capability.

10.4.2 Total Potential Electric and Gas Heat Pump Market Projection

Fundamental to projecting the heat pump market universe are assumptions that all buildings have electrical service and could therefore be space conditioned by an electric heat pump, that a heat pump would only be installed where comfort cooling is desired (an exception to this is the heat-only heat pump), and that with minor exception for certain nonresidential installations, a central air distribution system would be required for a heat pump installation. In addition to these general requirements, assumptions specific to each market are discussed along with the market projections in the following sub-sections.

Electric Heat Pump Market

It was previously stated that, on the average, annual residential construction is expected to be 1.75 million units yearly through the year 2000, distributed regionally similar to the 1971-1974 average for new construction (Commerce Department, 1974). Adjustment of the regional construction estimate by the saturation of central air conditioning, as shown in Exhibit 10-4, provides an annual potential market estimate of 1.079 million electric heat pump units for new residential construction for each year during the period through year 2000.

The replacement market in residential buildings is estimated to be two-thirds the new construction plus conversions (American Gas Association, 1977). Since replacements would occur some time after a building has been constructed, the warm air system saturation rate is used to estimate the replacement market and the prevalent regional saturation rate for central air conditioning. The potential replacement electric heat pump market for residential buildings is estimated to be 611,000 units annually to the year 2000.

Conversions of existing residential buildings to electric heat pumps is assumed in this study to continue at the 1975 level of 31,000 units. The total annual potential market for electric heat pumps--new construction, replacements and conversions--is estimated at 1.721 million units. This is summarized by region in Exhibit 10-4.

The nonresidential market estimate for electric heat pumps was derived from estimates of new construction (General Electric Co., 1976), adjusted for building type (Commerce Department, 1976) and light commercial heat pump service, i.e. centrally air conditioned with 25 ton or less cooling equipment requirement. Available information on nonresidential space-conditioning conversions and replacements would indicate this market to be about 10 percent of the new construction market for nonresidential buildings. Based on an estimate of one ton of cooling capacity required for every 300 ft^2 building area (Consolidated Natural Gas. Co., 1976) and with an average 4.8 tons cooling capacity per electric heat pump, the total nonresidential heat pump market for 1981 was estimated to be 581,000 units. This is summarized in Exhibit 10-5.

For the nonresidential market, a new construction growth rate of 2 percent per year through 2000 was used (Smith Barney, 1976). The potential electric heat pump market for nonresidential buildings in terms of area, tonnage and units for selected years is given below.

Potential Nonresidential Electric Heat Pump Market

Year	Building Square Footage $X\ 10^{-6}$	Heat Pump Tonnage $X\ 10^{-3}$	Heat Pump Units $X\ 10^{-3}$
1977	770	2,567	535
1981	836	2,787	581
1985	903	3,010	627
1990	997	3,323	692
1995	1,100	3,667	764
2000	1,215	4,050	844

Gas Heat Pump Market

The gas heat pump market is more limited than the electric heat pump market because not all areas are served by gas, nor is a major gas service expansion foreseen (AGA, 1975). Data from the American Gas Association on conversions to gas shows a continuing decline from 257,000 in 1974 to 100,600 in 1976, with a further drop to 85,000 expected by 1978). As discussed in Chapter 11, the available information indicates further reductions in overall gas supply for all except high priority users--i.e., the residential and commercial sectors--who will probably have gas available through reallocation of service away from industrial users, but a higher prices than at present. However, this gas availability situation is far from stable at this time. Obviously any change in gas

availability could affect the conclusions presented in this study on the market potential for the gas heat pump.

A summary of the residential building gas heat pump market by region is given in Exhibit 10-6. The total residential market potential during the 1981-2000 year period is estimated at 1,108,000 gas heat pump units per year.

As shown in Exhibit 10-7 the nonresidential building market potential for gas heat pumps is estimated to be 479,000 units in year 1981. Area, tonnage and units estimated for selected years is given below.

Potential Nonresidential Gas Heat Pump Market

Year	Building Area $ft^2 \times 10^{-6}$	Heat Pump Tonnage $\times 10^{-3}$	Heat Pump Units $\times 10^{-3}$
1981	690	2,300	479
1985	743	2,427	516
1990	814	2,713	565
1995	893	2,977	620
2000	922	3,240	675

As is the potential residential gas heat pump market, the nonresidential gas heat pump market is smaller than that of the electric heat pump because of regional limitations of gas coverage and availability.

10.4.3 Market Share Projections for Electric and Gas Heat Pumps

In the previous two Subsections, the heat pump market model and total heat pump market was developed for electric and gas heat pumps.

In this section, the model will be applied to produce estimates of the projected market share of the electric and gas heat pumps.

The market model assumptions and the market penetration rates used in this study are described in Exhibit D-1 in the Appendix. The conventional electric heat pump market penetrations are based on manufacturers' estimates and extrapolations, and are shown for each year through the year 2000. In the case of a presently marketed product such as the standard electric heat pump, some confidence in market penetration projections can be placed in the fact that a prior experience record is available, and near term projections generally will bear a very close relationship to the established market.

A much greater problem exists, however, for the advanced electric and gas systems since neither are currently marketed. The approach used in this report was to assume penetration rates based on historical market penetration data for analogous space conditioning systems. Two appliances were used; for the high growth scenario, the electric air conditioner which had relatively good initial market acceptance and growth, and for the low growth scenario, the electric heat pump which had numerous market acceptance problems when first introduced due, among other things, to poor reliability and inadequate manufacturer servicing capability. These appliances were selected because their operation and complexity would be more closely related to an advanced electric or gas heat pump than, for example, those of a gas furnace. The penetration rates used are smoothed data based on the actual historical penetration rates.

In using these growth rates to project the market size for the conventional and advanced systems, certain assumptions should be noted.

- Standard electric heat pumps will continue to be improved in efficiency and performance. They will constitute a basic or builder's model product which is expected to follow the growth trends projected by the industry;

- First generation advanced electric heat pumps may be introduced in 1979. Second generation continuously modulated advanced electric heat pumps may be introduced in mid-1982. Growth trends in the advanced electric heat pumps are expected to follow a pattern similar to historical growth patterns observed for similar comfort conditioning equipment;

- First generation gas heat pumps are projected to be introduced by 1981 and will follow growth rates as indicated above in those markets where gas is traditionally the heating fuel;

- The selling price increase for the advanced heat pumps will be within the maximum 30 percent premium expected by the manufacturers;

- There will be no significant insurmountable institutional barriers to the use of advanced heat pumps. (Various institutional related accelerating or retarding factors are discussed in Chapter 11). There are some factors, such as the possible future shortage of natural gas, which could severely limit the market potential for a gas heat pump (but may actually enhance the market potential for an advanced electric heat pump).

Market Projection for the Standard Electric Heat Pump

The historical and projected domestic market for standard electric heat pumps to the year 2000 is given in Exhibit 10-8. Also shown in this exhibit are forecasts for 1976-1981 made by the Air-Conditioning and Refrigeration Institute (ARI) based on individual manufacturer shipment estimates. The Gordian projection for total electric heat pump shipments is in good agreement with the Air-Conditioning and Refrigeration Institute's forecast for the period covered.

By 1990, it is estimated that the standard electric heat pump market share will be 46.5 percent of total residential and nonresidential building heating equipment market. This is expected to reach 63 percent by the year 2000. In terms of units, the projection is for shipments of 1,125,300 units in 1990 (800,300 residential; 325,000 nonresidential) and 1,621,300 units by year 2000 (1,084,200 residential and 537,100 nonresidential units). On a tonnage basis, the residential units would represent 2.16×10^6 ton cooling capacity, and the nonresidential units 1.56×10^6 ton cooling capacity in 1990; and in the year 2000, the tonnages projected are estimated to be 2.93×10^6 ton residential capacity and 2.58×10^6 ton nonresidential capacity. Thus, while the residential market shipments are projected to be 67 percent of total units shipped in 2000, they are projected to be only 53 percent of the tonnage cooling capacity shipped.

Market Projection for the Advanced Electric Heat Pump

The projected domestic market for advanced electric heat pumps is given in Exhibit 10-9. The nominal projection shown represents the average case between the projected high and low growth rates. A high-low range is used because of the uncertainty as to how the market place will accept the new technology. The expected units sold and the corresponding tonnage is shown below for selected years.

Unit Sales and Tonnage Projections for the Advanced Electric Heat Pump

	Low Projection		High Projection	
Year	Units X 10^{-3}	Tonnage X 10^{-3}	Units X 10^{-3}	Tonnage X 10^{-3}
1	11.1	25.9	18.9	61.9
5	40.8	132.4	104.8	240.6
10	71.8	236.4	222.6	732.2
20	146.1	493.4	609.7	2,059.1

As with the standard electric heat pump, the nonresidential tonnage, at 46 percent of all tonnage installed in the year 2000, represents a substantial share of the market. This relationship of units to tonnage is further illustrated by Exhibit 10-10, which shows the total potential market in 1979 for the advanced electric heat pump by size of units and by number of units compared to the tonnage distribution by size of unit shown in Exhibit 10-11 for the same year. The latter exhibit suggests that tonnage is a valid measure of market importance along with the estimate for number of units sold.

Market Projection for the Gas Heat Pump

The projected domestic market range for gas heat pumps is given in Exhibit 10-12. Also shown in this exhibit are market projections made independently by the American Gas Association (for the residential building market only) and by the General Electric Co. (for their nominal growth projection). Both independent projections lie within the range developed by the Gordian projection.

Given below are expected unit sales and corresponding tonnages for selected years based on the projections given in Exhibit 10-12.

Unit Sales and Tonnage Projections for the Gas Heat Pump

	Low Projection		High Projection	
Year	Units X 10^{-3}	Tonnage X 10^{-3}	Units X 10^{-3}	Tonnage X 10^{-3}
1	8.0	26.6	13.5	45.1
5	28.5	106.3	73.4	247.6
10	50.4	172.1	156.2	533.5
20	103.8	364.5	433.3	1,521.3

Although sizeable, the market prospects for the gas heat pump are projected to be less than for the advanced electric heat pump because the total potential market for gas-fired equipment is limited by gas availability in addition to the demand for comfort cooling, which is generally the limiting factor for electric heat pump market potential.

10.4.4 Implementation Time Scale

The foregoing analysis of the market prospects for advanced electric and gas heat pumps was based on the assumption that growth rate would follow the historical market share growth pattern regardless of the year introduced. While this approach permits projection of the expected upper and lower market bounds for the advanced heat pumps, it is necessary to estimate the expected calendar time of introduction of the advanced heat pumps in order to assess their potential for conserving scarce fossil fuels over specific periods in the future. A preliminary analysis of the impact of advanced heat pumps on electric and gas utilities is discussed in Chapter 13, based on the estimates for market size and time of implementation presented in this chapter.

As discussed in Chapters 3 through 7, there are a number of different heat pump concepts now in varying stages of development. In going from feasibility to commercialization, a number of stages may be involved, including:

Feasibility - establish technical viability of concept and build "breadboard" (i.e. laboratory constructed) working model. At this stage, the various components of the system may not be tested as one unit. A period of 1-2 years or longer is usually involved.

Prototype - hand build working units for testing of system in laboratory. A period of 12 to 18 months may be involved.

Preliminary Field Test - based on improvements made over the first prototype units, a limited number of units (from 5-20) may be put into the field to optimize the system performance and to check out possible design problems. A period of 12 to 18 months may be involved.

Full Scale Field Test - following the preliminary field test of hand built units, limited production is undertaken to build and field test between 100 and 1000 units. At least 2 full heating seasons are needed to ascertain system reliability and performance, a period of 18 to 24 months.

Commercialization - tooling-up to produce units in quantity of from 10,000 to 50,000 units. Part of this tooling-up can be concurrent with the field test.

Based on the foregoing a rough minimum time estimate for development of a new heat pump concept through commercialization may be 5 years, but more likely 7 or 8 or longer. Available funds are critical to the development time, because while the stages from prototype onward can be carried out simultaneously, entrepreneurial caution dictates that each stage be successfully completed before going to the next. Program acceleration of up to 2 years could be obtained by concurrent testing and tooling-up for commercialization, assuming that funds could be made available for concurrent programming of development stages.

A projected time scale for commercial introduction of advanced electric and gas heat pump products is given in Exhibit 10-13. This projection is based on the assessment of the status of technical developments of the systems in question as a result of discussions with various advanced heat pump concept developers and manufacturers.

Where a concept is presently in the experimental or pre-feasibility stage (or at an earlier theoretical concept stage), the assessment may be made that the time to commercialization is even greater than the 7 to 8 years previously indicated. It should also be noted that the time of product introduction shown in Exhibit 10-13 is the author's estimate for the earliest expected time for commercialization assuming no acceleration as a consequence of additional funding.

10.5 Impact of Economic and Institutional Factors On Market Prospects for Advanced Heat Pumps

The previous sections of this chapter analyzed the market potential for advanced heat pumps in terms of buildings and market share or penetration rate based on analogous comfort conditioning appliances. Projections of units installed to the year 2000 have been made for both advanced electric and gas heat pumps, as derived from a space conditioning point-of-view. In the actual market place, however, other considerations such as economic and institutional factors come into play. Although both aspects are discussed more fully in Chapters 9 and 11 respectively, a brief discussion is presented here as to the possible accelerating or retarding influences these economic and institutional factors may have on the market prospects for advanced heat pumps.

10.5.1 Economic and Institutional Factors Impact on Advanced Electric Heat Pumps

Impact of Economic Factors

The primary accelerating economic factor favoring advanced (higher efficiency) electric heat pumps is the rapidly increasing cost of energy which translates into higher utility bills (i.e. higher operating cost)

for space conditioning. Purchasers of space conditioning equipment have in recent surveys (Arkansas Power & Light Co., 1974; Professional Builder, 1976) indicated that lower operating and maintenance cost expectations were a major factor in selecting a heat pump, and that an added investment of up to $600 for energy conservation would be acceptable if the payout were within six years.

This added investment, for example, would be within the range of added first-cost found in this study for installing a high efficiency electric heat pump in place of the standard heat pump. However, under today's energy costs, a decision to purchase greater efficiency in a space conditioning device may not be economically justified on a life-cycle cost basis because of the higher first-cost of the more energy efficient equipment.

Obviously, if the advanced electric heat pump (taken to have an annual SPF of 3) is to reach its projected market penetration, installed cost cannot greatly exceed the buyer's limit for marginal investment in return for energy cost savings. It should be stressed that, although the saving of fuel may be an acknowledged goal for the country, in a free market the individual consumer is expected only to pay for efficiency to the extent that the added investment could potentially be recovered through the savings in fuel consumption. If the advanced electric heat pump (when commercially available) were to be much higher in first-cost than today's high efficiency electric heat pumps, cost would be an obvious potentially retarding factor. Since an advanced SPF 3 electric heat pump has not as yet been built, there is no adequate basis for assigning some actual quantitative effect to the higher first-cost vs. efficiency trade-off.

The foregoing discussion, while based primarily on the residential building comfort conditioning studies, would not be expected to be substantially different for nonresidential building space conditioning considerations.

Impact of Institutional Factors

There do not appear to be any significant retarding institutional factors related to the use of electric heat pumps in new residential buildings. Present accelerating factors are the shortage of natural gas and the high cost of electricity, with the former forcing the space conditioning appliance choice away from the gas furnace toward electric heat, and the latter forcing the choice toward the more efficient heat pump versus resistance heating. The increasing popularity of central air conditioning is a further accelerating factor to the use of a heat pump. Likewise, consumer attitudes, both of present owners and potential buyers, are favorable toward the heat pump. While an advanced electric heat pump is expected to be more efficient, it will still be in design and basic mode of operation fairly similar to today's heat pump. Therefore, no additional consumer learning or acceptance process is expected to be necessary for the new generation electric heat pump for residential buildings when it becomes available in the market place.

The above situation should be much the same for nonresidential building as for residential building electric heat pump installations, with one major exception. A recent legislative enactment in the State of California prohibits the use of strip heat from supplying more than 10 percent of the total heat requirement for a nonresidential building.

The net effect of this rule is to require oversized compressor capacity in order for the heat pump to meet a greater heating load than would normally be the case. This requirement may not only result in greater first cost for a larger appliance, but may be wasteful of energy because an over-sized compressor will cycle more frequently than a compressor that has been properly sized to meet the building cooling load. As discussed in the next chapter, the rule has not as yet been enforced by the State of California (California Energy Resources Conservation and Development Commission, 1976).

10.5.2 Economic and Institutional Factors Impact on Gas Heat Pumps

Impact of Economic Factors

The competitive choice vis-a-vis the gas heat pump is the gas warm air furnace with a high efficiency air conditioner. Excluding the baseboard convector with room air conditioner system because, although lowest in both first cost and life-cycle cost, it does not provide comfort conditioning comparable to a centrally ducted system, the gas warm air furnace with high efficiency air conditioner has the lowest system installed cost of all the systems studied. In addition, it generally has the lowest life-cycle cost of all systems studied, with the one exception as noted above. Equipment cost for the gas heat pump, on the other hand, has been estimated to be between 15 and 30 percent higher than a comparable-sized conventional gas furnace with high efficiency air conditioner. Installed gas heat pump system cost is even greater, since the contractor's fee is applied to a larger base equipment cost. (See Chapter 9 for a discussion of the calculation of equipment installed cost.)

Present system installed cost estimates for both residential and non-residential gas heat pumps are substantially over $600 greater in first-cost than the base gas furnace/air conditioner comparison, a situation which would be a definite retarding factor with respect to acceptance by the residential building consumer. Similarly, life-cycle costs for gas heat pumps only show cost equivalence under inflated energy considerations.

The payout on marginal installed cost for commercial-size gas heat pumps in particular appears to exceed for most locations the 5 year maximum that nonresidential building owners have indicated as the outer limit. However, it is possible that gas heat pump equipment costs will be less than presently estimated, and that gas fuel costs may be even higher than projected, in which case lower first-cost and greater operating cost savings will enhance the selection of the gas heat pump. In view of the economic uncertainty as to first-cost or life-cycle cost for gas heat pumps it is not possible at this time to estimate the impact of the economic factors on the market prospects for gas heat pumps. However, if the equipment cost does exceed 30 percent one could readily expect buyer resistance in spite of the fact that the gas heat pump actually saves gas in most locations while providing both heating and cooling space conditioning.

An even greater threat to the market prospects of the gas heat pump is the pulse gas warm air furnace currently under development by the American Gas Association. This furnace is expected to provide efficiency improvements in the order of 40 to 50 percent over the present conventional warm air furnace. However, it is not expected to cost much more than the present conventional gas furnace. If the consumer's interest is in

comfort heating only, the more efficient pulse gas furnace would be a logical choice. This development could, in particular, affect the replacement market where the consumer has gas heat and wishes to replace his-worn out or inefficient gas furnace with a new one. Unless the owner wishes to up-grade his space conditioning system to include comfort cooling, the choice of a much lower cost pulse gas furnace would not be unexpected. As noted, the pulse gas furnace is still under development, therefore any estimate of its impact on market penetration of the gas heat pump would be at this time speculative.

Impact of Institutional Factors

The most significant factor, obviously retarding, is the gas availability situation. In spite of various estimates that gas will be available for new residential or commercial buildings, there is substantial uncertainty as to the amount of gas that can be supplied. Weather obviously has a major influence on gas availability and if there are more winters as severe as the 1976-1977 winter, there is little doubt that more gas curtailments and restrictions will be imposed. However, since the number of gas heat pump units initially projected to be installed are not very large, it is likely that they could be supplied with gas regardless of restrictions in new gas hook-ups. In addition, restrictions would not be expected to the installation of a gas heat pump as a replacement for an existing gas furnace. The replacement market has been viewed by one gas heat pump developer as representing over half of all projected installations for their gas heat pump.

Another retarding factor, at least potentially, is restrictive code requirements due to the high working pressure of some of the gas

heat pump configurations under development (but not all, e.g.Brayton-Rankine gas engine driven heat pump operates at atmospheric or even slight negative pressure). As discussed in the next chapter, the high pressure involved with certain gas heat pump systems is not believed to be a serious problem to obtaining code approval.

10.6 Summary

The market prospects of advanced electric heat pumps and gas heat pumps are projected to be significant for both residential and nonresidential building applications. For the advanced electric heat pump, a market of between 72,000 and 223,000 units is estimated for the tenth year after introduction, while a market of between 50,000 and 156,000 gas heat pumps is estimated for a similar time after introduction, provided development results in a practical and cost-effective product. Because of limitations in gas distribution and gas availability, the potential market for a gas heat pump is less than for an advanced electric heat pump.

Development of a new heat pump concept from the feasibility stage through commercialization may require 7 to 8 years or longer. Acceleration of the time to commercialization of up to 2 years may be possible if adequate funds are provided to hasten the lengthy and costly full-scale field testing that is necessary before any manufacturer would be willing to introduce a new comfort conditioning appliance. Time scale estimates project the introduction of the organic absorption gas heat pump in 1981, followed by an advanced concept electric heat pump by mid-1982. A gas heat pump for commercial applications could be introduced by 1983.

The major driving force toward market acceptance of an advanced electric or gas heat pump is operating cost savings compared to a base system such as electric resistance heating or gas furnace with central electric air conditioner. However, the consumer, although energy-conservation-conscious, will not pay for additional space conditioning system efficiency if the payout period exceeds 5 or 6 years, or if the first cost of the system greatly exceeds what he is prepared to spend in additional investment. Based on recent consumer attitude surveys, this ceiling may be as low as $600 for a residential consumer, or 20 percent more than the cost of the base system for a nonresidential building consumer. Present data indicates that life-cycle costs still favor the base competing systems. Similarly, payout analyses for gas heat pumps versus the gas furnace with central air conditioner indicate that energy cost savings under present inflation expectations would represent at best a breakeven situation over the life of the equipment which is taken to be 9 years for a gas heat pump. This of course may change if the actual gas heat pump equipment first-cost is below present estimates, or if the cost of natural gas inflates higher than presently projected.

With the exception of natural gas shortages causing some possible uncertainty in the availability of new building gas hook-ups for the gas heat pump (and which still leaves a sizeable replacement market opportunity), there are no institutional factors that appear to have a significant retarding impact at this time for the advanced systems.

A possible competitive product which could greatly under-cut the market prospects for the gas heat pump is the pulse gas furnace currently under development. If the pulse gas furnace development is successful, it would provide efficiency not too far from a gas heat pump, yet likely have a life-cycle cost below the present gas warm air furnace with high efficiency

central air conditioner.

There does not appear to be a comparable competitive situation facing the advanced electric heat pump at this time. However, evolutionary improvements in conventional electric heat pump performance may narrow any marginal operating cost savings advantage of an advanced electric heat pump, thereby reducing the market prospects of the latter.

References to Chapter 10

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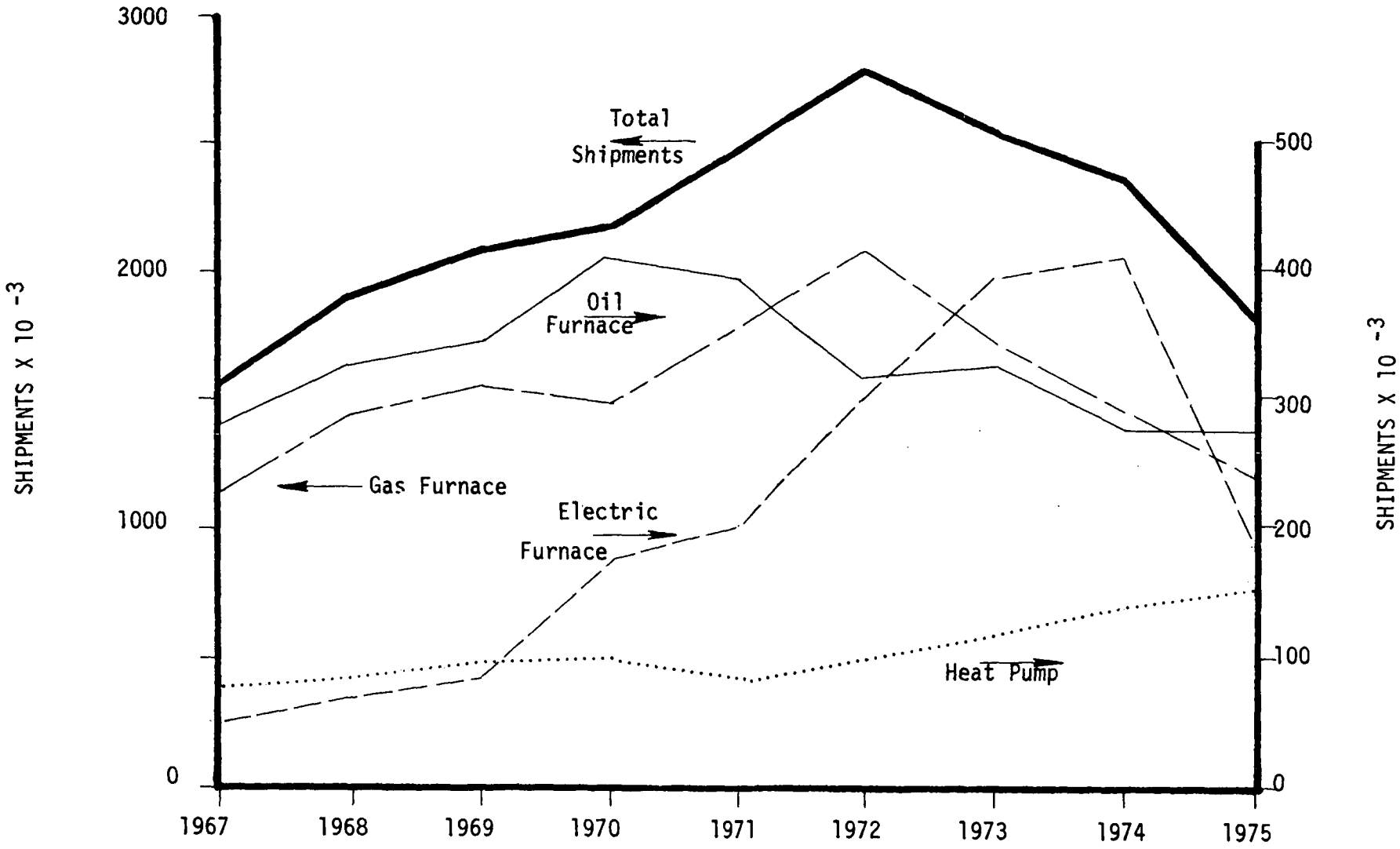
EXHIBIT 10-1

CENTRAL HEATING EQUIPMENT SHIPMENTS



Source: Gas Appliance Manufacturers Association. Air-Conditioning and Refrigeration Institute, Air Conditioning, Heating & Refrigeration News.

EXHIBIT 10-2

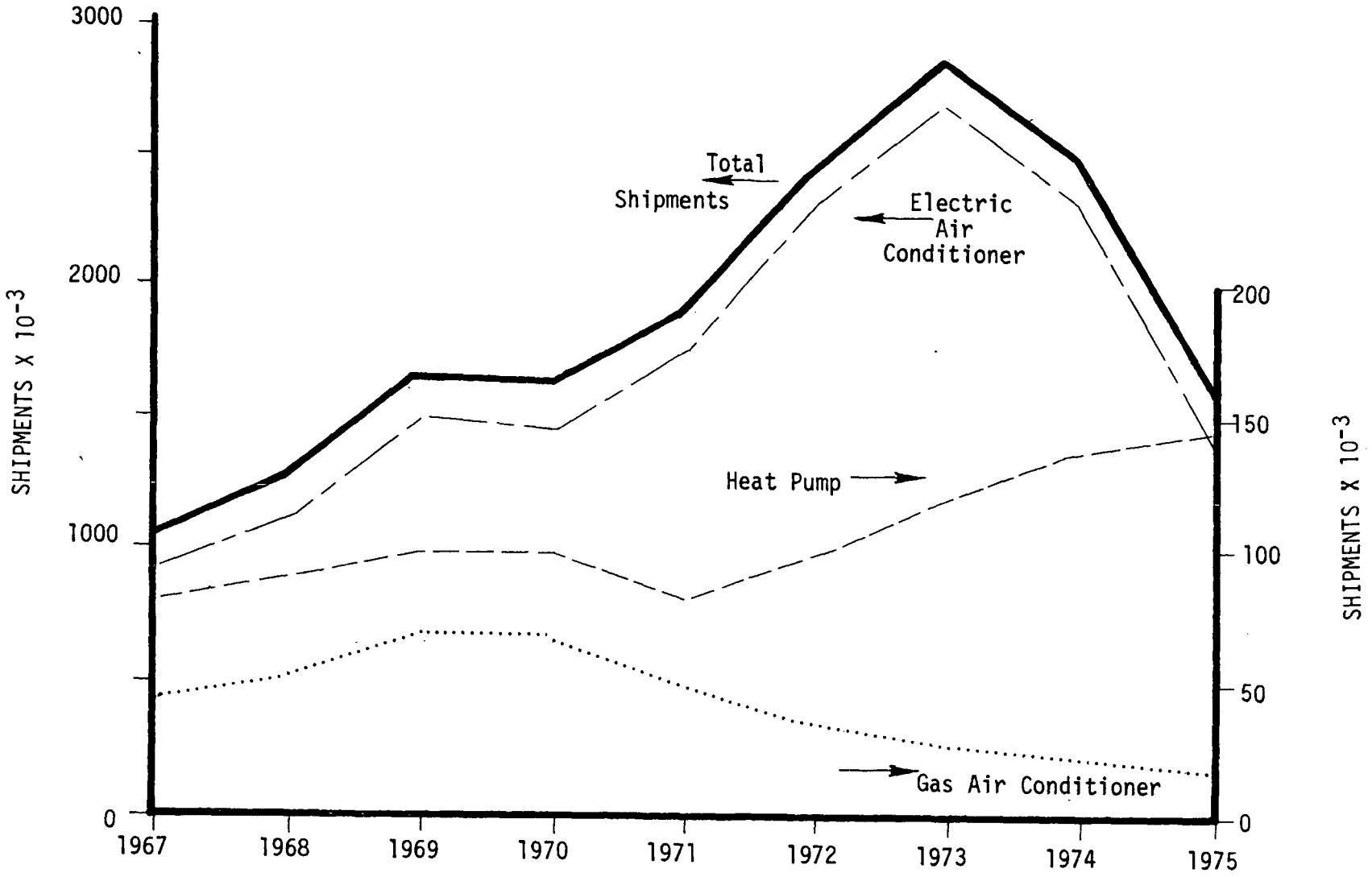
DISTRIBUTION OF RESIDENTIAL HEAT PUMP INSTALLATIONS

	Percent of Residential Installations					
	1976	1975	1974	1973	1972	1971
New installations in						
single family homes	61.1	57.5	53.1	50.4	44.5	44.7
multifamily homes	8.0	5.9	12.5	12.6	18.5	26.9
mobile homes	0.8	0.3	1.0	0.1	--	--
First installations in						
existing homes ("add-ons")	24.8	24.5	29.2	32.0	24.7	19.4
Replacements	5.3	11.8	4.2	4.9	12.3	9.0
	100.0	100.0	100.0	100.0	100.0	100.0
TOTALS	250,300	128,000	97,000	87,000	81,000	67,000

Source: Air-Conditioning and Refrigeration Institute, 1977.

EXHIBIT 10-3

UNITARY CENTRAL AIR-CONDITIONER SHIPMENTS



Source: Air-Conditioning and Refrigeration Institute, Air Conditioning, Heating and Refrigeration News

EXHIBIT 10-4

POTENTIAL RESIDENTIAL BUILDING MARKET FOR ELECTRIC HEAT PUMP (1981)

Region	Projected Total New Construction Units X 10^{-3}	Percent Central Air Conditioning Saturation	Total New Construction Potential Market Units X 10^{-3}	Replacement Market 1/ Units X 10^{-3}	Conversion Market 2/ Units X 10^{-3}	Total Potential Market Units X 10^{-3}
Northeast	228	22.7	52	16	4	72
North Central	367	48.3	178	97	7	282
South	788	86.0	678	398	13	1,089
West	367	46.6	171	100	7	278
Totals	1,750	--	1,079	611	31	1,721

1/ Replacement market based on warm air installations.

2/ Conversion market based on 1975 actual heat pump conversions.

Source: Gordian Associates Inc.

EXHIBIT 10-5

POTENTIAL NONRESIDENTIAL BUILDING MARKET FOR ELECTRIC HEAT PUMP(1981)

Building Type	New Construction $ft^2 \times 10^{-6}$(1981) 4/	Light Service Factor 5/	Potential Market: New Construction $ft^2 \times 10^{-6}$	Potential Market: Conversions/ Replacements $ft^2 \times 10^{-6}$ 6/	Total Potential Market: $ft^2 \times 10^{-6}$	Total Potential Market: Tonnage $\times 10^{-3}$ 7/	Total Potential Market: Units $\times 10^{-3}$ 7/
Office 1/	502	50	251	25	276	920	192
Commercial 2/	420	75	316	32	348	1160	242
Institutional 3/	618	31*	193	19	212	707	167
Total	1,540	--	760	76	836	2787	581

1./ Includes miscellaneous and public buildings.

2./ Includes stores, warehouses and shopping centers.

3./ Includes hospitals, educational and religious buildings.

4./ Gordian estimate excluding industrial construction.

5./ Estimated percent of building type that is air conditioned and a potential user of an electric heat pump up to 25 ton capacity. (*the institutional factor is weighted.)

6./ Estimated 10 percent of new construction market.

7./ Number of units based on 1 ton cooling requirement for every 300 ft^2, with each unit having an average 4.8 ton capacity.

Source: Gordian Associates Inc.

EXHIBIT 10-6

POTENTIAL RESIDENTIAL BUILDING MARKET FOR GAS HEAT PUMPS (1981-2000)

Region	Projected Total New Construction Units X 10^{-3}	Percent Central Air Conditioning Saturation	Total New Construction Potential Market Units X 10^{-3}	Replacement Market 2/ Units X 10^{-3}	Conversion Market 3/ Units X 10^{-3}	Total Potential Market Units X 10^{-3}
Northeast	228	22.7	52	16	5	73
North Central	367	48.3	178	91	21	290
South	788	38.0 1/	299	178	11	488
West	367	46.6	171	83	3	257
Totals	1,750		700	368	40	1,108

1./ Estimated gas warm air saturation rate.

2./ Replacement market based on new gas heat installations.

3./ Total gas heat conversions estimated at 85,000, adjusted to projected central air conditioning saturation for 1981.

Source: Gordian Associates Inc.

EXHIBIT 10-7

POTENTIAL NONRESIDENTIAL BUILDING MARKET FOR GAS HEAT PUMP (1981)

Building Type	New Construction 4/ $ft^2 \times 10^{-6}$ (1981)	Light Service Factor 5/	Potential Market: New Construction $ft^2 \times 10^{-6}$	Potential Market: Conversions/ Replacements 6/ $ft^2 \times 10^{-6}$	Total Potential Market: $ft^2 \times 10^{-6}$	Total Potential Market: Tonnage $\times 10$	Total Potential Market: Units 7/ $\times 10^{-6}$
Office 1/	502	41.3	207	21	228	760	158
Commercial 2/	420	62.0	261	26	287	957	199
Institutional 3/	618	25.6*	159	16	175	583	122
Total	1,540		627	63	690	2,300	479

1./ Includes miscellaneous and public buildings.

2./ Includes stores, warehouses and shopping centers.

3./ Includes hospitals, educational and religious buildings.

4./ Gordian estimate excluding industrial construction.

5./ Estimated percent of building type that is a potential light service heat pump user, adjusted for gas coverage. (* The institutional factor is weighted.)

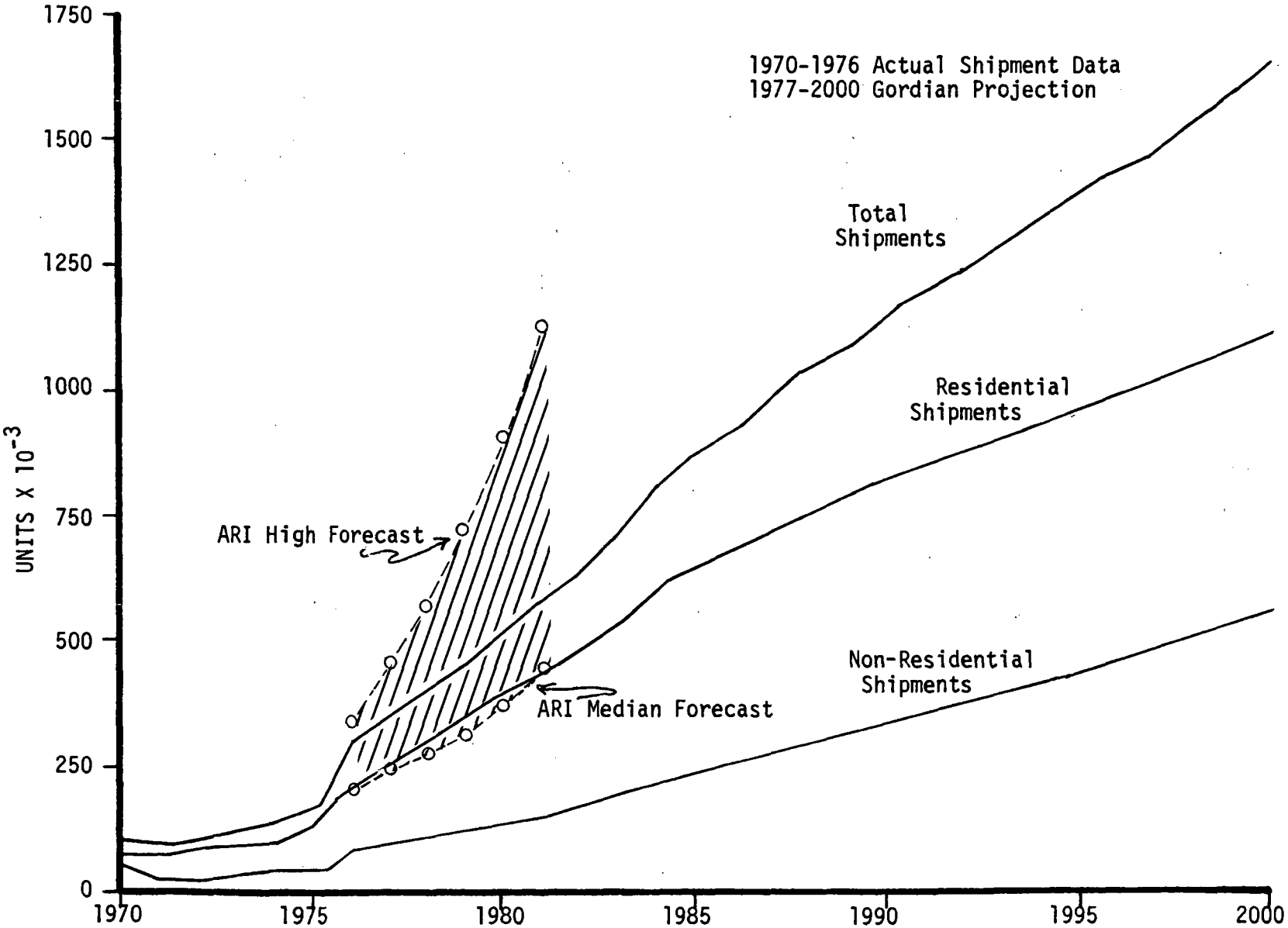
6./ Estimated 10 percent of new construction market.

7./ Number of units calculated based on 1 ton cooling requirement per 300 ft^2 space, with each unit having an average 4.8 ton capacity.

Source: Gordian Associates Inc.

EXHIBIT 10-8

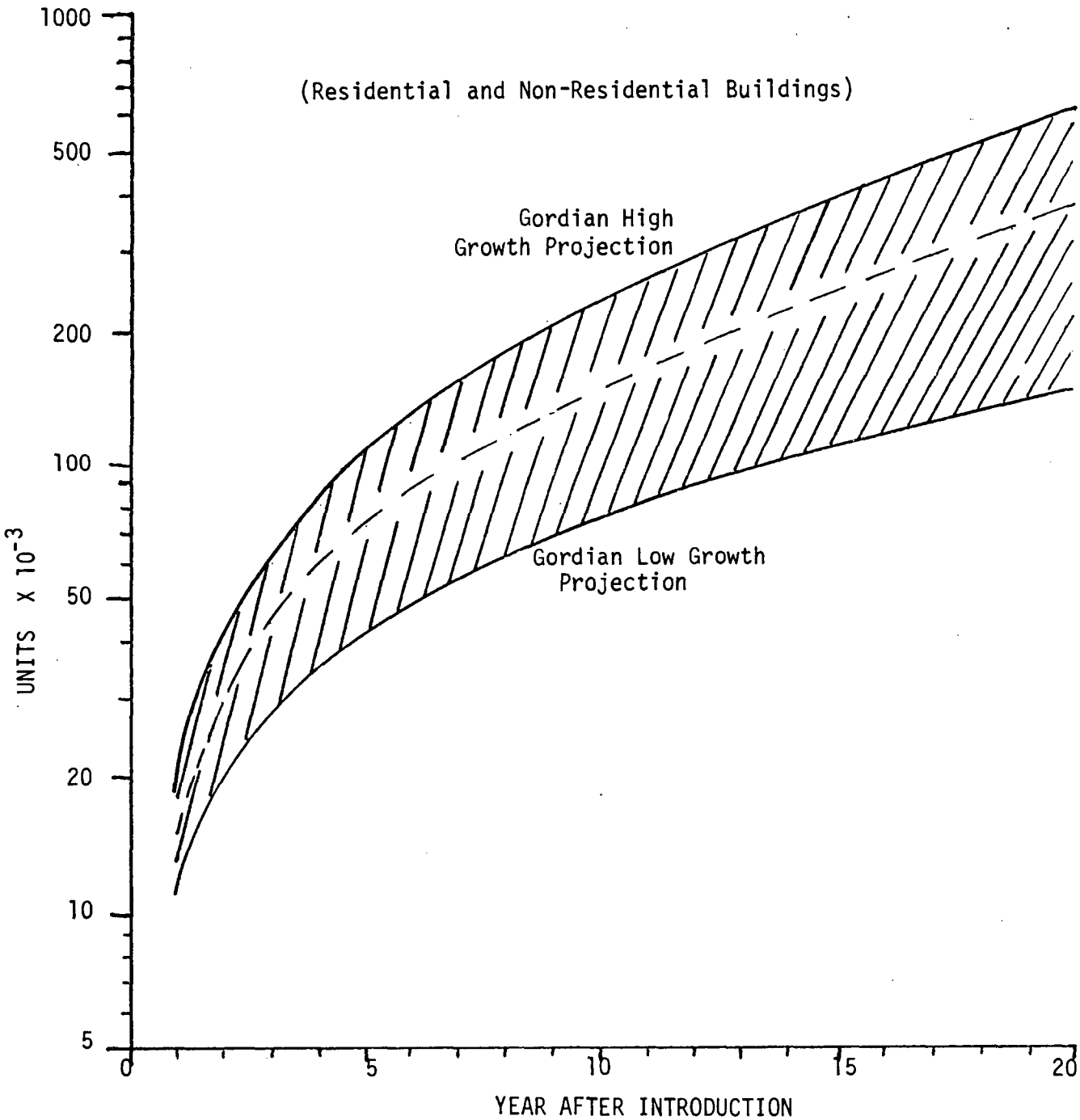
HISTORICAL AND PROJECTED DOMESTIC MARKET FOR STANDARD ELECTRIC HEAT PUMPS



Sources: Air-Conditioning and Refrigeration Institute, Gordian Associates, Inc.

EXHIBIT 10-9

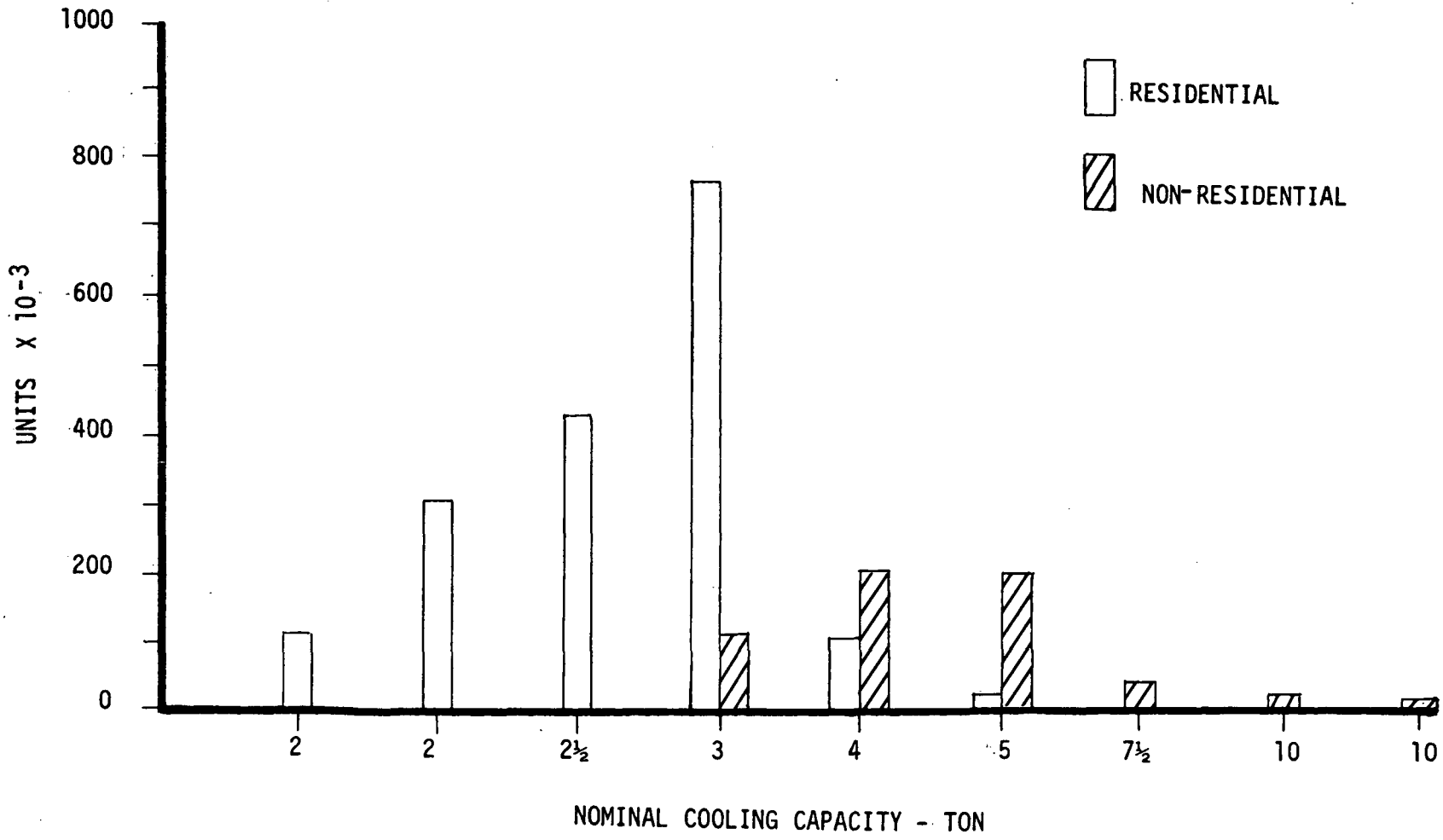
PROJECTED DOMESTIC MARKET FOR ADVANCED ELECTRIC HEAT PUMPS



Source: Gordian Associates, Inc.

EXHIBIT 10-10

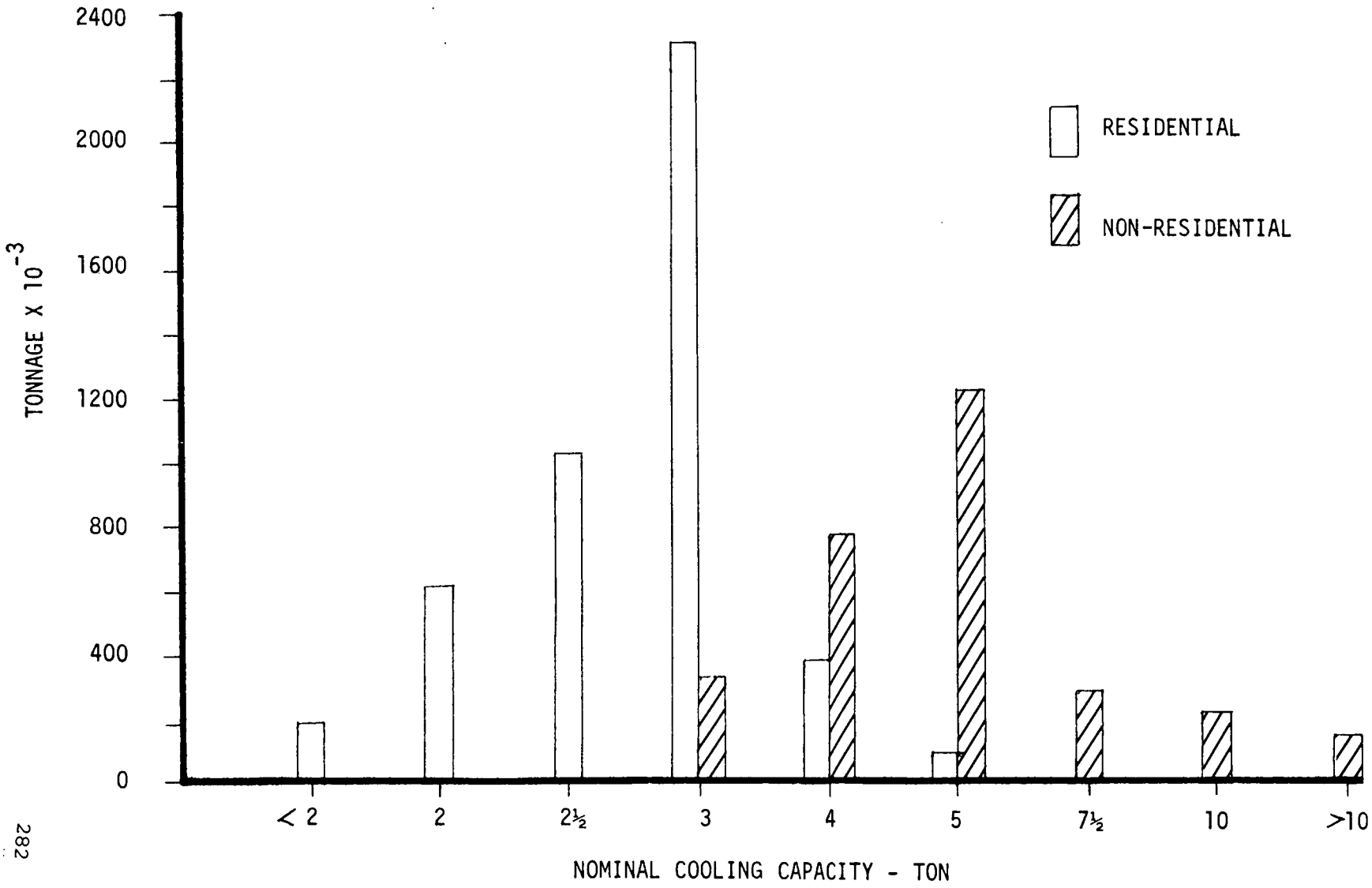
SIZE DISTRIBUTION OF TOTAL POTENTIAL MARKET FOR ADVANCED ELECTRIC HEAT PUMPS - 1979 ESTIMATE



Source: Gordian Associates, Inc.

EXHIBIT 10-11

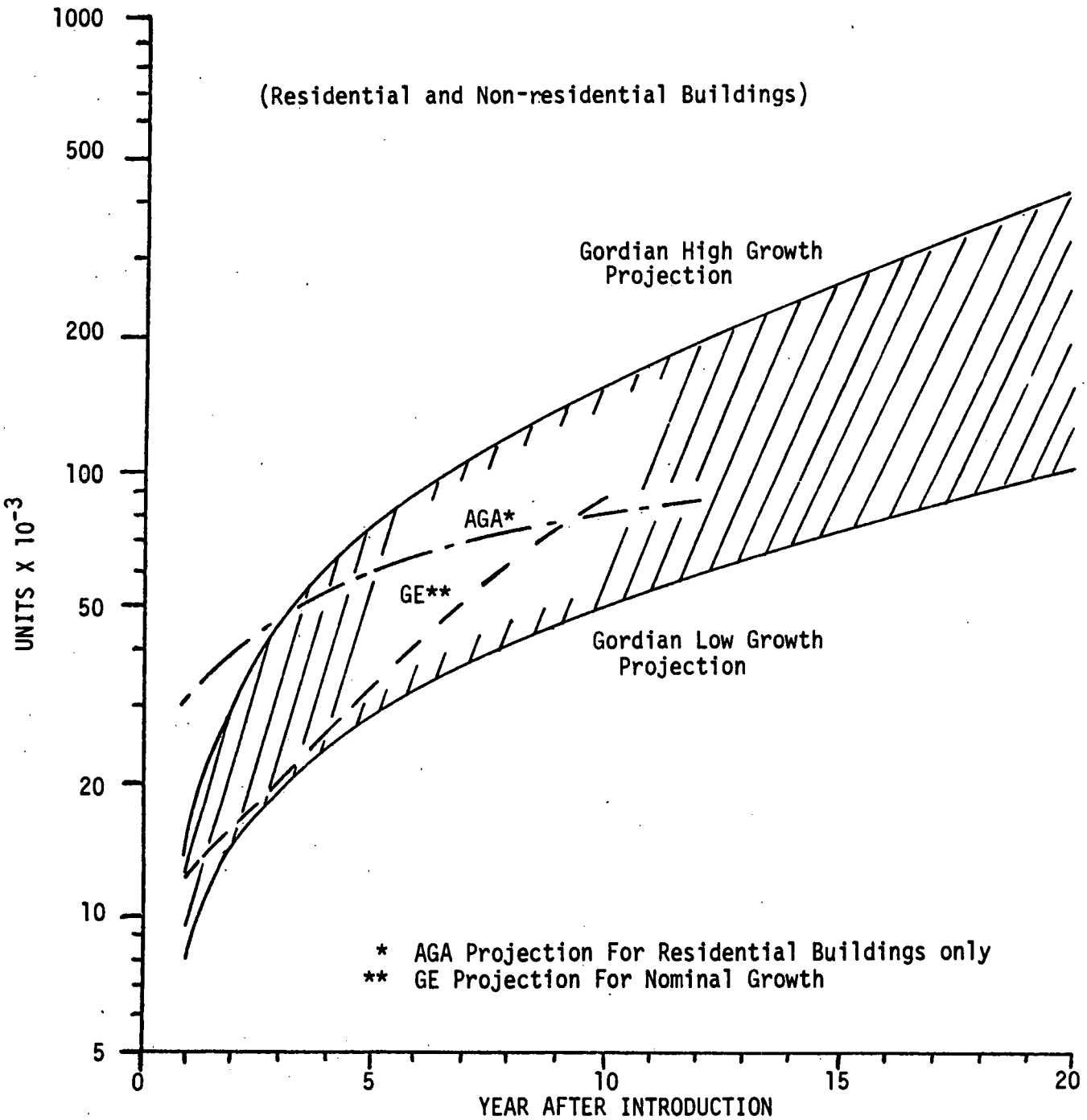
SIZE DISTRIBUTION OF TOTAL POTENTIAL TONNAGE FOR ADVANCED ELECTRIC HEAT PUMP -1979 ESTIMATE



Source: Gordian Associates Inc.

EXHIBIT 10-12

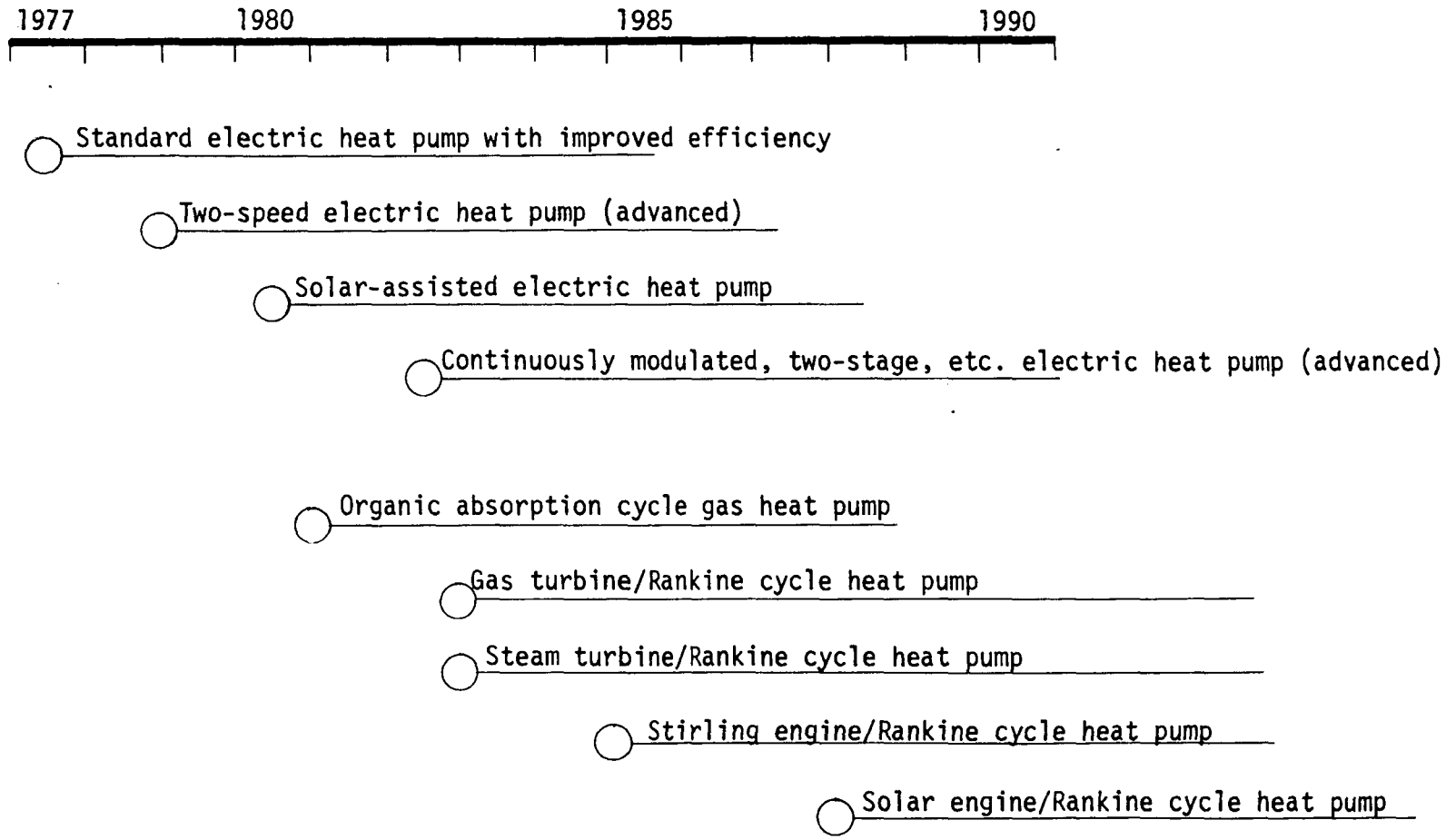
PROJECTED DOMESTIC MARKET FOR GAS HEAT PUMPS



Sources: American Gas Association; General Electric Co.; Gordian Associates, Inc.

EXHIBIT 10-13

PROJECTED TIME SCALE OF ADVANCED ELECTRIC AND GAS HEAT PUMP PRODUCT INTRODUCTION



Source: Gordian Associates Inc.

11. INSTITUTIONAL AND OTHER FACTORS

11.1 Introduction

The previous chapter considered the overall market prospects for advanced heat pumps from the viewpoint of market opportunity and market share without reference to the complex interplay of the factors involved in the decision-making process to purchase a particular space conditioning system. This chapter presents an evaluation of the influence of institutional and other factors on heat pump market penetration. The variables considered include consumer income level, traditional financing methods, building codes and consumers' and local officials' attitudes. Other factors included are energy availability and cost, real estate factors and building trends, and demographic factors as they apply to the purchase of heat pumps.

Where possible, the factors have been identified as to whether they have or would have an accelerating or retarding influence on heat pump market penetration. In some cases, factors that are presently identified as retarding influences may in the future become accelerating influences instead. While these factors are discussed separately here, it is recognized that in any particular transaction, several of the above cited factors, as well as others, may have an influence on the ultimate buying decision.

11.2 Lending Institution Factors

Whether the heat pump is purchased as part of a new home or as a replacement, the consumer will most likely finance this purchase through his mortgage or by a home improvement loan. Since a heat pump represents

an additional investment over a conventional gas or oil furnace, particularly if central air conditioning would not be purchased separately, a buyer may be forced to accept a lower first-cost option if the added down payment or monthly amortization costs exceed his economic resources. Through their mortgage or home improvement loan policies, banks and other lending institutions can directly influence the space conditioning appliance purchase decision.

The influence of lending institutions to encourage the use of energy conserving building construction methods and HVAC systems could be significant, should they exert it. Through favorable lending policies to either the builder or the buyer, and through direct assistance in free building construction planning and inspection services, a few banks have created programs that have successfully fostered the use of energy conserving devices such as heat pumps.

Some examples of favorable lending institution policies include:

- Additional construction loan to builder specifically for installing a heat pump (Cartersville Federal Savings & Loan Association, 1976).
- Lower mortgage interest rate for new housing that meets certain energy conserving criteria such as use of a heat pump for energy efficient heating (Seattle Trust & Savings Bank, 1976).
- Lower home improvement loan interest rate specifically for energy conserving improvements such as upgrading the heating system to a heat pump (Winters National Bank, 1977).
- Free planning and inspection service to enable homebuyer to lower utility cost by following energy conserving home plans which recommend a heat pump for smaller homes, and require a heat pump for larger homes (Home Federal Savings & Loan Association, 1976).

For the majority of banks and other lending institutions, no change has been found from the traditional conservative lending practices. Home buyers are qualified based on income and down payment without using

operating cost (i.e. cost of utilities) as a consideration (Lincoln Federal Savings and Loan Association, 1976).

Present conventional mortgage rates vary, depending on the down payment, from 8.5 percent for 50 percent down to 9.25 percent for 10 percent down. The average effective mortgage rate for 1976 was actually up slightly over 1975(9.06 percent versus 9.01 percent). (Federal Home Loan Bank, 1977). Home improvement loan rates vary with length of the loan with a term of 5 years having a 10.8 percent annual interest rate, while a term of 10 years has a 10.1 percent annual interest rate. A heat pump retrofit would be eligible for a home improvement loan.

Nonresidential building lending may reflect operating cost considerations to a much greater degree than is found in the residential sector. Commercial building owners now include a fuel adjustment clause in their lease, presumably to assure a return that would permit their fixed costs to be covered even if certain aspects of operating cost (i.e., cost of fuel) were to change considerably (Betz & Bischoff, 1976).

At this time, we are not aware of any special "energy conserving" lending institution programs directed toward the nonresidential building sector.

11.3 Energy Availability and Costs

The choice of a space conditioning system may not be dictated by economics alone. Recent shortages of natural gas have resulted in builders using alternative heating systems such as oil warm air furnaces, electric warm air furnaces, baseboard convectors or, for many, a heat pump (Ryan Homes, 1976).

The salient energy related factors are, on the accelerating factor side, the expected higher cost of energy of all forms, which would further enhance the prospects for energy conserving devices, and on the retarding side, the projected diminished supply of natural gas, which could adversely affect the market prospects for the gas heat pump.

The issues raised in the latter case include: gas reserves and gas well exploration, gas price deregulation, customer restrictions and curtailments, priority regulations, and reallocations of available supply. An extensive examination of these issues is beyond the scope of this report, however, a brief review of the current and projected outlook for natural gas is presented in subsection 11.3.1 below. Generally, the position that appears to be emerging is that there will be an overall reduction in future gas supply but that high priority users -- residential and commercial -- will be supplied through reallocation of the available gas, but at much higher prices.

A brief discussion of electricity and fuel oil costs is covered in the following subsection 11.3.2.

11.3.1 Natural Gas Situation

The current natural gas shortage has been developing for some period of time. For example, the reserve to production ratio for natural gas has been declining steadily from 32.5 in 1945 to 11.1 in 1974. Moreover, the actual net additions of new gas have been less that half the quantity produced, with the exception of one year, over the last seven years reported ending in 1974. Other indicators such as the productivity (in Mcf gas per foot) of successful gas well footage drilled has also generally shown a decline over the same 1969 to 1974 period (Zareski, 1976). Beginning in 1970, shortages began to develop which eventually resulted in the present restrictions in use. The effects of these developments are discussed below.

Restrictions and Curtailments

The American Gas Association (AGA) reported in their 1975 Gas Heating Survey that there were at that time extensive restrictions on the addition of new gas customers, particularly in areas of heavy population concentration. However, recent reports indicate that 341,000 new customers were added in 1976 by the gas utility industry, increasing the total homes and small commercial establishments served to 44.9 million. It is contended by the gas utilities that new customers are needed to avoid rate increases and loss of their gas allocation (New York Times, 1977).

Curtailments or demand deficiencies, were reported by the Federal Power Commission (FPC) for the 1976-77 heating season to be prevalent in most of the states. As shown in Exhibit 11-1, only a few states, mainly the less populous, have an expected curtailment below 10 percent. However, application of the curtailment appears generally to be made against those customers with the lowest priority standing. As discussed below, residential and small commercial customers are the last to be curtailed under present FPC policy.

Priorities and Reallocations

The FPC established in 1973 (Federal Power Commission, 1973) nine priority categories with residential and small commercial gas users (less than 50 Mcf on a peak day) being made the top priority. Large interruptible users, users with alternate fuel capabilities, and boiler fuel uses were assigned the lowest priorities. In establishing the priority plan, the FPC assigned residential and small commercial end uses the highest priority because interruption of service to these classes of service would require manual shut-off and reignition, with

the potential for disaster should any home or small commercial customer be overlooked or not be present for entry during the relighting process following a shut-off (Federal Power Commission, 1976).

The FPC priority plan also had conservation as its goal, with a reduction or elimination of the use of gas to fire utility boilers and other industrial uses where an alternate fuel could be employed. Since the industrial and utility end-use categories comprise over 50 percent of the total gas presently consumed, (Bureau of Mines, 1974) a reduction in their consumption could make substantial quantities available for the high priority end-uses.

A projected redistribution of gas utility sales by class of service is given in Exhibit 11-2. This projection, made by the American Gas Association, indicates that both residential and commercial service gas use is expected to increase, while industrial and other uses is expected to decline through 1990.

Gas Product Prices and Projections

The projected short-fall in domestic gas supply is expected to be met from various supplementary sources such as pipeline imports, LNG imports, and synthetic natural gas (SNG) from naphtha cracking and in the 1990's, from coal gasification (Federal Energy Administration, 1976). All of these sources, as well as others that are considered potential supply sources are expected to be substantially higher in price than the current regulated price of gas -- $0.52/Mcf for old gas and $1.42/Mcf for new gas (Bureau of Mines, 1976).

Price decontrol, to encourage greater gas well exploration, is also a strong possiblility. The effect of decontrol would be to further raise gas prices to a parity with oil prices. If gas is deregulated, it

is projected that the price of gas will be at $3/Mcf by 1980 and $3.50/Mcf by 1985 (Bureau of Mines, 1976). These well-head gas price projections are in agreement with an 8 percent energy inflation rate, which is used as the energy inflating factor in this report.

11.3.2 Electricity and Fuel Oil Situation

Electricity is expected to be available for the heating market as needed, but at higher prices. For example, the average cost of service per kWh increased 8.9 percent in 1974 and 13.4 percent in 1975 (Edison Electric Institute, 1976). An inflation rate of 8 percent per year is projected.

Home heating fuel oil (No. 2 oil) is expected to be available for the heating market, with possibly some local shortage situations. The market share of new homes equipped with an oil furnace during the 1971-1975 period was 9 percent, with most of the installations in the Northeast and North Central regions (Census Bureau, 1975). The projected market share of oil-fired equipment in new residential construction is expected to average 10 percent during the time frame of this report.

Fuel oil prices, as with the other energy forms, are also expected to inflate by 8 percent per year. A median price increase of 15.3 percent was found compared to a similar survey taken mid-1975 (Gordian Associates, 1975).

11.4 Consumer Attitudes On Heat Pumps and Energy Conservation

Heat pump equipment marketed in the 1950's and early 1960's had serious performance and reliability problems. As a counter-measure, a

Heat Pump Improvement Research Project was initiated in 1963 under the sponsorship of the Edison Electric Institute. The primary objective of the program was to improve the reliability and performance of heat pump equipment.

In addition to equipment shortcomings, there were also substantial problems associated with installation and servicing. The net result of this combined breakdown in equipment performance and poor service delivery led to a general rejection of the heat pump by utilities, HVAC contractors and the ultimate consumer. It was not until 1970 that the industry could report that new design and improved performance equipment had been successfully developed and tested (ASHRAE, 1970).

This early lack of adequate equipment reliability and poor service left a negative consumer attitude legacy that has only recently begun to change as a result of the pressures of energy shortages and higher energy prices. Recent surveys indicate that the consumer is receptive to buying an unconventional HVAC applicance such as a heat pump, if there are energy savings sufficient to justify the added investment.

While this positive attitude appears to be shared by residential and nonresidential building owners alike, it is derived along somewhat different lines of thought. The residential building owner, as discussed in subsection 11.4.1 below, does not view his purchase of a home as business with the need to minimize operating expenses, and he frequently has capital limitations. Neither is he generally a key buying influence, but rather one who is being influenced by the builder, contractor, utility company and the like.

In contrast the nonresidential building owner is much more cost conscious, even to the extent that economic analyses may be conducted for a variety of HVAC systems during the architectual planning stage, with system selection based on such factors as the builders assessment of payout and expected length of building ownership (Levin and Associates, 1976). The attitudes of nonresidential building owners is discussed in subsection 11.4.2.

11.4.1 Residential Building Purchasers

The general progression in formulating a buying decision is: need recognition, alternatives evaluation,and purchase action. The need recognition is generally apparent (e.g. you need to replace your heating or HVAC system or you are buying a new house) and the purchase action is the final outcome to satisfy the need, but the alternatives evaluation is the critical hurdle. To stand a chance at being selected, the system must be one that would be an acceptable alternative to the purchaser. The attitudinal and key buying influences that exert a force upon potential heat pump consumers are discussed below.

Residential Consumer Attitudes

A 1974 survey of recent heat pump owners found a high degree (83 percent) of customer satisfaction with their heat pump. A major factor in their purchase decision was the expected lower operating and maintenance costs of a heat pump compared to a gas furnace or electric resistance furnace (Arkansas Power & Light Co., 1974).

Surveys reported in 1975 and 1976 found that 78 and 89 percent respectively of active* new home buyers would be willing to spend at

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* Active buyer is one who has announced a serious intention to buy a new home in near future.

least $600 more in marginal investment if they could save $100 per year in energy costs. In addition, 87 percent of active home buyers surveyed in 1976 thought that a heat pump was a good or excellent idea. Similarly, 86 percent of potential buyers* in the prime home buying 25 to 44 age group viewed the heat pump as a good or excellent idea (Professional Builder, 1975, 1976). The apparent willingness of active new home buyers to pay more for energy savings -- with a simple 6 year payout -- and the positive attitude of present heat pump owners and of active home buyers toward heat pumps suggest that the heat pump may no longer need view itself in the past negative light. Furthermore, the consumer associates the heat pump with energy conservation, which provides additional reinforcement for the consumers positive attitude towards the heat pump.

Residential Key Buying Influence

Residential consumer heat pump purchases are influenced by third party authoritative sources such as: builders, developers, architects, utility representatives, engineers and HVAC contractors (Electric Comfort Conditioning News, 1975). Over 60 percent of heat pump buyers surveyed by Arkansas Power & Light Co. in 1974 first learned of the heat pump from one of these sources, and approximately the same percentage discussed the purchase with one of these key buying influences before making a decision to buy a heat pump.

Builders, developers and utilities(where they have a marketing program) appear to be responding to the energy conservation challenge with policies and programs designed to foster heat pump usage. For example: Ryan Homes expected to install heat pumps in over half the 6000 homes they were building in 1976 (Ryan Homes, 1976); Lowe's Companies

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* A "potential" buyer is one who would be a candidate from the general market, but does not presently intend to buy.

has made available a redesigned energy conserving home package, with sales to builders double the previous year (Lowe's Companies, 1976) and Mississippi Power and Light Co. (1977) has a heat pump promotional program (Easy on Energy) which has exceeded target sales objectives by 200 percent.

Generally, the findings are that third party key buying influences are both important and active in promoting the heat pump to the residential consumer. However, the previous subsection on consumer attitudes does also suggest that future home buyers will have greater awareness for the need to cut energy costs, which may create or induce a demand for heat pumps in addition to the acceleration given by the key buying influences discussed.

11.4.2 Nonresidential Building Purchasers

HVAC purchase decisions for nonresidential buildings primarily reflect the owners objective for short or long-term investment. If the owner expects to sell the building within a year or two after completion, first costs will be uppermost. Conversely, low operating cost becomes the more important criterion for a building owner who expects to continue ownership over a longer period. In general, the nonresidential consumer attitudes and key buying influences reflect a better informed and more sophisticated population that the residential building owner population.

Nonresidential Consumer Attitudes

A survey of nonresidential building owners (corporations) found a substantial majority believed that: heat pumps are practical and economical, a roof top gas heat pump would be considered for both new and replacement use, and life-cycle costing is usually employed in the decision making

process. The nonresidential building owner would most likely be willing to pay a premium of up to 20 percent for an energy conserving HVAC system, where the payback would be less than three years (Garrett Airesearch, 1976).

Nonresidential Key Buying Influences

The building architect or HVAC consulting engineer has the greatest influence on the nonresidential consumer's HVAC buying decision. The HVAC engineer typically conducts studies to show the payout of various HVAC system alternatives. A two-year payout is always acceptable while a five-year payout is acceptable only if the building owner expects to retain control for that period of time. Longer payout periods are not considered acceptable (Levin and Associates, 1976).

One further aspect of the architect's or engineer's influence is their control over the selection of alternatives, which may be exhibited by a preference for a decentralized heat pump system over for example, a central HVAC system. This bias would be less of a problem where the heat pump installation would be a conversion in an existing building. If the building is ducted, a central system would be a logical choice. If the building is not ducted, individual heat pump units which can be connected to room air diffusers with flexible connections would appear to be a better choice since it would avoid the requirement of installing large runs of ductwork.

11.5 Building Code and Government Regulatory Factors

Most communities, and certainly all cities of even modest size, have either a national or a local building code in effect. The code

frequently relies on standards developed by various testing organizations such as the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) who, for example, have developed standards for heat pumps. The codes are administered by local officials, who normally would not have any control over HVAC equipment standards or certifications. While this system would tend to protect the public from unsafe construction, building codes tend to change slowly, and in some instances may not reflect the latest developments, or may impose unnecessarily severe requirements (Hittman Associates, 1976). Thus, while well intentioned, the codes may act as a barrier to the use of a heat pump in some jurisdictions, especially with regard to advanced electric and gas heat pumps.

A brief discussion of the various model and national codes, as they apply to both residential and nonresidential buildings, is given in subsection 11.5.1.

The high internal working pressures of a gas heat pump may result in restrictive pressure vessel requirements being applied as prescribed by the ASME boiler code. This aspect is considered in subsection 11.5.2.

In one state, California, regulations have been promulgated that would appear to restrict the use of electric heat pumps in nonresidential buildings because of their auxiliary strip heat component. This is discussed in subsection 11.5.3.

11.5.1 Building Code Factors

The four model building codes recognized by the Departments of Housing and Urban Development, and Health, Education and Welfare are Building Officials and Code Administrators (BOCA), International Conference of Building Officials (ICBO), National Board of Fire Underwriters,

and Southern Building Code Congress International. These codes do not have separate heat pump standards, but rely on various related sections of the code such as those governing mechanical refrigeration, electric strip-heat and combustible furnace requirements. (BOCA, 1976; ICBO, 1976, National, 1976; Southern, 1976). All model codes, and in fact about 98 percent of all jurisdictions in the United States, use the National Electric Code (NEC) with regard to electrical requirements. The NEC requires the heat pump circuit to be rated at 125 percent of the current capacity for any continuous demand of 3 hours or more -- and 100 percent for a non-continuous demand. Included in this requirement would be the current demand for motors, compressor and all auxiliaries, and strip heat (National Fire Protection Association, 1976).

Although this electrical service requirement may result in some additional investment, it is not believed likely that service beyond 200A would be required (Town of Westfield, 1976). Cost estimates for electrical service from Means 1976 Building Construction Cost Data Book were given at $200 for 100 A service and $450 for 200 A service, hence the marginal investment for added electrical service would be $250 for an electric heat pump installation.

Separate code sections or supplements for heat pumps based on the American Society of Heating, Refrigeration and Air-Conditioning Engineers Standard 90-75 are now being developed by each of the model code organizations. Coefficient of performance (COP) requirements for heat pumps as provided by ASHRAE 90-75 are given below for the heating mode.

Minimum COP Requirement

Outdoor Temperature, ^{0}F	By 1/1/77	By 1/1/80
47 dry bulb, 43 wet bulb	2.2	2.5
17 dry bulb, 15 wet bulb	1.2	1.5

The ICBO is under contract to put ASHRAE 90-75 into code language.

The code requirements for a gas heat pump are not expected to be greatly different than for the electric heat pump. For the ICBO, the Uniform Mechanical Code (published jointly by ICBO and the International Association of Plumbing and Mechanical Officials) provides that local building officials can accept any alternate system if it can be shown to provide equivalent performance and equivalent safety to health and welfare.

The ASHRAE 90-75 standard not only appears to be the leading influence on the model codes, it is also the basis for the new Minimum Property Standards (MPS) to be established by HUD as a result of the Energy Conservation and Production Act of 1976 (P.L. 94-385) (Housing and Urban Development, 1976).

Local building codes frequently are patterned after one or more of the model codes. Some major jurisdictions such as New York City, Chicago, and San Francisco have their own codes. Smaller localities may have their own code, but would of necessity need to follow a model code since the locality would lack the resources of a large city in developing its own code modifications. Also, an increasing number of states are going to a single model code. New Jersey, in adopting the BOCA code statewide as of January, 1977, is the most recent example of this trend.

Finally, where local code requirements for heat pumps are found to be considerably more stringent than provided for in the model codes, legal action may be employed to have the unnecessary barrier removed. This would be of course reserved only for major market jurisdictions.

11.5.2 Potential Restrictions Due to High Working Pressures

Several of the gas heat pump configurations are reported to have high internal working pressures. As indicated in the table below, three of the systems currently under consideration have internal pressures greater than 100 psig, and one -- the Stirling-Rankine system -- is reported to exceed 1000 psig.

Appliance Configuration	Maximum Pressure psig	Maximum Temperature °F
Stirling-Rankine	1600	1200
Rankine-Rankine	285	1100
Brayton-Rankine	Sub-atm.	1500
Gas Absorption	approx. 100	approx. 280

Source: Colosimo, 1976.

The American Society of Mechanical Engineers (ASME) Boiler and Pressure Code provides that any fired boiler with pressures over 15 psig is subject to the ASME design code which among other things prescribes preferred materials of construction and design. However, the Code also states that not all possible details of design or construction are covered, but it is intended that the manufacturer shall provide for design and construction which will be as safe as otherwise required by the Code.

The position of at least one gas heat pump developer is that they will meet or exceed the Code, and that based on preliminary discussions with ASME and State officials, flexible application of the Code by the various state and local jurisdictions is expected. For example, they do not believe that the licensed stationary engineer requirement for fired boilers over 6 Hp would be enforced for systems such as a gas heat pump since it would not be reasonable application of the Code.

Also, there is precedent for the acceptance of high pressure devices without requiring a stationary engineer. For example, the pressure in the refrigeration loop in an electric heat pump or air conditioner may be as high as 450 psig, and an auto-steam-cleaner boiler may operate at pressures up to 300 psig, yet no special licensed engineer requirements are imposed on the homeowner or the car wash facility. Similarly, it is believed the gas heat pump will not be classed as an operating boiler, although the heat pump manufacturer will have to establish that the appliance's construction meets or exceeds the ASME code.

11.5.3 California Energy Code

The State of California recently promulgated certain regulations that, whether or not intended, do have a restrictive effect on electric heat pumps for nonresidential buildings. Under Title 24:6 Section T 20-1504, electric resistance heating cannot exceed 10 percent of the building space heating requirement. The strip heat of a heat pump would come under the 10 percent restriction.

It is conceded by the state's energy office that the effect of this rule would be to require the heat pump compressor to be oversized in order to minimize the strip heat necessary to provide auxiliary heating when the temperature falls below the balance point. However, the rule, scheduled to go into effect in February 1977, has not been enforced and may be changed. The intent was not to make heat pumps less desirable, but by forcing purchase of oversized equipment, it is expected that fewer heat pumps will be sold (California Energy Resources Conservation and Development Commission, 1976).

11.6 Real Estate Factors and Building Trends

In this section, real estate and building trend factors are considered as to their effect on heat pump market penetration. Installation of a heat pump may enhance or detract from the saleability of a building, depending on such factors as location and property taxes. This is discussed in subsection 11.6.1.

In subsection 11.6.2, the possible effect of current residential building trends are considered from the view-point of their accelerating or retarding potential to the market penetration of heat pumps. Nonresidential building trends are discussed briefly in subsection 11.6.3.

11.6.1 Real Estate Factors

Prior studies (Gordian Associates, 1976) had indicated that consumers who were unfamiliar with the heat pump would likely be wary of buying it. More recent information tends to reinforce this position to the extent that many people look upon the electric heat pump as "electric heat" and therefore as costly as electric resistance heat. Where heat pumps are generally not known to the consumer, such as in the Northeast, sale of real estate with a heat pump may be more difficult and may perhaps even sell at some discount (Betz & Bischoff, 1976; Newman, 1976).

However, where heat pumps are fairly well known, such as in the South, the available information would suggest a result the opposite of that found in the Northeast (Arkansas Power & Light Co., 1976). Other evidence supports this finding that where the consumer has been made aware of the heat pump through various educational and marketing programs, his receptivity has been very good, both as to the purchase of a heat pump and to maintenance of real estate values in housing containing a heat pump (Professional Builder, 1976).

A second real-estate-related factor is the added cost of purchasing a heat pump, and the likely added property tax because of a higher building valuation. To further assess this factor, a letter of inquiry was sent to six states that had recently announced passage or were in the process of passing legislation providing for property tax reductions for energy conservation heating systems (New York Times, 1976).

Replies from four of the states, Connecticut, Hawaii, Michigan and Vermont, all indicate that the tax relief is directed toward solar, wind and water energy conservation systems and does not include heat pumps (Connecticut, 1976; Hawaii, 1976; Michigan, 1976; Vermont, 1976). Based on these findings, the local property tax system would have a retarding effect on the use of heat pumps, at least to the extent that the added annual outlay would lengthen the payback period.

11.6.2 Residential Building Trends

Some of the important residential building trends and their effect on heat pump market penetration are:

High Consumer Preference for a Single Family Dwelling. New single family residential building construction is expected to comprise about 70 per cent of the current total housing market, an increase from the 55 percent share of new housing built during the 1960's and early 1970's. This is expected to occur due to the large number of Post-World-War-II babies that are now entering the prime 35 to 45 age category for single family home ownership, and the fact that over 60 percent of new housing is purchased by a previous homeowner who is trading on the equity in his present home to upgrade his housing (Smith Barney, 1976).

The influence of this trend would be to favor a larger overall market for heat pumps, with perhaps greater emphasis on the smaller size units due to new energy-conserving construction.

Emergence of Energy Efficient Housing

Various new construction techniques (e.g. 2 x 6 inch stud sidewalls, 24 inches on center) and the much greater use of insulating materials has enabled a reduction of up to two-thirds in the heat loss/heat gain for new single-family residential buildings compared to pre-1977 HUD MPS housing (Lowe's Companies, 1976). Others are obtaining from one-third to one-half heat loss/heat gain reduction in new construction by using more insulation and careful caulking alone, without altering the basic shell construction (Ryan Homes, 1976).

While the average new home size has been holding its own or perhaps even slightly increasing to 1638 ft^2 in 1976 (National Association of Home Builders, 1976), the size of heat pump necessary to adequately comfort condition the new "energy efficient home" will not need to be as large as in the past. Demand for heat pumps in the 1 and 1-1/2 ton nominal cooling-capacity sizes is expected to increase. Even now, there is some unfulfilled demand for units in these sizes (Lowe's Companies, 1976).

Resurgence of the Modular Home

The modular home is a home built in a factory using conventional construction and then delivered to a home site with prepared foundation. Unlike the mobile home, modular housing qualifies for conventional 30

year mortgage financing. The appeal of modular housing is its lower cost -- under $25,000 for a 1200 ft^2 conventional built home complete on a lot with foundation (Precision Built Homes, 1976). Mobile home prices, while they have been increasing rapidly, are on the average about half the price of the modular home (Manufactured Housing Institute, 1976).

The higher price and more permanent nature of the modular home makes it a prime candidate for the electric heat pump. One modular home manufacturer advised their problem was being unable to buy heat pumps direct from the factory, but having to deal with the local franchised contractors and thereby pay a higher price (Guerdon Industries, 1976).

The total market for manufactured housing is expected to average 250,000 units per year through 1980, with modular housing contributing up to 60,000 units per year during the same period. The present modular market is about 5 to 10 percent of the manufactured housing market.

11.6.3 Nonresidential Building Trends

Major potential heat pump user categories in the nonresidential building sector are office buildings, stores and institutional buildings, including schools and hospitals.

For the 1971-1977 period, office building construction has been declining in relative share of total construction, while stores and institutional construction has shown a slight increase. Present indications from construction industry experts are that they expect a leveling for 1977 in new nonresidential construction, with perhaps a slight 2 to 3 percent increase in office buildings and stores, and an even holding on

institutional building new construction. On a regional basis, the current share of new nonresidential building construction is 33 percent South, 27 percent West, 25 percent North Central and 15 percent Northeast.

As with residential homes, the concentration of nonresidential building in the South and West is an accelerating factor with respect to heat pump usage since electricity is the major heating fuel and the incidence of air conditioning would be expected to be high.

11.7 Consumer Income and Demographic Factors

In this section, consumer income and demographics are briefly considered with regard to their possible influence on the purchase of a heat pump for residential buildings. Consumer income and housing purchase correlations are discussed in 11.7.1, and consumer demographic factors are considered in 11.7.2.

11.7.1 Consumer Income Related Factors

As shown below, there is a long term correlation between income and housing unit value.

Housing Unit Value Compared to 1974 Median Income

Value of Housing Unit, Dollars	Median Income (1974) Dollars
< 5,000	4,000
5 - 10,000	6,100
10 - 15,000	8,700
15 - 20,000	10,700
20 - 25,000	12,200
25 - 35,000	14,500
> 35,000	20,300

Source: U.S. Department of Commerce: Annual Housing Survey: 1974

However, an even more important factor than income itself is the monthly or annual payment for housing compared to the income. As noted

in subsection 11.2.2, home buyers are qualified on the basis of income and down payment. Data reported for 1974 new home purchases show that the more expensive the home, the greater the down payment.

1974 Housing Sales Price (Dollars)			Percent Down Payment
	<	25,000	2.7
25	-	30,000	7.7
30	-	35,000	10.2
35	-	40,000	12.5
40	-	50,000	18.1
	>	50,000	22.8

Source: Census Bureau, Construction Report, 1974

These data further support the position that a substantial portion of the new home buying population is upgrading by trading on their equity from a previously owned residence. Only 13 percent of the new houses sold in 1974 were under $25,000, while 52 percent were in the $35,000 and over category, indicating that the consumer is not buying at the low end of the new housing market but is looking for more and better housing. For example, central air conditioning was installed in 58 percent of the new homes that sold for more than $50,000 in 1974, while those selling for less than $25,000 had a 38 percent installation rate (Census Bureau, 1976).

In terms of heat pump market influence, there has been a belief that the heat pump, being a higher first cost item, would be sold mainly to buyers of the more expensive housing. With the average new home costing $48,600 in 1976 (Federal Home Loan Bank, 1976), it would appear that the housing market has shifted to a position where the heat pump could be considered a candidate for the average new residential building.

11.7.2 Demographic Factors on Consumer Heat Pump Purchases

The heat pump buyer has been found, in one survey, to be better educated, have a higher than average income, hold a managerial, administrative, or professional job, be home-oriented and value-conscious. Age was not found to be a factor. The average buyer planned to live in his house for a least 8 years, while only 4 percent planned to live at their present location 5 years or less (Arkansas Power & Light Co., 1974).

This latter finding may be compared with the sale of existing single-family homes which was reported for 1976 to be at the annual rate of 3.3 million homes (National Association of Realtors, 1976). Based on a total of 52.5 million single-family homes, the turnover rate appears to be 6.3 percent for existing housing or 9 percent for new and existing housing combined. At 9 percent, the average length of single-family home ownership at one location would be about eleven years.

11.8 Summary

Examination of various institutional factors related to the market acceptance of heat pumps indicates that there are no insurmountable barriers to the use of advanced electric or gas heat pumps.

Consumer attitude toward the use of a heat pump is favorable because it has the potential for savings in lower operating cost. Greater promotion of heat pumps by the key buying influences such as utilities and HVAC contractors appears to be rapidly expanding the present heat pump market. Both residential and nonresidential building owners are willing to make an additional investment in an energy saving appliance

such as a heat pump, but the added investment should be recoverable from savings in lower operating cost. In particular, nonresidential buyers seek a fairly short term payout of 2 years, which could inhibit heat pump growth in that market.

There are no property tax advantages in using a heat pump, nor is there likely to be. Nevertheless, trends toward increased use of heat pumps can be identified in the upgrading of housing generally, the increased saturation of central air conditioning and the high cost of energy.

The market potential of the gas heat pump is clouded, to some extent, by present and projected shortages in natural gas. This situation may not be a major barrier to the market prospects if reallocation of the available supply to high priority use is achieved as projected by the American Gas Association.

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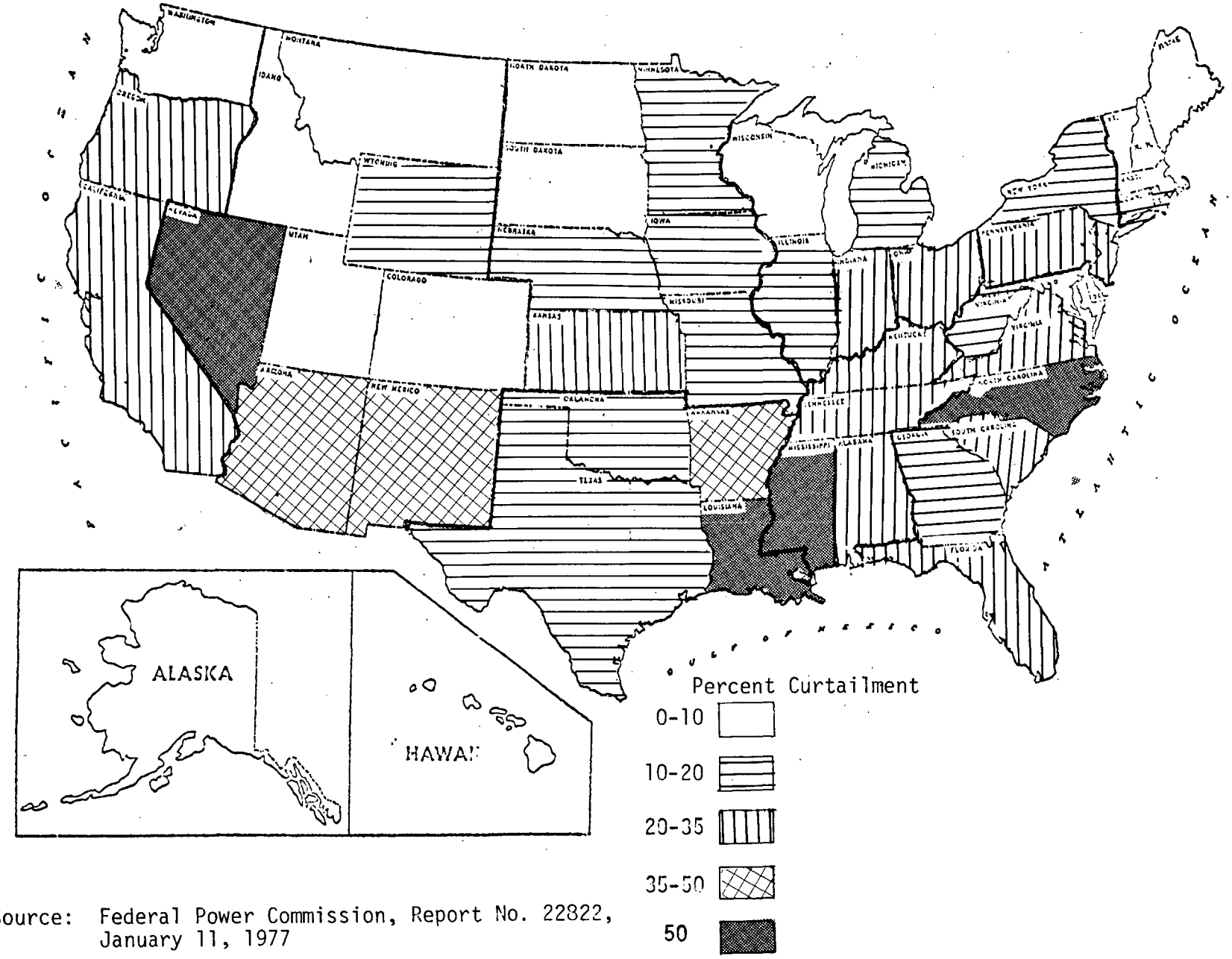
EXHIBIT 11-1

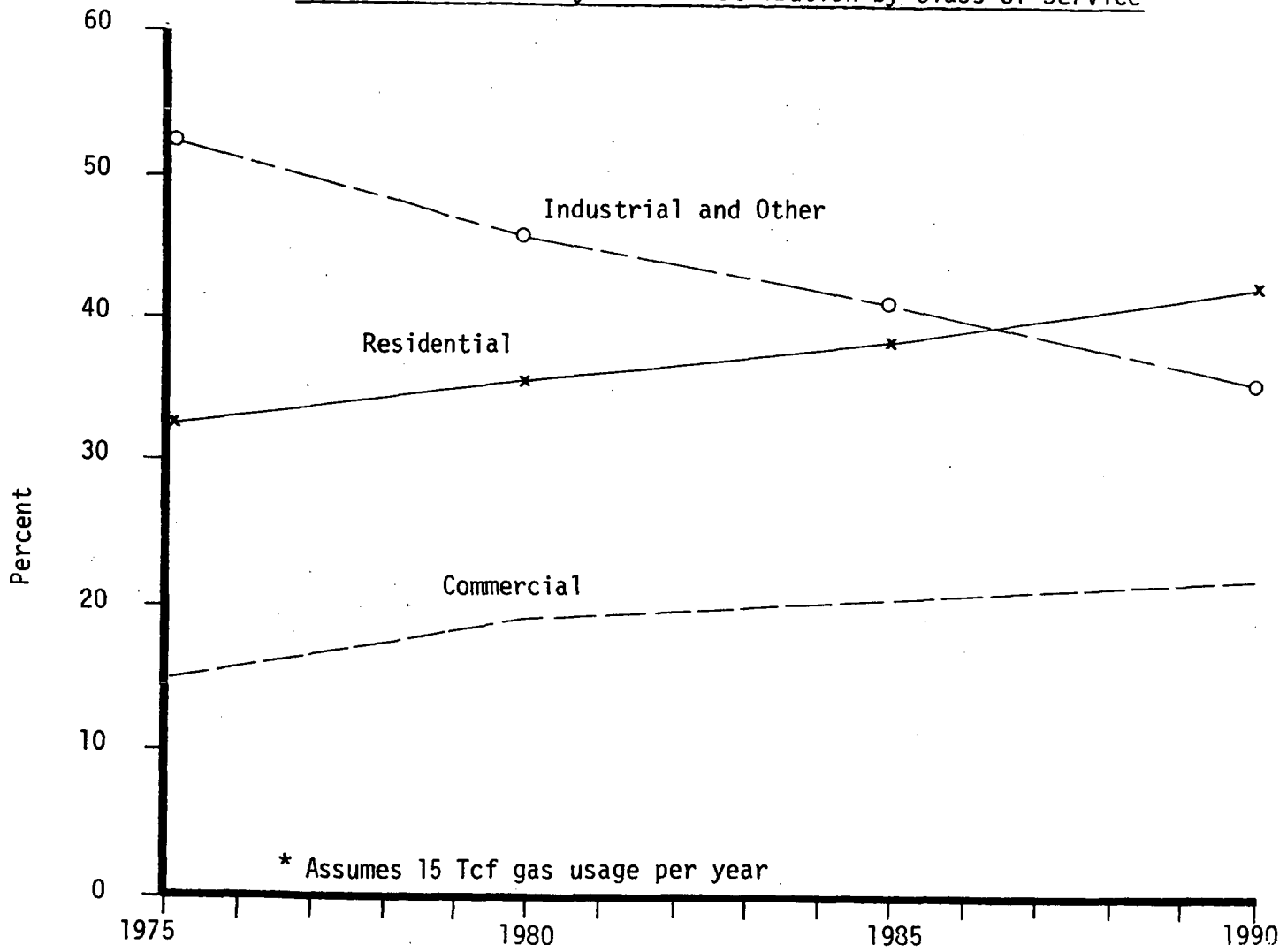
Natural Gas Curtailments By State - 1976/1977



Source: Federal Power Commission, Report No. 22822, January 11, 1977

EXHIBIT 11-2

Projected Gas Utility Sales Distribution by Class of Service*



Source: American Gas Association, Gas Utility Projections to 1990, September 1976

12. PRIMARY ENERGY UTILIZATION BY HEAT PUMPS

12.1 Introduction

In this chapter, a discussion of the impact of alternative space conditioning systems on primary fuel resources is presented. Although a reduction of total energy consumed on-site for space conditioning is desirable, this is not an end in itself. Rather, the effect of increased heat pump market saturation on the types of fuels consumed to produce the energy for on-site use should be examined. This is particularly important in light of the increasing scarcity of such primary fuels as oil and natural gas.

The sections below address this issue in two ways. First, primary fuel consumption by space conditioning systems is compared for each of the nine test cities. In a following section, the effect of increased heat pump market saturation on the nation's primary fuel resources is discussed on a regional basis.

12.2 Primary Energy Consumption by Individual HVAC Systems in the Nine Test Cities

12.2.1 Methodology

The energy consumption data presented in Exhibits 8-14 and 8-15 represent the equivalent energy values of the fuels and electric energy consumed on site for residential and commercial space conditioning. In this section, the basis of comparison is broadened to include the energy input to produce the energy consumed on-site. Using on-site energy consumption data presented in Chapter 8, primary energy consumption data were developed in the following manner:

Electric energy: average state plant heat rates were used to convert on-site consumption to resource energy consumption by the generating station; these heat rates are shown below for each city.

City	Plant Heat Rate (BTU/kWh)
Houston	10,304
Birmingham	10,194
Atlanta	10,696
Tulsa	10,283
Philadelphia	10,249
Seattle	9,717
Columbus	10,187
Cleveland	10,187
Concord	10,874

Source: Edison Electric Institute, 1975

In addition, transmission and distribution losses of 9% were assumed for electric energy.

Natural Gas: heating value of natural gas was assumed to be constant throughout the nation at 1,000 BTU/cft; 8% transmission and distribution losses were assumed.

Fuel Oil: heating value taken as 139,000 BTU/gal. No distribution losses were assumed.

From the primary energy consumption data, primary energy seasonal performance factors were computed for residential and commercial HVAC equipment. These SPF's were defined as the ratio of total seasonal heating or cooling load satisfied by the HVAC system to the total primary energy consumption. The primary energy consumption and SPF's are presented in sections 12.2.2 and 12.2.3 below for residential and commercial HVAC systems, respectively.

12.2.2 Residential Systems

The results of the primary energy comparisons on an individual system basis are shown in Exhibits 12-1 through 12-4. The SPF data appear

in Exhibits 12-1 and 12-2, for the conventional and advanced systems, respectively.

As expected, electric heat pumps were found to be more efficient than any of the all-electric HVAC systems examined. High efficiency residential electric heat pumps had performance factors in the 0.78-0.43 range for heating and the 0.61-0.82 range for cooling, constituting an average 9% improvement over the standard heat pump representing 1975 technology. In primary energy terms, electric heat pumps were comparable to or slightly less efficient than the combustion heating systems combined with electric cooling in regions with a significant heating load. In very cold regions, the combustion systems were more efficient. However, combustion system efficiencies were calculated based on insulated supply and return air ductwork. This is the observed practice with electric but not with combustion heating systems in general. Calculations showed that the performance of the combustion furnaces was considerably improved with duct insulation. With uninsulated ducts, electric heat pumps would probably have a slightly higher efficiency than the combustion systems.

Gas heat pumps now under development have a higher efficiency in a primary energy sense than electric heat pumps, combustion furnaces or resistance heating systems. Of the residential gas heat pumps examined, the gas-fired free-piston and the V-type single cylinder Stirling-Rankine heat pump were found to have performance factors in the 1.16-0.91 range for heating and in the 0.68-0.88 range for cooling. The corresponding results for the organic fluid absorption heat pump were lower: 0.95-0.87 for heating and 0.48-0.46 for cooling.

Total annual energy consumption shown in Exhibits 12-3 and 12-4 for residential space conditioning, is smallest for the combination of fossil fuel fired furnaces and high efficiency central air conditioners in colder parts of the country and is greatest for electric air conditioners. These results parallel heating season energy consumptions observations because of the relatively small impact of air conditioning requirements compared with heating season energy needs in colder climates. The hybrid heat pump, in colder regions, consumes about up to 15% more total primary energy annually than fossil fuel fired furnace/high efficiency air conditioners while the best high efficiency electric heat pump uses up to 20% more.

On an annual basis, the free piston Stirling-Rankine gas heat pump appears to consume the smallest amount of primary energy compared with the other heat activated heat pump systems. In the warmer cities, the pulse combustion gas furnace/high efficiency air conditioner combination is only slightly more energy consuming than the free piston Stirling-Rankine gas heat pump. The relative primary energy consumption of the combined system seems to increase in the colder cities, however, where the V-type single cylinder Stirling-Rankine gas heat pump can be ranked second to the free piston system. The advanced electric heat pump I also appears to consume more primary energy compared to other advanced systems as climatological conditions call for more heating in the colder cities. This behavior is experienced more dramatically with the advanced electric heat pump II, which is the most primary energy efficient system in the warm ciites (Houston, Birmingham, etc), but is the least efficient unit in the coldest city (Concord).

12.2.3 Commercial Systems

Commercial HVAC system primary energy SPF data are presented in Exhibit 12-5. The results show that the Rankine-Rankine and Brayton-Rankine heat pumps have a higher primary energy efficiency (performance factors 1.43-0.76 heating; 0.91-0.72, cooling) than electric heat pumps or rooftop air conditioners with gas or electric resistance heat. Primary energy consumption data, presented in Exhibit 12-6, clearly show that the new technology advanced unitary heat pump under development for light commercial building applications will consume less annual primary energy than any of the conventtional heating and cooling systems examined. The Rankine-Rankine gas heat pump performs better than the Brayton-Rankine gas heat pump in all the cities exmained. Among the conventional heating and cooling systems, the standard electric heat pumps uses the least energy in all climates compared with the single package rooftop gas furnace and electric resiatance heating combined with electric air conditioning. Generally, the Rankine-Rankine gas heat pump uses about 25-46% less energy for commercial building comfort conditioning than the standard electric heat pump in all cities examined. The larger savings are associated with colder climates.

12.3 Regional Primary Fuels and Energy Consumed by HVAC Systems

12.3.1 Methodology

The data presented in sections 12.3.2 and 12.3.3 below represent primary fuels and energy consumed on a regional basis for the operation of individual HVAC equipment and for estimated residential and commercial market penetration. These results were calculated in the following manner:

In order to estimate the quantities of fuel consumed by the various systems on a regional basis, energy consumption data specific to the nine test cities had to be extrapolated to the nine U.S. census regions. Since each census region may span several climatic or degree day belts, each region had to be characterized with appropriate weighted average indices which could be correlated against cooling and heating season on-site energy consumption of the HVAC system investigated.

The correlating parameters used were:

Heating: $$\frac{\text{Design Heating Load} \times \text{Heating Degree Days}}{\text{Design Temperature Difference}}$$

Cooling: $$\frac{\text{Design Cooling Load} \times \text{Cooling Degree Days}}{\text{Design Temperature Difference}}$$

A good correlation was found when heating and cooling energy consumption in each of the nine test cities was plotted against the correlating parameter for each residential and commercial HVAC systems studied. These plots were then used to determine the appropriate energy consumption of a given HVAC system, using values of the heating or cooling correlating parameters for each state in the contiguous United States by way of an intermediate correlation.

The intermediate correlation was one between the heating or cooling correlating parameter and statewide heating or cooling degree days. The procedure for determination of statewide average heating degree days is given elsewhere (Gordian Associates, 1976); calculations of cooling degree days was made in the same fashion.

The state-by-state data were then reduced to regional data by computing a weighted average consumption for each census region. This

average was determined by weighting the state-by-state results with the number of single family detached dwellings in each state for the appropriate census region:

$$E_r = \frac{\Sigma(E_s \times D_s)}{\Sigma \; D_s}$$

where E_r = regional average on-site energy consumption
E_s = state average on-site energy consumption
D_s = number of single family detached dwellings in state

This procedure was used for both residential and commercial HVAC systems. It was deemed acceptable for light commercial buildings, since the number of such buildings in each state would tend to reflect the population density of the state, and hence, the number of single family detached dwellings.

Where electric energy was used in heating or cooling, the regional values for on-site energy consumption developed above were translated into primary fuel consumption in the following manner. The fuel mix for electric utilities was determined regionally from published data compiled monthly by the Federal Power Commission(1976), shown in Exhibit E-1 and E-2 in Appendix E. Use of these data for heating and cooling months, in conjunction with plant heat rates and fuel heating values, compiled by the Edison Electric Institute (1976), enabled the determination of pounds of coal, gallons of fuel oil and cubic feet of natural gas burned at the generating plant to supply one kilowatt-hour of electric energy for on-site consumption. Fractions representing kilowatt hours of generation by nuclear and hydroelectric plants per kilowatt hour consumed on-site were also determined. Amounts of natural gas consumed on-site were increased by 8% to account for utility transmission and distribution

losses. Appendix E contains computer printouts showing primary fuels and electric energy consumed in each of nine census regions, for 23 different HVAC systems.

These data, developed per square foot of residential living space or commercial building floor area, used in conjunction with market penetration data (described in Chapter 10), enabled the analysis of the impact on primary fuel resources of increased system market saturation in the census regions.

The primary fuel data for each of the system on the nine-region basis were reweighted in terms of four major census regions, Northeast, North Central, South and West. Specific systems were compared to produce data (on a per square foot basis) exhibiting primary fuels and other energy sources which would be saved and/or consumed upon the substitution of one HVAC system for another. In conjunction with the market penetration data, which represents the number of square feet of building space expected to be conditioned by a particular system, projected through the year 2000, savings and or consumption of primary fuels and energy for alternative combinations of systems were calculated on 5 year intervals from 1985 through 2000.

12.3.2 Residential Systems

The information collected in Appendix E can be used to directly examine the differences in primary fuel usage for different HVAC systems on the same basis. It is appropriate to scrutinize the primary fuel data developed in this form prior to examining the impact on primary fuels for the U.S. due to market penetrations of the different types of heating and cooling systems.

A comparison of two different heating and cooling systems exclusively using the same type of on-site energy should reveal savings of all types of primary fuels for the more efficient systems. Thus, a comparison of primary fuels used to provide energy for a high efficiency heat pump using only electric energy shows a reduction in the quantities of each type compared with the standard electric heat pump (compare Appendix E, Exhibits E-3 and E-4 for example). Similarly a comparison of the primary fuels used to provide energy for the gas warm air furnace/high efficiency central air conditioner and the pulse combustion furnace/high efficiency central air conditioner reveals a significant decrease in primary gas usage for the pulse combustion system (Exhibits E-10 and E-19). Almost identical quantities of all other primary fuels are used for the two different systems. The reason for this, of course, is that electric energy usage by the two systems (to operate circulating fans and for air conditioning) are almost identical, whereas different quantities of gas are required by the different furnaces for heating.

While these are almost trivial and obvious comparisons, they do seem to suggest that efficiency and primary fuel usage are directly related, which is not at all correct. One example is to compare a gas-fired advanced heat pump (such as the free piston Stirling-Rankine gas heat pump, Exhibit E-16) with a conventional electric heat pump (such as a high efficiency all electric heat pump II, Exhibit E-5). Annual performance factors for the gas heat pump (Exhibits 12-1 and 12-2) are greater than those for the electric gas heat pump in all test cities, suggesting that it uses less total primary energy to space condition one square foot of residence on an annual basis. An estimation of primary fuels consumed reveals

that less of each type is used for the advanced gas heat pump than for the electric heat pump with the single exception of natural gas. For natural gas, all census regions show greater usage by the advanced gas heat pump than for the electric HVAC system. The primary fuel saving is due to the difference between the large amount of electric energy used by the electric heat pump and the small amount of electric energy for auxiliaries used by the advanced gas heat pump. The primary fuels that would have been used to generate this quantity of electric energy are not used. However, the amount of primary natural gas used by the advanced gas heat pump far exceeds the gas saved to generate electric energy and results in a net increase in gas consumption. The single exception involves the West South Central census region where natural gas is the principal fuel for power generation. In this case, the highly efficient Stirling-Rankine heat pump consumes less gas than the electric heat pump. For the nation as a whole, however, natural gas is minimally used to generate electricity and a net increase in gas consumption can be expected upon comparing primary fuels for the Stirling-Rankine heat pump with an all electric heat pump.

With this introduction, we compare below selected advanced and conventional heating and cooling systems (and between different conventional systems, in one case) indicating the impact on primary fuel usage of each based on the data compiled in Appendix E. Estimated first is the difference in primary fuel usage upon replacing a single conventional HVAC system with an advanaced (or different conventional) system. These are then aggregated for four regions of the U.S. based on estimated new starts for residential and commercial building and estimated penetration in these markets by different HVAC systems. These raw data are tabulated in Appendix F.

Advanced Electric Heat Pump II and Conventional Electric Heat Pump - Standard

Exhibits E-3 and E-15 indicate that the advanced electric heat pump will conserve all types of primary fuels in all census regions when compared with a standard electric heat pump. This is a direct consequence of the higher efficiency of the advanced electric heat pump compared with the type of conventional electric heat pump mainly in residential service today. Savings of coal and oil vary from 50-60% and gas savings vary from 55 to 70% on an annual basis. Of particular importance are the annual savings of critical fuels, fuel oil and natural gas, in all regions of the country. In some regions which use small quantities of natural gas for power generation the savings will be small in terms of cubic feet of gas conserved. However, the West North Central, Mountain and especially in the West South Central region where relatively large amounts of gas are used the total quantities saved could be quite significant.

The total annual savings are given in Exhibit 12-7 for the entire U.S. Natural gas and fuel oil savings by 1985 are projected to be less than one billion cubic feet (104,000 BFOE)* and 76,190 barrels, respectively. By the year 2000, these savings increase to 24 billion cubic feet of gas (3.9 million BFOE) and 2.9 million barrels of fuel oil.

Free Piston Stirling-Rankine Gas Heat Pump and a Conventional Electric Heat Pump - High Efficiency II

Because the conventional electric heat pump uses only electric energy and the gas heat pump uses electricity only to operate parasitic equipment, large savings of fuels used to produce electric energy are to be expected. Exhibits E-5 and E-16 show that approximately 20% of

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* One BFOE (barrel of fuel oil equivalent) is equal to 6.05 million Btu, assuming low sulfur fuel oil at about 20°API.

primary fuel quantities used by the conventional electric heat pumps are required by the Stirling-Rankine gas heat pump to condition one square foot of residence. The pattern of gas consumption, though, is quite different. In all but one of the nine census regions, natural gas usage is greater for the advanced gas heat pump than for the conventional electric heat pump. The largest increase is in the New England area where about 264 times more natural gas is required by the gas heat pump than by a high efficiency electric heat pump. The smallest increase is in the Mountain area where only 5 times more gas would be used by the gas heat pump. The difference in gas usage increases is due to the difference in fuel mix used to generate electric power, more gas being used in New England than in the Mountain states. A slight decrease in gas usage is observed for the West South Central region because of the large dependence on gas for power production in this area. For the nation as a whole, a net increase would be expected, and for gas-poor regions, such as the New England area, the increase in gas usage demanded by the Stirling-Rankine gas heat pump could be particularly serious. These observations are borne out by estimations of the impact on primary fuel quantities assuming a displacement of conventional electric heat pumps by the Stirling-Rankine gas heat pump in new single family residences for the period 1985-2000.

Exhibit 12-7 shows that as much as 555 million pounds of coal (1.2 million BFOE) and 255,000 barrels of oil could be saved by 1985 (assuming high estimates of building starts and market penetrations are met). By the year 2000, savings of coal would increase to 6700 million pounds of coal (14.5 million BFOE) and 3 million barrels of oil. However,

during this same time period, natural gas consumption would increase by 7 billion cubic feet (1.1 million BFOE) in 1985 and by 82 million cubic feet (13.5 million BFOE) by the year 2000.

Free Piston Stirling-Rankine Heat Pump Replacing a Gas Warm Air Furnace/ Conventional High Efficiency Central Air Conditioner

Replacing a gas warm air furnace and a high efficiency central air conditioner with a free piston Stirling-Rankine gas heat pump will, in general, reduce net annual consumption of each type of primary fuel resource, as shown in Exhibits E-10 and E-16. The gas heat pump was previously shown to have a higher heating season performance than a gas warm air furnace, but this is strictly true for gas usage,not for electric energy. The auxiliary electrical demand of this gas heat pump (air circulating fan, heat recovery system pumps and combustion blower) is greater than the parasitic electric demand of the gas warm air furnace (air circulating fan only), but the gas heat pump uses less gas for heating than the gas furnace.

Exhibit 12-7 shows that indeed savings of all types of primary energy may be expected. From high estimates of residential building starts and market penetration of gas heat pumps, gas and primary fuels saved on a national basis are shown. These data are reported below, adding BFOE equivalent saved as a measure of energy conserved.

	Coal Saved		Oil Saved		Gas Saved	
	lbs x 10^{-6}	BFOE x 10^{-6}	gal x 10^{-6}	BFOE x 10^{-6}	cft x 10^{-9}	BFOE x 10^{-6}
1985	100	0.2	2	0.4	3	0.6
1990	352	0.8	8	0.1	11	2.0
1995	729	1.6	17	0.5	22	3.7
2000	1213	2.7	28	0.7	37	6.1

Organic Fluid Absorption Gas Heat Pump Replacing a Gas Warm Air Furnace/ Conventional High Efficiency Central Air Conditioner

The organic fluid absorption gas heat pump is not as efficient as the free piston Stirling-Rankine gas heat pump. The absorption heat pump heating season performance factor is about 13% less than the Stirling-Rankine gas heat pump and about 33% less for the cooling season performance factor. One might, therefore, generally expect less savings of primary fuels when an absorption gas heat pump replaces a gas warm air furnace/high efficiency conventional central air conditioner than if the conventional system is replaced by a Stirling-Rankine gas heat pump. Exhibits E-10, E-16 and E-18 confirm this prediction although with some important exceptions.

During the heating season, the absorption gas heat pump uses more coal and oil, but less gas, than a conventional gas warm air furnace uses. This is identical to a comparison of the free-piston Stirling-Rankine gas heat pump with a gas warm air furnace. However, the savings in gas energy for the absorption gas heat pump is less, than that for the Stirling-Rankine gas heat pump. The reason for this is, as noted above, the lower heating season SPF for the absorption system than for the Stirling-Rankine system. However, although the absorption gas heat pump uses more coal and oil than the gas warm air furnace, the Stirling-Rankine gas heat pump uses even more. The explanation for this difference is that the demand for electric energy by the absorption gas heat pump is less than that for the Stirling-Rankine system (48% less) but only slightly greater than the electrical demand by the circulating fan of the gas warm air furnace.

During the cooling season, the absorption gas heat pump reduces the usage of all primary fuels except for natural gas, the consumption of which increases. The reason is the same as that offered above for the Stirling-Rankine gas heat pump -- electricity required by the absorption gas heat pump is

less than for a central electric air conditioner. More natural gas is used by the absorption heat pump than the central air conditioner because this primary fuel is burned on-site to drive the system prime mover. Compared to the Stirling-Rankine gas heat pump, however, more gas is consumed by the absorption heat pump, again because the absorption gas heat pump is not as efficient for cooling as the Stirling-Rankine gas heat pump. Also, more primary fuels are saved, other than natural gas, during the cooling season by the absorption system than the Stirling-Rankine system because of the smaller electrical demand by auxiliaries.

The net result on an annual basis is a reduction of all primary fuels, except for natural gas, when an absorption gas heat pump is used in place of a gas warm air furnace/central air conditioner. These savings are greater for the absorption heat pump than for the Stirling-Rankine heat pump because of the argument advanced above in terms of relative electrical requirements for parasitics for the two systems. The net result for natural gas usage annually is not clear cut. In four census regions, representing on the average mild summer weather conditions, the absorption gas heat pump uses less natural gas than a gas warm furnace/central air conditioner but the amount conserved is less than the more efficient Stirling-Rankine system. In five census regions representing quite warm summer conditions, the absorption heat pump uses more natural gas than a gas warm air furnace/central air conditioner, whereas the more efficient Stirling-Rankine system conserved natural gas in these regions. The net result estimated for the entire country, as

shown in Exhibit 12-7, is that all primary fuels, except for natural gas, are conserved, and to a greater extent than conserved by the Stirling-Rankine gas heat pump. Natural gas usage, however, is seen to be greater on a national basis for advanced absorption gas heat pumps replacing gas warm air furnaces/central air conditions. This result could not have been anticipated by comparing performance factors for the absorption heat pump and the gas warm air furnace/central air conditioner. Exhibits 12-1 and 12-2 show that the absorption heat pump annual performance factor is 20-40% better than for the conventional system. Indeed, there is a total net savings in primary fuels. But in terms of the mix of primary fuels used in this country, natural gas would not be conserved. The short table below, derived from Exhibit 12-7, emphasizes this point.

	Coal Saved		Oil Saved		Gas Saved	
	$lbx10^{-6}$	$BFOEx10^{-6}$	$galx10^{-6}$	$BFOEx10^{-6}$	$cftx10^{-9}$	$BFOEx10^{-6}$
1985	133	0.3	3	0.1	-1	0.1
1990	471	1.0	10	0.3	-2	0.3
1995	965	2.1	21	0.5	-4	0.6
2000	1604	3.5	36	0.8	-7	1.2

The interesting results observed for these systems illustrates the importance of the role played by auxiliary electric energy demands in determining the impact on primary fuels of advanced heat pumps and the need to examine the impact of system performance on individual primary fuels, particularly critical fuels.

Smaller savings of natural gas and other fuel resources are to be expected for the Stirling-Rankine system and greater consumption of natural gas for the absorption gas heat pump if these advanced systems replace gas warm air furnaces and high efficiency central air conditioners. This is due to the fact that the comparisons have been made including dynamic efficiency losses for the conventional system but not including these effects for the advanced systems. Using the approproximate factors shown in Section 8.2.5 to adjust energy usage by the advanced systems and then comparing with conventional systems would results in modest corrections to predicted fuel resource impacts, but would not substantially change the results described.

Pulse Combustion Furnace Replacing a Gas Warm Air Furnace

This comparison is essentially an examination of heating season performance. Both advanced and conventional furnaces are assumed to use central electric air conditioners of equal efficiency and therefore would use equal quantities of primary fuels for summer comfort conditioning. The analysis of the impact of these gas heated furnaces is rather simple, partly based on the manner of estimating the pulse combustion system energy requirements. Simply stated, electric energy consumption by auxiliaries were assumed slightly higher for the pulse combustion furnace, and gas consumption for the pulse combustion furnace decreased, based upon a 92.5% on-site combustion efficiency compared with an average of about 60% for the gas warm air furnace. Under these assumed conditions, there would be small differences in primary fuel consumption, except for natural gas, upon replacing a conventional gas furnace with a pulse combustion system. This prediction is borne out by examining Exhibits E-10 and E-19.

The savings of natural gas, however, for this assumed replacement are quite significant compared with other comparisons made up to this point. By 1985 about 125 billion cubic feet of gas (21 million BFOE) could possibly be saved annually by replacing gas warm air furnaces with pulse combustion furnaces, as shown in Exhibit 12-7. This saving is estimated to rise to about 498 billion cubic feet (82 million BFOE) by the year 2000.

Hybrid Electric Heat Pump/Gas Furnace Replacing a Conventional Standard Electric Heat Pump

This comparison is between two currently available conventional HVAC systems. Comparing Exhibits E-3 and E-7 shows that annual residential usage of natural gas would increase significantly in all regions, except for the West South Central area. This is to be expected because of the natural gas consumed on-site by the gas furnace of the hybrid system below the balance point, whereas the conventional electric heat pump uses electric energy for the heat pump compressor and supplemental electric resistance heating. The small increase in the West South Central region is again due to the fact that electric energy is predominantly generated by natural gas in that census region.

Exhibits E-3 and E-7 also show that all other primary fuels are conserved for winter heating as well as annually when a hybrid electric heat pump/gas furnace is used, compared with a standard electric heat pump. These savings are due solely to the reduction in electric energy used by the hybrid system which replaces electric energy below the balance point with natural gas burned by the gas furnace. These savings could possibly be larger than those shown except for the fact that the

hybrid system modelled uses an electric heat pump that is not as efficient as a standard electric heat pump (see Exhibit 8-7). Primary fuel savings other than natural gas would increase if a high efficiency electric heat pump were used as part of the hybrid system. This is also the reason for the increased usage of primary fuels for summer cooling by the hybrid system compared with a standard electric heat pump. The small differences shown are real and due simply to the lower efficiency of the electric heat pump used as part of the hybrid system compared with the standard electric heat pump.

Exhibit 12-7 shows the annual impact on primary fuel usage assuming the hybrid electric heat pump selectively replaces standard electric heat pumps. Natural gas consumption is seen to increase by 30 billion cubic feet (4.9 million BFOE) while fuel oil decreases by 548,000 barrels in 1985. By the year 2000, natural gas consumption would increase by 92 billion cubic feet (18 million BFOE) and fuel oil decrease by 1.7 million barrels.

One potentially interesting advanced system not included in this study would be a hybrid combination of an electrically driven high efficiency advanced heat pump with a pulse combustion furnace. Such a system would reduce primary fuels used to produce electric energy compared with conventional electric heat pumps, and reduce natural gas consumption burned on-site compared with a gas warm air furnace.

12.3.3 Commercial

Two comparisons were made of the impact on primary fuel resources by advanced heat pumps intended for light commercial applications. The first comparison assumes that a Rankine-Rankine gas heat pump is used in place of a combined single-package conventional gas furnace/electric air conditioner. The second assumes that the same advanced gas heat pump replaces a unitary standard electric heat pump.

As might be anticipated, replacing conventional HVAC systems that largely use electric energy with a gas fired advanced heat pump does not conserve natural gas. The greatest impact is caused by the Rankine-Rankine heat pump substituting for a conventional electric heat pump. Estimates of the square footage of light commercial buildings in place by 1985 and high and low estimates of market penetration, result in a forecast of increased natural gas use by 1985 of 420 million cubic feet (69,000 BFOE) as a low estimate and 850 million cubic feet (141,000 BFOE) as a high estimate. By the year 2000, additional natural gas consumption would be anticipated; a total increase to 10.4 billion feet (1.7 million BFOE) as a low estimate and about 25.6 billion cubic feet (4.2 million BFOE) as a high estimate. All other primary fuel resource usage decreases. For example, fuel oil saved could amount to 55,000 barrels of fuel oil in 1985 and 1.6 million barrels in the year 2000 (high estimate).

Exhibits E-22 and E-25 indicate that the Rankine-Rankine gas heat pump uses much less natural gas than the rooftop gas furnace/electric air conditioner for heating; however, cooling sees a much higher natural gas consumption by the gas heat pump. On an annual basis, the Rankine-Rankine gas heat pump uses less natural gas in the cooler regions.

As Exhibits F-7a shows, natural gas is conserved, on an annual basis, in all regions except the South, where cooling is the primary HVAC service. For the total U.S., a small net saving in natural gas would be realized by utilization of the gas heat pump instead of the rooftop gas furnace/electric air conditioner. Gas savings of 24.6 million cubic feet (4,000 BFOE) as a low estimate and 52.9 million cubic feet (8,700 BFOE) as a high estimate, could be realized by 1985. Forecasting to the year 2000, 650 million cubic feet (107,000 BFOE) as a low estimate and 1,598 million cubic feet (264,000 BFOE) as a high estimate, could be realized.

12.4 Summary

The greatest consumers of primary energy, on a total thermal equivalent basis, are electric resistance heating systems, represented by the central electric furnace and baseboard convectors. The smallest amount of total primary energy is consumed by fossil fuel warm air furnaces in mild winter areas of the country and by conventional electric heat pumps in colder winter areas. Conventional electric air conditioners are generally not as energy efficient with respect to primary energy usage as high efficiency electric heat pumps for air cooling.

The Stirling-Rankine gas heat pump uses the least amount of total primary energy for both heating and cooling in all cities examined and is matched by a highly efficient advanced electric heat pump only in mild winter regions. The organic fluid absorption gas heat pump is not as efficient as the Stirling-Rankine gas heat pump.

The regional analysis was performed by selectively examining the impact on primary fuels of replacing conventional HVAC systems with advanced

heating and cooling systems currently being developed. It was generally found that advanced systems replacing conventional systems using essentially the same type of on-site energy will generally conserve all types of primary fuel resources. Thus, an advanced electric heat pump replacing a conventional electric heat pump will conserve all forms of primary fuel resources. The quantities of primary fuels that could potentially be saved depend strongly on the relative efficiencies of the advanced and conventional system being examined and the region of the country considered. Local conditions determine fuel mix used to generate electric energy and the severity of winter and summer weather to which heat pumps are sensitive.

Advanced gas heat pumps examined were found to have potential for reducing all types of primary fuels, except natural gas, when replacing conventional electric heat pumps. However, this result is dependent upon the extent to which electrical energy is required for gas heat pump auxiliaries. Natural gas consumption is always found to increase for this replacement strategy. In replacing less efficient gas equipment, advanced gas heat pumps were found to have a potential for conserving all primary fuels, including natural gas, but this result must be carefully qualified. Because heat pump performance is generally not as efficient for cooling as for heating, only very high efficiency gas heat pumps will be acceptable in warmer regions of the country where cooling season energy is a major part of annual system energy requirements. For the two different generic gas heat pump systems examined here, the organic fluid absorption gas heat pump and gas heat pumps based on Stirling-Rankine cycle, only the Stirling-Rankine system appears to have potential for conserving natural gas in all regions of the country. The absorption

gas heat pump would conserve natural gas (and other primary fuels) in colder regions of the country, but not in warmer regions where its cooling efficiency is too low. In any case, the conservation potential for natural gas is greater for the Stirling-Rankine gas heat pumps than for the absorption gas heat pump in all parts of the country.

The hybrid heat pump with fossil-fuel-fired furnace appears to have potential for conserving all primary fuel resources except natural gas or fuel oil compared with currently available conventional electric heat pumps.

The pulse combustion furnace appears to have the greatest potential for reducing natural gas use for residential heating of all the gas fired advanced systems examined. Advanced electric heat pumps would probably reduce natural gas consumption even more dramatically and certainly to a greater extent than is estimated for advanced gas heat pumps.

The same general findings for residential HVAC systems may be applied to the light commercial HVAC systems. When compared to the all electric HVAC systems, gas heat pumps conserve all primary fuel resources except natural gas. When compared to the rooftop gas furnace/electric air conditioner system, gas heat pumps conserve all forms of primary fuel resources for heating, but consume more natural gas for cooling. Hence, in those regions where the primary HVAC service is for cooling, a net increase, on an annual basis, in natural gas consumption is observed.

References to Chapter 12

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EXHIBIT 12-1

CONVENTIONAL ELECTRIC HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF SEASONAL PERFORMANCE FACTORS ON A PRIMARY ENERGY BASIS

Heating	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	0.65 (0.72)*	0.59 (0.66)	0.58 (0.64)	0.51 (0.55)	0.57 (0.63)	0.59 (0.67)	0.52 (0.57)	0.52 (0.56)	0.41 (0.43)
Electric Heat Pump - High Efficiency I	0.78 (0.86)	0.69 (0.76)	0.68 (0.75)	0.59 (0.64)	0.66 (0.72)	0.65 (0.74)	0.58 (0.64)	0.57 (0.63)	0.43 (0.47)
Electric Heat Pump - High Efficiency II	0.73 (0.81)	0.67 (0.74)	0.61 (0.68)	0.60 (0.65)	0.64 (0.70)	0.68 (0.76)	0.58 (0.63)	0.57 (0.62)	0.48 (0.51)
Electric Heat Pump - High Efficiency III	0.69 (0.76)	0.62 (0.69)	0.61 (0.66)	0.52 (0.57)	0.60 (0.65)	0.67 (0.75)	0.54 (0.59)	0.53 (0.58)	0.46 (0.49)
Hybrid Electric Heat Pump-Gas Furnace	0.55 (0.61)	0.52 (0.57)	0.51 (0.55)	0.55 (0.60)	0.52 (0.56)	0.52 (0.55)	0.52 (0.55)	0.52 (0.55)	0.50 (0.53)
Hybrid Electric Heat Pump-Oil Furnace	0.55 (0.61)	0.53 (0.57)	0.51 (0.56)	0.56 (0.61)	0.52 (0.57)	0.53 (0.55)	0.53 (0.57)	0.54 (0.57)	0.52 (0.55)
Electric Heat Pump - Heat Only	0.78 (0.87)	0.72 (0.79)	0.69 (0.75)	0.65 (0.71)	0.75 (0.77)	0.75 (0.84)	0.63 (0.69)	0.62 (0.68)	0.51 (0.56)
Gas Warm Air Furnace	0.53	0.53	0.52	0.54	0.54	0.55	0.55	0.56	0.55
Oil Warm Air Furnace	0.59	0.59	0.59	0.61	0.60	0.61	0.61	0.62	0.62
Central Electric Furnace	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.28
Baseboard Convectors	0.30	0.31	0.29	0.30	0.31	0.32	0.31	0.31	0.29
Cooling									
Electric Heat Pump - Standard	0.60 (0.69)	0.60 (0.69)	0.57 (0.67)	0.58 (0.69)	0.60 (0.71)	0.67 (0.75)	0.61 (0.75)	0.61 (0.72)	0.57 (0.66)
Electric Heat Pump - High Efficiency I	0.70 (0.80)	0.70 (0.81)	0.67 (0.78)	0.70 (0.83)	0.71 (0.83)	0.82 (0.92)	0.71 (0.83)	0.73 (0.85)	0.69 (0.79)
Electric Heat Pump - High Efficiency II	0.65 (0.76)	0.65 (0.77)	0.61 (0.71)	0.66 (0.78)	0.66 (0.79)	0.73 (0.83)	0.66 (0.79)	0.67 (0.80)	0.61 (0.72)
Electric Heat Pump - High Efficiency III	0.69 (0.79)	0.68 (0.80)	0.66 (0.77)	0.58 (0.69)	0.69 (0.81)	0.79 (0.90)	0.69 (0.81)	0.70 (0.83)	0.71 (0.82)
Hybrid Electric Heat Pump	0.56 (0.65)	0.56 (0.66)	0.54 (0.64)	0.55 (0.67)	0.56 (0.67)	0.62 (0.72)	0.57 (0.67)	0.57 (0.68)	0.53 (0.64)
Central Air Conditioner - High Efficiency	0.66 (0.76)	0.66 (0.77)	0.63 (0.74)	0.60 (0.71)	0.67 (0.79)	0.75 (0.84)	0.67 (0.78)	0.68 (0.80)	0.64 (0.75)
Central Air Conditioner - Standard	0.55 (0.64)	0.55 (0.64)	0.53 (0.62)	0.57 (0.67)	0.56 (0.66)	0.63 (0.71)	0.56 (0.65)	0.57 (0.67)	0.57 (0.66)
Window Air Conditioners	0.67	0.71	0.67	0.69	0.70	0.68	0.71	0.70	0.67
Annual Heating and Cooling									
Electric Heat Pump - Standard	0.61 (0.69)	0.59 (0.68)	0.57 (0.65)	0.54 (0.60)	0.58 (0.65)	0.60 (0.68)	0.54 (0.59)	0.53 (0.58)	0.42 (0.45)
Electric Heat Pump - High Efficiency I	0.71 (0.81)	0.70 (0.79)	0.67 (0.77)	0.62 (0.70)	0.67 (0.75)	0.67 (0.75)	0.61 (0.67)	0.60 (0.66)	0.45 (0.49)
Electric Heat Pump - High Efficiency II	0.66 (0.77)	0.66 (0.75)	0.61 (0.70)	0.62 (0.69)	0.65 (0.72)	0.69 (0.77)	0.60 (0.66)	0.59 (0.65)	0.49 (0.53)
Electric Heat Pump - High Efficiency III	0.69 (0.78)	0.66 (0.75)	0.63 (0.72)	0.54 (0.61)	0.62 (0.69)	0.68 (0.77)	0.57 (0.63)	0.56 (0.61)	0.48 (0.51)
Hybrid Electric Heat Pump-Gas Furnace	0.56 (0.64)	0.54 (0.62)	0.52 (0.60)	0.55 (0.62)	0.53 (0.59)	0.53 (0.57)	0.53 (0.58)	0.53 (0.57)	0.50 (0.54)
Hybrid Electric Heat Pump-Oil Furnace	0.56 (0.65)	0.55 (0.62)	0.52 (0.60)	0.56 (0.63)	0.52 (0.59)	0.54 (0.57)	0.54 (0.59)	0.55 (0.58)	0.52 (0.56)
Gas Warm Air Furnace/Central Air Conditioner-High Efficiency	0.60	0.56	0.55	0.55	0.55	0.56	0.56	0.57	0.55
Oil Warm Air Furnace/Central Air Conditioner-High Efficiency	0.65	0.63	0.61	0.61	0.62	0.62	0.62	0.63	0.62
Central Electric Furnace/Central Air Conditioner-High Efficiency	0.54	0.44	0.41	0.37	0.36	0.35	0.35	0.33	0.30
Baseboard Convectors/Window Air Conditioners	0.49	0.41	0.38	0.35	0.34	0.34	0.34	0.33	0.30
Electric Heat Pump-Heat Only/Central Air Conditioner-High Efficiency	0.68 (0.78)	0.68 (0.78)	0.66 (0.74)	0.63 (0.71)	0.73 (0.78)	0.75 (0.84)	0.64 (0.71)	0.63 (0.70)	0.52 (0.57)

* Numbers in parenthese are on-site energy consumptions based on steady state or full load performance, i.e no dynamic efficiency losses are included.

EXHIBIT 12- 2

ADVANCED HEATING AND COOLING SYSTEMS: COMPARISON OF SEASONAL PERFORMANCE FACTORS ON A PRIMARY ENERGY BASIS

Residential Space Conditioning

HEATING	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Advanced Electric Heat Pump I	**	0.76	0.73	0.72	0.74	**	0.72	0.72	**
Advanced Electric Heat Pump II	1.04	0.86	0.86	0.69	0.81	0.96	0.74	0.73	0.58
Free Piston Stirling-Rankine Gas Heat Pump	1.10	1.07	1.06	1.04	1.05	1.09	1.03	1.03	0.98
V-Type Single Cylinder Stirling-Rankine Gas HP	1.16	1.11	1.10	1.01	1.10	1.11	1.07	1.07	0.91
Organic Fluid Absorption Gas Heat Pump	0.95	0.92	0.91	0.90	0.91	0.94	0.90	0.90	0.87
Pulse Combustion Furnace	0.78	0.79	0.79	0.81	0.81	0.82	0.82	0.82	0.82
COOLING									
Advanced Electric Heat Pump I	**	0.74	0.71	0.74	0.75	**	0.75	0.76	**
Advanced Electric Heat Pump II	1.00	1.01	0.96	0.98	1.02	1.15	1.02	1.04	0.97
Free Piston Stirling-Rankine Gas Heat Pump	0.72	0.69	0.69	0.71	0.70	0.71	0.69	0.71	0.68
V-Type Single Cylinder Stirling-Rankine Gas HP	0.81	0.80	0.81	0.73	0.84	0.88	0.82	0.85	0.82
Organic Fluid Absorption Gas Heat Pump	0.48	0.46	0.47	0.48	0.46	0.47	0.46	0.47	0.46
ANNUAL HEATING AND COOLING									
Advanced Electric Heat Pump I	**	0.75	0.72	0.73	0.74	**	0.73	0.73	**
Advanced Electric Heat Pump II	1.01	0.95	0.91	0.78	0.86	0.98	0.80	0.77	0.61
Free Piston Stirling-Rankine Gas Heat Pump	0.77	0.81	0.82	0.89	0.92	1.02	0.92	0.95	0.93
V-Type Single Cylinder Stirling-Rankine Gas HP	0.85	0.89	0.90	0.87	0.99	1.06	0.98	1.01	0.90
Organic Fluid Absorption Gas Heat Pump	0.53	0.59	0.60	0.68	0.71	0.83	0.73	0.78	0.80
Pulse Combustion Furnace/Central Air Conditioner-High Efficiency	0.67	0.71	0.70	0.72	0.76	0.81	0.78	0.79	0.80

* The energies presented may be underestimated for the advanced electric heat pump I, free piston Stirling-Rankine gas heat pump and the organic fluid absorption gas heat pump due to possible underestimation of cycling and defrost losses. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, then adjusted estimates of energies which include these effects can be obtained by multiplying the values in this table by a factor of 1.231 for the heating season, 1.181 for the cooling season and 1.252 for annual energy values.

** System is not sized for application in these cities.

EXHIBIT 12-3

CONVENTIONAL ELECTRIC HEAT PUMPS AND OTHER SYSTEMS: COMPARISON OF TOTAL FUEL AND ELECTRIC ENERGY CONSUMPTION ON A PRIMARY ENERGY BASIS

(Btu x 10-6)

Heating	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	15.75 (14.16)*	42.08 (37.94)	44.17 (40.08)	88.30 (81.29)	85.05 (77.73)	72.67 (64.62)	114.79 (105.35)	137.25 (126.16)	199.09 (186.61)
Electric Heat Pump - High Efficiency I	13.17 (11.86)	36.11 (32.67)	37.72 (34.21)	76.84 (70.66)	73.90 (67.56)	66.07 (58.24)	102.29 (93.66)	123.00 (112.66)	185.06 (173.18)
Electric Heat Pump - High Efficiency II	14.09 (12.64)	37.55 (33.42)	41.53 (37.67)	75.56 (69.48)	75.97 (69.43)	63.36 (56.49)	103.08 (94.53)	123.39 (113.31)	168.61 (157.71)
Electric Heat Pump - High Efficiency III	14.95 (13.44)	40.04 (36.19)	42.12 (38.29)	86.16 (79.24)	81.37 (74.44)	64.62 (57.53)	110.69 (101.56)	132.46 (121.73)	176.50 (165.26)
Hybrid Electric Heat Pump-Gas Furnace	18.67 (16.82)	47.96 (43.91)	50.41 (46.12)	81.43 (75.35)	94.04 (86.91)	83.22 (78.75)	114.58 (107.73)	135.02 (127.83)	160.16 (152.15)
Hybrid Electric Heat Pump-Oil Furnace	18.63 (16.79)	47.58 (43.63)	50.15 (45.90)	80.22 (74.31)	92.95 (85.92)	81.80 (78.30)	111.77 (105.40)	131.34 (124.95)	155.41 (147.96)
Electric Heat Pump - Heat Only	13.16 (11.74)	34.87 (31.46)	37.14 (33.80)	69.51 (63.82)	69.00 (63.07)	57.67 (51.42)	94.48 (86.34)	113.50 (103.87)	154.42 (144.50)
Gas Warm Air Furnace	19.58	47.09	48.25	83.78	89.69	78.46	107.94	12[illegible].53	144.60
Oil Warm Air Furnace	17.29	41.97	43.03	74.05	80.62	70.51	97.40	11[illegible].55	130.95
Central Electric Furnace	34.69	82.68	88.64	149.54	161.47	136.69	196.28	23[illegible].35	284.62
Baseboard Convectors	33.40	81.06	86.75	147.30	159.11	133.76	193.41	22[illegible].94	250.46
Cooling									
Electric Heat Pump - Standard	81.04 (70.46)	55.87 (47.79)	54.80 (46.65)	43.41 (36.55)	33.58 (28.47)	10.39 (9.27)	31.10 (26.60)	23.91 (20.33)	16.20 (14.04)
Electric Heat Pump - High Efficiency I	66.61 (58.16)	47.07 (40.63)	46.39 (39.72)	35.40 (29.91)	28.23 (24.07)	8.49 (7.62)	26.11 (22.46)	19.8[illegible] (17.01)	13.37 (11.59)
Electric Heat Pump - High Efficiency II	73.41 (69.98)	50.69 (42.98)	51.08 (44.04)	38.66 (32.85)	30.45 (25.51)	9.59 (8.42)	28.24 (23.85)	21.6[illegible] (18.14)	14.96 (12.76)
Electric Heat Pump - High Efficiency III	70.61 (61.53)	48.55 (41.77)	47.80 (40.78)	43.60 (36.74)	29.29 (24.89)	8.86 (7.83)	27.11 (23.24)	20.8[illegible] (17.77)	13.01 (11.16)
Hybrid Electric Heat Pump	85.02 (73.08)	58.59 (49.73)	57.82 (48.71)	45.10 (37.32)	35.42 (29.62)	11.10 (9.68)	32.82 (27.76)	25.2[illegible] (21.27)	16.96 (14.11)
Central Air Conditioner - High Efficiency	73.12 (63.62)	50.27 (43.18)	49.46 (42.14)	42.46 (35.81)	30.29 (25.70)	9.42 (8.32)	28.06 (24.01)	21.5[illegible] (18.34)	14.04 (12.06)
Central Air Conditioner - Standard	87.20 (76.02)	60.06 (51.70)	59.13 (50.48)	44.66 (37.68)	36.23 (30.80)	11.00 (9.87)	33.53 (28.75)	25.7[illegible] (21.98)	16.27 (14.03)
Window Air Conditioners	38.21	26.51	26.68	21.40	17.63	8.14	16.73	1[illegible].84	9.53
Annual Heating and Cooling									
Electric Heat Pump - Standard	96.79 (84.62)	97.75 (85.73)	98.97 (86.73)	131.71 (117.84)	118.63 (106.21)	83.06 (73.89)	145.89 (131.95)	161.2[illegible] (146.50)	215.28 (200.65)
Electric Heat Pump - High Efficiency I	79.78 (70.02)	83.18 (73.30)	84.11 (73.93)	112.24 (100.51)	102.13 (91.63)	74.56 (65.86)	128.40 (116.12)	142.8[illegible] (129.67)	198.43 (184.78)
Electric Heat Pump - High Efficiency II	87.50 (75.62)	88.23 (76.91)	92.61 (81.72)	114.23 (102.33)	106.40 (94.94)	72.95 (64.91)	131.32 (118.38)	144.9[illegible] (131.45)	183.58 (170.47)
Electric Heat Pump - High Efficiency III	85.56 (74.96)	88.60 (77.96)	89.92 (79.07)	129.76 (115.97)	110.66 (99.33)	73.48 (65.36)	137.80 (124.80)	153.3 (139.50)	189.51 (176.43)
Hybrid Electric Heat Pump-Gas Furnace	103.70 (89.90)	106.55 (93.64)	108.23 (94.83)	126.53 (112.67)	129.46 (116.60)	94.31 (88.43)	147.40 (135.50)	160.3 (149.09)	177.13 (166.27)
Hybrid Electric Heat Pump-Oil Furnace	103.65 (89.86)	106.17 (93.36)	107.97 (94.61)	125.32 (111.63)	128.37 (115.61)	92.90 (87.98)	144.59 (133.16)	156.6 (146.22)	172.37 (162.07)
Gas Warm Air Furnace/Central Air Conditioner - High Efficiency	92.70	97.36	97.71	126.24	119.98	87.88	136.00	8.08	158.64
Oil Warm Air Furnace/Central Air Conditioner - High Efficiency	90.41	92.23	92.48	116.51	110.92	79.93	125.46	5.10	144.99
Central Electric Furnace/Central Air Conditioner - High Efficiency	107.81	133.05	138.10	192.00	191.76	146.11	224.34	3.90	298.66
Baseboard Convectors/Window Air Conditioners	71.61	107.57	113.43	168.70	176.74	141.90	210.14	3.78	259.99
Electric Heat Pump-Heat Only/Central Air Conditioner - High Efficiency	86.28 (75.41)	85.14 (74.64)	86.60 (75.94)	111.97 (99.63)	99.29 (88.77)	67.09 (59.74)	122.54 (110.35)	135.[illegible] (122.21)	168.46 (156.56)

* Numbers in parenthese are on-site energy consumptions based on steady state or full load performance i.e., no dynamic efficiency losses are included.

EXHIBIT 12- 4

ADVANCED HEATING AND COOLING SYSTEMS: COMPARISON OF TOTAL FUEL AND ELECTRIC ENERGY CONSUMPTION ON A PRIMARY ENERGY BASIS

(Btu x 10^{-6})

HEATING	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Advanced Electric Heat Pump I	**	33.08	35.15	61.52	65.87	**	82.07	97.57	**
Advanced Electric Heat Pump II	8.11	22.16	23.33	49.97	45.91	36.45	61.24	73.26	108.05
Free Piston Stirling-Rankine Gas Heat Pump	9.32	23.47	24.17	43.30	46.43	39.43	58.05	68.97	83.05
V-Type Single Cylinder Stirling-Rankine Gas HP	8.83	22.51	23.12	44.46	44.51	38.77	55.67	66.02	87.19
Organic Fluid Absorption Gas Heat Pump	10.84	27.12	27.79	49.81	53.44	45.97	65.44	78.91	93.21
Pulse Combustion Furnace	13.09	31.34	32.21	55.97	60.37	53.39	73.04	86.34	98.31
COOLING									
Advanced Electric Heat Pump I	**	44.82	43.84	34.54	26.83	**	25.03	19.20	**
Advanced Electric Heat Pump II	40.68	27.89	27.22	21.33	16.57	5.41	15.52	11.87	8.19
Free Piston Stirling-Rankine Gas Heat Pump	66.82	48.00	45.31	35.59	28.87	9.99	26.95	20.64	13.41
V-Type Single Cylinder Stirling-Rankine Gas HP	75.64	55.71	51.72	45.36	32.62	11.20	30.89	23.11	15.48
Organic Fluid Absorption Gas Heat Pump	99.93	72.25	67.90	53.32	43.44	15.19	40.57	31.02	20.08
ANNUAL HEATING AND COOLING									
Advanced Electric Heat Pump I	**	77.90	78.99	96.06	92.70	**	107.10	116.77	**
Advanced Electric Heat Pump II	48.79	50.05	50.55	71.30	62.48	41.86	76.76	85.13	116.24
Free Piston Stirling-Rankine Gas Heat Pump	76.14	71.47	69.48	78.89	75.30	49.42	85.01	89.61	96.46
V-Type Single Cylinder Stirling-Rankine Gas HP	84.46	78.21	74.84	89.82	77.13	49.97	86.56	89.13	102.67
Organic Fluid Absorption Gas Heat Pump	110.77	99.37	95.69	103.13	96.88	61.16	107.01	109.93	113.29
Pulse Combustion Furnace/Central Air Conditioner-High Efficiency	76.71	74.53	74.34	91.77	86.08	61.72	97.06	104.68	110.36

* The performance factors presented may be underestimated for the advanced electric heat pump I, free piston Stirling-Rankine gas heat pump and organic fluid absorption gas heat pump due to possible underestimation of cycling and defrost losses. If the dynamic efficiency losses of the advanced heat pumps were similar to those of the conventional electric (Rankine cycle) heat pump, then adjusted estimates of performance factors which include these effects can be obtained by multiplying the values in this table by a factor of 1.097 for the heating season, 1.164 for the cooling season and 1.118 for annual performance factor values.

** System not sized for application in these cities.

EXHIBIT 12-5

ELECTRIC HEAT PUMP, CONVENTIONAL SYSTEMS, AND ADVANCED HEAT PUMPS FOR LIGHT COMMERCIAL BUILDING APPLICATION: COMPARISON OF AVERAGE HEATING, COOLING AND ANNUAL PERFORMANCE FACTOR - PRIMARY ENERGY BASIS

Electric Heat Pump and Conventional Systems	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard									
Heating	0.81	0.76	0.70	0.72	0.75	0.75	0.66	0.66	0.59
Cooling	0.65	0.65	0.62	0.65	0.64	0.70	0.67	0.68	0.63
Annual	0.66	0.68	0.64	0.67	0.69	0.73	0.66	0.66	0.60
Electric Resistance Heating/Electric Air Conditioning									
Heating	0.29	0.29	0.28	0.29	0.29	0.31	0.30	0.30	0.28
Cooling	0.65	0.65	0.62	0.65	0.54	0.71	0.68	0.68	0.61
Annual	0.59	0.50	0.47	0.45	0.39	0.39	0.39	0.37	0.34
Rooftop Gas Furnace/Electric Air Conditioning									
Heating	0.32	0.41	0.39	0.43	0.41	0.43	0.45	0.45	0.46
Cooling	0.60	0.60	0.58	0.60	0.60	0.66	0.63	0.63	0.59
Annual	0.56	0.54	0.51	0.53	0.50	0.49	0.51	0.50	0.50
Advanced Heat Pump Systems									
Brayton-Rankine Gas Heat Pump									
Heating	0.94	0.92	0.86	0.87	0.91	0.93	0.84	0.84	0.76
Cooling	0.78	0.76	0.76	0.74	0.77	0.73	0.77	0.77	0.72
Annual	0.79	0.79	0.78	0.78	0.83	0.85	0.81	0.82	0.75
Rankine-Rankine Gas Heat Pump									
Heating	1.43	1.35	1.32	1.30	1.34	1.35	1.28	1.27	1.27
Cooling	0.85	0.86	0.86	0.88	0.85	0.91	0.88	0.90	0.91
Annual	0.88	0.94	0.95	1.00	1.01	1.16	1.08	1.12	1.11

EXHIBIT 12-6

ELECTRIC HEAT PUMP, CONVENTIONAL SYSTEMS, AND ADVANCED HEAT PUMPS FOR LIGHT COMMERCIAL BUILDING APPLICATION: ANNUAL PRIMARY ENERGY CONSUMPTION - COMBINED HEATING AND COOLING (MILLIONS OF BTU'S)

Electric Heat Pump and Conventional Systems	Houston	Birmingham	Atlanta	Tulsa	Philadelphia	Seattle	Columbus	Cleveland	Concord
Electric Heat Pump - Standard	363.49	300.41	288.76	295.70	219.35	163.48	239.78	234.91	285.74
Electric Resistance Heating/Electric Air Conditioning	408.34	405.88	397.64	443.28	383.37	310.18	409.56	420.62	499.45
Rooftop Gas Furnace/Electric Air Conditioning	429.03	374.63	362.55	377.09	302.40	246.74	311.04	309.29	342.84
Advanced Heat Pump Systems									
Brayton-Rankine Gas Heat Pump	303.96	256.33	237.67	253.70	182.00	141.49	196.81	190.85	230.07
Rankine-Rankine Gas Heat Pump	274.23	215.46	195.96	199.18	148.46	103.73	147.31	139.30	154.27

EXHIBIT 12-7

ANNUAL PRIMARY FUEL AND ENERGY SAVINGS WITH ALTERNATIVE HVAC SYSTEMS - RESIDENTIAL SYSTEMS
(Basis: Total United States)

	Coal (lb x 10^{-6})		Oil (gal x 10^{-6})		Gas (cft x 10^{-9})		Hydro (kWh x 10^{-6})		Nuclear (kWh x 10^{-6})	
	Low	High	Low	High	Low	High	Low	High	Low	High
Hybrid Electric Heat Pump-Gas Furnace instead of Electric Heat Pump-Standard										
1985	(1674)		(23)		30		(246)		(258)	
1990	(3063)		(43)		55		(456)		(471)	
1995	(4207)		(59)		75		(626)		(646)	
2000	(5127)		(71)		92		(764)		(788)	
Advanced Electric Heat Pump II instead of Electric Heat Pump-Standard										
1985	(153)	(336)	(3)	(7)	(1)	(1)	(36)	(79)	(27)	(59)
1990	(613)	(1646)	(13)	(34)	(3)	(7)	(144)	(386)	(107)	(286)
1995	(1104)	(3386)	(23)	(72)	(5)	(14)	(260)	(815)	(193)	(593)
2000	(1592)	(5870)	(33)	(122)	(7)	(24)	(374)	(1381)	(280)	(1025)
Free Piston Stirling-Rankine Gas Heat Pump instead of Gas Warm Air Furnace/ Central Air Conditioner-High Efficiency										
1985	(44)	(100)	(1)	(2)	(1)	(3)	(15)	(33)	(8)	(20)
1990	(135)	(352)	(3)	(8)	(4)	(11)	(45)	(121)	(27)	(70)
1995	(226)	(729)	(5)	(17)	(7)	(22)	(75)	(242)	(45)	(144)
2000	(317)	(1213)	(7)	(28)	(10)	(37)	(105)	(400)	(63)	(240)
Pulse Combustion Furnace instead of Gas Warm Air Furnace										
1985	49		1		(125)		13		9	
1990	99		2		(250)		27		17	
1995	148		3		(374)		40		25	
2000	197		4		(498)		54		34	
Organic Fluid Absorption Gas Heat Pump instead of Gas Warm Air Furnace/ Central Air Conditioner-High Efficiency										
1985	(58)	(133)	(1)	(3)	0+	1	(18)	(41)	(11)	(25)
1990	(179)	(471)	(4)	(10)	0+	2	(56)	(150)	(34)	(90)
1995	(299)	(965)	(6)	(21)	1	4	(93)	(300)	(57)	(184)
2000	(420)	(1604)	(9)	(36)	2	7	(131)	(498)	(80)	(306)
Free Piston Stirling-Rankine Gas Heat Pump instead of Electric Heat Pump- High Efficiency II										
1985	(242)	(555)	(5)	(11)	3	7	(62)	(141)	(42)	(99)
1990	(750)	(2011)	(15)	(38)	9	24	(190)	(515)	(131)	(346)
1995	(1250)	(4035)	(24)	(77)	15	49	(317)	(1023)	(218)	(699)
2000	(1758)	(6700)	(34)	(129)	22	82	(446)	(1699)	(307)	(1164)

Note: Numbers in parentheses represent resource savings.

EXHIBIT 12-8

ANNUAL PRIMARY FUEL AND ENERGY SAVINGS WITH ALTERNATIVE HVAC SYSTEMS: COMMERCIAL SYSTEMS
(Basis: Total United States)

	Coal ($lb \times 10^{-6}$)		Oil ($gal \times 10^{-6}$)		Gas ($cft \times 10^{-9}$)		Hydro ($kWh \times 10^{-6}$)		Nuclear ($kWh \times 10^{-6}$)	
	Low	High	Low	High	Low	High	Low	High	Low	High
Rankine-Rankine Gas Heat Pump instead of Rooftop Gas Furnace/ Electric Air Conditioner										
1985	(25)	(51)	(1)	(1)	(0)	(0)	(8)	(15)	(5)	(10)
1990	(132)	(342)	(3)	(8)	(0)	(0)	(39)	(102)	(27)	(69)
1995	(326)	(701)	(8)	(17)	(0)	(1)	(97)	(209)	(66)	(142)
2000	(627)	(1539)	(15)	(37)	(1)	(2)	(187)	(459)	(127)	(311)
Rankine-Rankine Gas Heat Pump instead of Electric Heat Pump Standard										
1985	(45)	(91)	(1)	(2)	0	1	(14)	(28)	(9)	(19)
1990	(235)	(608)	(6)	(15)	2	6	(71)	(184)	(48)	(125)
1995	(579)	(1246)	(15)	(31)	5	12	(176)	(378)	(118)	(255)
2000	(1115)	(2737)	(28)	(69)	10	25	(338)	(831)	(228)	(561)

NOTE: Numbers in parentheses represent net savings.

13. PRELIMINARY ANALYSIS OF THE IMPACT OF HEAT PUMPS ON ELECTRIC AND GAS UTILITIES

13.1 Introduction

Before concluding the analyses of conventional and advanced heat pumps presented in this report, the effect of sufficiently large numbers of electric and gas heat pumps on electric and gas utility load characteristics must be considered. Although a quantitative or definitive analysis of this question was outside of the scope of this study, a preliminary analysis was, nevertheless, undertaken and some conclusions drawn.

The demand-statistical parameters of interest in considering the question of utility load impact are (a) annual or seasonal load factor (b) annual or seasonal performance factor, and (c) coincidence factor. Although other demand parameters are often defined and measured also, the above will suffice for our discussion. The load factor is defined as the ratio of the average demand to the maximum demand; the performance factor as the ratio of the heating or cooling load of the building to its on-site energy consumption. A system with a high load factor, then, represents a reasonably "level" demand; a system with a high seasonal performance factor uses less energy to meet a given seasonal building heat loss or gain than a system with a low seasonal performance factor.

In addition to the two parameters defined above, an additional quantity of interest is the coincidence factor. That is the ratio of the maximum demand imposed by a group or class of individual units to the sum of the individual demand maxima of all the units in the group or class under consideration. Central air conditioning, for example, represents a load class with relatively high coincidence factor: in

hot weather a large demand is imposed on the utility as almost all the air conditioners are on in response to the thermostats.

13.2 Effect of Electric Heat Pumps

In a previous study (Gordian Associates, 1976), the authors considered the question of the effect of electric heat pumps on electric utility load characteristics on the basis of a typical heat pump "balance point" curve. It was shown how stepwise rather than gradual increases in demand would occur if the outdoor temperature dropped below the first balance point, the second balance point, etc. An actual installation would not, of course, have a fixed balance point (or balance points, if more than one stage of supplemental heat is involved). Rather, the balance point would vary depending on the insolation, wind velocity, internal heat generation, thermostat setting, and, possibly, other variables. Nor would the balance points of different buildings (say, either residential or commercial) necessarily coincide, creating a demand diversity. This demand diversity is, of course, all-important to the utility, because it only "sees" the load-class average or diversified demand, not a multiple of the individual demand curves.

Unfortunately, the hour-by-hour simulation of the energy requirements (i.e. a hourly demand profile) of a heat pump conditioned building does not itself alone enable the computation of the diversified demand, unless a large number of simulations of building energy demand were undertaken. However, some load survey data and other information are available on heat pump demand diversity, permitting certain qualitative conclusions as to its impact on electric utility load characteristics. Some of these data are discussed below.

AEIC Load Research Committee reports (1971-72) have shown that electric heat pump diversified demand per customer (i.e. the demand the utility "sees") correlates reasonably well with heating and cooling degree-days, respectively. That is, daily maximum 30-minute integrated heating season electricity demand increases as a linear function of degree-days, whereas the analogous cooling season electricity demand increases as a linear function of cooling degree-days. The coincidence factor varies with the severity of the heating or cooling requirement as follows:

Heat Pump Coincidence Factors from Load Survey Data

	Heat Pump Coincidence Factor (Percent)	
	Heating	Cooling
Maximum	76.8	61.8
Month of maximum	February	June
Minimum	24.7	37.8
Month of minimum	September	October
Annual average	69.2	60.2

Source: Baltimore Gas and Electric Co. 1969 load survey (AEIC, 1971-72).

The results of this load survey showed that the heat pump has a significantly higher maximum heating coincidence factor than on cooling and that the heating coincidence factor appears to be more weather-sensitive.

Another interesting result was obtained in the Westinghouse northern climate heat pump improvement study (Kirschbaum and Veyo, 1977a): electric heat pumps expected to be installed in three selected utilities' service

territory by 1985 were shown to produce only a small increase in annual energy demand (i.e. generation) but a large increase in the annual peak. Annual load factors were shown to decrease from 2.2% to 17.3%, depending on the utility. A summary of the Westinghouse results is shown in the table below.

Effect of Projected Cumulative 1985 Heat Pump Saturation on Three Electric Utilities (Per Cent Change)

	Niagara-Mohawk Power Corp.	Boston Edison Co.	Public Service Co. of Colorado
Annual Peak	3.2	0.3	27.0
Annual Load Factor	-2.2	-3.9	-17.3
Energy Generation	1.0	1.0	10.5
Revenue	0.9	0.8	10.5

Source: Kirschbaum and Veyo (1977a)

It should be noted that solar assisted electric resistance heat supplemented systems are not even incorporated in the above result. These, of course, would be expected to cause even greater load factor deterioration.

Use of hybrid heat pumps (e.g. electric heat pumps using combustion furnaces for supplemental heat) would improve heat pump demand characteristics but deteriorate gas utility load factors.

13.3 Effect of Gas Heat Pumps

Little quantitative information is on hand on the pertinent factors affecting the load characteristics of gas space conditioning appliances on gas utility loads. However, it is well known that gas utilities experience a very season-and-weather dependent heating gas demand which essentially vanishes in the summer, the saturation of gas air conditioning

probably being insignificant in most areas. The effect of gas heat pumps in lieu of gas furnaces (with electric cooling, say) would clearly be to improve utility annual load factors by decreasing the heating season gas demand due to higher equipment efficiency and addition of load during the cooling season. It has been argued that gas heat pumps will increase electric utility load factors as well by drawing away some of their cooling load, but that argument appears specious with regard to regions where gas heat pumps would be most likely applied since in these areas electric utilities may well be winter-peaking in the near future anyway. Since a quantitative analysis of these questions did not fall within the scope of this study, little more can be said about the probable effect of gas heat pumps. In any case, however, no significant effect is likely to be felt for a long time even if commercially successful equipment were developed.

13.4 Summary

It is clear from the foregoing discussion that increased penetration of electric heat pumps with supplemental resistance heat will cause, in some cases, the deterioration of system annual load factors, unless appropriate thermal energy storage systems are also developed. Capacity modulation with heating or cooling load would mitigate the load factor decrease to some extent; increased use of solar assisted heat pumps (without appropriate thermal storage) would probably greatly accelerate load factor deterioration.

References to Chapter 13

Association of Edison Illuminating Companies, Load Research Committee; Residential Customer Load Study (1971-1972) pp. L-123-L-158.

Gordian Associates Inc. Evaluation of the Air-to-Air Heat Pump for Residential Space Conditioning (Report prep. for FEA, NTIS No. PB-255-652, April 1976).

Kirschbaum, H.S. and Veyo, S.E. An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate, Vol. I (EPRI Report No. EM-319, January, 1977a).

14. RESEARCH AND DEVELOPMENT NEEDS IN HEAT PUMP TECHNOLOGY

14.1 Introduction

The previous chapters have described new developments in heat pump technology and provided comparisons of on-site and primary energy efficiency and life cycle owning and operating cost of selected advanced and conventional or state-of-the art heat pumps. In this chapter, research needs in the area of advanced heat pump and related technology are identified. A suggested Federal Government sponsored research program complementing or supporting private sector research is outlined in Chapter 15. Again, this discussion makes no pretense of being comprehensive, since it has not been possible to investigate in depth all of the various heat pump concepts under development; certain important areas of heat pump technology such as applied or built-up heat pumps, of necessity, have been ignored altogether.

Before discussing heat pump research and development needs, a review of the status of existing programs and projects is necessary. Section 14.2 attempts to give a broad overview of ongoing research. Some related material may be obtained from a survey of heat pump research needs commissioned by the President's Council on Environmental Quality (Crane et al., 1976) and several EPRI publications on this subject (see Section 14.2.2).

14.2 Existing Research Programs

Heat pump research in the United States is being conducted by a number of firms, institutions or other organizations and by individual inventors. The diversity of the approaches considered and work ongoing

should be apparent from Chapters 3-7. For convenience the discussion of ongoing work in this section will be organized into broad segments of the private and public sector as follows:

- ERDA and other Federal Government programs
- Air conditioning industry programs
- Electric Power Research Institute programs
- Other electric utility industry programs
- American Gas Association programs
- Other gas utility industry programs
- University and independent research organization programs

14.2.1 ERDA and Other Federal Government Programs

The preceding sections have discussed two gas-fired thermal engine Rankine-cycle heat pump projects of which the Consumer Products and Technology Branch of ERDA's Office of Conservation is sponsor or co-sponsor. Additional information on these and other programs is available in a recently published Program Plan (ERDA, 1977) and will not be repeated here.

A number of heat pump or heat pump related projects are being funded by the Solar Energy Division of ERDA.

- Chemical heat pump
- Sulfuric acid heat pump
- Solar pivoting tip rotary vane expander-driven heat pump
- High temperature absorption heat pump

Projects included in the program plan but not necessarily funded are:

- A study to compare solar-assisted heat pump systems, direct solar heating systems and conventional heat pumps for use in various climates. Different configurations will be studied for performance advantages for both residential and commercial installations in new and old buildings,

- A study to evaluate the technical and economic feasibility of heat pump systems where the compressor is driven by a heat engine.

- A study to evaluate the technical and economic feasibility of using natural or otherwise available elements for thermal storage in a solar assisted heat pump system.

- A study of the performance and economic feasibility of incorporating wastewater heat recovery in a solar-assisted heat pump system.

- A study to identify the electric utility load characteristics of solar-assisted heat pumps. Also a study to investigate means of using thermal storage for load leveling in these systems.

- Configuration studies of various combinations of heat pumps (both water source and air source), collectors, and storage units. The characteristics of the best combinations and the sensitivity of each combination to changes in component performance are to be determined.

14.2.2 Air Conditioning Industry Programs

Only electric-motor-driven heat pumps are being manufactured today for residential and light commercial-industrial applications. Understandably, many, if not most, of the details of ongoing research and development work are highly proprietary. Chapter 3 describes what has been learned in our field investigations through contacts with various manufacturers and by following technical trends and developments in the industry. A summary of areas in which research or development work is believed to be ongoing is as follows:

- Two-speed heat pump compressor development -- a major manufacturer announced in 1973 a two-speed air-conditioner compressor in the 5 ton size. Others have developed two-speed hermetic refrigerant compressors. A two-speed heat pump has been built and is undergoing tests.

- Improvement of low-temperature capacity and efficiency - dramatic improvements in some manufacturer's lines have been observed since our FEA study (Gordian Associates, 1976);

- Solar assisted heat pump development - in comparison to resistance heat, the electric heat pump constitutes a low load-factor load to the utility; the solar-assisted heat pump even more so, unless substantial storage is provided. Several manufacturers have introduced solar assisted or boosted air-to-air heat pumps and have or are improving their water-to-air heat pump line;

- Continuous speed and capacity modulation;

- Improved heat pump controls and malfunction diagnostics.

14.2.3 Electric Power Research Institute Programs

Since its founding, EPRI has been sponsoring research into a number of areas involving heat pump systems, with the objective of fostering the development of so-called "preferred-systems", i.e. those that are efficient, reliable and cause the least deterioration of utility load factors.

Ongoing programs include:

- An analytical study to develop and optimize electric heat pump design to colder climates, with special consideration being given to solar assisted heat pumps. A final report on this project was issued recently (Kirschbaum and Veyo, 1977). (RP544-Contractor: Westinghouse Electric Corp.)

- A program to design and implement instrumentation for the performance evaluation of residential solar assisted heat pumps. (RP649-Contractor: Potomac Electric Power Co.)

- A program to design and implement instrumentation for the performance evaluation of a demonstration of solar assisted heat pump system in a commercial building. (RP650-Contractor: Boston Edison Co.)

- A study evaluating energy conservation and load management characteristics of a residential solar assisted heat pump

system. Direct solar augmentation of the solar assisted heat pump system is also being explored. (RP845-Contractor: New Bedford Gas and Edison Light Co.)

- A detailed analysis of cost and performance data for an installed solar assisted heat pump system. (RP924-Contractor: City of Colorado Springs Department of Public Utilities)

- A study of the performance and economic feasibility of a commercial building solar assisted heat pump which utilizes the building's rejected heat. (RP924-Contractor: Southern California Edison Co.)

- A study is being conducted by the Public Service Company of New Mexico to develop five residential solar augmented space conditioning systems, three of which will be heat pump systems. They are subcontractors to Arthur D. Little, Inc., which is developing a model to predict the compatibility of solar space conditioning systems with utility system. A paper describing the project has been released (Pepper, 1976). Another participant in this program is Long Island Lighting Co. with another five buildings.

- A project is being conducted in which instrumentation was developed by Carrier Corporation and Niagara Mohawk Power Company to monitor the performance of existing heat pump systems. The study concerns itself primarily with northern climate cities. A paper describing the work has been published (Reedy, et al., 1976).

14.2.4 Other Electric Utility Industry Programs

A number of electric utilities have ongoing projects evaluating the reliability and performance of heat pumps vs. competitive heating systems in mostly residential buildings. Some of the reliability data accumulation programs were described in our previous study (Gordian Associates, 1976); additional information on utility programs has been described in (Kirschbaum and Veyo, 1977) and a number of articles in Electrical World and other periodicals. To our knowledge no utility is by itself involved in heat pump equipment development, but may be cooperating with EPRI or heat pump manufacturers on joint projects testing heat pump performance in various applications (e.g. heat or cold storage and recovery, solar assisted systems, etc.). Much of this effort is ultimately designed to

produce equipment of improved performance and reliability and is of great value in assisting the industry to better define conditions under which heat pump equipment must operate in practice.

14.2.5 American Gas Association Programs

As already described, a number of heat pump studies and hardware development projects are being conducted under sponsorship of the AGA and by individual gas companies. A recent article in the ASHRAE Journal discusses the AGA programs (Sarkes *et al.*, 1977), although the description of the absorption cycle heat pump development is apparently dated. A list of selected projects is given below; however, since all of these have already been described in some detail in the preceding chapters, only the project participants are listed here:

- Organic fluid absorption gas heat pump: Sponsors: Allied Chemical Co., AGA; Systems Contractor: Phillips Engineering Co.
- Brayton-Rankine gas heat pump: Sponsor: AGA; Contractor: Garrett Airesearch Mfg. Co. of California.
- Stirling (free-piston)-Rankine gas heat pump: Sponsors: AGA, ERDA; Contractor: General Electric Co., Space Div.

14.2.6 Other Gas Utility Programs

Other gas-fired thermal engine and absorption cycle heat pump programs in progress under the sponsorship of individual gas utilities include the following:

- Steam turbine-Rankine heat pump: Sponsor: Consolidated Natural Gas Service Co.ERDA; Systems Contractor: Mechanical Technology Inc.
- (Aqua-ammonia) fluid absorption gas heat pump: Developer: Columbia Gas System Service Corp.
- Linear gas engine-Rankine heat pump: Sponsors: Columbia Gas System Service Corp. MTS Research Corp. Contractor: Tectonics Research Corp.

14.2.7 University and Other Private Sector Programs

Heat pump research, including (but to a lesser degree) equipment development, is also being carried out at a number of academic or research institutions, small companies and by private individuals. Some of these projects are sponsored by some of the organizations already mentioned and, thus, have already been described. Some of those not mentioned before include:

- Heat pump modelling and exploratory studies on a compressor or suction valve closing device (Hiller and Glicksman, 1976);
- Regional assessments of electric heat pump performance and reliability by state energy agencies (see, for example, Wildin *et al.*, 1976);
- Solar energy heating system demonstrations involving heat pumps (for examples, see *Proceedings of Second Annual Heat Pump Technology Conference*, 1976).

14.3 Research Needs Identified in this Study

As is evident from Section 14.2, a full spectrum of research projects and programs is already ongoing in the private and public sector.

In view of the decrease of electric heat pump capacity and COP with decreasing outdoor temperature, an obvious need exists to improve the low temperature capacity as well as the overall efficiency of the heat pump. A number of articles (Spofford, 1959; Ambrose, 1966) have discussed the reasons for the low efficiency of the actual heat pump in contrast to theoretical performance of a Carnot refrigeration machine operating under similar indoor-outdoor temperature conditions. Many of the proposed approaches to performance improvement center around developing a more efficient compressor capable of stepwise or continuous capacity modulation. There is an already voluminous and growing technical literature on

possibilities of COP improvement by improved compressor design or refrigeration loop modifications. The recently completed study by Westinghouse (Kirschbaum and Veyo, 1977), has identified most of the possible approaches to improvement of the state-of-the art electric heat pump. Other possibilities have been mentioned in the paper by Comly et al. (1975), including:

- Parallel compressors (possibly of different size);
- Cascaded refrigeration loops (with different refrigerants, if necessary);
- Series compressors
- Turbocharged compressor (i.e. precompressor);
- Two stage heat pumps.

A considerable amount of electric heat pump research is being carried on by the segment of the HVAC industry producing heat pumps or air conditioners. In many cases this may be best characterized as a process of product evolution and improvement - to increase performance, increase reliability and cut manufacturing costs. However, additional areas may be worthy of investigation. Research needs identified in this study are listed below. Clearly, many more could have been added to this list. Also, some projects have been or are being worked on and not all projects suggested are equally important. Nevertheless, these ideas are offered as means of furthering heat pump technology and market penetration.

Innovative Electric Heat Pump Designs

Although the heat pump and air conditioning industry is expected to continue electric heat pump improvement, including two-speed compressor development, refrigeration loop component and control system improvement and the other activities related to normal product evolution, it appears

that only moderate performance improvement is possible within constraints of the current design and costs. Therefore, there is a need for innovative design approaches to heat pump, which may for example deviate significantly from the conventional hermetic electric motor-compressor design with stationary heat exchangers. Some of these designs may incorporate other heat sources or sinks, use air or water instead of refrigerant switching, employ external stepwise or fully modulating (not necessarily induction) motors, heat pipes or other technology not now being used. Hybrid designs with furnaces, thermoelectric elements, and evaporative coolers should be explored, as added examples. Some of this involves higher risks or more significant deviations from the current product state-of-the-act than established manufacturers would be expected to undertake on their own.

Heat Pump Designs for Existing Housing

Electric heat pumps being manufactured today are principally aimed at the new construction market for reasons of (a) lower first cost of total installation in comparison to a refit, (b) ability to insure the adequacy of ductwork and (c) lack of competition in many areas from cheaper gas furnaces due to curtailments of new gas hookups. It is true that conversions of electric furnaces to heat pumps make up about 30% of the residential market and that hybrid or add-on heat pumps are designed for installation in existing buildings as well as new ones, nevertheless more development work is needed to devise equipment for older buildings with small-diameter warm air ducts or no ducts at all (i.e. hydronic or steam heated units) and more cost-effective equipment for areas where the cooling requirement is small or even nonexistent.

While the existing housing stock is potentially a significant market in terms of its energy conservation potential, it is obviously much more heterogeneous in comparison to new housing and hence offers more difficulty and less incentive for successful product development. These heat pump designs should proceed conjointly with reinsulation and other programs and consider possible combinations with thermal storage and solar supplementation.

Water Source Heat Pumps

The energy effectiveness, reliability, applicability and cost of unitary water source heat pumps needs to be investigated in greater detail and performance modelling and cost analysis procedures developed and verified.

Thermal Storage for Heat Pumps

Thermal energy storage has been and continues to be the subject of intensive research, much of it in conjunction with solar energy building heating systems. Various configurations of heat pumps and thermal storage have been examined in the Westinghouse study (Kirschbaum and Veyo, 1977b) as well. However, shorter-term storage to minimize supplemental resistance heat requirements also should be investigated. This storage might be replenished in periods of excess compressor capacity and used when compressor capacity is insufficient to meet the heating or cooling requirements of the structure. As, at least in the cooling mode, some efficiency loss is involved with all thermal storage systems*, the optimum combination of improved load factor and seasonal performance

* This loss can be minimized but not entirely eliminated by locating the storage system within the conditioned space.

factor should be investigated. Designs should be developed for existing as well as new buildings.

Refrigerant Development

Development of alternatives to the R-22 (chlorodifluoromethane) presently used in unitary heat pumps and air conditioners is another area where work is indicated. While dramatic efficiency improvement from that sector is probably unlikely, alternatives to potentially ozone-reactive fluorocarbons may be found,or more effective or flexible combinations of present refrigerants may be discovered. Among the systems investigated should certainly be two-fluid combinations in which additional fluid vaporization takes place at low evaporator temperature. Research is also needed to develop alternatives to R-11 (trichlorofluoromethane) and R-12(dichlorodifluoromethane), presently being considered for or used in centrifugal machines.

Frost Formation and Melting on Heat Exchangers

At the present time no theoretical method exists which would enable the prediction of evaporator freezing and its effect on heat pump performance as a function of coil temperature, ambient dry and wet bulb temperature, wind velocity, presence of precipitation, etc. Although a number of studies were conducted at Ohio State University under ASHRAE sponsorship, (Gates et al., 1967; Huffman and Sepsy, 1967; Parish and Sepsy, 1972) a considerable amount of both experimental and theoretical work needs to be done to (a) further the understanding of environmental and equipment operating conditions under which coil freezing is encountered, (b) achieve a quantitative understanding of frost layer thickness, density,

thermal conductivity and its effect on the overall coil heat transfer coefficients, (c) allow the calculation of the frequency and duration of defrost cycles encountered in typical application and (d) incorporate in heat pump performance estimation a reliable means of estimating seasonal defrost energy requirements.

While undoubtedly much of this work is probably being carried out by manufacturers, at least as far as empirical data-gathering and correlation is concerned, very little has filtered into the available technical literature. Yet a precise analysis of heat pump performance in typical applications would seem to require an understanding of energy requirements for defrost.

Control Systems

The York CHAMPION (York, 1975) solid state logic module with its adjustable balance point control represents a significant advance in heat pump control systems. However, additional innovation in control systems, especially controls for load sensing and capacity modulation are necessary.

On the negative side, increasing heat pump control system sophistication adds to system complexity and requires increased sophistication on the part of the servicer.

Externally Driven Hermetic Compressors

Given the apparently successful development of a magnetic coupling for the Brayton-Rankine heat pump (see Section 4.3.2), it is well worth investigating whether the same concept could not be applied to the hermetic compressor of an electric heat pump. That is, the motor would

be air-cooled and external to the refrigeration loop, driving the hermetic compressor through a magnetic coupling arrangement. This would eliminate the need for use of suction-gas motor cooling and thus increase compressor COP.

Thermal Engine Heat Pumps

ERDA is already sponsoring two thermal engine heat pump development projects (see Section 14.2.5). In the authors' view, thermal engine heat pumps appear most promising in commercial-size tonnages (up to 25 or 40 tons) where system complexity and attendant high first cost can be tolerated more readily than in smaller residential-scale installations. Designs using rotating rather than reciprocating machinery and employing hermetic refrigeration loops also appear to be the preferred ones in terms of assuring long-term in-service reliability and relatively maintenance-free operation. Therefore, a research program in the development and demonstration of thermal engine heat pumps especially for commercial applications should be continued. The main direction of this program should be the development of gas-burning equipment primarily for the refit market now served by year-round (usually roof-top) air conditioners with modular gas furnaces.

Absorption Cycle Heat Pumps

In view of the simplicity of gas-fired absorption cycle machines, further development and demonstration of these seems indicated primarily for residential refit applications. In addition to organic fluid based systems, further research into aqua-ammonia systems may be desirable to the extent required to obtain a clearer picture of the relative merits

(the probability of developing practical equipment) of each system. However, before use of absorption cycle machines can be justified in areas with a high cooling load, their cooling COP has to be improved significantly.

Brayton, Stirling and Other Refrigeration Cycles

Exploratory research should also be carried out on refrigeration cycles other than the Rankine cycle machine. These studies should be limited to proof-of-concept experiments, design feasibility analyses and similar studies, the object of these being not to overlook other promising (and perhaps longer-term) approaches. Brayton air cycle machines are being used in aircraft cooling and in the Advanced Concept Train air conditioning system design (Wurm, 1974; Garrett-Airesearch, 1976). In addition, a "hybrid" electrically driven Brayton and Rankine cycle heat pump concept has been proposed by the Rovac Corp. (1976), as already discussed in Section 3.6. These systems bear investigation as to their suitability for building HVAC system application.

Reliability Research and Field Data Consolidation

Although accelerated life testing in environmental test chambers of systems and components is an important part of the heat pump equipment development and quality assurance programs being conducted by reputable manufacturers, reliability assurance, nevertheless, remains an area where more research is necessary.

For example, there is a need for accumulation and analysis of field test data from equipment installations in various climatic regions. These data should be carefully screened and classified so that failure rates may be correlated with more of the important independent variables

and parameters expected to govern heat pump reliability than has been possible heretofore. More closely controlled data analyses on carefully stratified samples would provide the technical basis for industry reliability standard setting, equipment rating and consumer information programs.

Consumer Attitude Survey

The heat pump is still not widely known or recognized by the average buying public as an energy efficient space conditioning appliance. The present available literature on residential building owners' attitudes toward heat pumps include studies of a local or regional nature (Arkansas Power & Light Company, 1974) and published reports that may lack broader statistical validity. Even less published information is available on nonresidential heat pump buyer attitudes.

An important prelude to any heat pump consumer education program would be a better understanding of the average consumer's knowledge, understanding and attitude toward the heat pump. Any such survey, of course, should be done on a regional basis for the whole country.

While it is possible that various market and attitude surveys have been conducted by utilities and by heat pump manufacturers or developers, most of this information is proprietary and of a kind that may be limited to the manufacturer's interest (Garrett-Airesearch, 1976; General Electric Co. 1977). Therefore, a heat pump consumer attitude survey is needed that is statistically valid and national in scope to assist in consumer education and heat pump market identification and sales potential.

Potential Heat Pump Refit Market Survey

Of the total residential building stock in the U.S. in 1974, about half were heated by a warm air system (Census Bureau, 1976). A central

supply system is essential for refitting a residential building with a heat pump because altering an existing building to install a central air distribution is considered to be prohibitively expensive (Hittman, 1976).

However, accurate information on the warm air heated building population is not now available. The kinds of information that would permit much better estimation of the potential for heat pumps in a refit market application would include the present kind of heating systems in the population, the replacement rate for present heating systems in the population, and, most importantly, the suitability of the ducts for carrying the larger air volumes needed for a heat pump.

Similarly, information of a like nature should be obtained for the nonresidential building market. There is, in general, very little available information on nonresidential buildings. Therefore, any information such as discussed above would greatly improve the estimation for potential market for commercial service heat pumps.

Certification Procedures Study

A recent study (Hittman, 1976) reported that the heat only heat pump, because of design differences from a conventional heat pump, was unable to obtain Underwriters Laboratories' (UL) and Air-Conditioning and Refrigeration Institute (ARI) certification because no standards existed for the product. Certification is desirable because many building codes require it, and the local building code officers are loath to waive the requirement because UL and ARI certification offers assurance of product safety. Failure to have these certifications will result in lost heat pump business.

A survey of certification procedures and how they may inhibit non-conventional or advanced heat pumps would permit action to be taken, either for new standards development or through modification of an existing standard.

Building Code Study

As discussed in Chapter 11, the building codes have not, up to now, paid special attention to heat pumps. In general, the applicable sections for mechanical refrigeration, motors, etc. must be applied to the heat pumps. It would be preferable for the codes to provide explicit requirements rather than the present system, which would leave the acceptability of a heat pump open to the interpretation of the code official or fire inspector. Alternatively, additional research may be needed to establish fair requirements, particularly for advanced and nonconventional systems.

However, this leaves the question open as to how other systems different from the one gas heat pump system discussed might fare. Since it is not a settled matter by any means at this time, a need exists to review all potential systems against the ASME Boiler Code and to establish, possibly, a code modification applicable to the advanced gas heat pump system.

Development of an Improved Market Model for Heat Pumps

Several heat pump market models have been developed for the residential and the nonresidential building markets (General Electric Co., 1976; Hittman, 1976; Oak Ridge National Laboratory, 1977) in addition to the Gordian Associates market model used in this work. Further development of these models is needed as well as an examination of their predictive ability. One aspect of an improved market model would be the refining

of the various input data to the model, including improved definition of the refit market, better approximations for nonresidential new construction and the proportion of total non-residential construction that could be conditioned by unitary equipment.

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15. CONCLUSIONS AND RECOMMENDATIONS

15.1 Conclusions

The general conclusion of this investigation is that no space conditioning system is universally to be preferred to any other if energy effectiveness, life-cycle cost, conservation of specific scarce fuels or type of application is considered. Therefore, legislative or regulatory activity which favors one system over another would be counter-productive from an energy conservation standpoint if freedom of choice is to remain with the consumer. Another general conclusion is that new residential or commercial installations of gas heat pumps in lieu of electric heat pumps would accelerate the depletion of the fuel in scarcest supply, while possibly conserving more abundant energy resources.

More specific conclusions are offered in the sections that follow:

15.1.1 Conclusions Concerning Electric Heat Pumps

1. High efficiency electric heat pumps now available represent on the average an 18% COP improvement and 12% SPF improvement over the equipment available in 1975 and previously. Of the various all - electric heating and cooling systems studied, electric heat pumps are the most energy - efficient in terms of either on-site or primary energy consumption.

The above comparisons apply in case of electric furnace or baseboard heating systems with appropriate central or room cooling units. In baseboard heated residences with zone control, additional energy savings over the consumption data calculated here can be obtained by appropriate temperature set-point reductions below the levels assumed for the centrally

controlled systems. The room air conditioners in the test residence have a smaller total cooling capacity than the central units studied. The former thus meet only part of the cooling load of the house and have a commensurately smaller energy consumption.

The SPF's of the commercial-size heat pumps were higher than those of the high-efficiency residential units because the commercial building used in the simulations was well insulated, had relatively high lighting levels, was essentially operated only during daytime hours. Although a less energy-efficient structure may have penalized more the operation of the electric heat pump (as well as the resistance furnace), the building simulated was a recently built structure (rather than a hypothetical one) on which heat pump equipment has been actually installed.

2. The heat-only heat pump appears to be more energy-effective than the conventional (heating and cooling) heat pumps. However, the cost effectiveness of the former, either as a new installation or as an add-on appears to be questionable.

Because the heat-only heat pump contains most of the same components as the outdoor unit of a conventional split system heat pump, their first costs are similar. However, the first cost of the heat-only installation (in lieu of a resistance system, say) can be paid back in energy savings only over the heating season, making it uneconomical to own and operate. Although designs optimized primarily for heating or cooling may be justified, eliminating one or the other function altogether may not be economically prudent.

3. Hybrid heat pumps utilizing a fossil fired furnace for supplemental heat appear to be less attractive than expected in energy efficiency and life-cycle cost in comparison to conventional heat pumps. More advanced designs may reverse this conclusion, however.

In principle, hybrid heat pumps represent an interesting concept for retrofitting fossil fuel furnaces in newer homes. The system examined, however, was one of the earliest models commercially introduced (several years ago). Newer models already on the market have higher COP's and EER's and should prove more energy-and cost-effective. Indeed, one of the high-efficiency electric heat pumps considered is also available as an add-on unit to warm air furnaces.

4. In comparison to gas or oil furnaces combined with central air conditioning, under conditions of insulated supply and return air ductwork, the fossil fuel furnaces with electric cooling are more energy efficient in a primary energy sense in all but the warmer regions. However, current and past practice in furnace installations is not to insulate ductwork, in which case electric heat pumps (especially the high-efficiency type) would be superior almost everywhere.

Comparison of the energy-efficiency of competitive heating and cooling systems depends significantly on the underlying assumptions. If electric heat pumps and fossil fuel furnaces are compared on the basis of well-insulated ductwork, the furnaces may be slightly superior in moderate-to-cold regions. If compared on the basis of current new furnace installation practice, electric heat pumps would be the more energy efficient. Locations with a relatively long heating season but lacking temperature extremes (e.g. Seattle) represent ideal conditions for heat pump installation.

5. Electric heat pumps in most cases are still more costly to own and operate than available lower-first-cost fossil fuel or resistance heating systems coupled with high-efficiency air conditioners. In colder regions, the resistance systems are costlier than electric heat pumps.

Electric heat pumps are more complex and more costly than available HVAC system alternatives for both residential or commercial applications. To this must be added the fact that natural gas is not priced at its free-market energy value. Since our previous study, life-cycle costs of all systems have increased from 10-30% due to inflation in energy and equipment costs.

6. The added 5-20% installed cost of high-efficiency heat pumps in comparison to the lower-efficiency or standard unit appears only marginally cost-effective and only in colder regions and under circumstances of a high rate of inflation in electricity cost.

Marketplace realities could be an inhibiting factor to electric heat pump innovation or at least to the rate of efficiency improvement. As long as low-first-cost alternatives are available and natural gas is not priced at its free-market energy value, heat pump product evolution will not proceed as fast as technological advances would otherwise dictate. This is a consideration in evaluating any new design concept claiming significant performance improvement.

7. The commercial-size electric heat pumps are higher in life-cycle cost than any of the conventional or advanced electric or gas equipment examined in most regions. In colder climates the resistance systems are costlier.

A rooftop air conditioner with gas heat is still a lower-cost system for space conditioning of low-rise commercial buildings than an electric heat pump. When the effect of inflation on energy and maintenance costs is taken into consideration, the rooftop air conditioner with gas heat is also cheaper in most locations than the air conditioner with electric resistance heat. Electric heat pumps are also costlier than rooftop air conditioners with resistance heat, except in colder climates.

8. No insurmountable institutional (in contrast to economic) barriers were found that would inhibit the market acceptance of the electric or any other type of heat pump, which is otherwise technically feasible and cost-competitive with market alternatives.

While the electric heat pump in some sectors may still suffer from the image of poor reliability and association with electric resistance heat, this picture is rapidly changing.

In some cases, attempts to legislate against resistance heat may force cost-ineffective alterations in heat pump design or sizing procedures.

9. Review of research needs in electric heat pump technology indicates that a direct role for the Federal Government in electric heat pump research is not called for. However, the Government should support work in areas supportive of technical development and market acceptance and sponsor longer term innovative concepts or products for special markets.

This and previous studies conducted by the authors indicate that the air-conditioning industry itself is capable of and, indeed, most

suited for continuing electric heat pump performance and reliability improvement without significant Federal Government assistance. While the industry must remain sensitive to consumer requirements and market economics, it is expected to produce HVAC products for which there is sufficient demand. However, in many cases heat pump products for special applications or information activities in support of the technology and market could be productively undertaken or supported by the Goverment.

15.1.2 Conclusions Concerning Gas Heat Pumps

1. The free-piston as well as the V-type Stirling-Rankine gas heat pumps could potentially conserve all forms of primary fuel resources including natural gas if primarily used to displace conventional gas warm air furnaces combined with electric air conditioners. If used to displace electric heat pumps, especially of the advanced type, natural gas consumption would increase.

The cooling as well as the heating-mode primary energy efficiency of Stirling-Rankine heat pump is the highest of any of the systems considered in this investigation.

In locations with a large heating and relatively small cooling load, either type Stirling-Rankine heat pump would consume less gas for both heating and cooling than a gas furnace would consume for heating alone. On the other hand, since electric heat pumps are not dependent on specific fuels and since comparatively little natural gas is burned as a power plant fuel for electricity generation in most regions, electric heat pumps would be the preferred system if gas conservation were the objective. The above conclusion, as stated, is based on the types of

fuels presently burned by electric utility power plants. In the future, electric utilities will depend even less on scarce fuels (natural gas and oil). This would enhance the attractiveness of gas furnaces with electric cooling and electric heat pumps vs. gas heat pumps for new construction in regions with a significant cooling load.

2. The organic fluid absorption gas heat pump is also potentially capable of conserving primary energy resources if used to displace gas furnaces with electric air conditioning. However, natural gas savings are obtained only in regions with a low seasonal cooling requirement; in warm regions, significant market penetration of the absorption heat pump would cause substantial increases in natural gas consumption in comparison to the gas furnace-electric air conditioner combination.

The primary reason for the above is the still-too-low cooling COP of the absorption machine in comparison to that of the electric air conditioner. Therefore, development of the former is justifiable only if cooling performance can be cost-effectively improved. Other gas-fired absorption - cycle machines do not appear to be significantly different in equipment performance from the organic fluid absorption heat pumps and, therefore, would show a similar pattern of natural gas and other primary energy consumption.

3. The Brayton-Rankine and the Rankine-Rankine heat pumps are more energy efficient in a primary energy sense than any of the conventional commercial HVAC systems examined. However, if used to displace electric heat pumps, the gas heat pumps would increase the consumption of natural gas in all regions of the country.

The Rankine-Rankine heat pump, as simulated in this study, appears superior in both the heating and the cooling mode to the Brayton-Rankine heat pump. In comparison to the conventional systems, both gas heat pumps consume less total primary energy resources.

4. Unitary absorption cycle heat pumps are, in theory, highly reliable, making the concept attractive for residential applications where reliable, maintenance-free operation is paramount. Thermal engine heat pumps are more complex, offering greater opportunity for failure.

The above conclusion is based solely on theoretical considerations. No operation or even accelerated life testing experience is available. In contrast, the reliability of well-designed electric heat pumps in proper installations is a demonstrated fact.

5. Based on first-cost estimates which may be rather optimistic, the organic fluid gas absorption heat pump as well as the commercial-size thermal engine heat pumps appear to be cost-competitive in a life-cycle sense with conventional electric heat pumps and gas furnaces especially in colder regions of the country. This conclusion may require revision when the experimental systems proceed to a more advanced stage of development.

Due to their high energy efficiency and the comparatively low price of natural gas in comparison to other forms of purchased energy, advanced gas heat pumps appear to be economically attractive in comparison to available alternatives. These include state-of-the-art but not necessarily advanced gas furnaces. More realistic cost data based on operating experience or prototypes and field tests may change this conclusion.

6. An advanced high-efficiency gas furnace, if successfully developed, could pose a serious market threat to both residential and commercial gas heat pumps.

The pulse gas furnace or a furnace with condensing heat exchange in combination with a high-efficiency electric air conditioner would be considerably lower in life-cycle cost then a gas heat pump. Yet the efficiency of the latter would not be sufficiently superior to the former to pay out the added first costs in a reasonable time.

15.2 Recommendations

Based on the research needs identified in Chapter 14 and the conclusions of this study, summarized in the previous section, the following recommendations are presented concerning Federal Government sponsorship of heat pump research and development and other activities in furthering the technological evolution and increased market penetration of these systems.

7. Recognizing that the HVAC industry itself will continue to improve and develop electric heat pump technology in response to market requirements (or mandated efficiency standards), the Federal Government should solicit and sponsor research into those innovative electric heat pump design approaches which, while promising, entail too high a risk or too long a payout for industry to support on its own.

While the conventional hermetic motor-compressor single package or split system heat pump with stationary heat exchangers is expected to remain the basic product of the industry for the foreseeable future, only moderate performance improvement appears possible within the con-

straints of current design and costs. Therefore, innovation in design approaches for higher efficiency, lower cost, and improved reliability or marketability should be solicited and encouraged from a broader spectrum of contributors. However, the intent of this recommendation is not for sponsorship of programs that merely duplicate industry efforts, but of promising but long-range technologically innovative approaches.

2. The Federal Government should solicit and sponsor heat pump designs specifically for refit applications in existing buildings --especially residential housing.

As electric heat pump manufacturers continue to focus their product designs and development on the new construction market, the particular needs and greater problems of existing buildings that could be potential candidates for heat pumps should not be neglected. Since this market segment is much more heterogeneous and poses peculiar technological and cost constraints, it represents a higher risk and probably a lower profit opportunity to the industry.

3. The Federal Government should continue and expand research and development of gas and other novel heat pump systems. Gas heat pump programs should be specifically oriented toward the conversion market as a means of extending scarce gas supplies. Improved cooling performance of absorption systems should be a requirement.

The results of this and other analyses imply that electric heat pumps appear to be the preferred systems for new building applications. Gas heat pumps especially should be developed as efficient alternatives to the conventional gas furnace in existing structures already committed to gas heat. Because of their high efficiency, thermal engine systems should be the preferred ones for commercial buildings where higher first cost can be justified. Absorption systems, because of their potential reliability, should be investigated for residential applications. However, improved cooling mode performance of the latter should be a requirement in order to conserve gas supplies.

4. The Federal Government should sponsor fundamental and applied research into heat pump system and component reliability, establish a clearinghouse for in-service failure rate data available to consumer decision makers, and encourage the industry to establish minimum equipment reliability testing procedures and standards.

Since the HVAC systems for the future, even for residential buildings, are likely to be increasingly complex, greater stress should be placed on an understanding of the causes of equipment failure and failure statistics. At the present time this information is too fragmented and, as far as the equipment of specific manufacturers is concerned, unavailable to all but the manufacturers themselves. Formal standards are also indicated and the consumer should be given a simple means of differentiating good equipment from poor.

5. The Federal Government should sponsor research and development of equipment to meet specific regional needs or the needs of market segments unlikely to be fully served by the industry.

Large regions of the United States have rather special space conditioning needs due to climate and the architectural style it has evolved. Equipment development should consider these special needs: heat pumps with evaporative precoolers, hybrid evaporative cooler-heat pump combined systems, heat-only heat pumps, etc.

6. The Federal Government should assist in solving technical problems affecting the entire industry (some of which may result from regulatory actions such as the possible banning of fluorocarbon refrigerants) as well as sponsor supporting technical and market or consumer-oriented research.

These studies include ozone-nonreactive refrigerant development, consumer attitude surveys, market identification and penetration analyses and similar investigations of benefit to both the public and private sectors in achieving a better understanding of the full potential of heat pump technology.

7. The Federal Government should closely coordinate among its branches the sponsorship of all building energy system oriented research--solar energy building HVAC systems (especially solar heat pumps), thermal storage as well as (non-solar) heat pump development--to avoid duplication of activities.

The ERDA Solar Energy Division, especially, is sponsoring a number of projects involving heat pumps and has solicited more. Although it is stated that special temperature environments are encountered when heat pumps are combined with solar thermal energy collection systems, it is believed that conventional equipment can be (and some of it already has been) developed to have the added flexibility required. Dual-track heat pump development programs tapping the technical resources of essentially the same industry could be counter-productive and confusing.

APPENDIX A

THE ERICSSON - ERICSSON GAS HEAT PUMP

This appendix presents results for the on-site and primary energy consumption of the Ericsson - Ericsson gas heat pump. Because the machine is still in the proof-of-concept stage, it was decided to include results for this system in a separate appendix.

Exhibit A-1 presents results of the on-site energy consumption analysis. Comparing these results to those of Chapter 8, it can be seen that the system is extremely energy efficient, having heating season SPF's ranging from 2.12 to 2.51 and cooling season SPF's of 2.92 to 3.15. Since the system operates entirely with natural gas, low primary energy consumption and high primary energy SPF's would be expected. These are shown in Exhibit A-2. Primary energy SPF ranges from 1.96 to 2.32 for heating and 2.70 to 2.92 for cooling.

EXHIBIT A-1

PERFORMANCE DATA FOR ERICSSON-ERICSSON GAS HEAT PUMP: ON-SITE BASIS

	On-Site Gas Consumption (cft)			On-Site Energy Consumption (Btu x 10^{-6})			Seasonal Performance Factor		
City	Heating	Cooling	Total Annual	Heating	Cooling	Total Annual	Heating	Cooling	Total Annual
Houston	4,079	16,579	20,658	4.08	16.58	20.66	2.51	2.92	2.84
Birmingham	10,576	11,363	21,939	10.58	11.36	21.94	2.36	2.93	2.66
Atlanta	10,701	10,545	21,246	10.70	10.55	21.25	2.38	2.97	2.67
Tulsa	19,793	8,612	28,310	19.80	8.51	28.31	2.28	2.99	2.49
Philadelphia	21,047	6,696	27,743	21.05	6.70	27.74	2.31	3.02	2.48
Seattle	17,698	2,240	19,838	17.60	2.24	19.84	2.45	3.15	2.54
Columbus	26,656	6,302	32,958	26.66	6.30	32.96	2.23	2.98	2.37
Cleveland	31,674	4,792	36,466	31.67	4.79	36.47	2.23	3.06	2.34
Concord	36,895	3,018	39,913	36.90	3.02	39.91	2.12	3.05	2.19

EXHIBIT A-2

PERFORMANCE DATA FOR ERICSSON-ERICSSON GAS HEAT PUMP: PRIMARY ENERGY BASIS

City	Primary Energy Consumption ($Btu \times 10^{-6}$)			Seasonal Performance Factor		
	Heating	Cooling	Total Annual	Heating	Cooling	Total Annual
Houston	4.41	17.91	22.31	2.32	2.70	2.63
Birmingham	11.42	12.27	23.69	2.19	2.71	2.46
Atlanta	11.56	11.39	22.95	2.20	2.75	2.77
Tulsa	21.38	9.19	30.57	2.11	2.77	2.34
Philadelphia	22.73	7.23	29.96	2.14	2.80	2.30
Seattle	19.01	2.42	21.43	2.28	2.92	2.35
Columbus	23.79	6.81	35.59	2.06	2.76	2.20
Cleveland	34.21	5.18	39.38	2.06	2.83	2.17
Concord	39.85	3.26	43.11	1.96	2.82	2.03

APPENDIX B

ON-SITE ENERGY REQUIREMENTS FOR RESIDENTIAL SPACE CONDITIONING

EXHIBIT B-1

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY
ELECTRIC HEAT PUMP - STANDARD

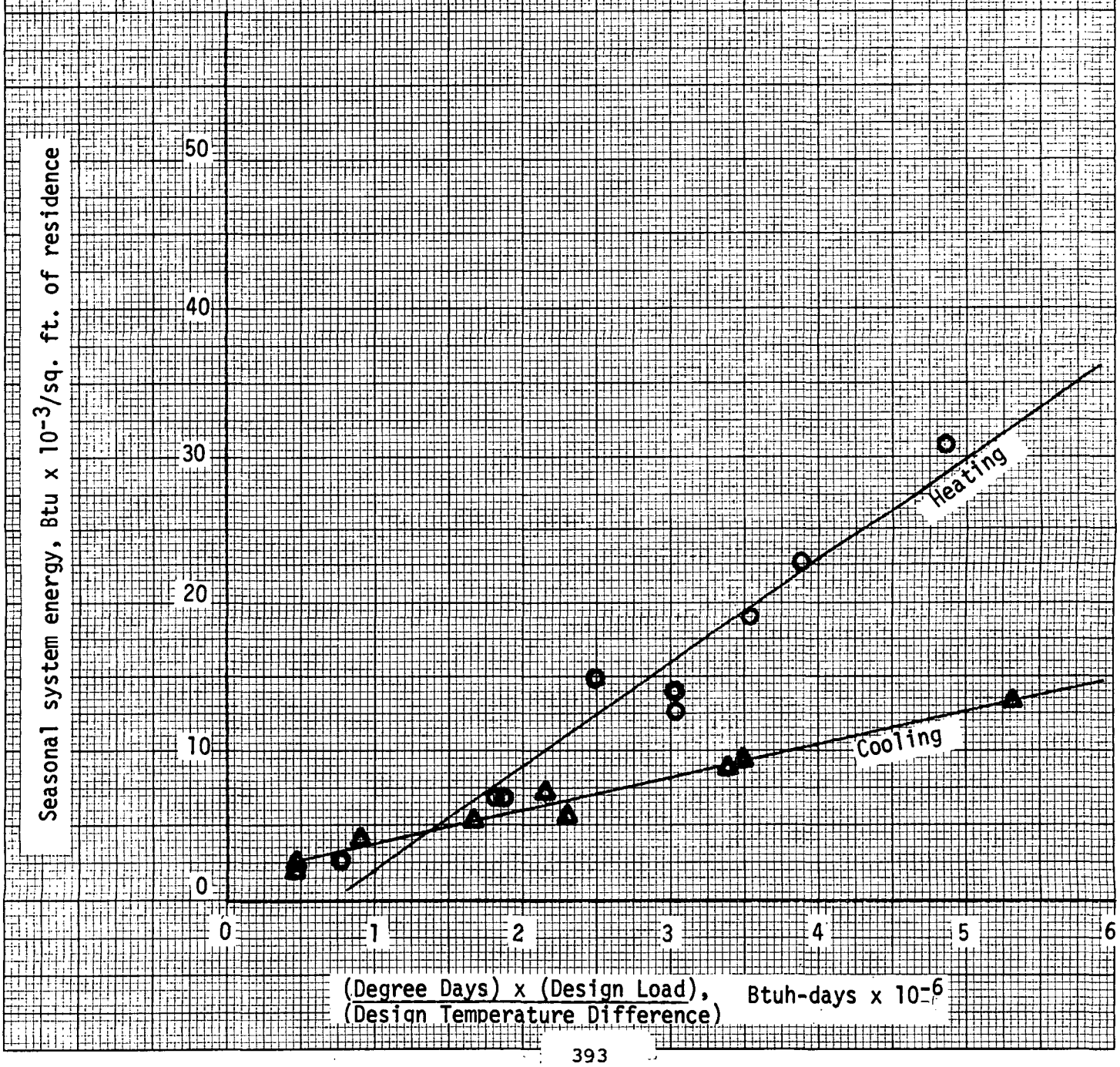


EXHIBIT B-2

CORRELATION OF RESIDENTIAL HEATING ENERGY
ELECTRIC HEAT PUMPS - HIGH EFFICIENCY SYSTEMS

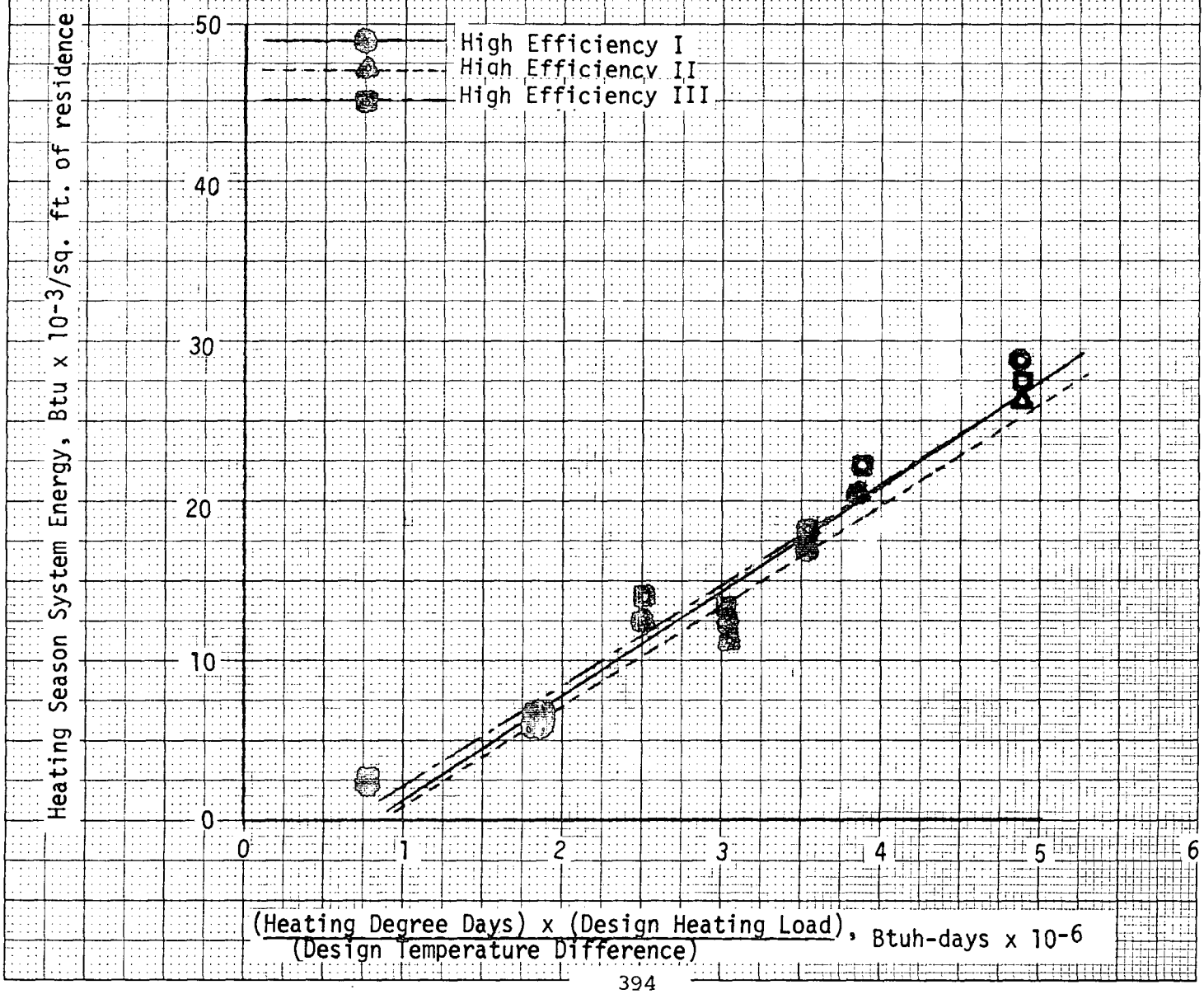


EXHIBIT B-3

CORRELATION OF RESIDENTIAL COOLING ENERGY
ELECTRIC HEAT PUMPS - HIGH EFFICIENCY SYSTEMS

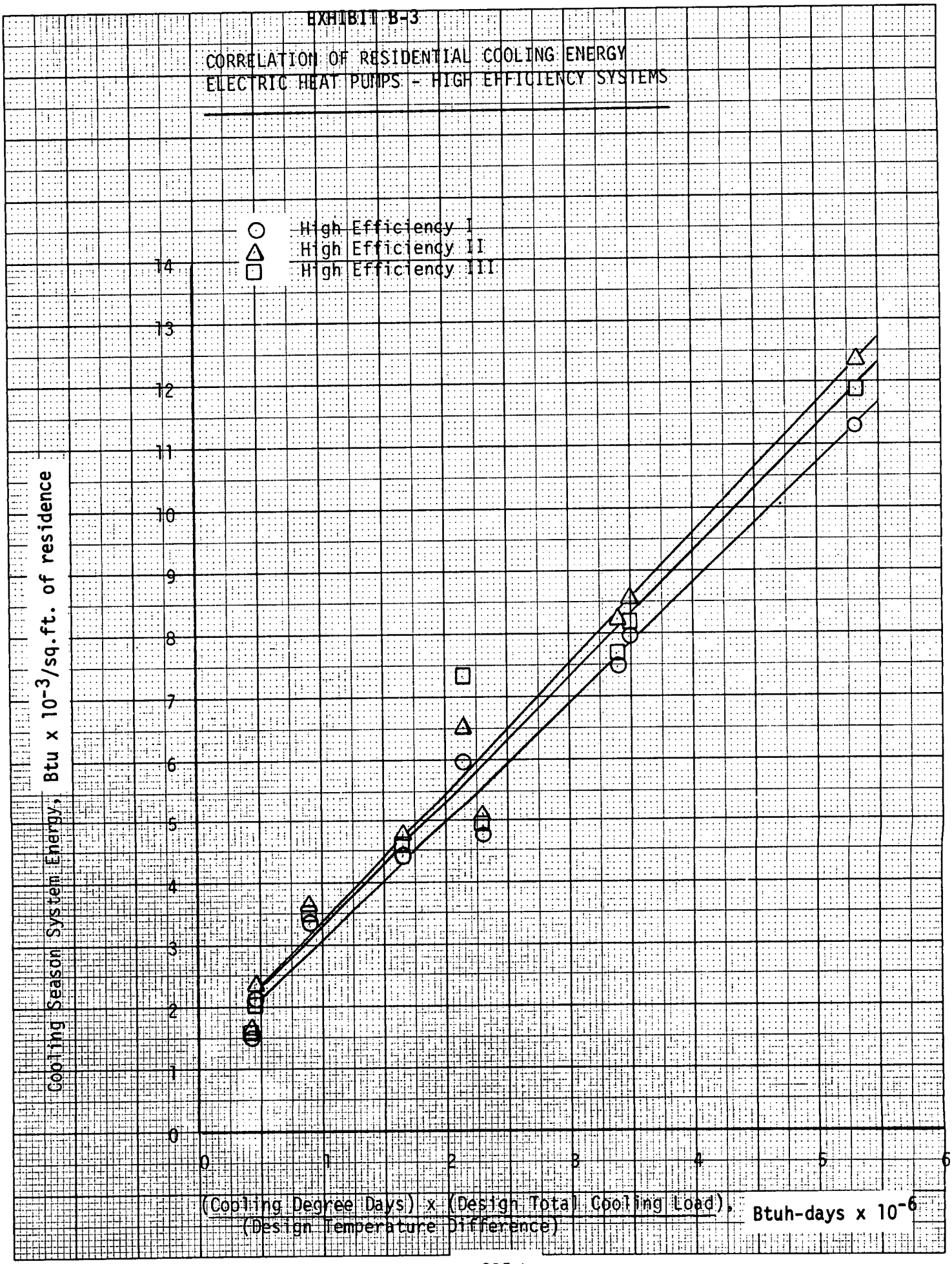


EXHIBIT B-4

CORRELATION OF HEATING AND COOLING ENERGY
HYBRID ELECTRIC HEAT PUMP - GAS OR OIL FURNACE

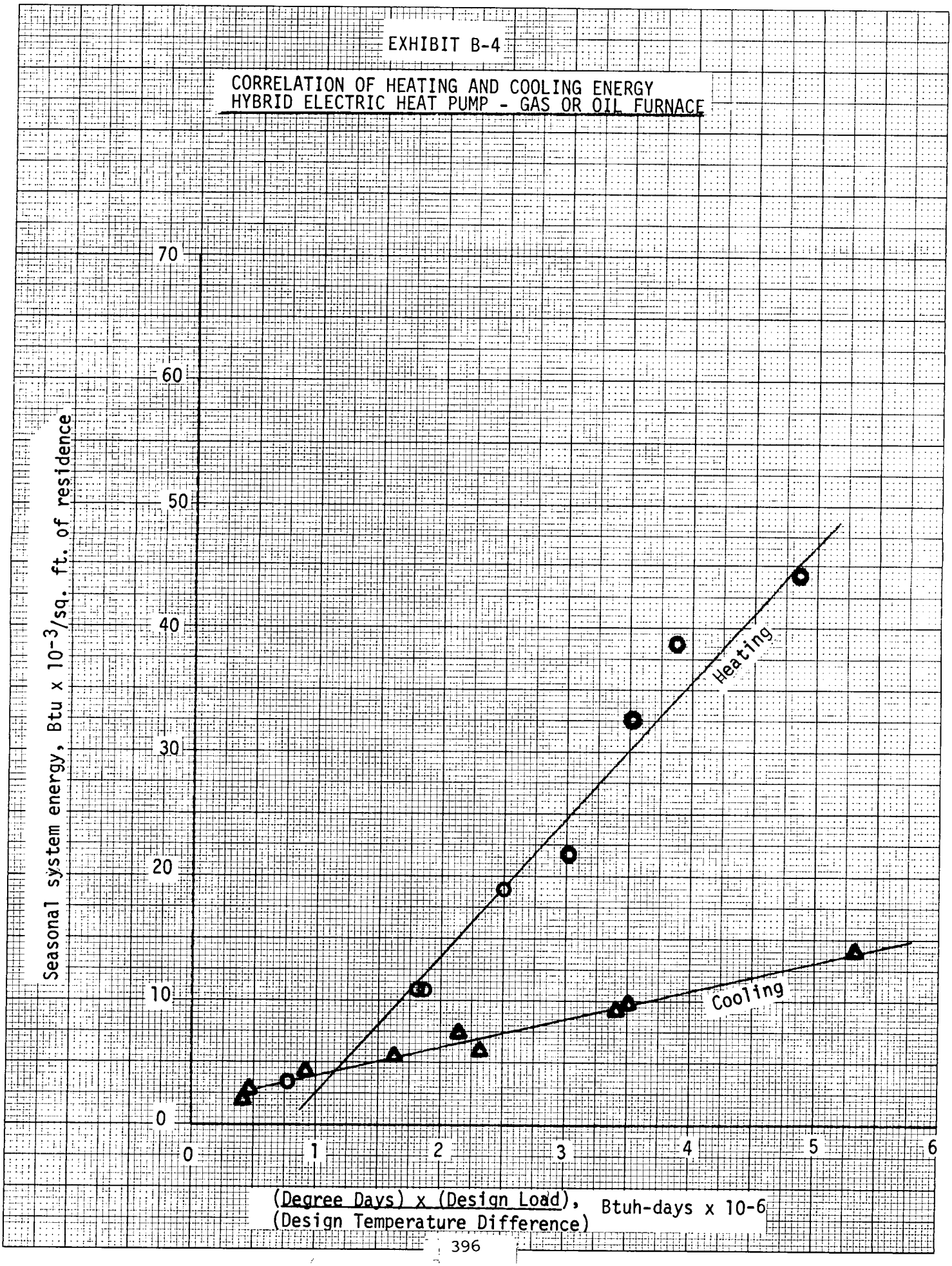


EXHIBIT B-5

CORRELATION OF HEATING AND COOLING ENERGY ELECTRIC HEAT PUMP - HEAT ONLY AND CENTRAL AIR CONDITIONER- HIGH EFFICIENCY

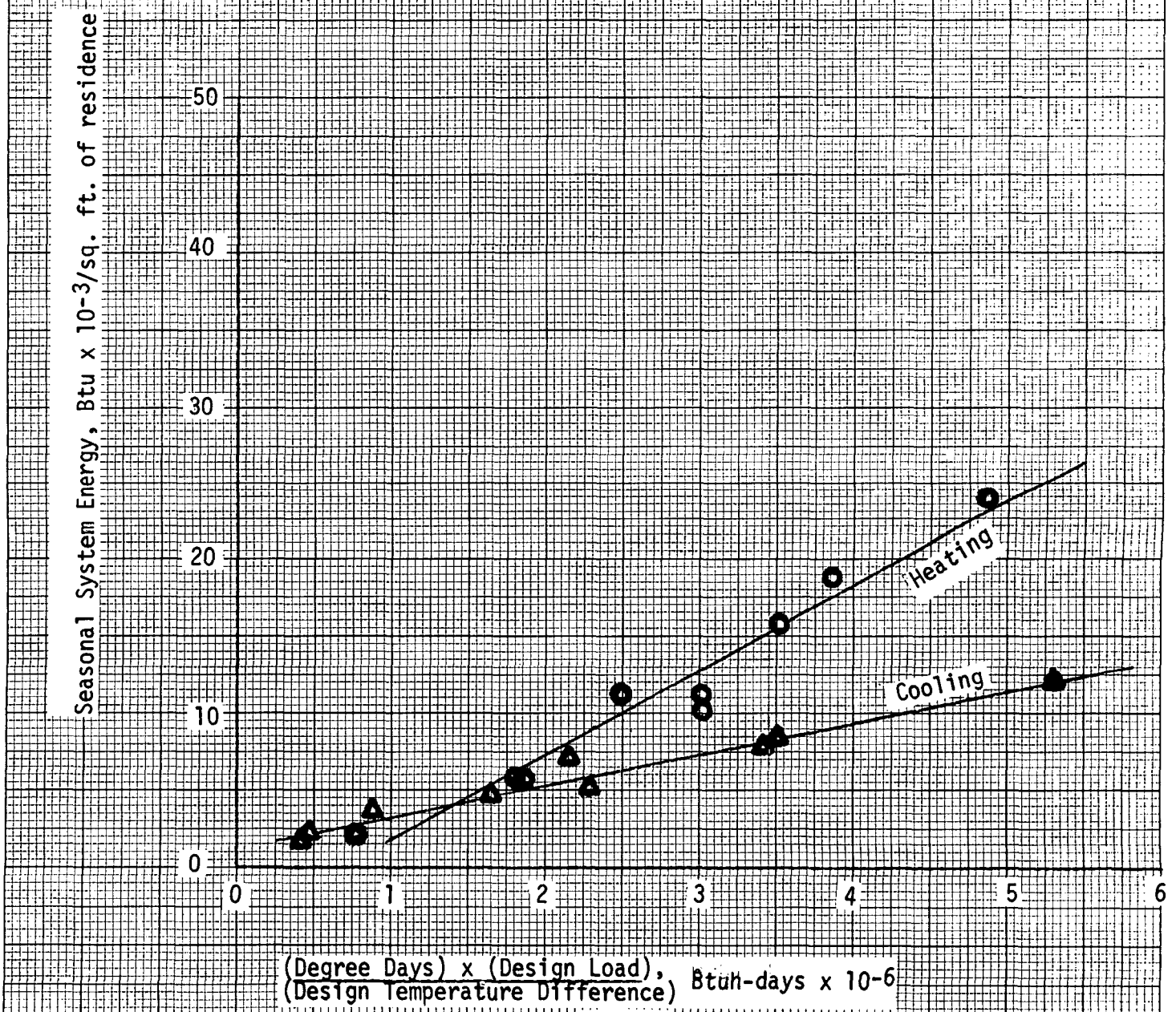


EXHIBIT B-6

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY GAS OR OIL WARM AIR FURNACE AND CENTRAL AIR CONDITIONER - HIGH EFFICIENCY

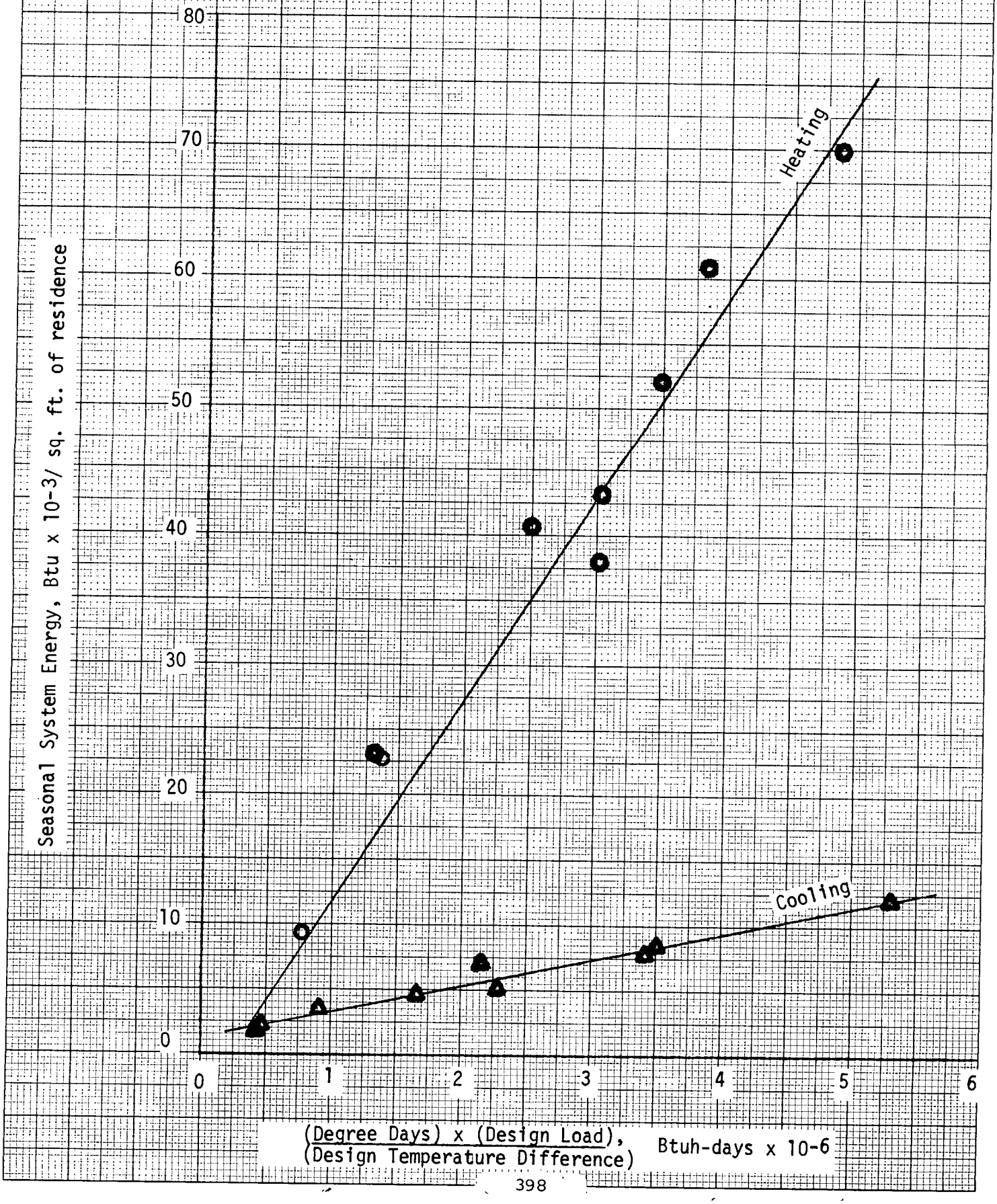


EXHIBIT B-7

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY

CENTRAL ELECTRIC FURNACE AND CENTRAL AIR CONDITIONER - HIGH EFFICIENCY

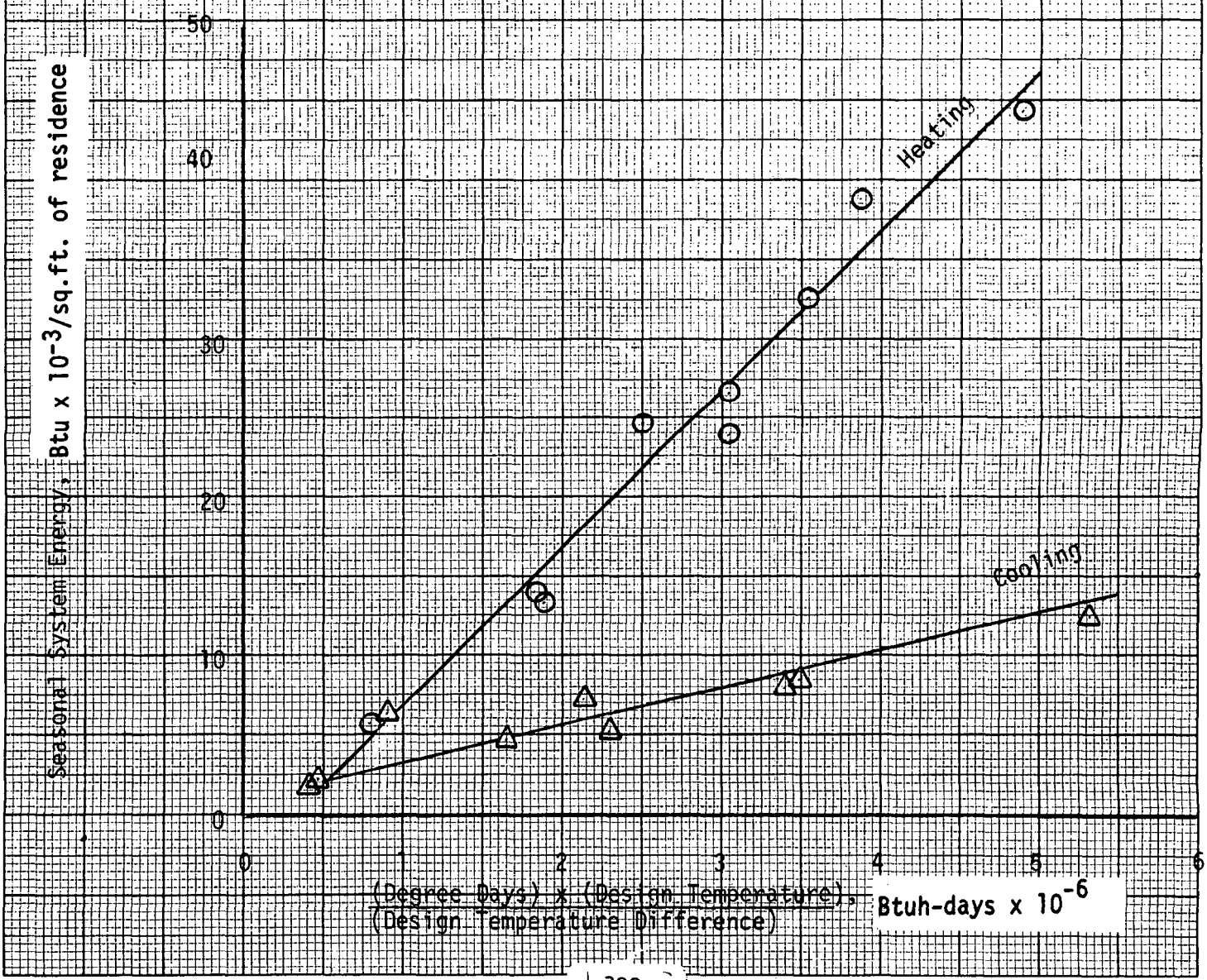


EXHIBIT B-8

CORRELATION OF HEATING AND COOLING ENERGY

BASEBOARD CONVECTORS AND ROOM AIR CONDITIONERS

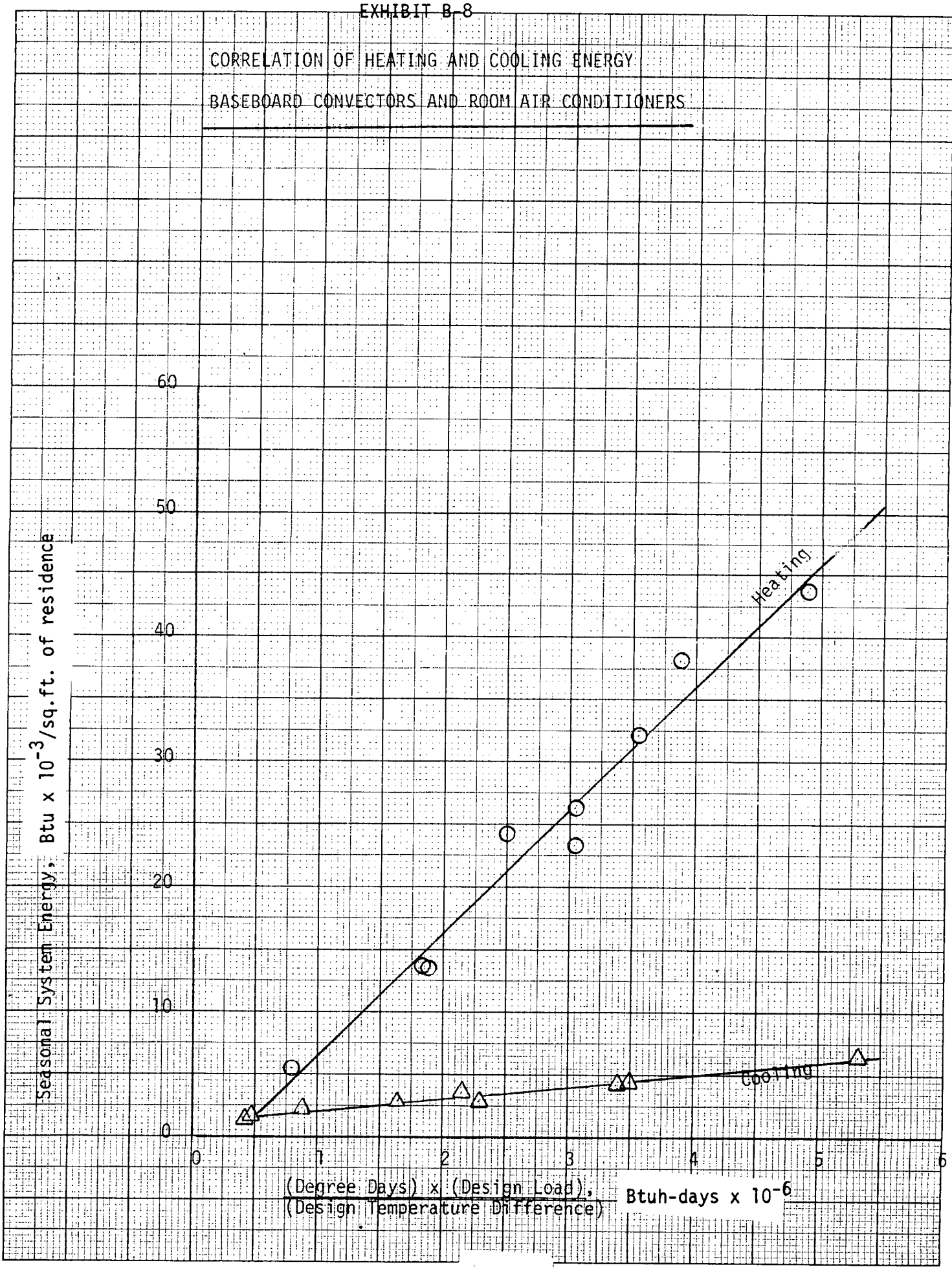


EXHIBIT B-9

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY
ADVANCED ELECTRIC HEAT PUMP I

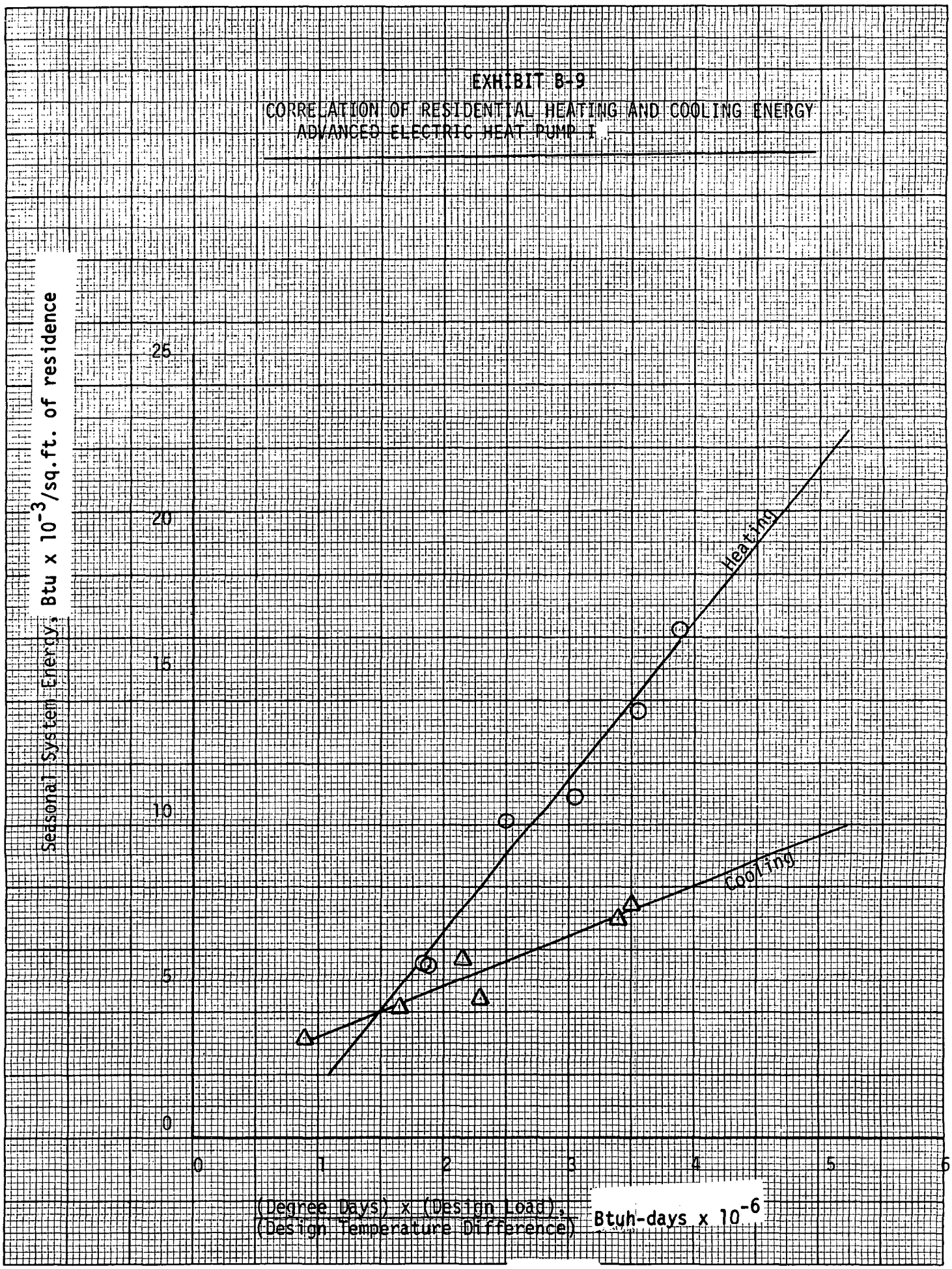


EXHIBIT B-10

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY
ADVANCED ELECTRIC HEAT PUMP II

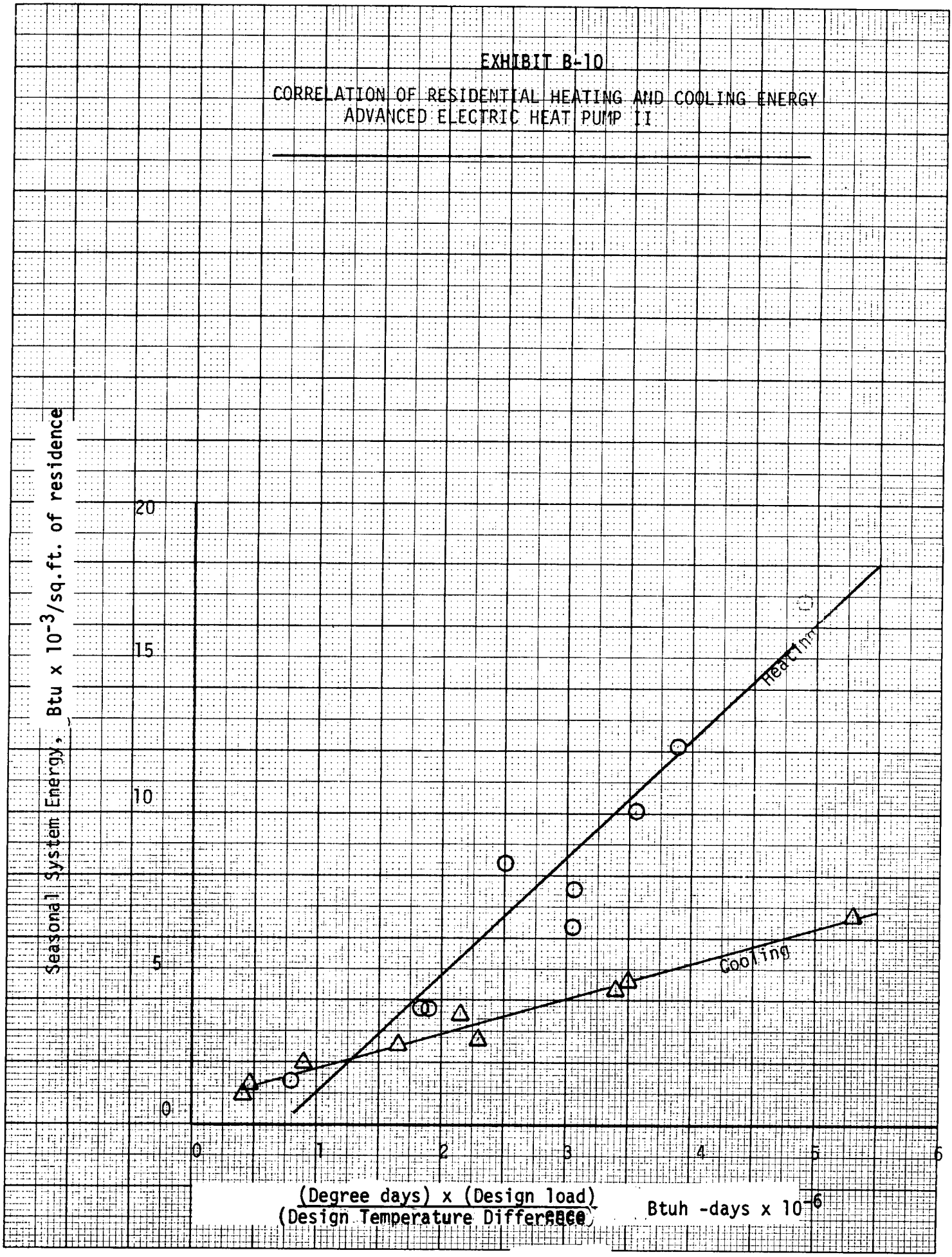
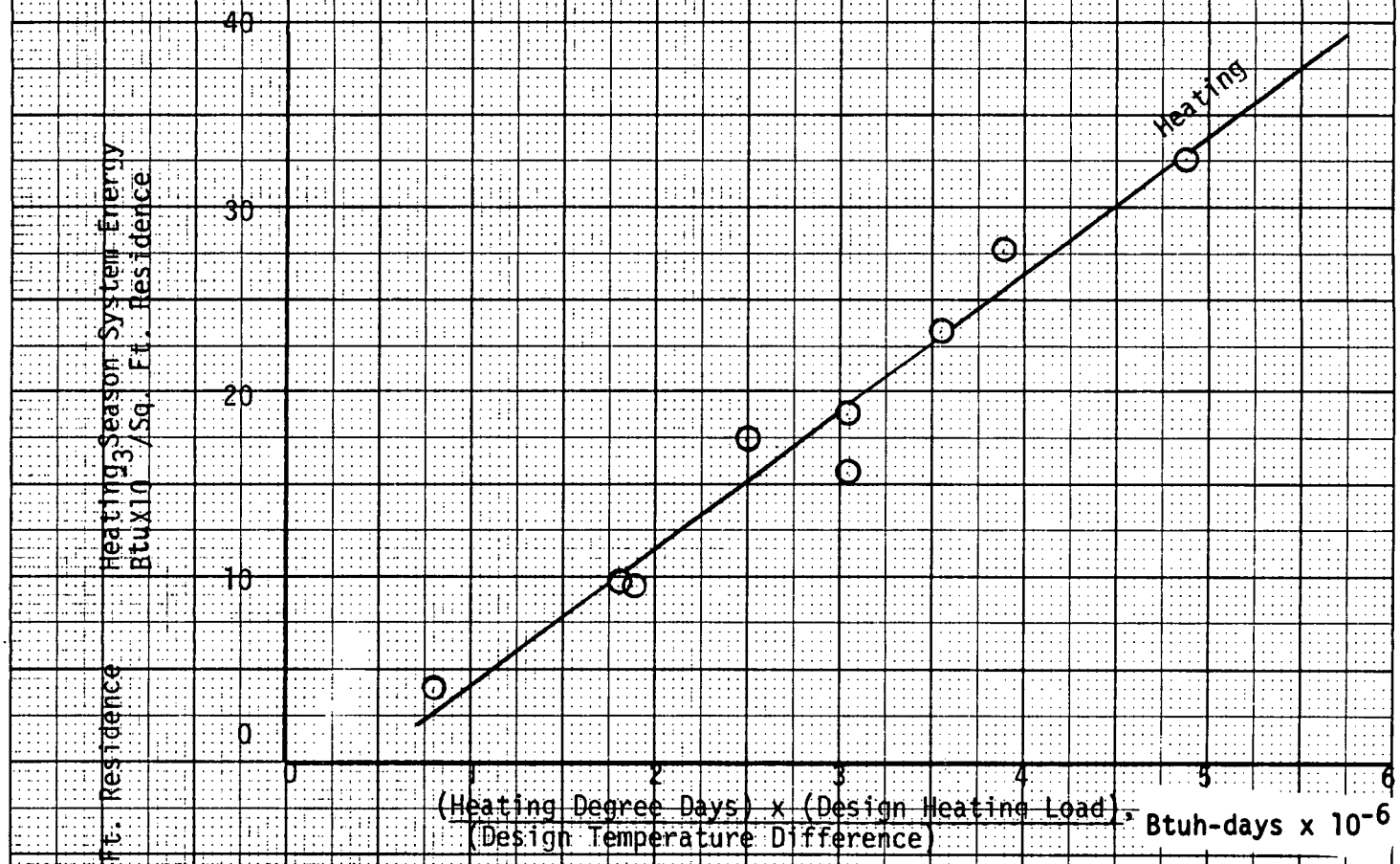


EXHIBIT B-11

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY
FREE PISTON STIRLING-RANKINE GAS HEAT PUMP



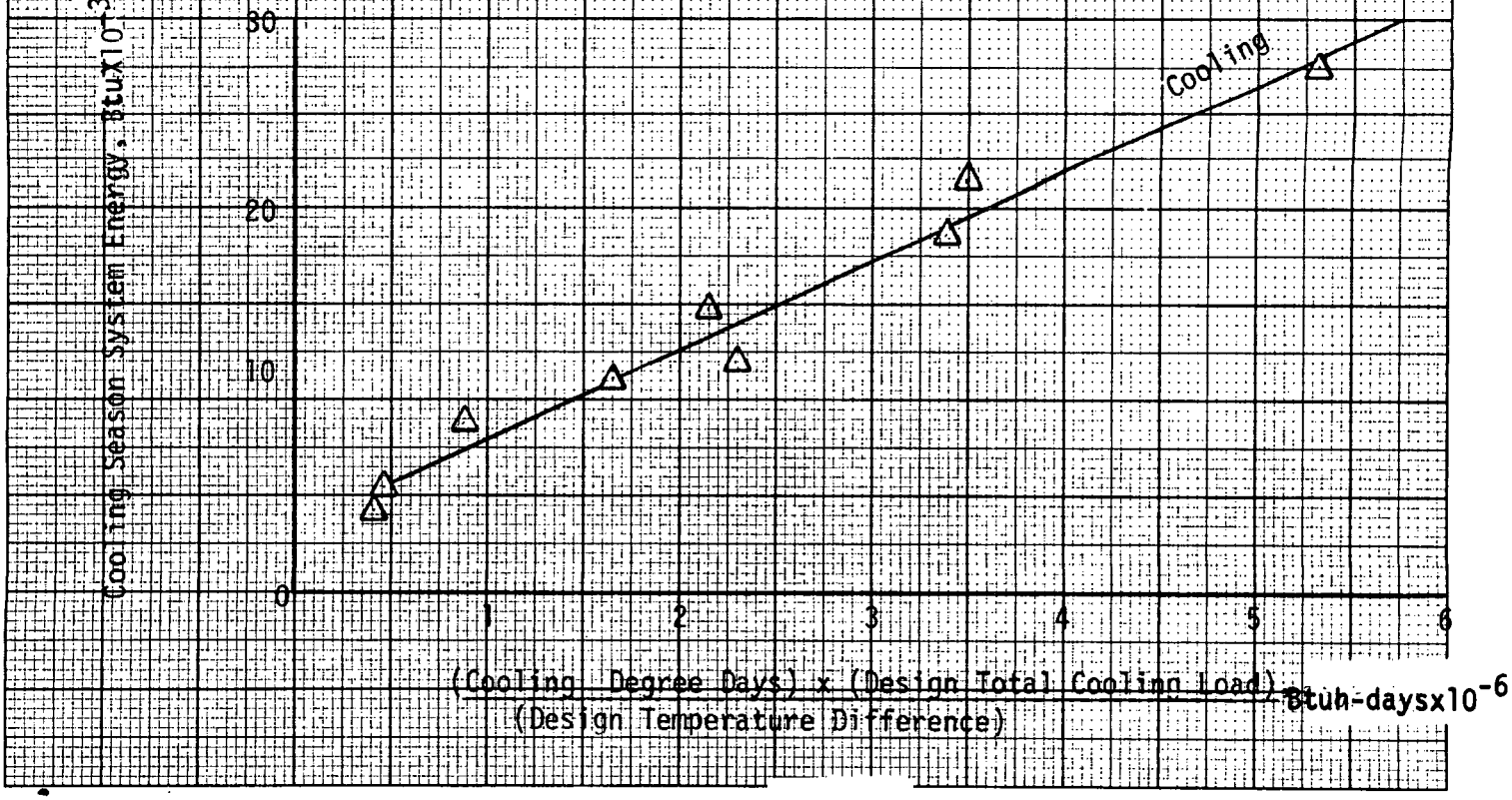
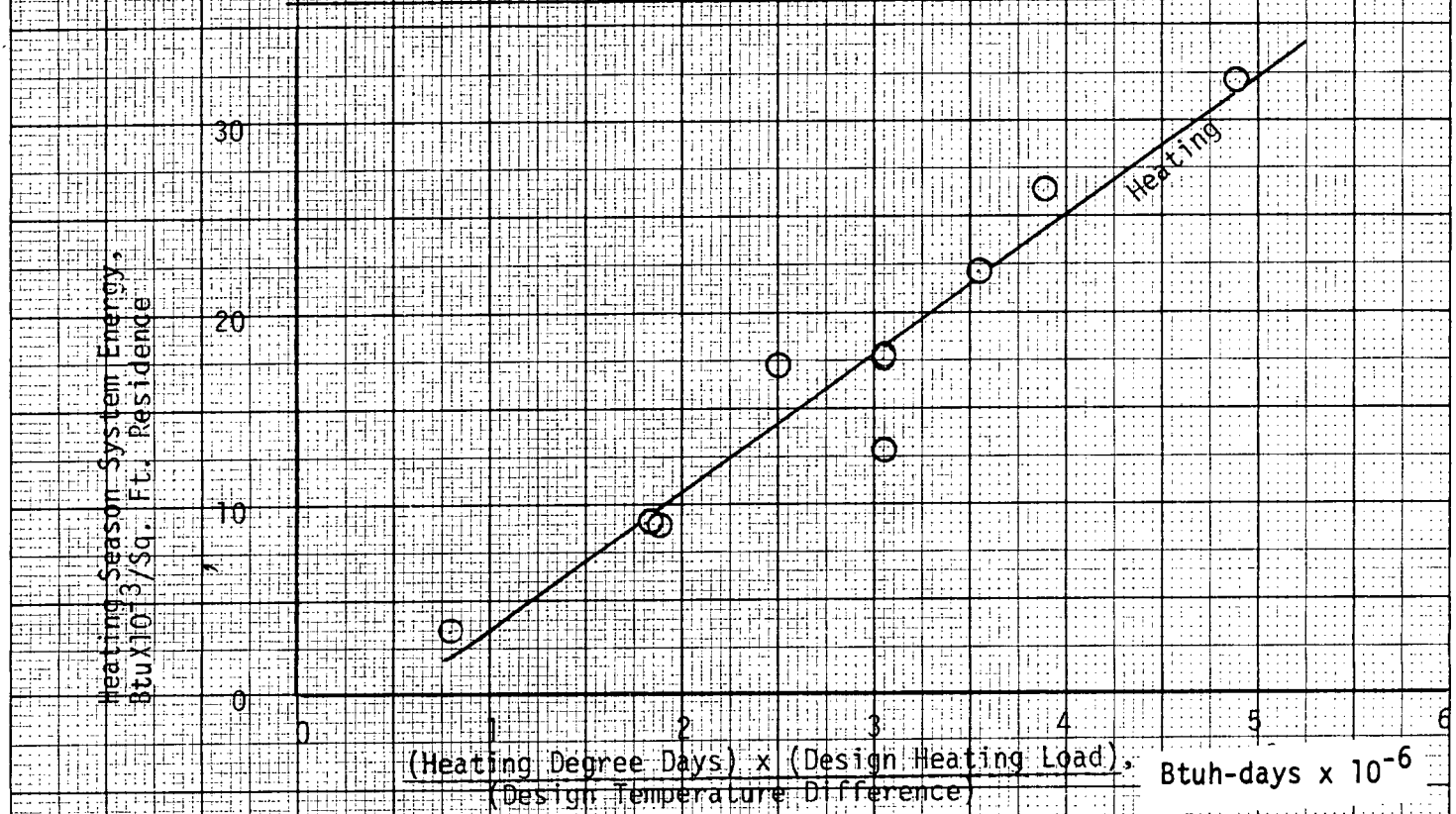


EXHIBIT B-12

CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY
V-TYPE SINGLE CYLINDER STIRLING-RANKINE GAS HEAT PUMP



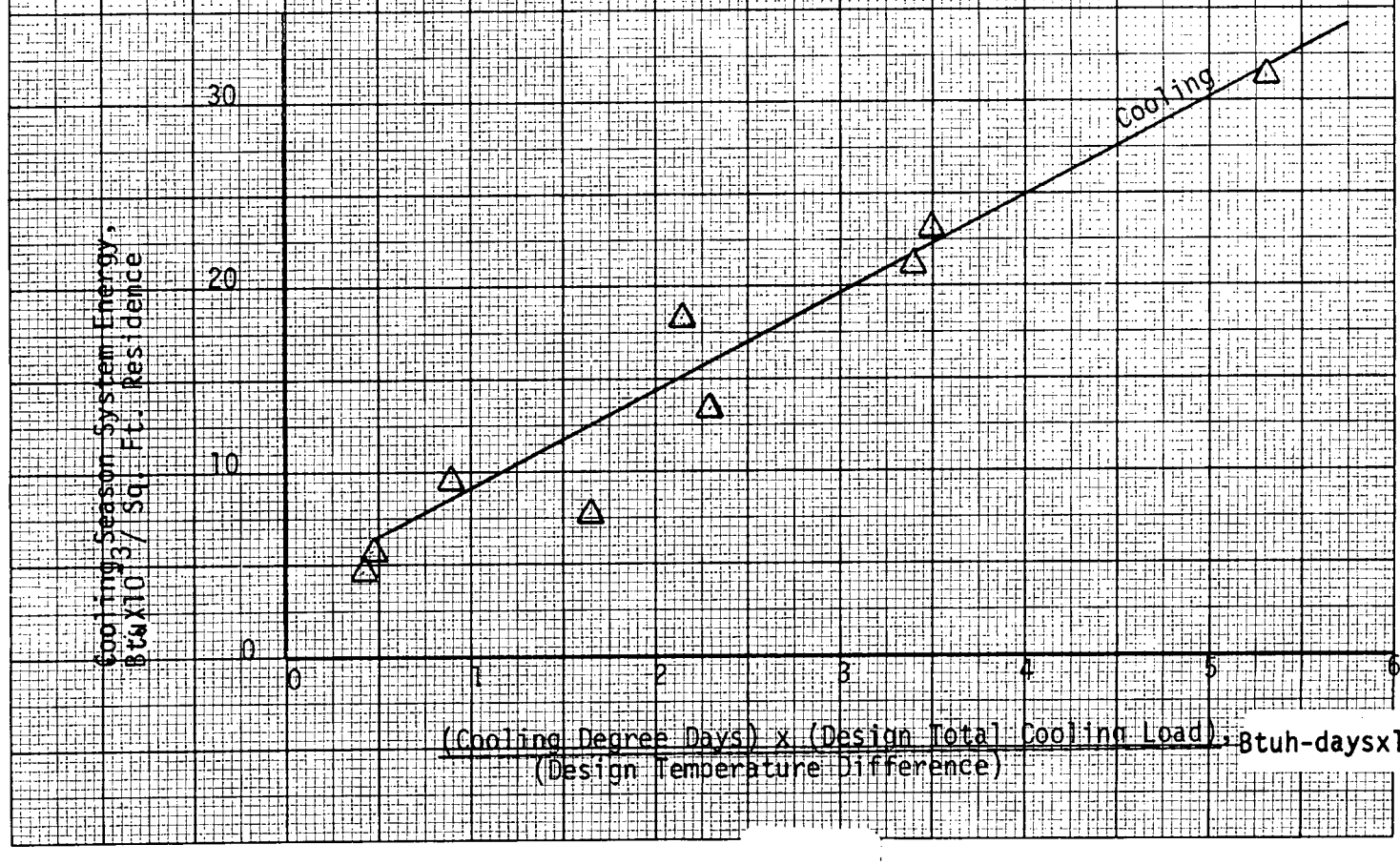
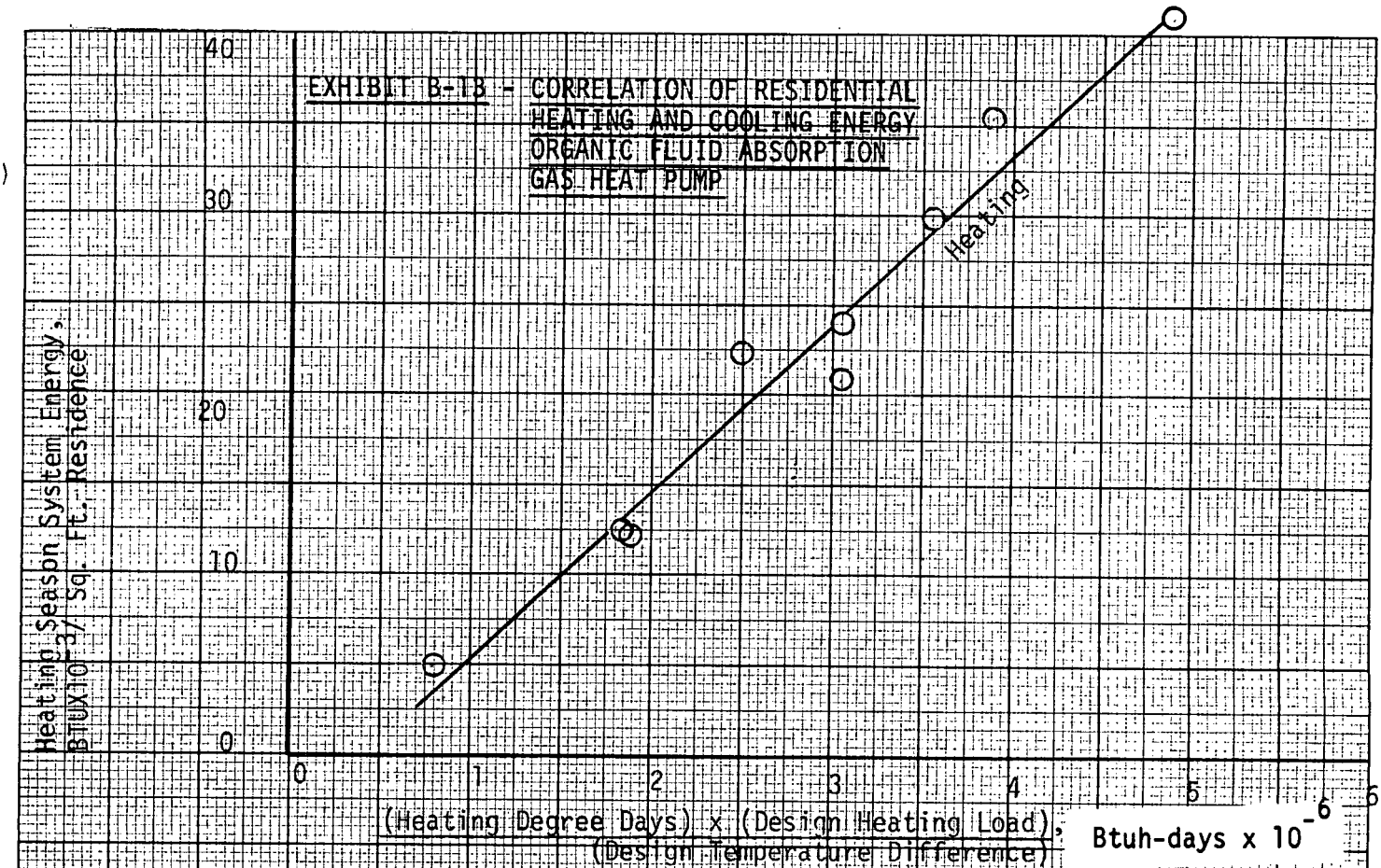


EXHIBIT B-13 - CORRELATION OF RESIDENTIAL HEATING AND COOLING ENERGY ORGANIC FLUID ABSORPTION GAS HEAT PUMP



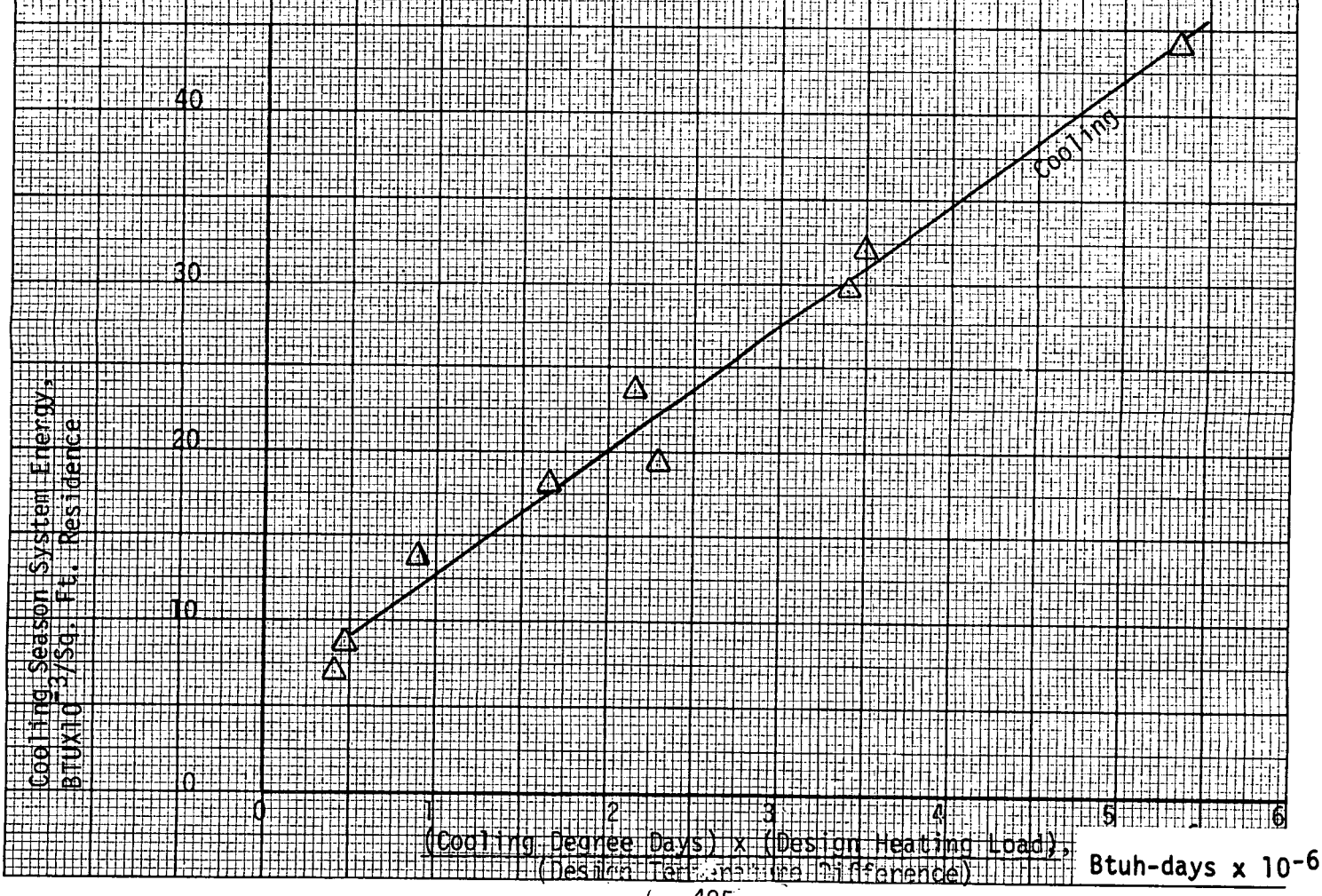
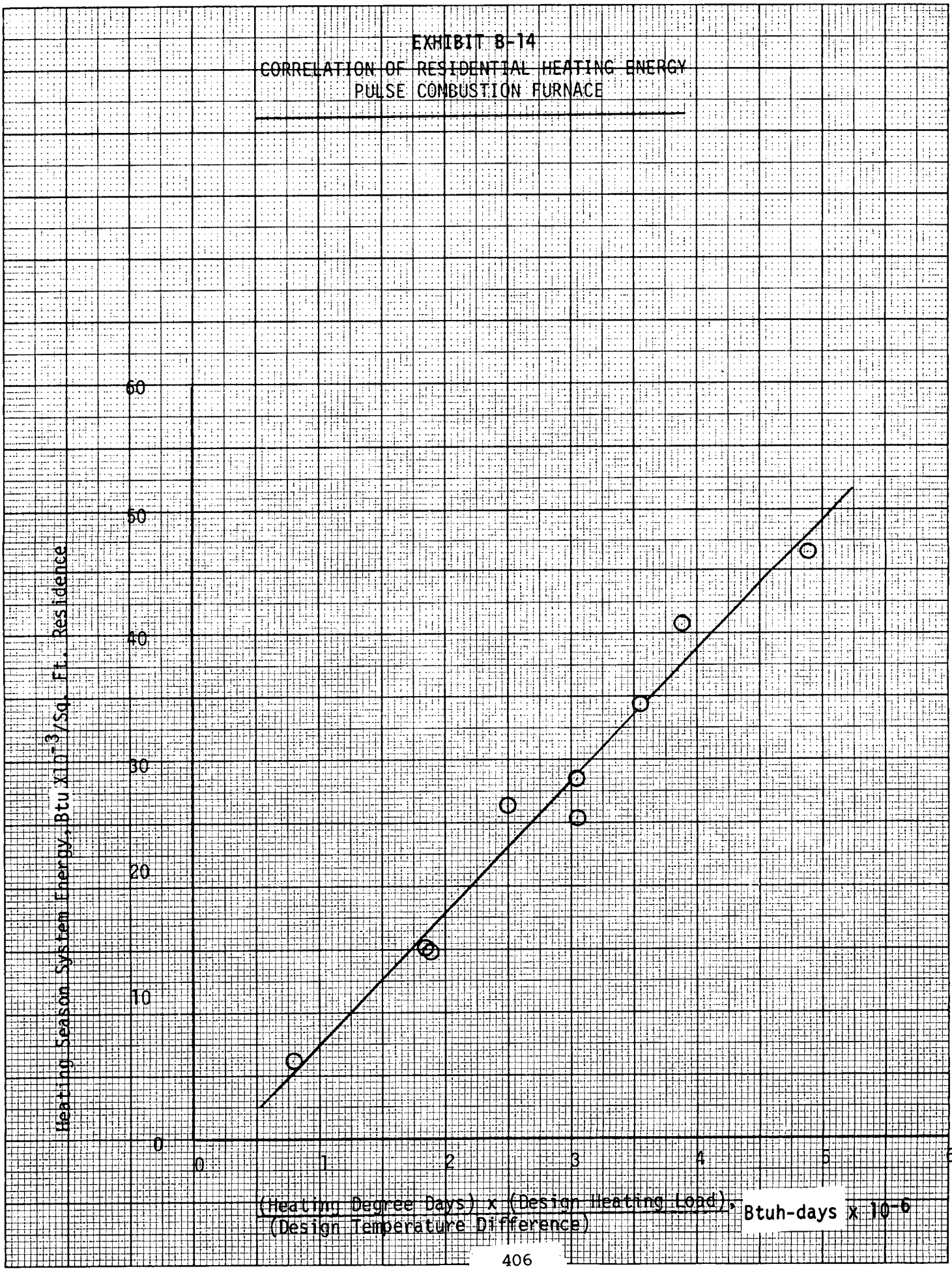


EXHIBIT B-14

CORRELATION OF RESIDENTIAL HEATING ENERGY
PULSE COMBUSTION FURNACE



APPENDIX C

HVAC SYSTEM PERFORMANCE SIMULATION

C.1 Residential Building HVAC System Simulation

As already described in Chapter 8, the calculation of residential building HVAC system energy consumption was accomplished in two parts: (a) a detailed hour-by-hour simulation of the building thermal load for 8760 hours of a typical weather year followed by (b) a calculation of the electrical energy and/or fossil fuel consumed by the HVAC system in meeting the building thermal (i.e., heating or cooling) load. The latter calculation was based on heat pump heating and cooling mode performance data provided by manufacturers,corrected to account for such factors as system cycling due to part load operation, defrost energy consumption (heat pumps), combustion air infiltration (fossil-fuel-fired furnaces), and ductwork heat losses and gains. The following paragraphs describe some of the features of the HVAC system simulation models.

C.1.1 Electric Heat Pump-Heating Mode

Manufacturers' integrated performance data were used in the hourly simulation of electric heat pump performance. These data were corrected to reflect the effects of frost buildup on the outdoor coil, and system cycling on system energy consumption. Although manufacturers' data include the effect of defrost cycling, results of experimental measurements by National Bureau of Standards indicate that further corrections are required to more accurately simulate system performance. Data from Parken *et al.* (1977)* were used in modeling system performance in the

* See References to Chapter 8.

defrost region (outdoor dry bulb temperature below 47°F with coincident outdoor relative humidity 60% or greater); data from Kelly and Bean (1976) were used in modeling system performance outside of the defrost region.

In the defrost region, manufacturers' integrated capacity and COP data represent time averages of these functions over an integral number of heating and defrost cycles. Factors detrimental to system performance include cycling, evaporator frosting, and the cooling effect of the defrost cycle. The compressor capacity and COP correction factors from Parken's data, shown graphically in Exhibit C-1,represent the ratio of observed to manufacturers' specified heating capacity on COP; these data were based on measurements taken on a heat pump in a laboratory investigation. The application of these corrections to the hourly calculational procedure is described below.

> The capacity and COP correction factors were computed for the outdoor temperature observed for the hour under investigation. Compressor on-time was computed as the ratio of house load and corrected compressor capacity (less losses). If house load exceeded compressor capacity, supplemental resistance heat was required to make up the difference. The system energy consumption was computed as the sum of manufacturers' specified input power requirements for the compressor and indoor and outdoor blowers multiplied by the on-time and by the ratio of compressor and COP correction factors. Supplemental resistance heat (if required) was added to this, as was the defrost energy requirement (the defrost energy calculation is described below).

Supplemental resistance heat was used to temper the air to maintain comfort conditions. The simulation model used in this study did not model the heat pump refrigerant circuit from first principles. Thus refrigerant temperature conditions in the outdoor coil for defrost initiation and termination control could not be supplied directly. Instead, an alternative

method was used, based on a 90 minute timed cycle procedure, with the duration of each defrost cycle having an average length of 3 1/2 minutes. The simulation model used accounts for a defrost cycle when, at the end of each 90 minute period, the outdoor dry bulb temperatures is less than 47°F with coincident outdoor relative humidity of 60% or greater. [Groff and Bullock's (1976) computer model utilizes a 1.8 minute defrost cycle; however, their measurements of defrost cycle duration and number for a residence in Minneapolis give an average defrost cycle length of 3.34 minutes. The 3 1/2 minute defrost cycle used in Gordian's simulation probably gives an overestimate of defrost cycle energy consumption.]

For systems with single stage electric resistance supplemental heat, initiating a defrost cycle caused the resistance element to be energized. For systems with two stages of supplemental resistance heating, the second stage was energized provided the first stage was already in operation.

For all other temperatures and coincident relative humidities (the no-defrost region), only cycling losses were assumed to be encountered, which were accounted for by a COP correction, given as a function of heating part load factor based on the data of Kelly and Bean. The correction factor is shown graphically in Exhibit C-1. The application of the correction factor to the hourly calculational procedure is given below.

> The heating part load factor was computed from the house load, duct losses and manufacturer's rated compressor capacity at the outdoor temperature for the hour. The COP correction factor was then computed. The compressor on-time was calculated as the ratio of house load and capacity. System energy consumption was calculated as the sum of compressor indoor blower and outdoor blower multiplied by the fractional on-time. If house load exceeded heat pump heating capacity, supplemental resistance heat to make up the difference was added.

C.1.2 Electric Heat Pump - Cooling Mode and Air Conditioners

Summer air conditioning involves two simultaneously occurring processes: the reduction of air temperature and water content (humidity) in the conditioned space. Equipment manufacturers list both sensible and total capacities for air cooling equipment, as functions of indoor wet bulb and indoor dry bulb temperature; the difference between total and sensible cooling capacity is a measure of the maximum extent to which entering air can be dehumidified (the latent capacity). If the full latent capacity is not utilized, air may be cooled in excess of the rated sensible capacity. The simulation model uses this effective sensible capacity,computed from manufacturers' performance data.

In the cooling mode it was assumed that only cycling losses are experienced. Data from Kelly and Bean were used to correct manufacturer's cooling performance data to reflect these losses. The COP correction factor used is shown graphically in Exhibit C-1 as a function of cooling part load factor. The application of this correction factor to the calculational method employed is discussed below.

> The manufacturers' effective sensible capacity was computed as described previously. The cooling part load factor was then computed and the COP correction factor calculated and applied to the effective sensible capacity. The system on-time is the ratio of sensible load to corrected sensible capacity, and was used to compute the total system energy consumption by multiplication by the sum of compressor indoor blower and outdoor blower input power requirement.

C.1.3 Fossil-Fuel-Fired Furnaces

Fossil-fuel-fired furnace ratings are based on full load operation. In reality, the performance of these furnaces suffers from the effects of such factors as extra infiltration due to the presence of

the chimney and flue, and cycling losses. Data and algorithms of Macriss et al. (1977) were used for the simulation of gas furnace performance and those of Bonne and co-workers (Bonne et al., 1976) for the simulation of oil furnace performance. Part load efficiency curves for the gas and oil warm furnaces as a function of the heating part load factor are shown in Exhibit C-2.

The simulation involved performing an energy balance around the furnace:

Furnace Heat Output = House Load + Chimney and Flue Losses + Duct Losses

From this relationship, the heating part load factor (numerically equal to furnace on-time) and hence, the part load efficiency was determined. The fuel consumption was calculated as the product of the manufacturers' rated fuel input (Btuh) and time on, divided by the fuel heating value (1000 Btu/cft for natural gas; 139,000 Btu/gal for No. 2 fuel oil). Indoor blower time on was computed as a function of furnace on time. The relationship between furnace time on and indoor blower time on is shown in Exhibit C-3.

C.2 Commercial Building HVAC System Simulation

The AXCESS Version "6" building thermal load and HVAC system energy consumption model was used for analysis of the commercial building HVAC systems. As documentation of that model is already available to users, only the means of HVAC system part-load operation will be described here.

C.2.1 Electric Heat Pump-Heating and Cooling Mode

In AXCESS electric heat pumps in heating mode operation are characterized by a user-specified maximum output/maximum input rating. Provision is also made for modification of the rated values by two outdoor-

temperature-dependent functions: one giving the fraction of maximum output and another giving the fraction of maximum input. These functions were constructed from manufacturer's detailed performance data and the part load data of Kelly and Bean (1976). The latter include the effect of one stage of resistance heat but probably penalize the performance too severely at low temperatures. There is no other means of modelling defrost with AXCESS at this time.

For cooling mode operation, the user is required to specify, in addition to fixed rated input and output, a full-load system COP as a function of temperature and a part load curve consisting of a functional relationship between fractional input as a function of fractional load (pressured to equal the output). The COP vs. temperature function was constructed from manufacturers' performance data; the part load curve from Kelly and Bean (1976). Although, AXCESS has no specific gas heat pump simulation model, the Rankine-Rankine heat pump performance data obtained from CNG were detailed enough to be cast into the required form for the electric heat pump. Part-load data were unavailable for the Brayton-Rankine heat pump, hence the variation of capacity and COP for that system was modelled only on a full-load-operation basis. In neither case was there an ability to account for defrost cycling.

C.2.2 Electric Air Conditioning

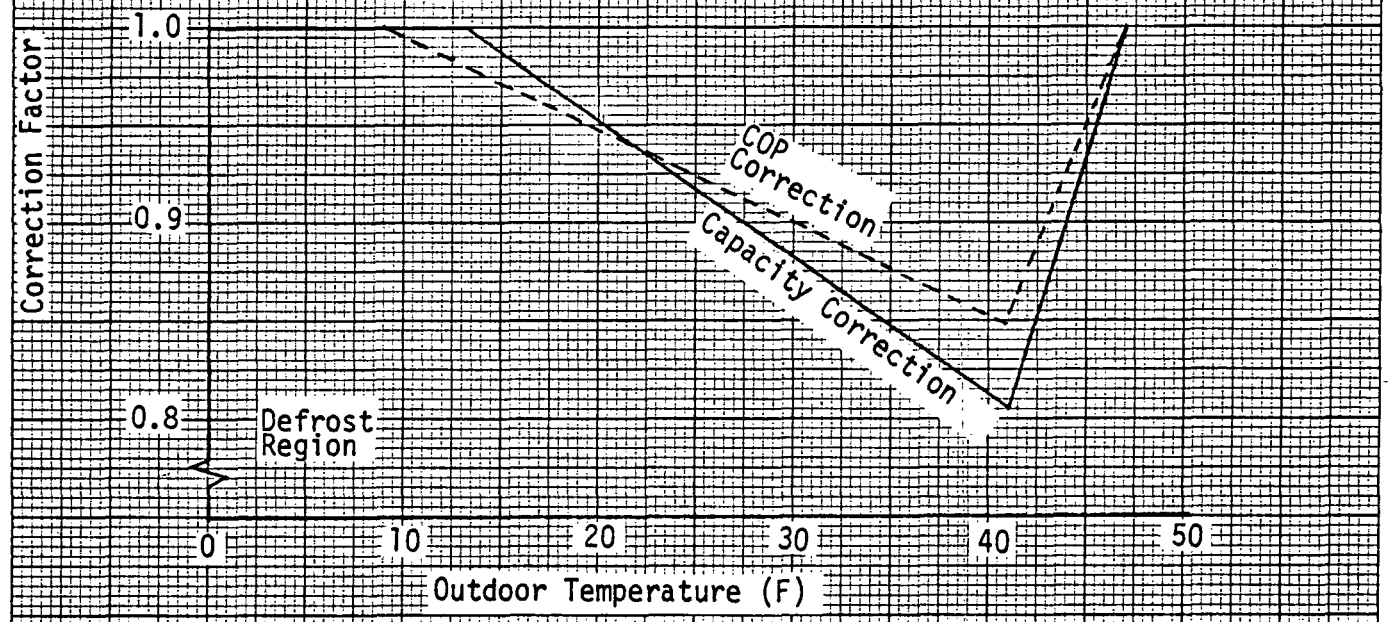
The electric air conditioner was modelled in the same way as the electric heat pump in the cooling mode.

C.2.3 Gas Furnace

The gas furnace model in AXCESS requires the specification of maximum output capacity, the thermal value of the fuel consumption rate at maximum capacity and a part load efficiency curve. The former data were, of course, obtained from manufacturers' literature; the part load efficiency data are from Macriss *et al.*(1977).

EXHIBIT C-1

PART LOAD AND DEFROST SYSTEM EFFICIENCY CORRECTIONS FOR ELECTRIC HEAT PUMP



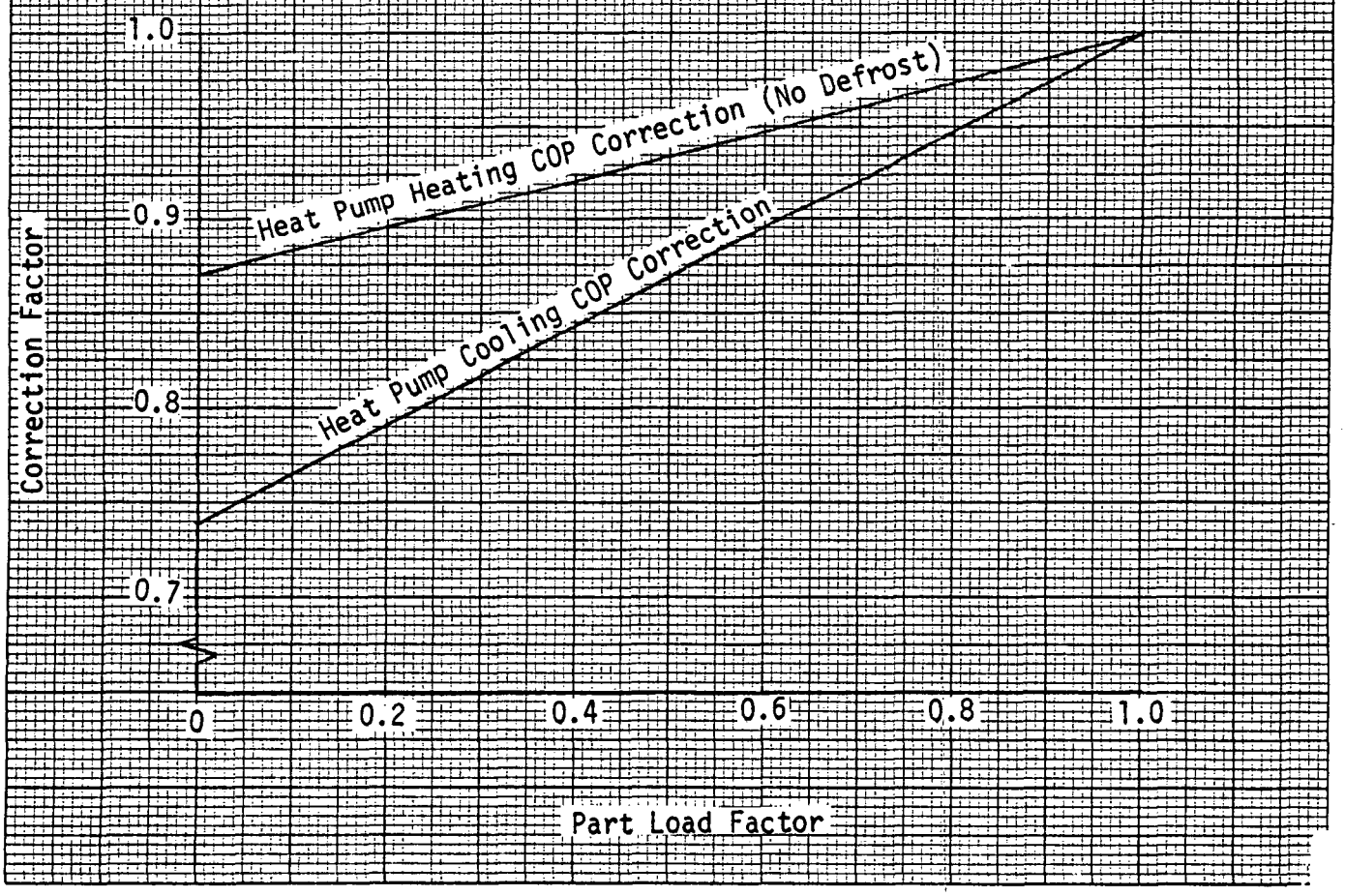


EXHIBIT C-2

PART LOAD EFFICIENCY CURVES FOR FOSSIL FUEL FIRED FURNACES

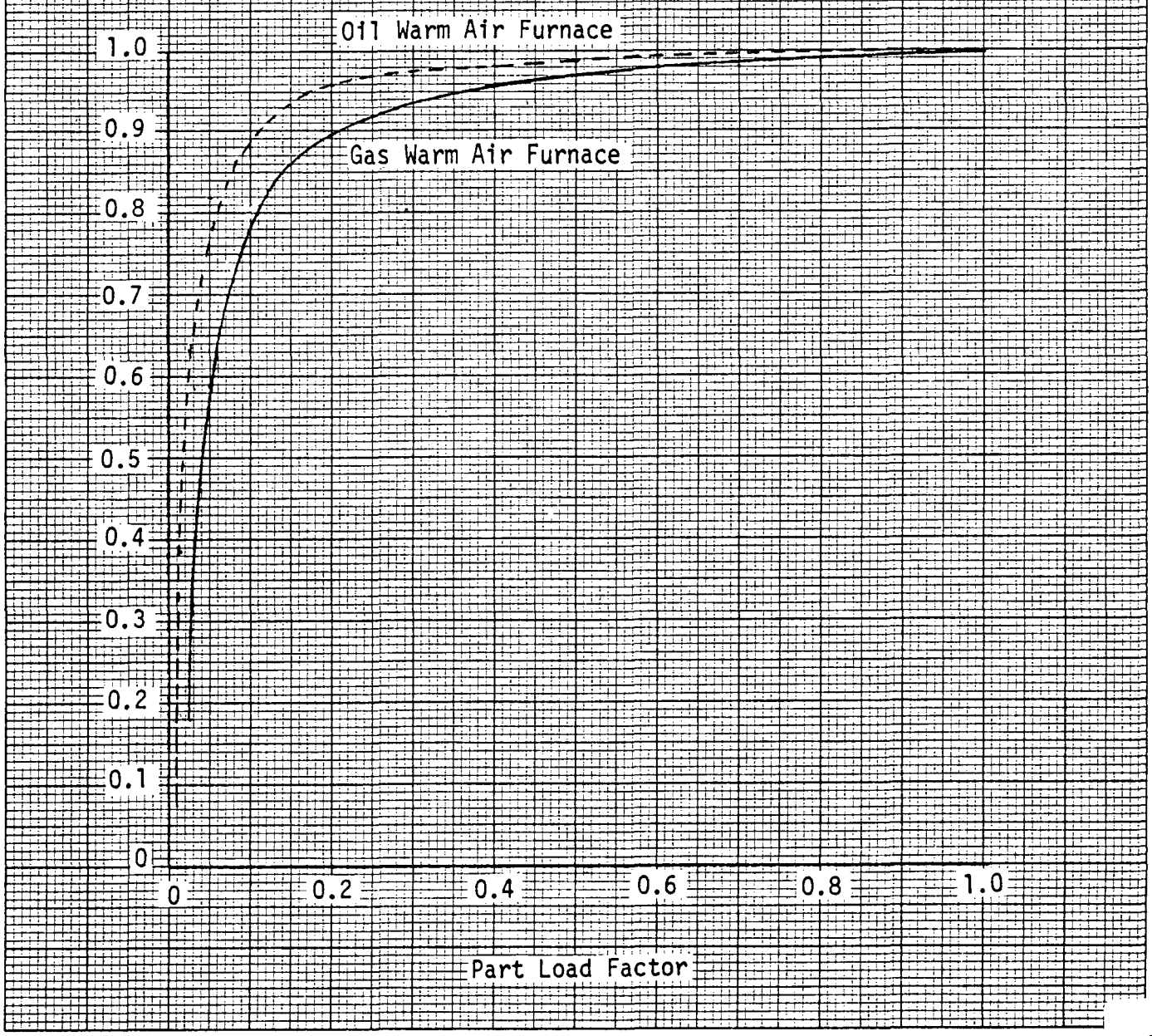
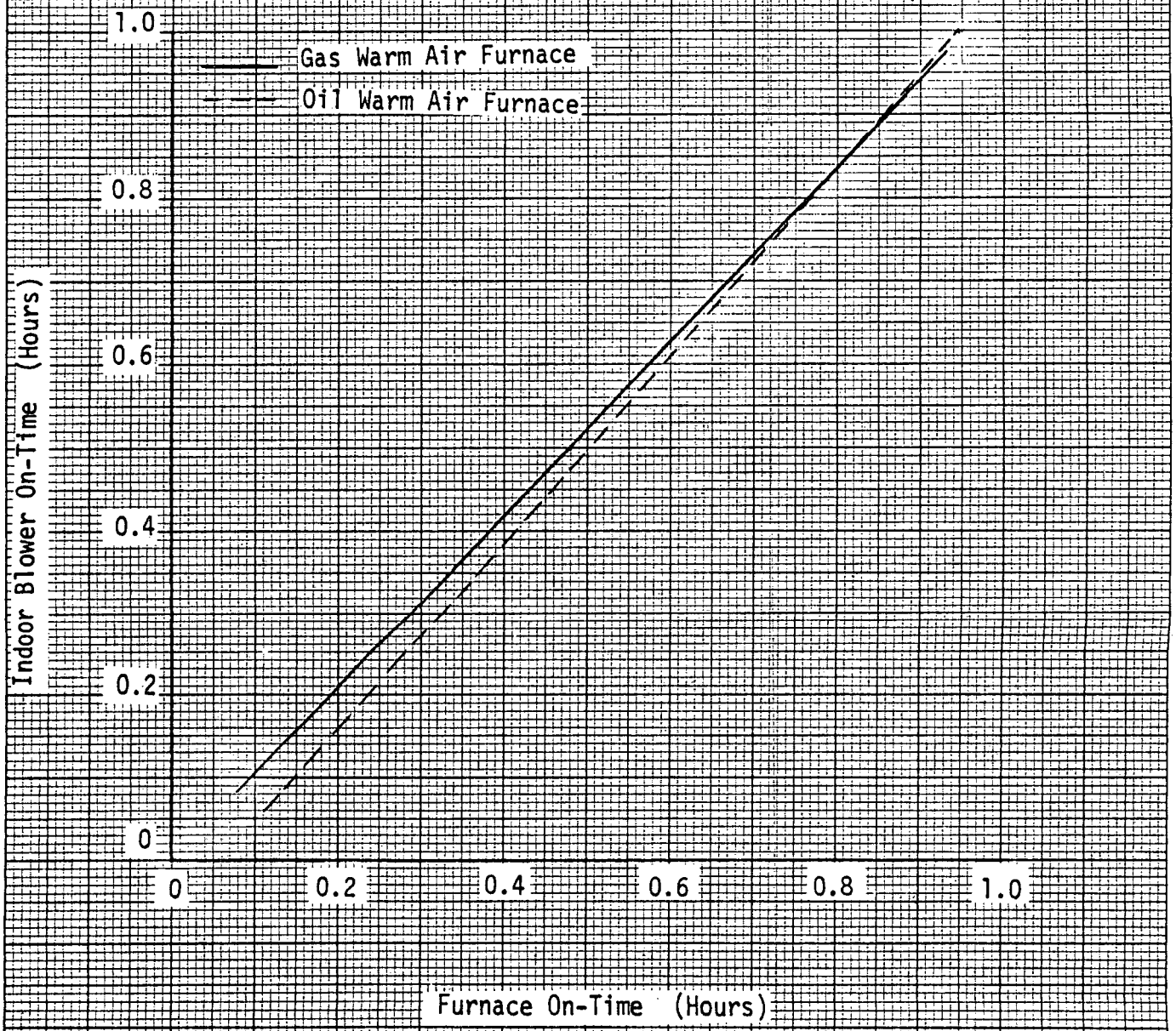


EXHIBIT C-3

RELATIONSHIP BETWEEN FURNACE ON-TIME AND INDOOR BLOWER ON-TIME



APPENDIX D

DETAILED DESCRIPTION OF THE MARKET MODELS

This appendix presents the major assumptions of the residential and the commercial heat pump market models utilized in the market forecasts of Chapter 10 and in the primary energy consumption estimates of Chapter 12.

D.1 Major Assumptions for the Residential Market

The major assumptions for the residential building market for the 1977-2000 year period are:

a. The annual conventional housing production for the 1977 to 2000 period is estimated to be between 1.3 and 2.0 million units, with an average of 1.75 million units per year. High rise apartments will constitute only 5 percent of this total

b. Housing growth is not expected to increase beyond the average 1.75 million units per year because of declining population growth and an increase in the number of people 65 and older, who are not prime home buying candidates.

c. The ratio of new construction by region will be similar to the 1971-1974 average for new home completions. These were:

Northeast	13 percent
North Central	21 percent
South	45 percent
West	21 percent

d. The installation rate for central warm air furnaces by region will be similar to the 1971-1974 average for new home completions, except in the South, where the warm air installation rate will be assumed equal to the installed central air conditioning rate. The installation rates used in this report are:

Northeast	37 percent
North Central	78 percent
South	77 percent
West	84 percent

e. Gas will be available as a heating fuel, with its use by region similar to the usage found in the 1971-1974 new home completion average. These gas heating usage levels were reported as:

Northeast	35 percent
North Central	65 percent
South	38 percent
West	71 percent

f. Central air conditioning is the limiting factor on new electric heat pump installations. Historic and projected saturation rates used in this report are given below.

Regional Central Air Conditioning Saturation Rates
(percent of new construction)

Region	1971-1974 Average	1975	1976	1977	1978	1979	1980-2000
Northeast	13	17.0	18.0	19.1	20.2	21.4	22.7
North Central	32	37.8	39.7	41.7	43.8	46.0	48.3
South	62	74.2	76.4	78.7	81.1	83.5	86.0
West	39	40.2	41.4	42.6	43.9	45.2	46.6

Sources: U.S. Department of Commerce, Gordian Associates Inc.

D.2 Assumptions for the Nonresidential Market

The key assumptions for the nonresidential building market for the 1977-2000 year period are:

a. Total square footage for all nonresidential buildings will be 1895 x 10^6 ft^2 in 1981.

b. The average growth rate for nonresidential building new construction will be 2 percent per year from 1977 to year 2000.

c. The proportion of new construction by building type will be similar to the average of the 1971-1975 period. This distribution, based on value-put-in-place, is given below.

Building Category	Percent
Office	14.1
Commercial	22.2
Industrial	18.7
Public and miscellaneous	12.4
Religious	2.2
Educational	19.2
Hospital	11.2
	100.0

d. The proportion of candidate buildings suitable for light commercial heat pump installations (25 ton or less) will vary by building category as follows:

Building Category	Estimated Percent Light Service	Comments
Office	50	Remainder high rise
Commercial	75	Remainder built-up systems
Hospital	40	Remainder built-up systems
Religious	40	Remainder not air-conditioned
Industrial	0	No air conditioning
Educational	25	Remainder not air-conditioned or use built-up system
Public and miscellaneous	50	Remainder high rise and /or uses built-up system.

Source: Gordian Associates Inc.

e. The rate of market penetration by heat pumps will be identical for both the residential and nonresidential building sectors.

f. Gas coverage factor will decline from 83.7 percent to 81.7 percent during the 1981-2000 period. The factor represents that proportion of new structures that are situated in an area served by natural gas.

g. The replacement/conversion potential market will be 10 percent of the new construction market.

h. One ton of cooling capacity will be required for every 300 ft^2 building space.

D.3 Market Penetration Rates

The market penetration rates for standard and advanced electric and gas heat pumps are presented in Exhibit D-1. The standard or conventional electric heat pump market penetrations are based on manufacturer's estimates and extrapolations. Where the projections are based on a currently marketed product such as the standard electric heat pump, some degree of confidence can be placed in the fact that a prior experience record is available. Generally, near term projections will hear a very close relationships to the established market.

The market penetrations rates employed in projecting the market prospects for the advanced heat pumps are based on historical market penetration for analagous space conditioning systems. Two appliances were selected based on their historical performance to represent a high growth and a low growth scenario. For the high growth scenario, the central electric air conditioner was selected. This appliance had relatively good initial market accpetance and growth. The electric heat pump was used to represent a low growth rate appliance. This appliance had numerous market acceptance problems when first introduced due, among other things, to poor reliability and inadequate manufacturer servicing capability.

These particular appliances were selected because their operation and complexity would be more closely related to an advanced electric heat pump or gas heat pump, than, for example, those of a gas furnace.

Three sets of market penetration projections are given in Exhibit D-1, with the high and low penetration rates based on smoothed actual data. The nominal projection represents the mean of the high and low projections.

EXHIBIT D-1

MARKET PENETRATION RATES FOR STANDARD AND ADVANCED HEAT PUMPS

Standard Electric Heat Pumps

Year	Percent Market Penetration
1977	16.3
1978	18.3
79	19.9
80	22.2
81	24.5
82	26.8
83	20.4
84	34.0
85	36.5
86	38.0
87	41.0
88	43.0
89	44.4
1990	46.5
91	48.5
92	50.0
93	51.8
95	55.0
96	56.6
97	58.0
98	59.7
99	61.2
2000	63.0

Advanced Electric and Gas Heat Pumps

Year	Percant Market Penetration		
	Low	Nominal	High
1	0.50	0.68	0.85
2	0.85	1.28	1.7
3	1.2	1.90	2.6
4	1.45	2.48	3.5
5	1.75	3.13	4.5
6	2.0	3.70	5.4
7	2.25	4.33	6.4
8	2.5	4.9	7.3
9	2.75	5.53	8.3
10	3.0	6.15	9.3
11	3.3	6.90	10.5
12	3.55	7.58	11.6
13	3.8	8.35	12.9
14	4.1	9.05	14.0
15	4.35	9.93	15.5
16	4.6	10.80	17.0
17	4.95	11.73	18.5
18	5.15	12.58	20.0
19	5.40	13.70	22.0
20	5.75	14.88	24.0

Sources: Air-Conditioning and Refrigeration Institute, General Electric Company, Dept. of Commerce, Gordian Associates, Inc.

APPENDIX E

PRIMARY FUELS AND ELECTRIC ENERGY CONSUMED BY SPACE CONDITIONING SYSTEMS ON A REGIONAL BASIS

This appendix contains data on primary fuels and electric energy consumed on a regional basis for residential and commercial HVAC systems. Exhibits E-1 and E-2 present the regional fuel mix used to produce one kilowatt hour of electric energy for on-site consumption in each indicated region. Exhibits E-3 through E-20 present primary fuel and energy consumption for residential systems, and Exhibits E-21 through E-25 present equivalent data for commercial units.

EXHIBIT E - 1

Primary Fuels Consumed and Kilowatt Hours Generated by Nuclear and Hydroelectric Plants to Provide One Kilowatt Hour of Energy

Region	Coal (LBS)	Oil (GALS)	Gas (CFT)	Hydro (KWH)	Nuclear (KWH)
			HEATING SEASON		
New England	0.011	0.047	0.001	0.051	0.273
Middle Atlantic	0.383	0.025	0.011	0.108	0.116
East North Central	0.788	0.004	0.181	0.008	0.110
West North Central	0.749	0.005	0.591	0.091	0.120
South Atlantic	0.466	0.016	0.283	0.053	0.150
East South Central	0.718	0.002	0.104	0.170	-
West South Central	0.115	0.006	8.215	0.014	0.015
Mountain	0.665	0.004	0.802	0.289	-
Pacific Contiguous	0.048	0.018	0.375	0.624	0.033
			COOLING SEASON		
New England	0.033	0.041	0.118	0.059	0.318
Middle Atlantic	0.384	0.021	0.148	0.114	0.155
East North Central	0.758	0.003	0.265	0.006	0.148
West North Central	0.622	0.002	1.948	0.107	0.140
South Atlantic	0.476	0.016	0.337	0.040	0.139
East South Central	0.733	0.004	0.191	0.117	-
West South Central	0.096	0.004	8.264	0.032	0.022
Mountain	0.593	0.004	1.766	0.265	-
Pacific Contiguous	0.027	0.012	1.807	0.609	0.011

EXHIBIT E - 2

Primary Fuels Consumed and Kilowatt Hours Generated by Nuclear and Hydroelectric Plants to Provide One Kilowatt Hour of Electric Energy

	Coal (Lbs)	Oil (Gals)	Gas (Cuft)	Hydro (KWH)	Nuclear (KWH)
New England	0.022	0.042	0.035	0.069	0.316
Middle Atlantic	0.394	0.022	0.054	0.119	0.131
East North Central	0.776	0.003	0.211	0.007	0.127
West North Central	0.698	0.003	1.360	0.097	0.123
South Atlantic	0.493	0.016	0.307	0.046	0.124
East South Central	0.643	0.011	0.138	0.099	0.022
West South Central	0.112	0.005	8.177	0.026	0.020
Mountain	0.656	0.003	1.174	0.274	0.000
Pacific Contiguous	0.049	0.016	1.097	0.584	0.036

EXHIBIT E-3

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ELECTRIC HEAT PUMP -- STANDARD

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.093	0.397	0.008	0.431	2.308
2	MID-ATLANTIC	2.635	0.172	0.076	0.743	0.798
3	EAST NORTH CENTRAL	6.355	0.032	1.469	0.045	0.893
4	WEST NORTH CENTRAL	5.905	0.039	4.560	0.717	0.906
5	SOUTH ATLANTIC	1.284	0.044	0.780	0.146	0.413
6	EAST SOUTH CENTRAL	2.482	0.007	0.360	0.598	0.0
7	WEST SOUTH CENTRAL	0.234	0.012	16.710	0.028	0.031
8	MOUNTAIN	3.634	0.022	4.382	1.579	0.0
9	PACIFIC CONTIGUOUS	0.121	0.045	0.947	1.576	0.083

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.039	0.049	0.140	0.070	0.377
2	MID-ATLANTIC	0.594	0.032	0.229	0.176	0.200
3	EAST NORTH CENTRAL	1.088	0.004	0.380	0.009	0.212
4	WEST NORTH CENTRAL	1.193	0.004	3.735	0.205	0.268
5	SOUTH ATLANTIC	1.427	0.048	1.011	0.120	0.417
6	EAST SOUTH CENTRAL	2.125	0.012	0.554	0.339	0.0
7	WEST SOUTH CENTRAL	0.368	0.015	31.692	0.123	0.084
8	MOUNTAIN	1.427	0.010	4.251	0.638	0.0
9	PACIFIC CONTIGUOUS	0.034	0.016	2.388	0.805	0.015

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.132	0.446	0.149	0.501	2.685
2	MID-ATLANTIC	3.229	0.204	0.305	0.919	1.038
3	EAST NORTH CENTRAL	7.442	0.037	1.849	0.074	1.105
4	WEST NORTH CENTRAL	7.098	0.043	8.395	0.923	1.215
5	SOUTH ATLANTIC	2.712	0.092	1.791	0.266	0.830
6	EAST SOUTH CENTRAL	4.607	0.019	0.913	0.927	0.0
7	WEST SOUTH CENTRAL	0.602	0.028	48.393	0.151	0.115
8	MOUNTAIN	5.061	0.031	8.633	2.217	0.0
9	PACIFIC CONTIGUOUS	0.157	0.061	3.335	2.381	0.098

EXHIBIT E-4

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ELECTRIC HEAT PUMP -- HIGH EFFICIENCY I

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.083	0.354	0.008	0.384	2.056
2	MID-ATLANTIC	2.334	0.152	0.067	0.658	0.707
3	EAST NORTH CENTRAL	5.655	0.029	1.307	0.058	0.794
4	WEST NORTH CENTRAL	5.251	0.035	4.143	0.638	0.841
5	SOUTH ATLANTIC	1.087	0.037	0.650	0.124	0.350
6	EAST SOUTH CENTRAL	2.134	0.006	0.309	0.505	0.0
7	WEST SOUTH CENTRAL	0.193	0.010	13.759	0.023	0.025
8	MOUNTAIN	3.194	0.019	3.852	1.388	0.0
9	PACIFIC CONTIGUOUS	0.102	0.038	0.796	1.325	0.070

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.033	0.040	0.117	0.058	0.314
2	MID-ATLANTIC	0.495	0.027	0.191	0.147	0.200
3	EAST NORTH CENTRAL	0.906	0.004	0.317	0.007	0.177
4	WEST NORTH CENTRAL	0.923	0.003	3.110	0.171	0.224
5	SOUTH ATLANTIC	1.188	0.040	0.841	0.100	0.347
6	EAST SOUTH CENTRAL	1.768	0.010	0.461	0.282	0.0
7	WEST SOUTH CENTRAL	0.306	0.013	26.352	0.102	0.070
8	MOUNTAIN	1.189	0.008	3.538	0.541	0.0
9	PACIFIC CONTIGUOUS	0.030	0.013	1.991	0.671	0.012

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.115	0.394	0.124	0.442	2.370
2	MID-ATLANTIC	2.829	0.179	0.258	0.805	0.907
3	EAST NORTH CENTRAL	6.561	0.032	1.624	0.065	0.971
4	WEST NORTH CENTRAL	6.244	0.038	7.254	0.809	1.065
5	SOUTH ATLANTIC	2.275	0.077	1.501	0.223	0.697
6	EAST SOUTH CENTRAL	3.902	0.016	0.770	0.788	0.0
7	WEST SOUTH CENTRAL	0.499	0.023	40.112	0.125	0.095
8	MOUNTAIN	4.382	0.027	7.391	1.919	0.0
9	PACIFIC CONTIGUOUS	0.132	0.051	2.787	1.996	0.082

EXHIBIT E-5

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ELECTRIC HEAT PUMP -- HIGH EFFICIENCY II

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.079	0.339	0.007	0.368	1.972
2	MID-ATLANTIC	2.259	0.147	0.065	0.637	0.684
3	EAST NORTH CENTRAL	5.832	0.028	1.256	0.055	0.763
4	WEST NORTH CENTRAL	5.050	0.034	3.985	0.614	0.809
5	SOUTH ATLANTIC	1.132	0.039	0.697	0.129	0.354
6	EAST SOUTH CENTRAL	2.167	0.006	0.314	0.513	0.0
7	WEST SOUTH CENTRAL	0.209	0.011	14.955	0.026	0.027
8	MOUNTAIN	3.130	0.019	3.775	1.360	0.0
9	PACIFIC CONTIGUOUS	0.107	0.040	0.838	1.395	0.074

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.127	0.064	0.343
2	MID-ATLANTIC	0.540	0.030	0.208	0.160	0.218
3	EAST NORTH CENTRAL	0.990	0.004	0.346	0.008	0.193
4	WEST NORTH CENTRAL	1.085	0.003	3.399	0.187	0.244
5	SOUTH ATLANTIC	1.209	0.044	0.920	0.109	0.379
6	EAST SOUTH CENTRAL	1.934	0.011	0.504	0.309	0.0
7	WEST SOUTH CENTRAL	0.335	0.014	28.834	0.112	0.077
8	MOUNTAIN	1.299	0.009	3.869	0.581	0.0
9	PACIFIC CONTIGUOUS	0.032	0.014	2.173	0.732	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.115	0.384	0.134	0.432	2.314
2	MID-ATLANTIC	2.799	0.177	0.273	0.797	0.902
3	EAST NORTH CENTRAL	6.821	0.032	1.602	0.063	0.956
4	WEST NORTH CENTRAL	6.135	0.037	7.384	0.800	1.053
5	SOUTH ATLANTIC	2.331	0.083	1.607	0.238	0.744
6	EAST SOUTH CENTRAL	4.101	0.017	0.818	0.822	0.0
7	WEST SOUTH CENTRAL	0.544	0.025	43.798	0.137	0.104
8	MOUNTAIN	4.429	0.028	7.644	1.941	0.0
9	PACIFIC CONTIGUOUS	0.140	0.055	3.012	2.127	0.087

EXHIBIT E-6

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ELECTRIC HEAT PUMP -- HIGH EFFICIENCY III

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.085	0.351	0.008	0.392	2.098
2	MID-ATLANTIC	2.408	0.157	0.069	0.679	0.729
3	EAST NORTH CENTRAL	5.783	0.030	1.337	0.059	0.812
4	WEST NORTH CENTRAL	5.377	0.036	4.243	0.653	0.862
5	SOUTH ATLANTIC	1.222	0.042	0.742	0.139	0.393
6	EAST SOUTH CENTRAL	2.330	0.006	0.338	0.552	0.0
7	WEST SOUTH CENTRAL	0.228	0.012	16.275	0.028	0.030
8	MOUNTAIN	3.348	0.020	4.053	1.455	0.0
9	PACIFIC CONTIGUOUS	0.116	0.044	0.907	1.509	0.080

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.035	0.043	0.124	0.062	0.335
2	MID-ATLANTIC	0.526	0.029	0.203	0.156	0.213
3	EAST NORTH CENTRAL	0.965	0.004	0.337	0.008	0.188
4	WEST NORTH CENTRAL	1.056	0.003	3.306	0.182	0.238
5	SOUTH ATLANTIC	1.260	0.042	0.892	0.106	0.368
6	EAST SOUTH CENTRAL	1.875	0.010	0.489	0.299	0.0
7	WEST SOUTH CENTRAL	0.324	0.014	27.924	0.108	0.074
8	MOUNTAIN	1.261	0.009	3.756	0.564	0.0
9	PACIFIC CONTIGUOUS	0.032	0.014	2.121	0.715	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.119	0.404	0.132	0.454	2.433
2	MID-ATLANTIC	2.935	0.186	0.272	0.835	0.942
3	EAST NORTH CENTRAL	6.748	0.033	1.674	0.067	1.001
4	WEST NORTH CENTRAL	6.433	0.039	7.549	0.835	1.099
5	SOUTH ATLANTIC	2.482	0.084	1.634	0.245	0.761
6	EAST SOUTH CENTRAL	4.206	0.017	0.826	0.851	0.0
7	WEST SOUTH CENTRAL	0.552	0.025	44.198	0.136	0.104
8	MOUNTAIN	4.605	0.029	7.789	2.017	0.0
9	PACIFIC CONTIGUOUS	0.148	0.058	3.028	2.224	0.093

EXHIBIT E-7

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: HYBRID HEAT PUMP -- GAS FURNACE

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.055	0.234	30.620	0.234	1.358
2	MID-ATLANTIC	1.625	0.106	20.077	0.458	0.492
3	EAST NORTH CENTRAL	3.773	0.019	29.162	0.059	0.550
4	WEST NORTH CENTRAL	3.523	0.024	31.547	0.428	0.564
5	SOUTH ATLANTIC	1.049	0.036	6.325	0.119	0.338
6	EAST SOUTH CENTRAL	1.871	0.005	6.771	0.445	0.0
7	WEST SOUTH CENTRAL	0.220	0.011	18.695	0.027	0.029
8	MOUNTAIN	2.362	0.014	19.462	1.027	0.0
9	PACIFIC CONTIGUOUS	0.103	0.039	5.476	1.340	0.071

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.041	0.051	0.147	0.074	0.397
2	MID-ATLANTIC	0.625	0.034	0.241	0.166	0.252
3	EAST NORTH CENTRAL	1.145	0.005	0.400	0.009	0.224
4	WEST NORTH CENTRAL	1.254	0.004	3.928	0.216	0.282
5	SOUTH ATLANTIC	1.499	0.050	1.062	0.126	0.438
6	EAST SOUTH CENTRAL	2.232	0.012	0.582	0.356	0.0
7	WEST SOUTH CENTRAL	0.386	0.016	33.266	0.129	0.089
8	MOUNTAIN	1.500	0.010	4.468	0.670	0.0
9	PACIFIC CONTIGUOUS	0.038	0.017	2.515	0.848	0.015

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.096	0.285	30.767	0.327	1.755
2	MID-ATLANTIC	2.250	0.140	20.317	0.644	0.744
3	EAST NORTH CENTRAL	4.918	0.024	29.562	0.048	0.754
4	WEST NORTH CENTRAL	4.777	0.028	35.475	0.644	0.847
5	SOUTH ATLANTIC	2.548	0.086	7.386	0.245	0.775
6	EAST SOUTH CENTRAL	4.104	0.017	7.353	0.799	0.0
7	WEST SOUTH CENTRAL	0.606	0.028	51.959	0.156	0.117
8	MOUNTAIN	3.863	0.024	23.929	1.697	0.0
9	PACIFIC CONTIGUOUS	0.141	0.055	7.991	2.188	0.086

EXHIBIT E-8

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: HYBRID HEAT PUMP -- OIL FURNACE

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.055	0.453	0.005	0.254	1.358
2	MID-ATLANTIC	1.625	0.236	0.047	0.458	0.492
3	EAST NORTH CENTRAL	3.773	0.203	0.872	0.039	0.530
4	WEST NORTH CENTRAL	3.523	0.210	2.780	0.428	0.564
5	SOUTH ATLANTIC	1.049	0.073	0.637	0.119	0.338
6	EAST SOUTH CENTRAL	1.871	0.047	0.271	0.443	0.0
7	WEST SOUTH CENTRAL	0.220	0.031	15.698	0.027	0.029
8	MOUNTAIN	2.362	0.122	2.849	1.027	0.0
9	PACIFIC CONTIGUOUS	0.103	0.069	0.805	1.540	0.071

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.041	0.051	0.147	0.074	0.397
2	MID-ATLANTIC	0.625	0.034	0.241	0.186	0.252
3	EAST NORTH CENTRAL	1.145	0.005	0.400	0.009	0.224
4	WEST NORTH CENTRAL	1.254	0.004	3.928	0.216	0.282
5	SOUTH ATLANTIC	1.499	0.050	1.062	0.126	0.458
6	EAST SOUTH CENTRAL	2.232	0.012	0.562	0.356	0.0
7	WEST SOUTH CENTRAL	0.386	0.016	53.266	0.129	0.089
8	MOUNTAIN	1.500	0.010	4.468	0.670	0.0
9	PACIFIC CONTIGUOUS	0.038	0.017	2.515	0.848	0.015

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.096	0.454	0.152	0.327	1.755
2	MID-ATLANTIC	2.250	0.270	0.288	0.644	0.744
3	EAST NORTH CENTRAL	4.918	0.208	1.272	0.048	0.754
4	WEST NORTH CENTRAL	4.777	0.215	6.708	0.644	0.847
5	SOUTH ATLANTIC	2.548	0.123	1.699	0.245	0.775
6	EAST SOUTH CENTRAL	4.104	0.060	0.833	0.799	0.0
7	WEST SOUTH CENTRAL	0.606	0.047	48.964	0.156	0.117
8	MOUNTAIN	3.863	0.132	7.317	1.697	0.0
9	PACIFIC CONTIGUOUS	0.141	0.086	3.321	2.188	0.086

EXHIBIT E-9

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ELECTRIC HEAT PUMP - HEAT ONLY CENTRAL AIR CONDITIONER -- HIGH EFFICIENCY

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.073	0.311	0.007	0.337	1.807
2	MID-ATLANTIC	2.070	0.135	0.059	0.584	0.627
3	EAST NORTH CENTRAL	4.977	0.025	1.151	0.051	0.699
4	WEST NORTH CENTRAL	4.627	0.031	3.651	0.562	0.741
5	SOUTH ATLANTIC	1.037	0.036	0.630	0.118	0.334
6	EAST SOUTH CENTRAL	1.985	0.006	0.268	0.470	0.0
7	WEST SOUTH CENTRAL	0.192	0.010	13.707	0.023	0.025
8	MOUNTAIN	2.868	0.017	3.459	1.246	0.0
9	PACIFIC CONTIGUOUS	0.098	0.037	0.768	1.278	0.068

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.128	0.064	0.344
2	MID-ATLANTIC	0.542	0.030	0.209	0.151	0.219
3	EAST NORTH CENTRAL	0.993	0.004	0.347	0.008	0.194
4	WEST NORTH CENTRAL	1.086	0.003	3.402	0.187	0.204
5	SOUTH ATLANTIC	1.297	0.044	0.918	0.109	0.379
6	EAST SOUTH CENTRAL	1.931	0.011	0.503	0.308	0.0
7	WEST SOUTH CENTRAL	0.334	0.014	28.758	0.111	0.077
8	MOUNTAIN	1.293	0.009	3.866	0.580	0.0
9	PACIFIC CONTIGUOUS	0.033	0.014	2.181	0.735	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.109	0.355	0.134	0.401	2.151
2	MID-ATLANTIC	2.611	0.165	0.268	0.744	0.845
3	EAST NORTH CENTRAL	5.970	0.029	1.498	0.059	0.893
4	WEST NORTH CENTRAL	5.713	0.034	7.053	0.749	0.986
5	SOUTH ATLANTIC	2.334	0.079	1.548	0.227	0.712
6	EAST SOUTH CENTRAL	3.916	0.016	0.771	0.778	0.0
7	WEST SOUTH CENTRAL	0.526	0.024	42.465	0.135	0.102
8	MOUNTAIN	4.166	0.026	7.325	1.826	0.0
9	PACIFIC CONTIGUOUS	0.131	0.051	2.949	2.013	0.081

EXHIBIT E-10

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: GAS WARM AIR FURNACE -- CENTRAL AIR CONDITIONER -- HIGH EFFICIENCY

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.004	0.017	59.075	0.019	0.101
2	MID-ATLANTIC	0.119	0.008	57.689	0.034	0.036
3	EAST NORTH CENTRAL	0.279	0.001	66.699	0.003	0.039
4	WEST NORTH CENTRAL	0.260	0.002	65.153	0.032	0.042
5	SOUTH ATLANTIC	0.075	0.003	27.889	0.009	0.024
6	EAST SOUTH CENTRAL	0.134	0.000	32.933	0.032	0.0
7	WEST SOUTH CENTRAL	0.015	0.001	23.727	0.002	0.002
8	MOUNTAIN	0.173	0.001	47.646	0.075	0.0
9	PACIFIC CONTIGUOUS	0.007	0.003	26.234	0.095	0.005

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.128	0.064	0.344
2	MID-ATLANTIC	0.542	0.030	0.209	0.161	0.219
3	EAST NORTH CENTRAL	0.993	0.004	0.347	0.008	0.194
4	WEST NORTH CENTRAL	1.086	0.003	3.402	0.187	0.204
5	SOUTH ATLANTIC	1.297	0.044	0.918	0.109	0.379
6	EAST SOUTH CENTRAL	1.931	0.011	0.503	0.308	0.0
7	WEST SOUTH CENTRAL	0.334	0.014	28.758	0.111	0.077
8	MOUNTAIN	1.298	0.009	3.856	0.580	0.0
9	PACIFIC CONTIGUOUS	0.033	0.014	2.181	0.735	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.040	0.062	59.203	0.083	0.445
2	MID-ATLANTIC	0.661	0.037	57.898	0.194	0.255
3	EAST NORTH CENTRAL	1.271	0.005	67.036	0.011	0.233
4	WEST NORTH CENTRAL	1.386	0.005	68.555	0.218	0.286
5	SOUTH ATLANTIC	1.372	0.046	28.807	0.118	0.403
6	EAST SOUTH CENTRAL	2.065	0.011	33.437	0.340	0.0
7	WEST SOUTH CENTRAL	0.350	0.015	52.485	0.113	0.079
8	MOUNTAIN	1.471	0.010	51.512	0.655	0.0
9	PACIFIC CONTIGUOUS	0.040	0.017	28.415	0.830	0.018

EXHIBIT E-11

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: OIL WARM AIR FURNACE -- CENTRAL AIR CONDITIONER -- HIGH EFFICIENCY

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.003	0.456	0.000	0.015	0.078
2	MID-ATLANTIC	0.091	0.383	0.003	0.026	0.028
3	EAST NORTH CENTRAL	0.215	0.437	0.050	0.002	0.030
4	WEST NORTH CENTRAL	0.201	0.427	0.158	0.024	0.032
5	SOUTH ATLANTIC	0.053	0.183	0.032	0.006	0.017
6	EAST SOUTH CENTRAL	0.096	0.215	0.014	0.023	0.0
7	WEST SOUTH CENTRAL	0.010	0.147	0.748	0.001	0.001
8	MOUNTAIN	0.130	0.311	0.156	0.056	0.0
9	PACIFIC CONTIGUOUS	0.005	0.172	0.040	0.056	0.003

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.128	0.064	0.344
2	MID-ATLANTIC	0.542	0.030	0.209	0.161	0.219
3	EAST NORTH CENTRAL	0.993	0.004	0.347	0.008	0.194
4	WEST NORTH CENTRAL	1.086	0.003	3.402	0.187	0.244
5	SOUTH ATLANTIC	1.297	0.044	0.918	0.109	0.379
6	EAST SOUTH CENTRAL	1.031	0.011	0.503	0.308	0.0
7	WEST SOUTH CENTRAL	0.334	0.014	28.758	0.111	0.077
8	MOUNTAIN	1.298	0.009	3.866	0.580	0.0
9	PACIFIC CONTIGUOUS	0.033	0.014	2.181	0.735	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.039	0.510	0.128	0.078	0.422
2	MID-ATLANTIC	0.633	0.413	0.211	0.186	0.246
3	EAST NORTH CENTRAL	1.208	0.441	0.397	0.010	0.224
4	WEST NORTH CENTRAL	1.267	0.430	3.560	0.211	0.277
5	SOUTH ATLANTIC	1.349	0.226	0.950	0.115	0.396
6	EAST SOUTH CENTRAL	2.027	0.225	0.517	0.331	0.0
7	WEST SOUTH CENTRAL	0.345	0.161	29.507	0.113	0.078
8	MOUNTAIN	1.428	0.320	4.022	0.636	0.0
9	PACIFIC CONTIGUOUS	0.038	0.186	2.221	0.801	0.017

EXHIBIT E-12

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: CENTRAL ELECTRIC FURNACE -- CENTRAL AIR CONDITIONER -- HIGH EFFICIENCY

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.102	0.609	0.013	0.660	3.536
2	MID-ATLANTIC	4.121	0.269	0.118	1.162	1.248
3	EAST NORTH CENTRAL	9.771	0.050	2.259	0.100	1.373
4	WEST NORTH CENTRAL	9.105	0.061	7.185	1.106	1.459
5	SOUTH ATLANTIC	2.338	0.080	1.420	0.266	0.752
6	EAST SOUTH CENTRAL	4.302	0.012	0.623	1.019	0.0
7	WEST SOUTH CENTRAL	0.461	0.024	32.951	0.056	0.060
8	MOUNTAIN	5.843	0.035	7.047	2.539	0.0
9	PACIFIC CONTIGUOUS	0.225	0.085	1.761	2.930	0.155

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.128	0.064	0.344
2	MID-ATLANTIC	0.542	0.030	0.209	0.161	0.219
3	EAST NORTH CENTRAL	0.903	0.004	0.347	0.008	0.194
4	WEST NORTH CENTRAL	1.086	0.003	3.402	0.187	0.204
5	SOUTH ATLANTIC	1.297	0.044	0.918	0.109	0.379
6	EAST SOUTH CENTRAL	1.031	0.011	0.503	0.308	0.0
7	WEST SOUTH CENTRAL	0.334	0.014	28.758	0.111	0.077
8	MOUNTAIN	1.298	0.009	3.866	0.580	0.0
9	PACIFIC CONTIGUOUS	0.033	0.014	2.181	0.735	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.178	0.653	0.141	0.724	3.880
2	MID-ATLANTIC	4.642	0.299	0.327	1.323	1.467
3	EAST NORTH CENTRAL	10.764	0.054	2.606	0.108	1.567
4	WEST NORTH CENTRAL	10.192	0.064	10.586	1.293	1.703
5	SOUTH ATLANTIC	3.634	0.124	2.338	0.375	1.131
6	EAST SOUTH CENTRAL	5.233	0.023	1.126	1.327	0.0
7	WEST SOUTH CENTRAL	0.795	0.038	61.709	0.168	0.137
8	MOUNTAIN	7.142	0.044	10.913	3.120	0.0
9	PACIFIC CONTIGUOUS	0.258	0.099	3.942	3.665	0.168

EXHIBIT E-13

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: BASEBOARD CONVECTORS -- CENTRAL AIR CONDITIONER -- HIGH EFFICIENCY

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.112	0.607	0.013	0.659	3.526
2	MID-ATLANTIC	4.114	0.269	0.118	1.150	1.246
3	EAST NORTH CENTRAL	9.717	0.050	2.253	0.100	1.369
4	WEST NORTH CENTRAL	9.084	0.051	7.168	1.194	1.455
5	SOUTH ATLANTIC	2.348	0.081	1.426	0.267	0.756
6	EAST SOUTH CENTRAL	4.313	0.012	0.625	1.021	0.0
7	WEST SOUTH CENTRAL	0.465	0.024	33.196	0.057	0.061
8	MOUNTAIN	5.840	0.035	7.043	2.538	0.0
9	PACIFIC CONTIGUOUS	0.227	0.085	1.770	2.946	0.156

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.128	0.064	0.344
2	MID-ATLANTIC	0.542	0.030	0.209	0.161	0.219
3	EAST NORTH CENTRAL	0.293	0.004	0.347	0.008	0.194
4	WEST NORTH CENTRAL	1.096	0.003	3.402	0.197	0.204
5	SOUTH ATLANTIC	1.297	0.044	0.918	0.109	0.379
6	EAST SOUTH CENTRAL	1.931	0.011	0.503	0.308	0.0
7	WEST SOUTH CENTRAL	0.334	0.014	28.758	0.111	0.077
8	MOUNTAIN	1.220	0.009	3.856	0.580	0.0
9	PACIFIC CONTIGUOUS	0.033	0.014	2.181	0.735	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.178	0.651	0.141	0.723	3.870
2	MID-ATLANTIC	4.655	0.298	0.327	1.321	1.464
3	EAST NORTH CENTRAL	10.739	0.054	2.600	0.107	1.563
4	WEST NORTH CENTRAL	10.170	0.064	10.569	1.290	1.700
5	SOUTH ATLANTIC	3.645	0.124	2.344	0.376	1.135
6	EAST SOUTH CENTRAL	6.244	0.023	1.128	1.330	0.0
7	WEST SOUTH CENTRAL	0.799	0.038	61.954	0.168	0.137
8	MOUNTAIN	7.138	0.044	10.910	3.118	0.0
9	PACIFIC CONTIGUOUS	0.259	0.099	3.951	3.681	0.169

Exhibit E-14

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ADVANCED ELECTRIC HEAT PUMP - I

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY NOT INCLUDED
(INSULATED DUCTS)

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.052	0.264	0.006	0.286	1.532
2	MIDDLE ATLANTIC	1.776	0.116	0.051	0.501	0.539
3	EAST NORTH CENTRAL	4.230	0.022	0.978	0.043	0.594
4	WEST NORTH CENTRAL	3.939	0.026	3.108	0.479	0.631
5	SOUTH ATLANTIC	0.958	0.033	0.588	0.110	0.312
6	EAST SOUTH CENTRAL	1.934	0.005	0.261	0.427	0.000
7	WEST SOUTH CENTRAL	0.197	0.010	13.398	0.023	0.024
8	MOUNTAIN	2.499	0.015	3.414	1.066	0.000
9	PACIFIC CONTIGUOUS	0.093	0.035	0.725	1.207	0.064

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.031	0.039	0.112	0.056	0.302
2	MIDDLE ATLANTIC	0.477	0.026	0.184	0.142	0.192
3	EAST NORTH CENTRAL	0.973	0.003	0.305	0.007	0.178
4	WEST NORTH CENTRAL	0.959	0.003	3.004	0.155	0.215
5	SOUTH ATLANTIC	1.151	0.039	0.815	0.097	0.335
6	EAST SOUTH CENTRAL	1.713	0.009	0.446	0.273	0.000
7	WEST SOUTH CENTRAL	0.297	0.012	25.574	0.099	0.068
8	MOUNTAIN	1.130	0.009	3.424	0.514	0.000
9	PACIFIC CONTIGUOUS	0.029	0.013	1.915	0.545	0.012

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.083	0.303	0.118	0.342	1.834
2	MIDDLE ATLANTIC	2.253	0.142	0.235	0.642	0.730
3	EAST NORTH CENTRAL	5.103	0.025	1.283	0.050	0.765
4	WEST NORTH CENTRAL	4.898	0.029	6.113	0.644	0.847
5	SOUTH ATLANTIC	2.119	0.072	1.403	0.207	0.649
6	EAST SOUTH CENTRAL	3.517	0.014	0.709	0.701	0.000
7	WEST SOUTH CENTRAL	0.435	0.022	38.953	0.122	0.093
8	MOUNTAIN	3.549	0.023	6.438	1.580	0.000
9	PACIFIC CONTIGUOUS	0.121	0.048	2.640	1.853	0.076

Exhibit E-15

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ADVANCED ELECTRIC HEAT PUMP - II

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.052	0.265	0.006	0.288	1.542
2	MIDDLE ATLANTIC	1.754	0.114	0.054	0.495	0.531
3	EAST NORTH CENTRAL	4.243	0.022	0.981	0.043	0.596
4	WEST NORTH CENTRAL	3.941	0.026	3.109	0.479	0.031
5	SOUTH ATLANTIC	0.938	0.028	0.504	0.094	0.267
6	EAST SOUTH CENTRAL	1.528	0.005	0.235	0.383	0.000
7	WEST SOUTH CENTRAL	0.143	0.005	10.595	0.018	0.019
8	MOUNTAIN	2.406	0.014	2.002	1.046	0.000
9	PACIFIC CONTIGUOUS	0.079	0.029	0.569	1.013	0.054

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.023	0.029	0.003	0.041	0.222
2	MIDDLE ATLANTIC	0.352	0.019	0.130	0.164	0.142
3	EAST NORTH CENTRAL	0.544	0.003	0.225	0.005	0.126
4	WEST NORTH CENTRAL	0.767	0.002	2.215	0.122	0.159
5	SOUTH ATLANTIC	0.349	0.029	0.501	0.071	0.246
6	EAST SOUTH CENTRAL	1.254	0.007	0.329	0.202	0.000
7	WEST SOUTH CENTRAL	0.219	0.009	18.560	0.073	0.050
8	MOUNTAIN	0.948	0.006	2.525	0.379	0.000
9	PACIFIC CONTIGUOUS	0.021	0.009	1.412	0.476	0.009

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.085	0.294	0.088	0.329	1.764
2	MIDDLE ATLANTIC	2.106	0.134	0.196	0.599	0.673
3	EAST NORTH CENTRAL	4.585	0.024	1.206	0.048	0.722
4	WEST NORTH CENTRAL	4.549	0.029	5.325	0.600	0.191
5	SOUTH ATLANTIC	1.579	0.057	1.105	0.166	0.515
6	EAST SOUTH CENTRAL	2.583	0.011	0.564	0.585	0.000
7	WEST SOUTH CENTRAL	0.357	0.017	29.458	0.091	0.070
8	MOUNTAIN	3.254	0.020	5.427	1.425	0.000
9	PACIFIC CONTIGUOUS	0.099	0.039	2.021	1.489	0.062

Exhibit E-16

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: FREE PISTON STIRLING RANKINE GAS HEAT PUMP

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY NOT INCLUDED
(INSULATED DUCTS)

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.013	0.054	27.550	0.059	0.315
2	MIDDLE ATLANTIC	0.353	0.024	22.791	0.102	0.110
3	EAST NORTH CENTRAL	0.358	0.004	26.724	0.009	0.122
4	WEST NORTH CENTRAL	0.308	0.005	25.458	0.098	0.129
5	SOUTH ATLANTIC	0.192	0.007	10.405	0.022	0.062
6	EAST SOUTH CENTRAL	0.352	0.001	12.464	0.056	0.000
7	WEST SOUTH CENTRAL	0.037	0.002	10.721	0.004	0.005
8	MOUNTAIN	0.508	0.003	19.104	0.221	0.000
9	PACIFIC CONTIGUOUS	0.018	0.007	9.734	0.239	0.013

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.009	0.011	7.912	0.016	0.085
2	MIDDLE ATLANTIC	0.132	0.007	10.178	0.039	0.053
3	EAST NORTH CENTRAL	0.243	0.001	9.521	0.002	0.049
4	WEST NORTH CENTRAL	0.254	0.001	13.258	0.045	0.059
5	SOUTH ATLANTIC	0.312	0.010	19.364	0.026	0.091
6	EAST SOUTH CENTRAL	0.454	0.003	18.546	0.074	0.000
7	WEST SOUTH CENTRAL	0.058	0.003	31.209	0.027	0.019
8	MOUNTAIN	0.313	0.002	16.405	0.144	0.000
9	PACIFIC CONTIGUOUS	0.003	0.004	9.270	0.181	0.003

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.022	0.065	35.461	0.075	0.400
2	MIDDLE ATLANTIC	0.495	0.031	32.970	0.142	0.163
3	EAST NORTH CENTRAL	1.112	0.005	36.244	0.011	0.170
4	WEST NORTH CENTRAL	1.072	0.006	39.715	0.144	0.189
5	SOUTH ATLANTIC	0.504	0.017	29.770	0.048	0.153
6	EAST SOUTH CENTRAL	0.825	0.004	31.010	0.150	0.000
7	WEST SOUTH CENTRAL	0.117	0.005	41.930	0.031	0.023
8	MOUNTAIN	0.822	0.005	35.509	0.361	0.000
9	PACIFIC CONTIGUOUS	0.025	0.010	19.005	0.420	0.015

Exhibit E-17

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: V TYPE SINGLE CYLINDER STIRLING RANKINE HEAT PUMP

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY NOT INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.015	0.062	26.531	0.058	0.361
2	MIDDLE ATLANTIC	0.414	0.027	21.965	0.117	0.125
3	EAST NORTH CENTRAL	0.935	0.005	25.775	0.010	0.140
4	WEST NORTH CENTRAL	0.925	0.006	25.602	0.112	0.148
5	SOUTH ATLANTIC	0.285	0.007	10.286	0.023	0.066
6	EAST SOUTH CENTRAL	0.334	0.001	12.055	0.093	0.000
7	WEST SOUTH CENTRAL	0.039	0.002	10.556	0.005	0.005
8	MOUNTAIN	0.572	0.003	18.525	0.249	0.000
9	PACIFIC CONTIGUOUS	0.019	0.007	9.443	0.252	0.013

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.011	0.014	8.827	0.020	0.110
2	MIDDLE ATLANTIC	0.155	0.009	11.483	0.049	0.067
3	EAST NORTH CENTRAL	0.307	0.001	10.716	0.002	0.060
4	WEST NORTH CENTRAL	0.324	0.001	15.134	0.056	0.073
5	SOUTH ATLANTIC	0.372	0.012	22.247	0.031	0.109
6	EAST SOUTH CENTRAL	0.535	0.003	21.404	0.089	0.000
7	WEST SOUTH CENTRAL	0.094	0.004	36.165	0.031	0.022
8	MOUNTAIN	0.379	0.003	18.509	0.159	0.000
9	PACIFIC CONTIGUOUS	0.010	0.005	10.469	0.230	0.004

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.025	0.076	35.358	0.078	0.471
2	MIDDLE ATLANTIC	0.579	0.036	33.448	0.166	0.192
3	EAST NORTH CENTRAL	1.303	0.006	36.491	0.013	0.200
4	WEST NORTH CENTRAL	1.250	0.007	40.735	0.168	0.221
5	SOUTH ATLANTIC	0.577	0.020	32.533	0.055	0.175
6	EAST SOUTH CENTRAL	0.949	0.004	33.459	0.182	0.000
7	WEST SOUTH CENTRAL	0.132	0.006	46.721	0.036	0.027
8	MOUNTAIN	0.951	0.006	37.334	0.418	0.000
9	PACIFIC CONTIGUOUS	0.030	0.012	19.912	0.482	0.017

Exhibit E-18

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ORGANIC FLUID ABSORPTION GAS HEAT PUMP

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY NOT INCLUDED
(INSULATED DUCTS)

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.008	0.032	37.827	0.035	0.189
2	MIDDLE ATLANTIC	0.219	0.014	31.305	0.061	0.065
3	EAST NORTH CENTRAL	0.519	0.003	36.542	0.005	0.073
4	WEST NORTH CENTRAL	0.483	0.003	35.842	0.059	0.077
5	SOUTH ATLANTIC	0.119	0.004	14.264	0.013	0.039
6	EAST SOUTH CENTRAL	0.220	0.001	17.130	0.052	0.040
7	WEST SOUTH CENTRAL	0.023	0.001	12.824	0.003	0.003
8	MOUNTAIN	0.306	0.002	25.793	0.133	0.030
9	PACIFIC CONTIGUOUS	0.011	0.004	13.323	0.147	0.009

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.008	0.010	13.269	0.015	0.081
2	MIDDLE ATLANTIC	0.127	0.007	17.079	0.038	0.051
3	EAST NORTH CENTRAL	0.233	0.001	15.944	0.002	0.045
4	WEST NORTH CENTRAL	0.255	0.001	21.717	0.044	0.057
5	SOUTH ATLANTIC	0.306	0.010	32.457	0.026	0.089
6	EAST SOUTH CENTRAL	0.455	0.002	31.315	0.073	0.040
7	WEST SOUTH CENTRAL	0.079	0.003	47.764	0.026	0.019
8	MOUNTAIN	0.305	0.002	26.955	0.137	0.000
9	PACIFIC CONTIGUOUS	0.003	0.003	15.196	0.172	0.003

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.015	0.043	51.096	0.050	0.269
2	MIDDLE ATLANTIC	0.345	0.021	48.383	0.099	0.117
3	EAST NORTH CENTRAL	0.752	0.004	52.485	0.007	0.118
4	WEST NORTH CENTRAL	0.739	0.004	57.559	0.103	0.135
5	SOUTH ATLANTIC	0.423	0.014	46.720	0.039	0.127
6	EAST SOUTH CENTRAL	0.675	0.023	48.446	0.125	0.080
7	WEST SOUTH CENTRAL	0.101	0.004	60.584	0.029	0.021
8	MOUNTAIN	0.511	0.004	52.748	0.269	0.000
9	PACIFIC CONTIGUOUS	0.019	0.008	28.515	0.319	0.011

EXHIBIT E-19

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: PULSE COMBUSTION FURNACE / CENTRAL A.C. - HIGH EFFICIENCY

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.004	0.018	44.791	0.020	0.106
2	MID-ATLANTIC	0.125	0.008	37.295	0.035	0.038
3	EAST NORTH CENTRAL	0.203	0.001	43.245	0.003	0.041
4	WEST NORTH CENTRAL	0.273	0.002	42.298	0.033	0.044
5	SOUTH ATLANTIC	0.079	0.003	17.689	0.009	0.025
6	EAST SOUTH CENTRAL	0.140	0.000	21.001	0.033	0.0
7	WEST SOUTH CENTRAL	0.016	0.001	15.363	0.002	0.002
8	MOUNTAIN	0.181	0.001	30.762	0.079	0.0
9	PACIFIC CONTIGUOUS	0.008	0.003	16.604	0.100	0.005

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.036	0.044	0.128	0.064	0.344
2	MID-ATLANTIC	0.542	0.030	0.209	0.161	0.219
3	EAST NORTH CENTRAL	0.993	0.004	0.547	0.008	0.194
4	WEST NORTH CENTRAL	1.086	0.003	3.402	0.187	0.244
5	SOUTH ATLANTIC	1.297	0.044	0.918	0.109	0.379
6	EAST SOUTH CENTRAL	1.931	0.011	0.503	0.308	0.0
7	WEST SOUTH CENTRAL	0.334	0.014	28.758	0.111	0.077
8	MOUNTAIN	1.298	0.009	3.856	0.580	0.0
9	PACIFIC CONTIGUOUS	0.033	0.014	2.181	0.735	0.013

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.040	0.053	44.918	0.084	0.450
2	MID-ATLANTIC	0.667	0.038	37.504	0.196	0.256
3	EAST NORTH CENTRAL	1.295	0.005	43.592	0.011	0.235
4	WEST NORTH CENTRAL	1.353	0.005	45.690	0.220	0.288
5	SOUTH ATLANTIC	1.376	0.046	18.608	0.118	0.404
6	EAST SOUTH CENTRAL	2.071	0.011	21.504	0.341	0.0
7	WEST SOUTH CENTRAL	0.350	0.015	44.121	0.113	0.079
8	MOUNTAIN	1.479	0.010	34.526	0.659	0.0
9	PACIFIC CONTIGUOUS	0.040	0.017	18.785	0.835	0.019

Exhibit E-20

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR RESIDENTIAL SPACE CONDITIONING

SYSTEM: ERICSSON - ERICSSON HEAT PUMP (400 CFM/TON)

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA
DYNAMIC EFFICIENCY NOT INCLUDED

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.215	0.917	0.020	0.995	5.325
2	MIDDLE ATLANTIC	5.152	0.402	0.177	1.738	1.866
3	EAST NORTH CENTRAL	14.597	0.075	3.397	0.150	2.065
4	WEST NORTH CENTRAL	13.593	0.091	10.796	1.062	2.192
5	SOUTH ATLANTIC	3.327	0.114	2.021	0.378	1.071
6	EAST SOUTH CENTRAL	5.219	0.017	0.931	1.472	0.000
7	WEST SOUTH CENTRAL	0.541	0.033	45.782	0.076	0.084
8	MOUNTAIN	8.555	0.052	17.439	3.762	0.000
9	PACIFIC CONTIGUOUS	0.319	0.120	2.490	4.143	0.219

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.037	0.188	0.310	0.155	0.035
2	MIDDLE ATLANTIC	1.329	0.073	0.512	0.394	0.536
3	EAST NORTH CENTRAL	2.427	0.010	0.949	0.219	0.474
4	WEST NORTH CENTRAL	2.592	0.009	8.399	0.451	0.504
5	SOUTH ATLANTIC	3.235	0.109	2.291	0.272	0.945
6	EAST SOUTH CENTRAL	4.915	0.026	1.255	0.769	0.000
7	WEST SOUTH CENTRAL	0.037	0.035	72.051	0.279	0.192
8	MOUNTAIN	3.225	0.022	9.603	1.441	0.000
9	PACIFIC CONTIGUOUS	0.079	0.035	5.315	1.791	0.032

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.381	1.024	0.329	1.150	6.160
2	MIDDLE ATLANTIC	7.490	0.475	0.689	2.132	2.402
3	EAST NORTH CENTRAL	17.124	0.085	4.246	0.169	2.539
4	WEST NORTH CENTRAL	15.355	0.100	19.195	2.124	2.795
5	SOUTH ATLANTIC	5.553	0.223	4.312	0.650	2.015
6	EAST SOUTH CENTRAL	11.033	0.044	2.155	2.241	0.000
7	WEST SOUTH CENTRAL	1.479	0.068	117.832	0.357	0.275
8	MOUNTAIN	11.980	0.074	20.042	5.203	0.000
9	PACIFIC CONTIGUOUS	0.398	0.155	7.803	5.935	0.251

EXHIBIT E-21

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR COMMERCIAL SPACE CONDITIONING

SYSTEM: ELECTRIC HEAT PUMP - STANDARD

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.031	0.134	0.003	0.146	0.781
2	MID-ATLANTIC	0.909	0.059	0.026	0.256	0.275
3	EAST NORTH CENTRAL	2.158	0.011	0.499	0.022	0.303
4	WEST NORTH CENTRAL	2.011	0.013	1.587	0.244	0.322
5	SOUTH ATLANTIC	0.509	0.017	0.309	0.058	0.154
6	EAST SOUTH CENTRAL	0.941	0.003	0.156	0.223	0.0
7	WEST SOUTH CENTRAL	0.100	0.005	7.139	0.012	0.013
8	MOUNTAIN	1.286	0.008	1.551	0.559	0.0
9	PACIFIC CONTIGUOUS	0.049	0.018	0.583	0.637	0.034

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.092	0.102	0.294	0.147	0.792
2	MID-ATLANTIC	1.136	0.052	0.458	0.337	0.459
3	EAST NORTH CENTRAL	2.134	0.008	0.746	0.017	0.417
4	WEST NORTH CENTRAL	2.139	0.007	6.700	0.368	0.482
5	SOUTH ATLANTIC	2.303	0.077	1.650	0.174	0.672
6	EAST SOUTH CENTRAL	3.452	0.019	0.899	0.551	0.0
7	WEST SOUTH CENTRAL	0.568	0.024	48.907	0.189	0.130
8	MOUNTAIN	2.415	0.016	7.193	1.079	0.0
9	PACIFIC CONTIGUOUS	0.072	0.032	4.822	1.625	0.029

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.119	0.237	0.297	0.293	1.573
2	MID-ATLANTIC	2.045	0.121	0.464	0.594	0.734
3	EAST NORTH CENTRAL	4.292	0.019	1.245	0.039	0.720
4	WEST NORTH CENTRAL	4.150	0.020	8.286	0.612	0.804
5	SOUTH ATLANTIC	2.812	0.095	1.940	0.251	0.836
6	EAST SOUTH CENTRAL	4.393	0.021	1.056	0.774	0.0
7	WEST SOUTH CENTRAL	0.668	0.029	56.047	0.202	0.143
8	MOUNTAIN	3.701	0.024	8.745	1.638	0.0
9	PACIFIC CONTIGUOUS	0.121	0.050	5.205	2.263	0.063

EXHIBIT E-22

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR COMMERCIAL SPACE CONDITIONING

SYSTEM: ROOFTOP GAS FURNACE/ELECTRIC AC

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.002	0.009	40.195	0.010	0.051
2	MID-ATLANTIC	0.065	0.004	54.566	0.018	0.020
3	EAST NORTH CENTRAL	0.144	0.001	39.017	0.001	0.020
4	WEST NORTH CENTRAL	0.136	0.001	58.251	0.016	0.022
5	SOUTH ATLANTIC	0.057	0.002	19.842	0.006	0.018
6	EAST SOUTH CENTRAL	0.093	0.000	22.329	0.022	0.0
7	WEST SOUTH CENTRAL	0.013	0.001	18.155	0.002	0.002
8	MOUNTAIN	0.102	0.001	29.619	0.044	0.0
9	PACIFIC CONTIGUOUS	0.006	0.002	19.028	0.074	0.004

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.054	0.067	0.191	0.096	0.516
2	MID-ATLANTIC	0.819	0.045	0.316	0.243	0.331
3	EAST NORTH CENTRAL	1.498	0.006	0.524	0.012	0.292
4	WEST NORTH CENTRAL	1.653	0.005	5.177	0.284	0.372
5	SOUTH ATLANTIC	1.992	0.057	1.410	0.167	0.582
6	EAST SOUTH CENTRAL	2.964	0.016	0.772	0.473	0.0
7	WEST SOUTH CENTRAL	0.515	0.021	44.325	0.172	0.118
8	MOUNTAIN	1.996	0.013	5.915	0.888	0.0
9	PACIFIC CONTIGUOUS	0.049	0.022	3.282	1.106	0.020

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.056	0.075	40.386	0.105	0.567
2	MID-ATLANTIC	0.884	0.049	54.881	0.262	0.350
3	EAST NORTH CENTRAL	1.642	0.007	39.540	0.013	0.313
4	WEST NORTH CENTRAL	1.789	0.006	43.439	0.301	0.394
5	SOUTH ATLANTIC	2.049	0.059	21.253	0.174	0.600
6	EAST SOUTH CENTRAL	3.057	0.016	23.101	0.495	0.0
7	WEST SOUTH CENTRAL	0.528	0.022	62.480	0.173	0.120
8	MOUNTAIN	2.088	0.014	35.534	0.932	0.0
9	PACIFIC CONTIGUOUS	0.055	0.024	22.311	1.180	0.024

EXHIBIT E-23

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR COMMERCIAL SPACE CONDITIONING

SYSTEM: ELECTRIC RESISTANCE HEATING/ELECTRIC AC

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.069	0.295	0.006	0.320	1.714
2	MID-ATLANTIC	2.041	0.153	0.059	0.576	0.618
3	EAST NORTH CENTRAL	4.757	0.024	1.100	0.049	0.668
4	WEST NORTH CENTRAL	4.445	0.030	3.508	0.540	0.712
5	SOUTH ATLANTIC	1.323	0.045	0.804	0.151	0.426
6	EAST SOUTH CENTRAL	2.343	0.007	0.339	0.555	0.0
7	WEST SOUTH CENTRAL	0.276	0.014	19.751	0.034	0.036
8	MOUNTAIN	2.976	0.018	3.589	1.293	0.0
9	PACIFIC CONTIGUOUS	0.130	0.049	1.013	1.685	0.099

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.052	0.064	0.185	0.092	0.498
2	MID-ATLANTIC	0.781	0.043	0.301	0.232	0.315
3	EAST NORTH CENTRAL	1.432	0.006	0.501	0.011	0.280
4	WEST NORTH CENTRAL	1.562	0.005	4.893	0.269	0.352
5	SOUTH ATLANTIC	1.859	0.062	1.316	0.156	0.543
6	EAST SOUTH CENTRAL	2.768	0.015	0.721	0.442	0.0
7	WEST SOUTH CENTRAL	0.478	0.020	41.167	0.159	0.110
8	MOUNTAIN	1.864	0.013	5.550	0.833	0.0
9	PACIFIC CONTIGUOUS	0.047	0.021	3.151	1.062	0.019

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.121	0.359	0.191	0.413	2.212
2	MID-ATLANTIC	2.822	0.176	0.360	0.807	0.933
3	EAST NORTH CENTRAL	6.199	0.030	1.600	0.060	0.948
4	WEST NORTH CENTRAL	6.008	0.035	8.400	0.809	1.064
5	SOUTH ATLANTIC	3.182	0.108	2.120	0.307	0.969
6	EAST SOUTH CENTRAL	5.111	0.022	1.061	0.997	0.0
7	WEST SOUTH CENTRAL	0.755	0.034	60.918	0.193	0.146
8	MOUNTAIN	4.839	0.030	9.139	2.126	0.0
9	PACIFIC CONTIGUOUS	0.177	0.070	4.164	2.747	0.108

EXHIBIT E-24

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR COMMERCIAL SPACE CONDITIONING

SYSTEM: BRAYTON-RANKINE GAS HEAT PUMP

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.006	0.024	18.097	0.026	0.100
2	MID-ATLANTIC	0.163	0.011	13.106	0.046	0.089
3	EAST NORTH CENTRAL	0.386	0.002	17.592	0.004	0.054
4	WEST NORTH CENTRAL	0.360	0.002	17.295	0.044	0.058
5	SOUTH ATLANTIC	0.093	0.003	7.308	0.011	0.030
6	EAST SOUTH CENTRAL	0.170	0.000	8.611	0.040	0.0
7	WEST SOUTH CENTRAL	0.018	0.001	7.195	0.002	0.002
8	MOUNTAIN	0.231	0.001	12.685	0.100	0.0
9	PACIFIC CONTIGUOUS	0.009	0.003	6.894	0.116	0.006

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.008	0.010	11.519	0.014	0.076
2	MID-ATLANTIC	0.118	0.006	14.762	0.035	0.048
3	EAST NORTH CENTRAL	0.217	0.001	13.799	0.002	0.042
4	WEST NORTH CENTRAL	0.237	0.001	18.766	0.041	0.053
5	SOUTH ATLANTIC	0.282	0.009	27.858	0.024	0.082
6	EAST SOUTH CENTRAL	0.419	0.002	26.880	0.067	0.0
7	WEST SOUTH CENTRAL	0.072	0.003	41.329	0.024	0.017
8	MOUNTAIN	0.282	0.002	23.229	0.126	0.0
9	PACIFIC CONTIGUOUS	0.007	0.003	13.194	0.161	0.003

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.013	0.034	29.616	0.040	0.215
2	MID-ATLANTIC	0.281	0.017	29.868	0.081	0.097
3	EAST NORTH CENTRAL	0.604	0.003	31.341	0.006	0.097
4	WEST NORTH CENTRAL	0.597	0.003	36.062	0.084	0.111
5	SOUTH ATLANTIC	0.374	0.013	35.167	0.034	0.112
6	EAST SOUTH CENTRAL	0.590	0.003	35.491	0.107	0.0
7	WEST SOUTH CENTRAL	0.091	0.004	48.514	0.026	0.019
8	MOUNTAIN	0.514	0.003	35.913	0.227	0.0
9	PACIFIC CONTIGUOUS	0.016	0.007	20.078	0.277	0.009

EXHIBIT E-25

PRIMARY FUELS AND OTHER ENERGY SOURCES REQUIRED FOR COMMERCIAL SPACE CONDITIONING

SYSTEM: RANKINE-RANKINE GAS HEAT PUMP

BASIS: PER SQUARE FOOT OF RESIDENCE LIVING AREA

HEATING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.0	0.0	15.812	0.0	0.0
2	MID-ATLANTIC	0.0	0.0	13.423	0.0	0.0
3	EAST NORTH CENTRAL	0.0	0.0	15.298	0.0	0.0
4	WEST NORTH CENTRAL	0.0	0.0	14.946	0.0	0.0
5	SOUTH ATLANTIC	0.0	0.0	7.164	0.0	0.0
6	EAST SOUTH CENTRAL	0.0	0.0	8.228	0.0	0.0
7	WEST SOUTH CENTRAL	0.0	0.0	6.069	0.0	0.0
8	MOUNTAIN	0.0	0.0	11.274	0.0	0.0
9	PACIFIC CONTIGUOUS	0.0	0.0	6.815	0.0	0.0

COOLING SEASON

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.0	0.0	11.495	0.0	0.0
2	MID-ATLANTIC	0.0	0.0	15.221	0.0	0.0
3	EAST NORTH CENTRAL	0.0	0.0	14.074	0.0	0.0
4	WEST NORTH CENTRAL	0.0	0.0	19.041	0.0	0.0
5	SOUTH ATLANTIC	0.0	0.0	30.166	0.0	0.0
6	EAST SOUTH CENTRAL	0.0	0.0	29.140	0.0	0.0
7	WEST SOUTH CENTRAL	0.0	0.0	38.756	0.0	0.0
8	MOUNTAIN	0.0	0.0	24.079	0.0	0.0
9	PACIFIC CONTIGUOUS	0.0	0.0	12.911	0.0	0.0

TOTAL ANNUAL

	CENSUS REGION	COAL(LBS)	OIL(GALS)	GAS(CFT)	HYDRO(KWH)	NUCLEAR(KWH)
1	NEW ENGLAND	0.0	0.0	27.308	0.0	0.0
2	MID-ATLANTIC	0.0	0.0	28.644	0.0	0.0
3	EAST NORTH CENTRAL	0.0	0.0	29.372	0.0	0.0
4	WEST NORTH CENTRAL	0.0	0.0	33.987	0.0	0.0
5	SOUTH ATLANTIC	0.0	0.0	37.330	0.0	0.0
6	EAST SOUTH CENTRAL	0.0	0.0	37.368	0.0	0.0
7	WEST SOUTH CENTRAL	0.0	0.0	44.824	0.0	0.0
8	MOUNTAIN	0.0	0.0	35.353	0.0	0.0
9	PACIFIC CONTIGUOUS	0.0	0.0	19.725	0.0	0.0

APPENDIX F

DETAILED CALCULATION OF PRIMARY ENERGY IMPACTS BASED ON REGIONAL MARKET PENETRATION PROJECTIONS

EXHIBIT F-1a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

ADVANCED ELECTRIC HEAT PUMP II VS ELECTRIC HEAT PUMP - STANDARD

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-4.8	-0.5	-0.5	-1.6	-3.0
1985	NORTH CENTRAL	-56.4	-0.3	-29.2	-2.5	-8.8
1985	SOUTH	-82.0	-1.9	-561.3	-12.4	-14.4
1985	WEST	-10.2	-0.5	-37.6	-19.6	-0.6
1985	TOTAL U.S.	-153.3	-3.2	-628.6	-36.1	-26.8
1990	NORTH EAST	-18.9	-2.1	-2.1	-6.4	-11.8
1990	NORTH CENTRAL	-225.6	-1.2	-116.6	-9.9	-35.0
1990	SOUTH	-327.8	-7.6	-2244.4	-49.5	-57.7
1990	WEST	-40.6	-1.8	-150.3	-78.4	-2.6
1990	TOTAL U.S.	-613.0	-12.7	-2513.4	-144.2	-107.0
1995	NORTH EAST	-34.1	-3.9	-3.8	-11.5	-21.2
1995	NORTH CENTRAL	-406.3	-2.1	-210.0	-17.9	-63.0
1995	SOUTH	-590.2	-13.6	-4040.9	-89.1	-103.8
1995	WEST	-73.1	-3.3	-270.3	-141.1	-4.6
1995	TOTAL U.S.	-1103.6	-22.9	-4525.0	-259.6	-192.7
2000	NORTH EAST	-49.2	-5.6	-5.5	-16.6	-30.7
2000	NORTH CENTRAL	-586.2	-3.1	-302.9	-25.8	-91.0
2000	SOUTH	-851.4	-19.6	-5829.2	-128.5	-149.7
2000	WEST	-105.4	-4.8	-389.9	-203.5	-6.7
2000	TOTAL U.S.	-1592.1	-33.1	-6527.5	-374.4	-278.0

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-10.3	-1.2	-1.2	-3.5	-6.5
1985	NORTH CENTRAL	-123.9	-0.7	-64.0	-5.4	-19.2
1985	SOUTH	-179.8	-4.1	-1231.1	-27.1	-31.6
1985	WEST	-22.3	-1.0	-82.4	-43.0	-1.4
1985	TOTAL U.S.	-336.3	-7.0	-1378.7	-79.1	-58.7
1990	NORTH EAST	-50.6	-5.7	-5.7	-17.1	-31.5
1990	NORTH CENTRAL	-604.0	-3.2	-312.2	-26.6	-93.7
1990	SOUTH	-877.3	-20.2	-6006.3	-132.4	-154.3
1990	WEST	-108.6	-4.9	-401.7	-209.7	-6.9
1990	TOTAL U.S.	-1640.4	-34.0	-6725.8	-385.7	-286.4
1995	NORTH EAST	-108.4	-12.3	-12.2	-36.6	-67.6
1995	NORTH CENTRAL	-1176.3	-6.2	-607.9	-51.7	-182.5
1995	SOUTH	-1864.2	-43.0	-12763.5	-281.4	-327.9
1995	WEST	-230.8	-10.4	-853.7	-445.6	-14.6
199[illegible]	[illegible]AL U.S.	-3379.7	-71.9	-14237.3	-815.3	-592.6
200[illegible]	[illegible]TH EAST	-181.9	-20.6	-20.4	-61.4	-113.5
2000	NORTH CENTRAL	-2160.9	-11.4	-1116.8	-95.1	-335.3
2000	SOUTH	-3138.6	-72.4	-21488.6	-473.7	-552.0
2000	WEST	-389.0	-17.6	-1438.9	-751.1	-24.6
2000	TOTAL U.S.	-5870.3	-121.9	-24064.6	-1381.3	-1025.3

COAL - LBX10-6 OIL - GALX10-6 GAS - CFTX10-6 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-1b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

ADVANCED ELECTRIC HEAT PUMP II VS ELECTRIC HEAT PUMP - STANDARD

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-48.8	-78.5	-0.5	-1.6	-5.0
1985	NORTH CENTRAL	-577.3	-45.1	-29.6	-2.5	-8.8
1985	SOUTH	-839.1	-274.5	-569.2	-12.4	-14.4
1985	WEST	-104.1	-66.7	-38.2	-19.6	-0.6
1985	TOTAL U.S.	-1569.3	-462.6	-637.4	-36.1	-26.8
1990	NORTH EAST	-193.9	-311.5	-2.2	-6.4	-11.8
1990	NORTH CENTRAL	-2309.2	-172.2	-118.2	-9.9	-35.0
1990	SOUTH	-3355.2	-1097.7	-2275.8	-49.5	-57.7
1990	WEST	-415.8	-266.4	-152.4	-78.4	-2.6
1990	TOTAL U.S.	-6274.1	-1847.6	-2548.6	-144.2	-107.0
1995	NORTH EAST	-348.6	-559.6	-3.9	-11.5	-21.2
1995	NORTH CENTRAL	-4158.0	-310.1	-212.9	-17.9	-63.0
1995	SOUTH	-6040.8	-1976.5	-4097.5	-89.1	-103.8
1995	WEST	-747.9	-479.1	-274.1	-141.1	-4.6
1995	TOTAL U.S.	-11295.3	-3325.1	-4588.4	-259.6	-192.7
2000	NORTH EAST	-503.2	-807.8	-5.6	-16.6	-30.7
2000	NORTH CENTRAL	-5999.5	-447.5	-307.2	-25.8	-91.0
2000	SOUTH	-8714.1	-2850.9	-5910.8	-128.5	-149.7
2000	WEST	-1078.7	-691.1	-395.4	-203.5	-6.7
2000	TOTAL U.S.	-16295.6	-4797.3	-6618.9	-374.4	-278.0

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-105.9	-170.0	-1.2	-3.5	-6.5
1985	NORTH CENTRAL	-1267.9	-94.6	-64.9	-5.4	-19.2
1985	SOUTH	-1840.3	-602.1	-1248.3	-27.1	-31.6
1985	WEST	-228.0	-146.1	-83.6	-43.0	-1.4
1985	TOTAL U.S.	-3442.1	-1012.7	-1398.0	-79.1	-58.7
1990	NORTH EAST	-517.5	-830.7	-5.7	-17.1	-31.5
1990	NORTH CENTRAL	-6182.2	-461.1	-316.5	-26.6	-93.7
1990	SOUTH	-8979.0	-2937.5	-6090.4	-132.4	-154.3
1990	WEST	-1111.3	-711.9	-407.3	-209.7	-5.9
1990	TOTAL U.S.	-16790.0	-4941.3	-6820.0	-385.7	-286.4
1995	NORTH EAST	-1109.9	-1781.8	-12.3	-36.6	-67.6
1995	NORTH CENTRAL	-12039.2	-897.9	-616.4	-51.7	-182.5
1995	SOUTH	-19080.5	-6242.3	-12942.2	-281.4	-327.9
1995	WEST	-2361.9	-1513.1	-865.6	-445.6	-14.6
1995	TOTAL U.S.	-34591.5	-10435.1	-14436.6	-815.3	-592.6
2000	NORTH EAST	-1861.8	-2988.8	-20.7	-61.4	-113.5
2000	NORTH CENTRAL	-22116.3	-1649.5	-1132.4	-95.1	-335.3
2000	SOUTH	-32123.8	-10509.6	-21789.5	-473.7	-552.0
2000	WEST	-3991.0	-2550.3	-1459.0	-751.1	-24.6
2000	TOTAL U.S.	-60082.8	-17698.1	-24401.5	-1381.3	-1025.3

COAL - BTUX10-9 OIL - BTUX10-9 GAS - BTUX10-9 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-2a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

FREE PISTON STIRLING-RANKINE HEAT PUMP VS ELECTRIC HEAT PUMP - HIGH EFFICIENCY II

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-10.7	-1.2	246.7	-3.6	-6.7
1985	NORTH CENTRAL	-130.4	-0.7	988.9	-5.7	-20.2
1985	SOUTH	-80.5	-1.9	1152.3	-12.2	-14.1
1985	WEST	-20.9	-0.9	580.3	-40.0	-1.3
1985	TOTAL U.S.	-242.6	-4.7	2968.3	-61.6	-42.3
1990	NORTH EAST	-33.1	-3.8	763.7	-11.2	-20.7
1990	NORTH CENTRAL	-403.3	-2.1	3058.1	-17.7	-62.3
1990	SOUTH	-248.5	-5.7	3555.8	-37.6	-43.7
1990	WEST	-64.7	-2.9	1793.7	-125.8	-4.1
1990	TOTAL U.S.	-749.7	-14.5	9171.3	-190.3	-130.8
1995	NORTH EAST	-55.2	-6.3	1272.9	-18.7	-34.5
1995	NORTH CENTRAL	-672.0	-3.5	5095.5	-29.5	-103.9
1995	SOUTH	-414.5	-9.5	5929.6	-62.8	-72.8
1995	WEST	-108.0	-4.8	2992.0	-206.4	-6.8
1995	TOTAL U.S.	-1249.7	-24.2	15290.0	-317.3	-218.0
2000	NORTH EAST	-77.7	-8.8	1789.9	-26.2	-48.4
2000	NORTH CENTRAL	-945.9	-5.0	7172.7	-41.5	-146.2
2000	SOUTH	-583.2	-13.4	8342.9	-88.5	-102.4
2000	WEST	-151.8	-6.8	4207.9	-290.3	-9.6
2000	TOTAL U.S.	-1758.6	-34.1	21513.3	-446.4	-306.7

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-24.5	-2.8	564.0	-8.3	-15.3
1985	NORTH CENTRAL	-298.3	-1.6	2263.8	-13.1	-46.1
1985	SOUTH	-184.1	-4.2	2633.4	-27.9	-32.3
1985	WEST	-48.0	-2.2	1328.9	-91.7	-3.0
1985	TOTAL U.S.	-555.0	-10.7	6790.2	-140.9	-96.8
1990	NORTH EAST	-81.4	-9.3	1876.0	-27.5	-50.8
1990	NORTH CENTRAL	-1112.5	-5.9	9435.6	-48.8	-172.0
1990	SOUTH	-639.3	-14.7	9146.6	-96.8	-112.3
1990	WEST	-178.7	-8.0	4951.5	-341.6	-11.3
1990	TOTAL U.S.	-2011.9	-37.9	24409.7	-514.7	-346.3
1995	NORTH EAST	-168.8	-19.2	3889.2	-57.0	-105.3
1995	NORTH CENTRAL	-2175.2	-11.5	15493.9	-95.4	-336.2
1995	SOUTH	-1341.2	-30.9	19188.3	-203.2	-235.6
1995	WEST	-349.3	-15.7	9679.4	-567.8	-22.1
1995	TOTAL U.S.	-4034.5	-77.2	49250.7	-1023.4	-699.2
2000	NORTH EAST	-286.2	-32.5	6595.5	-96.7	-178.5
2000	NORTH CENTRAL	-3603.5	-19.0	27324.4	-158.0	-557.0
2000	SOUTH	-2232.4	-51.4	31937.6	-338.2	-392.1
2000	WEST	-578.6	-26.0	16035.1	-1106.3	-36.6
2000	TOTAL U.S.	-6700.7	-128.9	81892.6	-1699.1	-1164.2

COAL - LBX10-6 OIL - GALX10-6 GAS - CFIX10-6 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-2b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

FREE PISTON STIRLING-RANKINE HEAT PUMP VS ELECTRIC HEAT PUMP - HIGH EFFICIENCY II

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-109.6	-176.6	250.2	-5.6	-6.7
1985	NORTH CENTRAL	-1334.8	-99.9	1002.8	-5.7	-20.2
1985	SOUTH	-824.4	-269.1	1168.4	-12.2	-14.1
1985	WEST	-214.5	-136.5	588.4	-40.0	-1.5
1985	TOTAL U.S.	-2483.1	-682.1	3009.8	-61.6	-42.5
1990	NORTH EAST	-339.2	-546.7	774.4	-11.2	-20.7
1990	NORTH CENTRAL	-4127.8	-308.9	3100.9	-17.7	-62.5
1990	SOUTH	-2543.9	-830.5	3605.5	-37.6	-43.7
1990	WEST	-662.4	-421.8	1818.8	-123.8	-4.1
1990	TOTAL U.S.	-7673.5	-2108.0	9299.7	-190.5	-130.8
1995	NORTH EAST	-565.4	-911.2	1290.7	-18.7	-34.5
1995	NORTH CENTRAL	-6877.3	-514.8	5166.9	-29.3	-103.9
1995	SOUTH	-4242.1	-1385.0	6012.6	-62.8	-72.8
1995	WEST	-1105.0	-703.6	3033.9	-206.4	-6.8
1995	TOTAL U.S.	-12790.3	-3514.6	15504.0	-317.3	-218.0
2000	NORTH EAST	-795.0	-1281.3	1814.9	-26.2	-48.4
2000	NORTH CENTRAL	-9681.5	-724.6	7273.1	-41.5	-146.2
2000	SOUTH	-5968.7	-1948.7	8459.7	-88.3	-102.4
2000	WEST	-1554.0	-989.6	4266.8	-290.3	-9.6
2000	TOTAL U.S.	-17999.2	-4944.1	21814.5	-446.4	-306.7

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-250.5	-403.7	571.9	-8.3	-15.3
1985	NORTH CENTRAL	-3055.6	-228.7	2295.5	-13.1	-46.1
1985	SOUTH	-1884.0	-615.1	2670.3	-27.9	-32.3
1985	WEST	-490.8	-312.5	1347.5	-91.7	-3.0
1985	TOTAL U.S.	-5680.9	-1560.1	6885.2	-140.9	-96.8
1990	NORTH EAST	-833.3	-1343.0	1902.3	-27.5	-50.8
1990	NORTH CENTRAL	-11386.2	-852.2	8553.7	-48.8	-172.0
1990	SOUTH	-6543.6	-2136.4	9274.6	-96.8	-112.3
1990	WEST	-1828.6	-1164.4	5020.8	-341.6	-11.3
1990	TOTAL U.S.	-20591.7	-5496.0	24751.4	-514.7	-346.3
1995	NORTH EAST	-1727.5	-2784.1	3943.6	-57.0	-105.3
1995	NORTH CENTRAL	-22263.1	-1666.3	16724.9	-95.4	-336.2
1995	SOUTH	-13727.6	-4481.8	19456.9	-203.2	-235.6
1995	WEST	-3574.6	-2276.3	9814.9	-667.8	-22.1
1995	TOTAL U.S.	-41292.8	-11208.5	49940.2	-1023.4	-699.2
2000	NORTH EAST	-2929.6	-4721.4	6687.8	-96.7	-178.5
2000	NORTH CENTRAL	-36881.9	-2760.4	27707.0	-158.0	-557.0
2000	SOUTH	-22848.7	-7459.7	32384.7	-338.2	-392.1
2000	WEST	-5921.8	-3771.0	16259.6	-1106.3	-36.6
2000	TOTAL U.S.	-68581.9	-18712.5	83039.1	-1699.1	-1164.2

COAL - BTUX10-9 OIL - BTUX10-9 GAS - BTUX10-9 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-3a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

FREE PISTON STIRLING-RANKINE HEAT PUMP VS GAS FURNACE/CENTRAL AIR COND HIGH EFF

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-0.8	-0.0	-172.2	-0.3	-0.5
1985	NORTH CENTRAL	-4.8	0.0	-752.5	-0.5	-1.8
1985	SOUTH	-34.6	-0.9	-159.4	-4.5	-6.3
1985	WEST	-3.6	-0.2	-258.7	-9.2	-0.0
1985	TOTAL U.S.	-43.8	-1.0	-1342.8	-14.5	-8.7
1990	NORTH EAST	-2.4	-0.1	-533.1	-0.8	-1.5
1990	NORTH CENTRAL	-14.9	0.0	-2326.9	-1.7	-5.7
1990	SOUTH	-106.7	-2.6	-491.8	-13.9	-19.4
1990	WEST	-11.2	-0.5	-799.8	-28.4	-0.1
1990	TOTAL U.S.	-135.3	-3.1	-4151.5	-44.8	-26.8
1995	NORTH EAST	-4.1	-0.1	-888.4	-1.3	-2.6
1995	NORTH CENTRAL	-24.8	0.0	-3877.1	-2.8	-9.4
1995	SOUTH	-177.9	-4.4	-820.1	-23.2	-32.4
1995	WEST	-18.8	-0.8	-1334.1	-47.3	-0.2
1995	TOTAL U.S.	-225.6	-5.2	-6919.8	-74.7	-44.6
2000	NORTH EAST	-5.7	-0.2	-1249.3	-1.8	-3.6
2000	NORTH CENTRAL	-34.9	0.1	-5457.6	-4.0	-13.3
2000	SOUTH	-250.3	-6.2	-1154.0	-32.6	-45.6
2000	WEST	-26.4	-1.1	-1876.2	-66.6	-0.3
2000	TOTAL U.S.	-317.4	-7.4	-9737.1	-105.0	-62.8

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-1.8	-0.1	-393.6	-0.6	-1.1
1985	NORTH CENTRAL	-11.0	0.0	-1722.5	-1.2	-4.2
1985	SOUTH	-79.0	-1.9	-364.2	-10.3	-14.4
1985	WEST	-8.3	-0.3	-592.5	-21.0	-0.1
1985	TOTAL U.S.	-100.2	-2.3	-3072.9	-33.2	-19.8
1990	NORTH EAST	-6.0	-0.2	-1309.4	-1.9	-3.8
1990	NORTH CENTRAL	-41.1	0.1	-6418.6	-4.7	-15.6
1990	SOUTH	-274.4	-6.8	-1265.1	-35.8	-50.0
1990	WEST	-31.1	-1.3	-2207.8	-78.3	-0.4
1990	TOTAL U.S.	-352.6	-8.1	-11200.9	-120.7	-69.8
1995	NORTH EAST	-12.4	-0.4	-2714.5	-4.0	-7.8
1995	NORTH CENTRAL	-80.3	0.1	-12550.1	-9.1	-30.5
1995	SOUTH	-575.7	-14.2	-2654.0	-75.0	-105.0
1995	WEST	-60.7	-2.5	-4315.9	-155.1	-0.8
1995	TOTAL U.S.	-729.2	-16.9	-22234.5	-241.3	-144.0
2000	NORTH EAST	-21.0	-0.6	-4603.5	-6.8	-13.2
2000	NORTH CENTRAL	-133.1	0.2	-20790.9	-15.1	-50.5
2000	SOUTH	-958.3	-23.6	-4417.4	-124.9	-174.7
2000	WEST	-100.6	-4.2	-7149.8	-253.7	-1.2
2000	TOTAL U.S.	-1213.0	-28.1	-36961.6	-400.5	-239.7

COAL - LBX10-6 OIL - GALX10-6 GAS - CFTX10-6 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-3b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

FREE PISTON STIRLING-RANKINE HEAT PUMP VS GAS FURNACE/CENTRAL AIR COND HIGH EFF

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-8.1	-3.4	-174.6	-0.3	-0.5
1985	NORTH CENTRAL	-49.3	1.2	-765.0	-0.5	-1.8
1985	SOUTH	-353.9	-123.5	-161.6	-4.5	-6.3
1985	WEST	-37.3	-21.8	-262.4	-9.2	-0.0
1985	TOTAL U.S.	-448.5	-147.5	-1361.6	-14.5	-8.7
1990	NORTH EAST	-24.9	-10.6	-540.5	-0.8	-1.5
1990	NORTH CENTRAL	-152.4	3.7	-2359.5	-1.7	-5.7
1990	SOUTH	-1092.0	-381.0	-498.7	-15.9	-19.4
1990	WEST	-115.1	-67.5	-811.0	-28.4	-0.1
1990	TOTAL U.S.	-1384.5	-455.4	-4209.7	-44.8	-26.8
1995	NORTH EAST	-41.6	-17.6	-900.9	-1.3	-2.6
1995	NORTH CENTRAL	-254.0	6.1	-3931.4	-2.8	-9.4
1995	SOUTH	-1821.0	-535.4	-831.6	-23.2	-32.4
1995	WEST	-192.1	-112.5	-1352.8	-47.3	-0.2
1995	TOTAL U.S.	-2308.5	-759.4	-7016.7	-74.7	-44.6
2000	NORTH EAST	-58.4	-24.8	-1266.8	-1.8	-3.6
2000	NORTH CENTRAL	-357.5	8.6	-5534.0	-4.0	-13.3
2000	SOUTH	-2562.2	-894.0	-1170.1	-32.6	-45.6
2000	WEST	-270.1	-158.3	-1902.5	-66.6	-0.3
2000	TOTAL U.S.	-3248.2	-1068.4	-9873.4	-105.0	-62.8

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-18.4	-7.8	-399.2	-0.6	-1.1
1985	NORTH CENTRAL	-112.8	2.7	-1746.6	-1.2	-4.2
1985	SOUTH	-808.7	-282.2	-369.3	-10.3	-14.4
1985	WEST	-85.3	-50.0	-600.8	-21.0	-0.1
1985	TOTAL U.S.	-1025.3	-337.3	-3116.0	-33.2	-19.8
1990	NORTH EAST	-61.2	-26.0	-1327.8	-1.9	-3.8
1990	NORTH CENTRAL	-420.4	10.1	-6508.5	-4.7	-15.6
1990	SOUTH	-2808.9	-980.1	-1282.8	-35.3	-50.0
1990	WEST	-317.8	-186.2	-2238.7	-78.3	-0.4
1990	TOTAL U.S.	-3608.5	-1182.2	-11357.7	-120.7	-69.8
1995	NORTH EAST	-127.0	-53.9	-2752.5	-4.0	-7.8
1995	NORTH CENTRAL	-822.0	19.8	-12725.8	-9.1	-30.5
1995	SOUTH	-5892.8	-2056.2	-2691.2	-75.0	-105.0
1995	WEST	-621.3	-364.0	-4376.3	-153.1	-0.8
1995	TOTAL U.S.	-7463.1	-2454.2	-22545.8	-241.3	-144.0
2000	NORTH EAST	-215.3	-91.3	-4667.9	-6.8	-13.2
2000	NORTH CENTRAL	-1361.8	32.9	-21082.0	-15.1	-50.5
2000	SOUTH	-9808.2	-3422.4	-4479.3	-124.9	-174.7
2000	WEST	-1029.3	-603.1	-7249.9	-253.7	-1.2
2000	TOTAL U.S.	-12414.6	-4083.9	-37479.1	-400.5	-239.7

COAL - BTUX10-9 OIL - BTUX10-9 GAS - BTUX10-9 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-4a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

ORGANIC FLUID ABSORPTION GAS HEAT PUMP VS GAS FURNACE/CENTRAL AIR COND HIGH EFF

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-1.5	-0.1	-74.7	-0.5	-0.9
1985	NORTH CENTRAL	-13.6	-0.0	-336.4	-0.9	-5.1
1985	SOUTH	-38.0	-0.9	659.7	-5.1	-6.9
1985	WEST	-4.9	-0.2	8.3	-11.5	-0.1
1985	TOTAL U.S.	-58.0	-1.5	257.0	-18.0	-11.1
1990	NORTH EAST	-4.6	-0.5	-231.3	-1.5	-2.9
1990	NORTH CENTRAL	-42.0	-0.1	-1040.1	-2.8	-9.7
1990	SOUTH	-117.4	-2.9	2035.9	-15.7	-21.5
1990	WEST	-15.1	-0.6	25.7	-35.6	-0.4
1990	TOTAL U.S.	-179.1	-4.0	790.1	-55.6	-34.5
1995	NORTH EAST	-7.7	-0.6	-385.5	-2.5	-4.8
1995	NORTH CENTRAL	-70.0	-0.2	-1735.1	-4.7	-16.1
1995	SOUTH	-195.8	-4.8	3395.0	-26.1	-35.5
1995	WEST	-25.1	-1.1	42.8	-59.5	-0.7
1995	TOTAL U.S.	-298.5	-6.6	1319.1	-92.8	-57.1
2000	NORTH EAST	-10.8	-0.8	-542.1	-3.6	-6.8
2000	NORTH CENTRAL	-98.5	-0.5	-2439.6	-6.6	-22.6
2000	SOUTH	-275.4	-6.7	4776.7	-36.8	-49.9
2000	WEST	-35.4	-1.5	60.2	-83.6	-1.0
2000	TOTAL U.S.	-420.1	-9.5	1855.2	-130.5	-80.4

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-3.4	-0.2	-170.8	-1.1	-2.1
1985	NORTH CENTRAL	-31.1	-0.1	-770.0	-2.1	-7.1
1985	SOUTH	-86.9	-2.1	1507.8	-11.6	-15.8
1985	WEST	-11.2	-0.5	19.0	-26.4	-0.5
1985	TOTAL U.S.	-132.6	-2.9	586.0	-41.2	-25.4
1990	NORTH EAST	-11.5	-0.8	-568.2	-3.7	-7.1
1990	NORTH CENTRAL	-115.8	-0.5	-2869.2	-7.7	-26.6
1990	SOUTH	-302.0	-7.4	5236.9	-40.5	-54.8
1990	WEST	-41.6	-1.8	70.8	-98.4	-1.2
1990	TOTAL U.S.	-470.7	-10.5	1870.5	-150.2	-89.7
1995	NORTH EAST	-23.4	-1.7	-1178.0	-7.8	-14.7
1995	NORTH CENTRAL	-226.5	-0.7	-3610.0	-15.1	-32.0
1995	SOUTH	-633.5	-15.4	10986.2	-84.5	-114.9
1995	WEST	-81.5	-3.5	138.5	-192.4	-2.5
1995	TOTAL U.S.	-964.7	-21.5	4336.8	-299.8	-183.9
2000	NORTH EAST	-39.8	-2.9	-1997.7	-13.2	-24.9
2000	NORTH CENTRAL	-375.2	-1.1	-9293.7	-25.1	-86.2
2000	SOUTH	-1034.4	-25.7	18285.9	-140.7	-191.2
2000	WEST	-134.7	-5.8	229.4	-318.7	-3.8
2000	TOTAL U.S.	-1604.0	-35.5	7223.9	-497.6	-306.2

COAL = LBX10-6 OIL = GALX10-6 GAS = CFIX10-6 HYDRO = KWHX10-6 NUCLEAR = KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-4b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

ORGANIC FLUID ABSORPTION GAS HEAT PUMP VS GAS FURNACE/CENTRAL AIR COND HIGH EFF

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-15.2	-15.5	-75.8	-0.5	-0.9
1985	NORTH CENTRAL	-139.0	-5.8	-341.1	-0.9	-3.1
1985	SOUTH	-389.3	-134.6	669.0	-5.1	-6.9
1985	WEST	-49.9	-30.4	8.4	-11.5	-0.1
1985	TOTAL U.S.	-593.5	-186.3	260.5	-18.0	-11.1
1990	NORTH EAST	-47.1	-48.0	-234.6	-1.5	-2.9
1990	NORTH CENTRAL	-429.8	-17.9	-1054.7	-2.8	-9.7
1990	SOUTH	-1201.5	-413.3	2064.4	-15.7	-21.3
1990	WEST	-154.3	-94.0	26.0	-35.6	-0.4
1990	TOTAL U.S.	-1832.6	-573.3	801.1	-55.6	-34.3
1995	NORTH EAST	-78.5	-80.0	-390.9	-2.5	-4.8
1995	NORTH CENTRAL	-716.1	-29.9	-1757.4	-4.7	-16.1
1995	SOUTH	-2003.6	-692.6	3442.5	-26.1	-35.5
1995	WEST	-257.3	-156.8	43.4	-59.5	-0.7
1995	TOTAL U.S.	-3055.5	-959.3	1337.6	-92.8	-57.1
2000	NORTH EAST	-110.4	-112.6	-549.7	-3.6	-6.8
2000	NORTH CENTRAL	-1008.0	-42.1	-2473.7	-6.6	-22.6
2000	SOUTH	-2819.0	-974.4	4843.6	-36.8	-49.9
2000	WEST	-361.9	-220.6	61.0	-83.6	-1.0
2000	TOTAL U.S.	-4299.3	-1349.8	1881.2	-130.9	-80.4

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-34.8	-35.5	-173.2	-1.1	-2.1
1985	NORTH CENTRAL	-318.1	-13.3	-780.8	-2.1	-7.1
1985	SOUTH	-889.8	-307.6	1528.9	-11.6	-15.8
1985	WEST	-114.3	-69.7	19.3	-26.4	-0.3
1985	TOTAL U.S.	-1357.0	-426.0	594.2	-41.2	-25.4
1990	NORTH EAST	-115.7	-118.0	-576.2	-3.7	-7.1
1990	NORTH CENTRAL	-1185.5	-49.3	-2909.3	-7.7	-26.6
1990	SOUTH	-3090.6	-1068.3	5310.2	-40.3	-54.8
1990	WEST	-425.8	-259.6	71.8	-98.4	-1.2
1990	TOTAL U.S.	-4817.6	-1495.3	1896.5	-150.2	-89.7
1995	NORTH EAST	-239.9	-244.6	-1194.5	-7.8	-14.7
1995	NORTH CENTRAL	-2317.9	-96.7	-5688.5	-15.1	-32.0
1995	SOUTH	-6483.6	-2241.1	11140.0	-84.3	-114.9
1995	WEST	-832.4	-507.4	140.4	-192.4	-2.3
1995	TOTAL U.S.	-9873.9	-3089.8	4397.5	-299.8	-163.9
2000	NORTH EAST	-406.9	-414.8	-2025.6	-13.2	-24.9
2000	NORTH CENTRAL	-3839.9	-160.2	-9423.8	-25.1	-86.2
2000	SOUTH	-10791.5	-3730.2	18541.9	-140.7	-191.2
2000	WEST	-1379.1	-840.6	232.6	-318.7	-3.8
2000	TOTAL U.S.	-16417.3	-5145.8	7325.0	-497.6	-306.2

COAL = BTUX10-9 OIL = BTUX10-9 GAS = BTUX10-9 HYDRO = KWHX10-6 NUCLEAR = KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-5a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

PULSE COMBUSTION FURNACE VS GAS WARM AIR FURNACE

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	2.4	0.3	-11801.5	0.8	1.5
1985	NORTH CENTRAL	29.4	0.2	-50505.5	1.2	4.3
1985	SOUTH	13.1	0.3	-39017.6	2.2	2.3
1985	WEST	4.5	0.2	-23270.0	9.2	0.4
1985	TOTAL U.S.	49.3	1.0	-124594.7	13.4	8.5
1990	NORTH EAST	4.8	0.6	-23603.1	1.6	2.9
1990	NORTH CENTRAL	58.8	0.3	-101011.1	2.4	8.6
1990	SOUTH	26.2	0.6	-78045.2	4.3	4.6
1990	WEST	8.9	0.5	-46540.1	18.5	0.8
1990	TOTAL U.S.	98.7	2.0	-249199.3	26.8	16.9
1995	NORTH EAST	7.2	0.8	-35404.6	2.4	4.4
1995	NORTH CENTRAL	88.1	0.5	-151516.6	3.6	12.9
1995	SOUTH	39.3	0.9	-117092.7	6.5	6.9
1995	WEST	13.4	0.7	-69810.1	27.7	1.2
1995	TOTAL U.S.	148.0	3.0	-373824.0	40.3	25.4
2000	NORTH EAST	9.5	1.1	-47206.1	3.2	5.8
2000	NORTH CENTRAL	117.5	0.7	-202022.1	4.8	17.2
2000	SOUTH	52.4	1.2	-156090.4	8.6	9.2
2000	WEST	17.8	1.0	-93080.1	37.0	1.6
2000	TOTAL U.S.	197.3	3.9	-498398.7	53.7	33.8

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	2.4	0.3	-11801.5	0.8	1.5
1985	NORTH CENTRAL	29.4	0.2	-50505.5	1.2	4.3
1985	SOUTH	13.1	0.3	-39017.6	2.2	2.3
1985	WEST	4.5	0.2	-23270.0	9.2	0.4
1985	TOTAL U.S.	49.3	1.0	-124594.7	13.4	8.5
1990	NORTH EAST	4.8	0.6	-23603.1	1.6	2.9
1990	NORTH CENTRAL	58.8	0.3	-101011.1	2.4	8.6
1990	SOUTH	26.2	0.6	-78045.2	4.3	4.6
1990	WEST	8.9	0.5	-46540.1	18.5	0.8
1990	TOTAL U.S.	98.7	2.0	-249199.3	26.8	16.9
1995	NORTH EAST	7.2	0.8	-35404.6	2.4	4.4
1995	NORTH CENTRAL	88.1	0.5	-151516.6	3.6	12.9
1995	SOUTH	39.3	0.9	-117092.7	6.5	6.9
1995	WEST	13.4	0.7	-69810.1	27.7	1.2
1995	TOTAL U.S.	148.0	3.0	-373824.0	40.3	25.4
2000	NORTH EAST	9.5	1.1	-47206.1	3.2	5.8
2000	NORTH CENTRAL	117.5	0.7	-202022.1	4.8	17.2
2000	SOUTH	52.4	1.2	-156090.4	8.6	9.2
2000	WEST	17.8	1.0	-93080.1	37.0	1.6
2000	TOTAL U.S.	197.3	3.9	-498398.7	53.7	33.8

COAL = LBX10-6 OIL = GALX10-6 GAS = CFIX10-6 HYDRO = KWHX10-6 NUCLEAR = KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-5b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

PULSE COMBUSTION FURNACE VS GAS WARM AIR FURNACE

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	24.4	41.0	-11966.7	0.8	1.5
1985	NORTH CENTRAL	300.7	23.6	-51212.6	1.2	4.3
1985	SOUTH	134.2	43.1	-39563.9	2.2	2.3
1985	WEST	45.6	35.0	-23595.8	9.2	0.4
1985	TOTAL U.S.	504.9	142.8	-126339.0	13.4	8.5
1990	NORTH EAST	48.8	82.1	-23933.5	1.6	2.9
1990	NORTH CENTRAL	601.4	47.3	-102425.2	2.4	8.6
1990	SOUTH	268.4	86.2	-79137.8	4.3	4.6
1990	WEST	91.1	70.1	-47191.6	18.5	0.8
1990	TOTAL U.S.	1009.7	285.6	-252688.1	26.8	16.9
1995	N TH EAST	73.3	123.1	-35900.2	2.4	4.4
1995	N TH CENTRAL	902.1	70.9	-153637.8	3.6	12.9
1995	SOUTH	402.6	129.3	-118732.0	6.5	6.9
1995	WEST	136.7	105.1	-70787.4	27.7	1.2
1995	TOTAL U.S.	1514.7	428.5	-379057.5	40.3	25.4
2000	NORTH EAST	97.7	164.1	-47867.0	3.2	5.8
2000	NORTH CENTRAL	1202.8	94.6	-204850.4	4.8	17.2
2000	SOUTH	536.7	172.4	-158275.7	8.6	9.2
2000	WEST	182.3	140.2	-94383.2	37.0	1.6
2000	TOTAL U.S.	2019.5	571.3	-505376.2	53.7	33.8

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	24.4	41.0	-11966.7	0.8	1.5
1985	NORTH CENTRAL	300.7	23.6	-51212.6	1.2	4.3
1985	SOUTH	134.2	43.1	-39563.9	2.2	2.3
1985	WEST	45.6	35.0	-23595.8	9.2	0.4
1985	TOTAL U.S.	504.9	142.8	-126339.0	13.4	8.5
1990	NORTH EAST	48.8	82.1	-23933.5	1.6	2.9
1990	NORTH CENTRAL	601.4	47.3	-102425.2	2.4	8.6
1990	SOUTH	268.4	86.2	-79137.8	4.3	4.6
1990	WEST	91.1	70.1	-47191.6	18.5	0.8
1990	TOTAL U.S.	1009.7	285.6	-252688.1	26.8	16.9
1995	NORTH EAST	73.3	123.1	-35900.2	2.4	4.4
1995	NORTH CENTRAL	902.1	70.9	-153637.8	3.6	12.9
1995	SOUTH	402.6	129.3	-118732.0	6.5	6.9
1995	WEST	136.7	105.1	-70787.4	27.7	1.2
1995	TOTAL U.S.	1514.7	428.5	-379057.5	40.3	25.4
2000	NORTH EAST	97.7	164.1	-47867.0	3.2	5.8
2000	NORTH CENTRAL	1202.8	94.6	-204850.4	4.8	17.2
2000	SOUTH	536.7	172.4	-158275.7	8.6	9.2
2000	WEST	182.3	140.2	-94383.2	37.0	1.6
2000	TOTAL U.S.	2019.5	571.3	-505376.2	53.7	33.8

COAL = BTUX10-9 OIL = BTUX10-9 GAS = BTUX10-9 HYDRO = KWHX10-6 NUCLEAR = KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-6a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

HYBRID ELECTRIC HEAT PUMP/GAS FURNACE VS ELECTRIC HEAT PUMP - STANDARD

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-82.1	-10.5	2518.9	-28.2	-53.3
1985	NORTH CENTRAL	-1094.8	-6.1	10629.5	-44.4	-158.3
1985	SOUTH	-373.1	-4.7	13895.2	-77.0	-43.5
1985	WEST	-124.0	-2.1	2964.8	-96.0	-2.8
1985	TOTAL U.S.	-1674.0	-23.2	30008.4	-245.7	-258.0
1990	NORTH EAST	-150.6	-19.0	4621.8	-51.8	-97.8
1990	NORTH CENTRAL	-1995.9	-11.1	19378.9	-80.9	-288.6
1990	SOUTH	-680.7	-8.5	25347.4	-140.5	-79.3
1990	WEST	-236.3	-4.0	5650.8	-183.0	-5.4
1990	TOTAL U.S.	-3063.5	-42.7	54999.0	-456.3	-471.2
1995	NORTH EAST	-205.6	-25.9	6308.8	-70.7	-133.5
1995	NORTH CENTRAL	-2742.1	-15.3	26624.0	-111.2	-396.5
1995	SOUTH	-935.1	-11.7	34821.7	-193.1	-109.0
1995	WEST	-324.7	-5.6	7765.6	-251.5	-7.5
1995	TOTAL U.S.	-4207.6	-58.5	75520.1	-626.4	-646.5
2000	NORTH EAST	-250.8	-31.6	7695.3	-86.2	-162.9
2000	NORTH CENTRAL	-3341.2	-18.7	32440.2	-135.5	-483.1
2000	SOUTH	-1139.9	-14.3	42447.5	-235.3	-132.9
2000	WEST	-395.8	-6.8	9465.7	-306.6	-9.1
2000	TOTAL U.S.	-5127.7	-71.3	92048.6	-763.6	-788.0

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-82.1	-10.5	2518.9	-28.2	-53.3
1985	NORTH CENTRAL	-1094.8	-6.1	10629.5	-44.4	-158.3
1985	SOUTH	-373.1	-4.7	13895.2	-77.0	-43.5
1985	WEST	-124.0	-2.1	2964.8	-96.0	-2.8
1985	TOTAL U.S.	-1674.0	-23.2	30008.4	-245.7	-258.0
1990	NORTH EAST	-150.6	-19.0	4621.8	-51.8	-97.8
1990	NORTH CENTRAL	-1995.9	-11.1	19378.9	-80.9	-288.6
1990	SOUTH	-680.7	-8.5	25347.4	-140.5	-79.3
1990	WEST	-236.3	-4.0	5650.8	-183.0	-5.4
1990	TOTAL U.S.	-3063.5	-42.7	54999.0	-456.3	-471.2
1995	NORTH EAST	-205.6	-25.9	6308.8	-70.7	-133.5
1995	NORTH CENTRAL	-2742.1	-15.3	26624.0	-111.2	-396.5
1995	SOUTH	-935.1	-11.7	34821.7	-193.1	-109.0
1995	WEST	-324.7	-5.6	7765.6	-251.5	-7.5
1995	TOTAL U.S.	-4207.6	-58.5	75520.1	-626.4	-646.5
2000	NORTH EAST	-250.8	-31.6	7695.3	-86.2	-162.9
2000	NORTH CENTRAL	-3341.2	-18.7	32440.2	-135.5	-483.1
2000	SOUTH	-1139.9	-14.3	42447.5	-235.3	-132.9
2000	WEST	-395.8	-6.8	9465.7	-306.6	-9.1
2000	TOTAL U.S.	-5127.7	-71.3	92048.6	-763.6	-788.0

COAL - LBX10-6 OIL - GALX10-6 GAS - CFTX10-6 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-6b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

HYBRID ELECTRIC HEAT PUMP/GAS FURNACE VS ELECTRIC HEAT PUMP - STANDARD

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-840.2	-1500.5	2554.2	-28.2	-53.3
1985	NORTH CENTRAL	-11205.2	-887.1	10778.4	-44.4	-158.3
1985	SOUTH	-3819.1	-677.6	14089.7	-77.0	-43.5
1985	WEST	-1268.9	-308.0	3006.3	-96.0	-2.8
1985	TOTAL U.S.	-17133.5	-3373.1	30428.5	-245.7	-258.0
1990	NORTH EAST	-1541.6	-2753.2	4686.5	-51.8	-97.8
1990	NORTH CENTRAL	-20428.4	-1617.3	19650.2	-80.9	-288.6
1990	SOUTH	-6966.8	-1236.1	25702.3	-140.5	-79.3
1990	WEST	-2418.5	-587.0	5729.9	-183.0	-5.4
1990	TOTAL U.S.	-31355.4	-6193.5	55768.9	-456.3	-471.2
1995	NORTH EAST	-2104.3	-3758.1	6397.1	-70.7	-133.5
1995	NORTH CENTRAL	-28065.9	-2221.9	26996.7	-111.2	-396.5
1995	SOUTH	-9570.9	-1698.1	35309.2	-193.1	-109.0
1995	WEST	-3323.7	-806.6	7874.4	-251.5	-7.5
1995	TOTAL U.S.	-43064.7	-8484.7	76577.5	-626.4	-646.5
2000	NORTH EAST	-2566.8	-4584.0	7803.1	-86.2	-162.9
2000	NORTH CENTRAL	-34197.0	-2707.3	32894.3	-135.5	-483.1
2000	SOUTH	-11666.8	-2070.0	43041.8	-235.3	-132.9
2000	WEST	-4051.3	-983.2	9598.2	-306.6	-9.1
2000	TOTAL U.S.	-52482.0	-10344.5	93337.2	-763.6	-788.0

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-840.2	-1500.5	2554.2	-28.2	-53.3
1985	NORTH CENTRAL	-11205.2	-887.1	10778.4	-44.4	-158.3
1985	SOUTH	-3819.1	-677.6	14089.7	-77.0	-43.5
1985	WEST	-1268.9	-308.0	3006.3	-96.0	-2.8
1985	TOTAL U.S.	-17133.5	-3373.1	30428.5	-245.7	-258.0
1990	NORTH EAST	-1541.6	-2753.2	4686.5	-51.8	-97.8
1990	NORTH CENTRAL	-20428.4	-1617.3	19650.2	-80.9	-288.6
1990	SOUTH	-6966.8	-1236.1	25702.3	-140.5	-79.3
1990	WEST	-2418.5	-587.0	5729.9	-183.0	-5.4
1990	TOTAL U.S.	-31355.4	-6193.5	55768.9	-456.3	-471.2
1995	NORTH EAST	-2104.3	-3758.1	6397.1	-70.7	-133.5
1995	NORTH CENTRAL	-28065.9	-2221.9	26996.7	-111.2	-396.5
1995	SOUTH	-9570.9	-1698.1	35309.2	-193.1	-109.0
1995	WEST	-3323.7	-806.6	7874.4	-251.5	-7.5
1995	TOTAL U.S.	-43064.7	-8484.7	76577.3	-626.4	-646.5
2000	NORTH EAST	-2566.8	-4584.0	7803.1	-86.2	-162.9
2000	NORTH CENTRAL	-34197.0	-2707.3	32894.3	-135.5	-483.1
2000	SOUTH	-11566.8	-2070.0	43041.8	-235.3	-132.9
2000	WEST	-4051.3	-983.2	9598.2	-306.6	-9.1
2000	TOTAL U.S.	-52482.0	-10344.5	93337.2	-763.6	-788.0

COAL - BTUX10-9 OIL - BTUX10-9 GAS - BTUX10-9 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-7a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

RANKINE-RANKINE GAS HEAT PUMP VS ROOFTOP GAS FURNACE/ELECTRIC AC

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-1.6	-0.1	-19.5	-0.5	-1.0
1985	NORTH CENTRAL	-6.6	-0.0	-38.8	-0.4	-1.3
1985	SOUTH	-15.2	-0.4	42.0	-2.1	-2.8
1985	WEST	-2.0	-0.1	-8.3	-4.5	-0.1
1985	TOTAL U.S.	-25.4	-0.6	-24.6	-7.5	-5.1
1990	NORTH EAST	-8.4	-0.7	-103.8	-2.8	-5.2
1990	NORTH CENTRAL	-34.7	-0.1	-205.1	-2.0	-6.9
1990	SOUTH	-73.1	-1.9	215.2	-10.6	-14.1
1990	WEST	-10.6	-0.5	-43.8	-23.9	-0.4
1990	TOTAL U.S.	-131.9	-3.2	-137.5	-39.4	-26.7
1995	NORTH EAST	-20.7	-1.8	-255.4	-6.9	-12.9
1995	NORTH CENTRAL	-85.6	-0.3	-505.9	-5.0	-17.1
1995	SOUTH	-193.2	-4.7	531.9	-26.3	-34.9
1995	WEST	-26.2	-1.1	-107.9	-59.0	-1.0
1995	TOTAL U.S.	-325.7	-7.9	-337.3	-97.1	-65.9
2000	NORTH EAST	-39.9	-3.4	-491.3	-13.3	-24.8
2000	NORTH CENTRAL	-165.0	-0.6	-974.9	-9.6	-32.9
2000	SOUTH	-371.9	-9.0	1024.3	-50.6	-67.2
2000	WEST	-50.5	-2.2	-208.0	-113.8	-1.9
2000	TOTAL U.S.	-627.3	-15.3	-650.0	-187.1	-126.9

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-3.2	-0.3	-39.7	-1.1	-2.0
1985	NORTH CENTRAL	-13.5	-0.1	-79.7	-0.8	-2.7
1985	SOUTH	-30.3	-0.7	83.4	-4.1	-5.5
1985	WEST	-4.1	-0.2	-17.0	-9.3	-0.2
1985	TOTAL U.S.	-51.1	-1.2	-52.9	-15.3	-10.3
1990	NORTH EAST	-21.8	-1.9	-268.3	-7.2	-13.5
1990	NORTH CENTRAL	-90.0	-0.3	-531.8	-5.2	-18.0
1990	SOUTH	-202.9	-4.9	558.7	-27.6	-36.7
1990	WEST	-27.5	-1.2	-113.5	-62.0	-1.0
1990	TOTAL U.S.	-342.2	-8.3	-354.9	-102.1	-69.2
1995	NORTH EAST	-44.6	-3.8	-549.6	-14.8	-27.8
1995	NORTH CENTRAL	-184.5	-0.7	-1090.5	-10.7	-36.8
1995	SOUTH	-415.7	-10.1	1144.8	-55.5	-75.1
1995	WEST	-56.5	-2.5	-232.7	-127.2	-2.1
1995	TOTAL U.S.	-701.3	-17.1	-726.0	-209.3	-141.9
2000	NORTH EAST	-97.9	-8.4	-1206.3	-32.6	-60.9
2000	NORTH CENTRAL	-405.1	-1.5	-2394.0	-23.5	-80.9
2000	SOUTH	-912.6	-22.1	2513.1	-124.1	-165.0
2000	WEST	-124.0	-5.4	-510.8	-279.3	-4.6
2000	TOTAL U.S.	-1539.5	-37.4	-1598.0	-459.4	-311.4

COAL - LBX10-6 OIL - GALX10-6 GAS - CFTX10-6 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-7b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

RANKINE-RANKINE GAS HEAT PUMP VS ROOFTOP GAS FURNACE/ELECTRIC AC

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-16.2	-19.6	-19.7	-0.5	-1.0
1985	NORTH CENTRAL	-67.3	-3.7	-39.4	-0.4	-1.3
1985	SOUTH	-156.0	-53.6	42.6	-2.1	-2.6
1985	WEST	-20.6	-12.7	-8.4	-4.5	-0.1
1985	TOTAL U.S.	-260.0	-89.6	-25.0	-7.5	-5.1
1990	NORTH EAST	-86.2	-104.5	-105.2	-2.8	-5.2
1990	NORTH CENTRAL	-355.2	-19.5	-298.0	-2.0	-6.9
1990	SOUTH	-799.8	-274.7	218.2	-10.6	-14.1
1990	WEST	-108.8	-67.2	-44.4	-23.9	-0.4
1990	TOTAL U.S.	-1350.0	-465.9	-139.4	-39.4	-26.7
1995	NORTH EAST	-212.1	-257.1	-258.9	-6.9	-12.9
1995	NORTH CENTRAL	-876.1	-48.2	-513.0	-5.0	-17.1
1995	SOUTH	-1976.9	-679.0	539.3	-26.3	-34.9
1995	WEST	-268.2	-165.6	-109.5	-59.0	-1.0
1995	TOTAL U.S.	-3333.2	-1149.9	-342.0	-97.1	-65.9
2000	NORTH EAST	-408.0	-494.6	-498.1	-13.3	-24.8
2000	NORTH CENTRAL	-1688.3	-92.9	-988.6	-9.6	-32.9
2000	SOUTH	-3806.9	-1307.6	1038.6	-50.6	-67.2
2000	WEST	-516.9	-319.2	-210.9	-113.8	-1.9
2000	TOTAL U.S.	-6420.1	-2214.2	-659.1	-187.1	-126.9

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-33.0	-40.0	-40.3	-1.1	-2.0
1985	NORTH CENTRAL	-138.0	-7.6	-80.8	-0.8	-2.7
1985	SOUTH	-310.1	-106.5	84.6	-4.1	-5.5
1985	WEST	-42.2	-26.1	-17.2	-9.3	-0.2
1985	TOTAL U.S.	-523.3	-180.2	-53.7	-15.3	-10.3
1990	NORTH EAST	-222.8	-270.1	-272.1	-7.2	-13.6
1990	NORTH CENTRAL	-920.9	-50.6	-539.2	-5.2	-18.0
1990	SOUTH	-2076.6	-713.3	566.6	-27.6	-36.7
1990	WEST	-282.0	-174.1	-115.1	-62.0	-1.0
1990	TOTAL U.S.	-3502.3	-1208.2	-359.8	-102.1	-69.2
1995	NORTH EAST	-456.5	-553.3	-557.3	-14.8	-27.8
1995	NORTH CENTRAL	-1888.3	-103.9	-1105.7	-10.7	-36.8
1995	SOUTH	-4254.9	-1461.5	1160.8	-56.5	-75.1
1995	WEST	-578.2	-357.0	-235.9	-127.2	-2.1
1995	TOTAL U.S.	-7177.8	-2475.6	-738.2	-209.3	-141.9
2000	NORTH EAST	-1001.8	-1214.4	-1223.2	-32.6	-60.9
2000	NORTH CENTRAL	-4145.7	-228.0	-2427.5	-23.5	-80.9
2000	SOUTH	-9340.4	-3208.3	2548.3	-124.1	-165.0
2000	WEST	-1269.3	-783.7	-518.0	-279.3	-4.6
2000	TOTAL U.S.	-15757.2	-5434.4	-1620.4	-459.4	-311.4

COAL - BTUX10-9 OIL - BTUX10-9 GAS - BTUX10-9 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-8a

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

RANKINE-RANKINE GAS HEAT PUMP VS ELECTRIC HEAT PUMP - STANDARD

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-3.6	-0.4	66.9	-1.2	-2.3
1985	NORTH CENTRAL	-16.6	-0.1	106.9	-0.8	-2.9
1985	SOUTH	-21.1	-0.5	177.5	-3.0	-3.8
1985	WEST	-3.6	-0.2	69.1	-8.6	-0.2
1985	TOTAL U.S.	-45.0	-1.1	420.4	-13.6	-9.2
1990	NORTH EAST	-19.4	-2.0	356.6	-6.5	-12.3
1990	NORTH CENTRAL	-87.6	-0.4	564.8	-4.3	-15.3
1990	SOUTH	-108.3	-2.6	910.1	-15.2	-19.4
1990	WEST	-19.2	-0.9	364.9	-45.2	-1.0
1990	TOTAL U.S.	-234.5	-5.9	2196.5	-71.2	-48.1
1995	NORTH EAST	-47.8	-4.8	877.6	-16.1	-30.3
1995	NORTH CENTRAL	-215.9	-1.0	1392.7	-10.5	-37.8
1995	SOUTH	-267.6	-6.4	2249.6	-37.6	-47.9
1995	WEST	-47.4	-2.3	900.0	-111.5	-2.6
1995	TOTAL U.S.	-578.8	-14.5	5419.8	-175.7	-118.6
2000	NORTH EAST	-92.0	-9.3	1688.3	-31.0	-58.4
2000	NORTH CENTRAL	-416.1	-1.9	2684.0	-20.3	-72.9
2000	SOUTH	-515.4	-12.2	4331.9	-72.4	-92.2
2000	WEST	-91.4	-4.5	1734.4	-214.8	-5.0
2000	TOTAL U.S.	-1114.9	-28.0	10438.6	-338.4	-228.4

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-7.4	-0.7	136.5	-2.5	-4.7
1985	NORTH CENTRAL	-34.0	-0.2	219.3	-1.7	-6.0
1985	SOUTH	-42.0	-1.0	352.9	-5.9	-7.5
1985	WEST	-7.5	-0.4	141.7	-17.6	-0.4
1985	TOTAL U.S.	-90.9	-2.3	850.5	-27.6	-18.6
1990	NORTH EAST	-50.2	-5.1	922.2	-16.9	-31.9
1990	NORTH CENTRAL	-227.0	-1.1	1464.0	-11.3	-39.8
1990	SOUTH	-281.2	-6.7	2363.1	-39.5	-50.3
1990	WEST	-49.9	-2.5	946.0	-117.2	-2.7
1990	TOTAL U.S.	-608.2	-15.3	5695.3	-184.6	-124.6
1995	NORTH EAST	-102.9	-10.4	1888.9	-34.7	-65.3
1995	NORTH CENTRAL	-465.4	-2.2	3002.0	-22.7	-81.5
1995	SOUTH	-576.1	-13.7	4841.7	-80.9	-103.0
1995	WEST	-102.3	-5.0	1939.9	-240.2	-5.6
1995	TOTAL U.S.	-1246.6	-31.3	11672.5	-378.5	-255.4
2000	NORTH EAST	-225.8	-22.8	4145.6	-74.1	-143.3
2000	NORTH CENTRAL	-1021.8	-4.7	6590.7	-49.8	-179.0
2000	SOUTH	-1264.5	-30.1	10628.6	-177.6	-226.1
2000	WEST	-224.5	-11.1	4258.9	-527.4	-12.2
2000	TOTAL U.S.	-2736.7	-68.5	25623.9	-830.9	-560.6

COAL - LB/X10-6 OIL - GAL/X10-6 GAS - CFT/X10-6 HYDRO - KWH/X10-6 NUCLEAR - KWH/X10-6

Note: Negative numbers represent resource savings.

EXHIBIT F-8b

DIFFERENCE IN PRIMARY ENERGY CONSUMPTION BASED ON PROJECTION OF REGIONAL MARKET PENETRATIONS

RANKINE-RANKINE GAS HEAT PUMP VS ELECTRIC HEAT PUMP - STANDARD

LOW ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-37.3	-53.3	67.8	-1.2	-2.3
1985	NORTH CENTRAL	-169.7	-11.2	108.4	-0.8	-2.9
1985	SOUTH	-216.1	-72.8	180.0	-3.0	-3.8
1985	WEST	-37.3	-26.1	70.1	-8.6	-0.2
1985	TOTAL U.S.	-460.3	-163.4	426.3	-13.6	-9.2
1990	NORTH EAST	-198.8	-284.1	361.6	-6.5	-12.3
1990	NORTH CENTRAL	-896.2	-58.9	572.7	-4.3	-15.3
1990	SOUTH	-1108.3	-373.5	922.9	-15.2	-19.4
1990	WEST	-196.9	-137.7	370.1	-45.2	-1.0
1990	TOTAL U.S.	-2400.2	-854.4	2227.2	-71.2	-48.1
1995	NORTH EAST	-489.2	-699.2	889.9	-16.1	-30.3
1995	NORTH CENTRAL	-2209.9	-145.4	1412.2	-10.5	-37.8
1995	SOUTH	-2739.4	-923.3	2281.0	-37.6	-47.9
1995	WEST	-485.5	-339.7	912.6	-111.5	-2.6
1995	TOTAL U.S.	-5924.1	-2107.5	5495.7	-175.7	-118.6
2000	NORTH EAST	-941.2	-1345.2	1712.0	-31.0	-58.4
2000	NORTH CENTRAL	-4258.9	-280.1	2721.6	-20.3	-72.9
2000	SOUTH	-5275.2	-1777.9	4392.6	-72.4	-92.2
2000	WEST	-935.7	-654.6	1758.7	-214.8	-5.0
2000	TOTAL U.S.	-11411.0	-4057.8	10584.8	-338.4	-228.4

HIGH ESTIMATE

YEAR	REGION	COAL	OIL	GAS	HYDRO	NUCLEAR
1985	NORTH EAST	-76.1	-108.8	138.4	-2.5	-4.7
1985	NORTH CENTRAL	-348.0	-22.9	222.4	-1.7	-6.0
1985	SOUTH	-429.8	-144.8	357.9	-5.9	-7.5
1985	WEST	-76.5	-53.5	143.7	-17.6	-0.4
1985	TOTAL U.S.	-930.3	-330.0	862.4	-27.6	-18.6
1990	NORTH EAST	-514.1	-734.8	935.1	-16.9	-31.9
1990	NORTH CENTRAL	-2323.0	-152.8	1484.5	-11.1	-39.8
1990	SOUTH	-2877.6	-969.8	2396.1	-39.5	-50.3
1990	WEST	-510.4	-357.0	959.3	-117.2	-2.7
1990	TOTAL U.S.	-6225.1	-2214.4	5775.0	-184.6	-124.6
1995	NORTH EAST	-1053.0	-1505.0	1915.4	-34.7	-65.3
1995	NORTH CENTRAL	-4763.5	-313.3	3044.0	-22.7	-81.5
1995	SOUTH	-5896.0	-1987.1	4909.5	-80.9	-103.0
1995	WEST	-1046.6	-732.1	1967.0	-240.2	-5.6
1995	TOTAL U.S.	-12759.1	-4537.6	11835.9	-378.5	-255.4
2000	NORTH EAST	-2311.0	-3303.1	4203.7	-76.1	-143.3
2000	NORTH CENTRAL	-10459.1	-687.9	6683.0	-49.8	-179.0
2000	SOUTH	-12943.0	-4362.2	10777.4	-177.6	-226.1
2000	WEST	-2297.7	-1607.0	4318.5	-527.0	-12.2
2000	TOTAL U.S.	-28009.7	-9960.5	25982.6	-830.9	-560.6

COAL - BTUX10-9 OIL - BTUX10-9 GAS - BTUX10-9 HYDRO - KWHX10-6 NUCLEAR - KWHX10-6

Note: Negative numbers represent resource savings.

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