CONF-841231

ornl

Proceedings of the DOE/ORNL Heat Pump Conference: RESEARCH AND DEVELOPMENT ON HEAT PUMPS FOR SPACE CONDITIONING APPLICATIONS

WASHINGTON MARRIOTT HOTEL
WASHINGTON, D.C.
DECEMBER 11-13, 1984

Sponsored by

U.S. DEPARTMENT OF ENERGY

OFFICE OF BUILDING ENERGY RESEARCH AND DEVELOPMENT

Conference Committee

Chairpersons

Ronald J. Fiskum, **DOE**
Frederick A. Creswick, **ORNL**
Kay H. Zimmerman, **ORNL**

Staff

Secretary — *Pamela J. Lewis*
Coordinator — *Norma F. Callaham*
Technical Editors — *Vivian A. Jacobs*
Raleigh H. Powell, Jr.

Proceedings Committee

Chairperson

Kay H. Zimmerman

In-house Manuscripts Staff

Teresa J. Simmons
Pamela J. Lewis
Illustrator — *LeRoy Gilliam*
Editors — *Carolyn Srite*
Nancy L. Prince
Technical Editors — *Vivian A. Jacobs*
Raleigh H. Powell, Jr.

GRAPHICS and DESIGN – Alice D. Richardson

CONF-841231

Proceedings of the
DOE/ORNL Heat Pump Conference:
RESEARCH AND DEVELOPMENT ON HEAT PUMPS FOR SPACE CONDITIONING APPLICATIONS

WASHINGTON MARRIOTT HOTEL
WASHINGTON, D.C.
DECEMBER 11-13, 1984

PUBLISHED: AUGUST 1985

Prepared by the
Oak Ridge National Laboratory
Oak Ridge, Tennessee 37831
operated by
Martin Marietta Energy Systems, Inc.
for the
U.S. DEPARTMENT OF ENERGY

CONTENTS

Page

Foreword v

Introduction

History and Objective of the Energy Conversion Equipment Program
Ronald J. Fiskum 5

Session 1: Electric Heat Pump Technology—I

EPRI Heat Pump R&D Overview
Arvo Lannus 15

ORNL Electric Systems Program Overview
Phillip D. Fairchild 19

Dual-Stroke Heat Pump Field Performance
Stephen E. Veyo 27

ORNL Air-Source Heat Pump Field Experiments
Van D. Baxter 39

Session 2: Electric Heat Pump Technology—II

Laboratory Experiments of Heat Pump Dynamic Losses
William A. Miller 49

A Comparative Analysis of Single- and Continuously Variable-Capacity Heat Pump Concepts
C. Keith Rice and *Steven K. Fischer* 57

Scoping Evaluation of Potential Benefits of Zoning with Residential Space-Conditioning Systems
Edmund A. Nephew and *John C. Moyers* 69

Condensing Coefficients When Using Refrigerant Mixtures
Wilbert F. Stoecker 77

The Performance of a Residential Heat Pump Operating with a Non-Azeotropic Binary Refrigerant Mixture—An Interim Report
David A. Didion and *William J. Mulroy* 91

Stirling Cycle Heat Pump Studies
Norberto Domingo 101

Session 3A: Ground-Coupled Heat Pump Technology

Optimized Ground-Coupled Heat Pump Mechanical Package
Mark A. Catan 113

Ground-Coupled Heat Pump Research at the University of Tennessee
William S. Johnson, Bernie A. McGraw, Robert N. Baugh, and *William A. Griffith* 123

ORNL Ground Coil Analytical and Experimental Studies
Vince C. Mei 133

Session 3B: International Activities

U.S. Department of Energy Cooperation in Advanced Heat Pump Research Through the International Energy Agency
John D. Ryan ... 143

The International Energy Agency Heat Pump Center
Frederick A. Creswick ... 151

Session 4: Heat-Actuated Heat Pumps—I

Stirling-Engine-Driven Heat Pump Program Overview
Fang C. Chen ... 157

Breadboard Development of a Hydraulically Coupled Free-Piston Stirling Engine Heat Pump Compressor
Thomas J. Marusak ... 163

Initial Results of Sensitivity Tests Performed on the RE-1000 Free-Piston Stirling Engine
Jeffrey G. Schreiber ... 171

Simplified Analysis of Stirling Engines and Heat Pumps
Norbert C.J. Chen, Frederick P. Griffin, and *Colin D. West* ... 183

Assessment of Free-Piston Stirling Engines as Heat Pump Drives
W. Peter Teagan ... 195

Session 5: Heat-Actuated Heat Pumps—II

Engine-Driven and Absorption Heat Pump Programs
George T. Privon ... 211

Status of the Braun Free-Piston Engine-Driven Heat Pump
A. Tom Braun ... 217

Laboratory Experiments on Absorption Heat Pumps
Reinhard Radermacher ... 227

Absorption Fluids Data Survey
Robert A. Macriss and *Thomas S. Zawacki* ... 235

The GRI Gas Heat Pump Program—An Update
Arthur R. Maret ... 247

Session 6: Heat-Actuated Heat Pumps—III

Advanced Absorption Heat Pump Cycles
Robert C. Reimann ... 259

Analyses of Advanced Residential Absorption Heat Pump Cycles
Benjamin A. Phillips ... 265

Evaluation of Advanced Design Concepts for Absorption Heat Pumps
Floyd C. Hayes and *Robert J. Modahl* ... 291

Author Index ... 305

Author Affiliations ... 307

Conference Attendees ... 309

FOREWORD

This conference was planned to provide information on current activities in the U.S. Department of Energy (DOE) and Oak Ridge National Laboratory (ORNL) Building Equipment Research (BER) Program. It was primarily for the benefit of HVAC equipment manufacturers and other interested parties, including utilities, independent research and development organizations, universities, other government groups, and research funding and management organizations.

Because this conference was intended to be a significant part of our technology transfer efforts, the presentations emphasized objectives, current status, accomplishments, and expected benefits. In addition, the conference was organized to facilitate information exchange among attendees, speakers, and the DOE and ORNL Program staff.

The technical presentations were grouped into two principal subject areas: electric systems and thermally activated systems. Electric-system topics included field performance studies, laboratory experiments on cycling performance, analytical estimates of the benefits of variable capacity and zone control, nonazeotropic refrigerant mixtures, ground-coupled systems, and an analysis of Stirling-cycle heat pumps. In the area of thermally activated heat pumps, presentations centered on the development of absorption systems, Stirling-engine-driven systems, and a linear, free-piston IC-engine compressor.

Because of the strong ties between the BER Program and the Electric Power Research Institute (EPRI) and Gas Research Institute (GRI) heat pump research and development programs, overview presentations on the EPRI and GRI programs were included in the conference. There was also a short session on international activities (conducted under this program) that consisted primarily of participation in an International Energy Agency group on Advanced Heat Pumps.

The official attendance of 108 persons represented 54 organizations. The ORNL staff judged it to be a "quality" audience, with many of the leading equipment manufacturers represented by personnel who are involved in advanced product development. The feedback from participants has been valuable to ORNL in planning both program content and interaction with the private sector.

On behalf of DOE and ORNL, the undersigned wish to take this opportunity to again express our appreciation to all participants in this conference.

Conference Committee

F. A. Creswick, ORNL
R. J. Fiskum, DOE
K. H. Zimmerman, ORNL

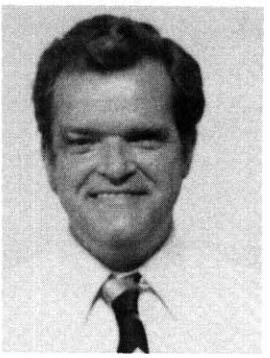
INTRODUCTION

History and Objective of the Energy Conversion Equipment Program

Ronald J. Fiskum

U.S. Department of Energy

Washington, D.C. 20585



Ronald J. Fiskum has been with the U.S. Department of Energy in Washington, D.C., since 1980. As manager of the Heat Pump Research and Development Program, he is responsible for developing research activities in the residential and commercial building equipment area. A graduate of Hofstra University, he received his B.S. degree in engineering science in 1968. Previous work has been in satellite thermal design with the U.S. Naval Research Laboratory and Grumann Aerospace Corporation.

HISTORY AND OBJECTIVE OF THE ENERGY CONVERSION EQUIPMENT PROGRAM

Ronald J. Fiskum
Program Manager
Energy Conversion Equipment Branch
U.S. Department of Energy
Washington, DC 20585

INTRODUCTION

The Department of Energy is responsible for conducting research and development, and promoting projects that will accelerate the private sector utilization and acceptance of energy saving technologies. Within this charter, energy supply achieved through conservation is viewed as a resource that is subject to the same market mechanisms as are other energy resources. Energy conservation gained recognition as an energy resource when the cost of conserving existing supplies of energy became economically competitive with the cost of finding and developing new energy sources or developing new energy supply technologies. The Departments stated role in energy conservation is to foster the technical and economic environment that ensure continued development of conservation energy resources potential through research and development programs and technology transfer activities.

The Office of Building Energy Research and Development (BERD) conducts research and development, promotes technology transfer and develops and implements statutory regulation designed to improve the efficient use of energy is residential and commercial buildings and community systems.

The Office operates its building energy research programs through three functional divisions: Building Systems, Building Services, Building Equipment; and a staff office: Analysis and Technology Transfer. Programs are conducted to the maximum extent possible in conjunction with trade associations, and private industry with the objective to increase our scientific and technical understanding of new energy conserving technologies to reduce the high risk of introducing them in the free marketplace, and to increase the potential of the private sector to assume responsibility for development and marketing of improved products.

HISTORY

Since 1977, BERD has brought a focus to its efforts by defining specific objectives. These objectives were to achieve a 20% reduction in energy consumption in existing buildings from 1978 to 1985 and a 30 percent reduction in energy use in new buildings. The equipment research program objectives was to introduce household appliances that used 30 percent less energy, space heating equipment that used 50 percent less energy, and lighting systems that used 50 percent less energy than the average efficiencies of 1978 products. The strategy for reaching these objectives was to determine the energy use in the building sector by different function (e.g. heating, cooling, lighting etc.), inventory the research that could increase the efficiency in each of these functional areas and in the system as a whole, identify the most promising technologies, determine the need for a federal role and design a RD&D effort that would lead to the commercial introduction of new, more energy efficient practices and products. The objective was to accelerate the introduction and market penetration of such practices and products. As the commercial value of these measures become evident, industry were expected to assume a larger share of the cost.

These active, varied BERD programs were mandated by a series of energy Legislation starting in 1973.. The Non-Nuclear Energy Research and Development Act, (P.L. 93-577), directed the Energy Research and Development Agency (ERDA) to initiate a national energy conservation RD&D program. The Energy Production and Conservation Act (P.L. 94-163) directed the newly created Federal Energy Administration (FEA) to develop test procedures and energy efficient targets for major home appliances and establish an energy conservation program for Federal buildings and operations. The Energy Conservation and Production Act, (P.L. 94-385), directed HUD to develop and promulgate building performance standards for new residential and commercial building and mobile homes. The Department of Energy Organization Act (P.L. 95-91), gave these responsibilities to the Department of Energy. The National Energy Conservation Policy Act (P.L. 95-619(established the Residential Conservation Service (RCS) program, called for life-cycle cost procedures for Federal conservation investments and required the enforcement of minimum efficiency standards for all major home appliances. The Power Plant and Industrial Fuel Act (P.L. 95-620),

provided grants for areas impacted by coal and uranium mining. The Energy Security Act (P.L. 95-294), expanded RCS to include multi-family apartments and small commercial buildings authorized auditor training grants to States (funding rescinded) and authorized Residential End-Use Energy Planning (REEP) as one element of BERD's building retrofit demonstration program (partially funded).

The current view of the administration is that although BERD can point to cost-effective achievements, the demonstration and commercialization role is best left to the private sector, and that the Federal government continue funding in areas of generic research, that lead to technology advancement.

Accordingly, the BERD research objective, has been redirected toward long-term, generic, high-risk technologies that offer large potential savings, but are too risky for the private investors. None of the earlier objectives remain, however, BERD continues to be responsible for many statutory responsibilities which DOE intends to meet. (FEMP and RCS to be moved out of BERD).

ORGANIZATION AND MANAGEMENT PLAN

1.2 Organization and Management Plan

The Energy Conversion Equipment Program is part of the Office of Building and Community Systems, one of four major program offices in the Office of Conservation. The organizational structure, shown in Figure 1, reflects the program's mandated mission. ECEP in response to (PL 93-438) is funded through the technology and consumer products budget line item under the Energy Conversion area of the Federal Budget.

In recognition of the technical resources residing in the national laboratories, ECEP has established a decentralized management plan (Figure 2). Programmatic direction and policy making is the responsibility of DOE headquarters in Washington, D.C. However, four of the national laboratories assume all other normal program management functions. The national laboratories issue Requests for Proposals (RFPs), evaluate proposals, execute contracts, monitor contractor performance, and interact with the private sector in making programmatic recommendations to ECEP personnel.

The staff of the Energy Conversion Equipment Program have the responsibility for planning and administering the R&D programs in building energy conversion technology. The program's direction is based on guidance and advance from researchers at the national laboratories and extensive interactions with industry. Input from the private sector is actively pursued to ensure that private sector research is not duplicated and that the research programs address areas of greatest technical need.

1.3 Program Scope

The ECEP addresses technologies related to energy-consuming equipment used in buildings. Thus, the program's scope extends from the ubiquitous 100-watt light bulb in a residence to a 1000-ton chiller used to cool a large office complex. Partly due to its position in the DOE organization, the program, does not address regulation of appliance efficiency or the technology of the building structure, both of which are the purview of other Programs within the Office of Building and Community Systems. Likewise, the program does not address renewable technology or efficiency improvements in industrial processes (although the technologies under development by ECEP may be of significant value to these areas). These areas are the responsibility of other programs within the Office of Conservation and Renewable Energy.

2.0 Program Overview

The basic philosophy of the ECEP has always been to encourage private industry involvement in research efforts to develop and market energy-efficient building equipment. This is a particularly important role because the nature of the industry in which a large number of firms each having a small market share, makes it difficult for private industry to develop new technologies on its own due to the lack of resources or facilities. In the past, ECEP's budget (Table 1) has allowed for both basic and applied research of which 50% has been allocated to heat pump and refrigeration research. Some of this applied research has already resulted in the development and marketing of several pieces of energy conversion equipment with vastly improved energy efficiencies. As a result the change in administration philosophy, specific product development is now generally left to industry.

HEAT PUMP RESEARCH AND DEVELOPMENT PROGRAM

The Heat Pump Research and Development Program as part of the ECEP is divided in two segments. One is Thermally-Activated Heat Pump Research, the other Refrigeration System and Appliance Research.

In the Termally Activated Heat Pump Program, research efforts are focused on the technical opportunities presented by using gas and oil in a heat pump cycle to drive the prime mover to produce heat energy greater than the energy available in the fuel utilized. It addresses a range of technologies applicable to space heating, space cooling and water heating with gaseous and liquid fuels.

In the Refrigeration and Appliances Technology Program, research efforts are

focused to address a broad range of generic technical opportunities for energy savings in electrical-driven heat pumps for space and water heating, space cooling, refrigeration-freezers, commercial refrigeration and appliances. Results of generic research in this area could also be applied to the refrigeration cycle of thermally-activated heat pumps. This program is also responsible for developing advanced insulation that will be used in broad range of Energy Conversion Equipment.

REFRIGERATION AND APPLIANCE TECHNOLOGY PROGRAM

The objective of ECEP's Refrigeration Systems and Appliance Technology Program is to bring heat pumps, refrigeration systems and appliances to the practical limits of efficiency usefulness and effectiveness. The approach is to carry out the needed evaluation of commercial success.

Research is directed toward equipment in residential and commercial buildings. However, the technologies developed are applicable to other uses such as industrial processes, transport refrigeration etc. Specific areas of R&D are:

1) Ground Coupled Heat Transfer Research
2) Heat Pump Dynamic Loss Research
3) Novel Cycle Research and Development
4) Non-Azeotropic Refrigerants Mixture Research
5) Advanced Insulation Materials Research
6) Advanced Commercial HVAC Research

THERMALLY ACTIVATED HEAT PUMP PROGRAM

The objective of this program is to identify advanced concepts in thermally-activated prime movers for use in heat pumps and refrigeration systems for residential and commercial application. To determine analytically and experimentally the energy savings potential and economic viability of each candidate concept, and to bring the most promising concepts to a proof-of-concept development. Five principal areas of research have developed under this program:

1) Advanced Cycle Absorption Heat Pump
2) Absorption Fluids and Supporting Technologies
3) Free Piston Stirling Engine Driven Heat Pump
4) Internal Combustion Engine Driven Heat Pump
5) Engine Driven Supporting Research

It is anticipated that as a result of the research effort that more efficient and economically viable fossil-fired systems for heating and cooling buildings will become available. These will be attractive alternatives to electric systems in areas of high electricity prices and abundant natural gas supplies, and could replace conventional oil and gas furnaces and boilers or the fossil-fuel system of choice. This technology is well suited for northern climate application and where water heating demand is high in commercial establishments.

Technology Transfer

In accordance with the policy of the Office of Conservation and Renewable Energy, BERD is pacing increased emphasis on technology transfer. Technology transfer is viewed as a reciprocal process which serves as a vital link between the public and private sectors through the priorities of private industry are considered early in the Federal planning process and the results of DOE R&D programs are communicated to the appropriate users.

Technology transfer takes place throughout the BERD program. Research planning frequently involves the private sector--particularly where R&D is cost-shared. This helps to assure that the research will meet a real-world need and that a potential market for the technology exists, should the R&D prove successful.

Research is reported on to the user community for further R&D or application via conferences, workshops and technical reports. Tools are developed to assist the buildings and equipment industries in the use or understanding of new technologies, i.e. design and diagnostic tools, prediction and monitoring technologies. Data on field performance to establish potential impacts of new techniques and validate their potential is obtained and shared via workshops, reports, and clearing houses or "data bases."

Past Success

Since the establishment of the ECEP research program, several research efforts have culminated in energy efficient heat pumps and appliance concepts which have been incorporated into systems in the market plaee today.

The most prominent successful accomplishments are:

1 Steady State Computer Model for Residential Heat Pumps
2 Heat Pump Water Heater
3 Energy Efficient Compressor for Refrigerator-Freezers
4 Energy Efficient High Back Pressure Compressor for Room-Air Conditioners and Cold Drink Vending Machines
5 One $18Ft^3$ Frost-free Refrigerator Freezer
6 Bi-Radiant Oven
7 High Efficiency Gas-Fired Residential Water Heater
8 High Efficiency Gas-Fired Residential Water Heater
9 Dual Stoke High Efficiency Electric Driven Heat Pump
10 Micro Processor Controlled Multiple Parallel Compressor System for Supermarket Refrigeration

11 Development of a LIBR/H^2O Residential Gas-Fired Absorption Heat Pump

12 Development of Ammonia Water Absorption Heat Pump

CONCLUSION

The ECEP through its mandate to develop energy efficient building equipment and appliances will continue to search for novel concepts in building equipment and to develop those concepts with assistance of the private research community, universities, national laboratories and research institutions.

The ECEP is committed to sharing its research function with the private sector as well as encourage to participation of the private sector in its research endeavor.

ORGANIZATION AND MANAGEMENT PLAN

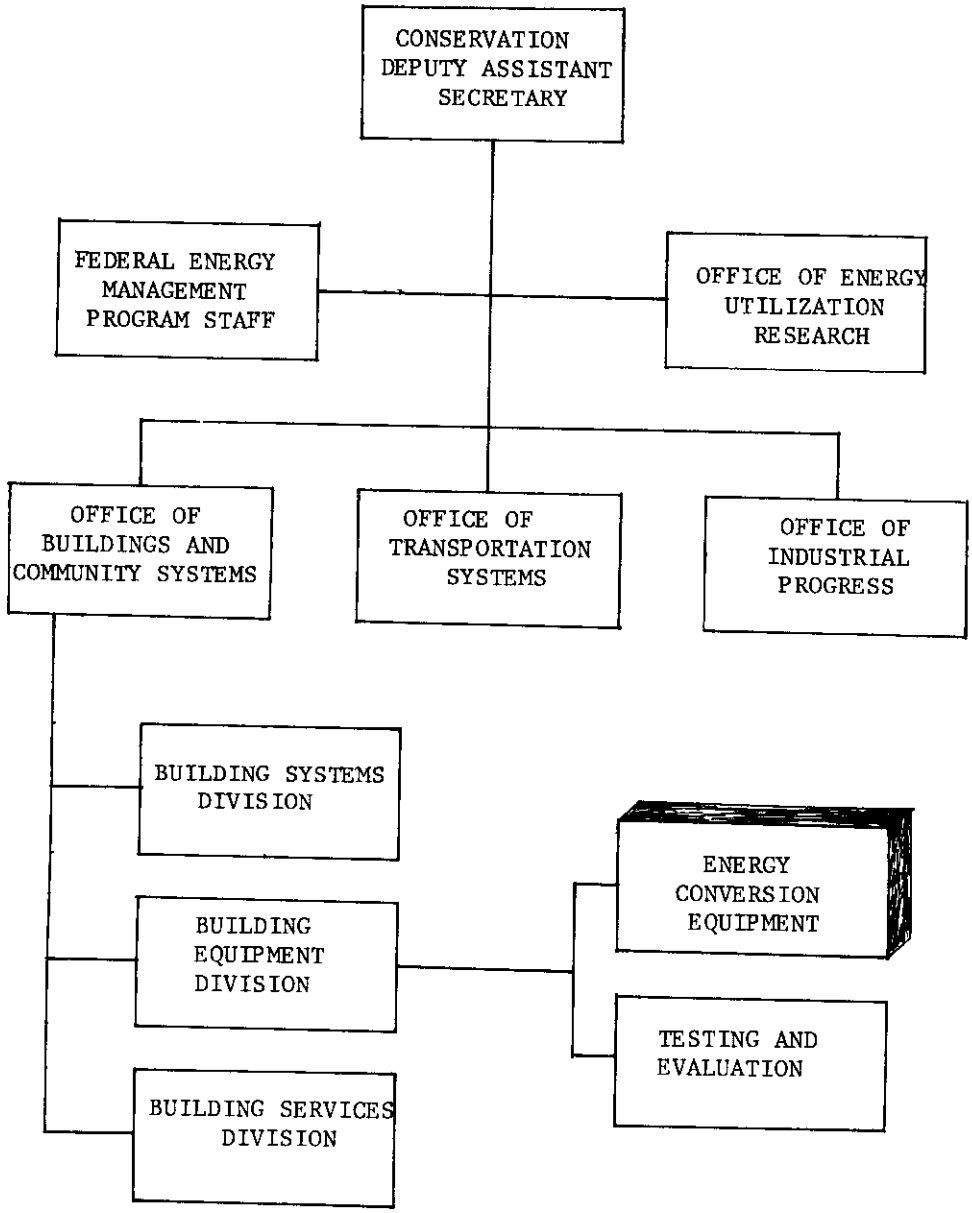


FIGURE 1

STRUCTURE OF ENERGY CONVERSION
EQUIPMENT PROGRAM

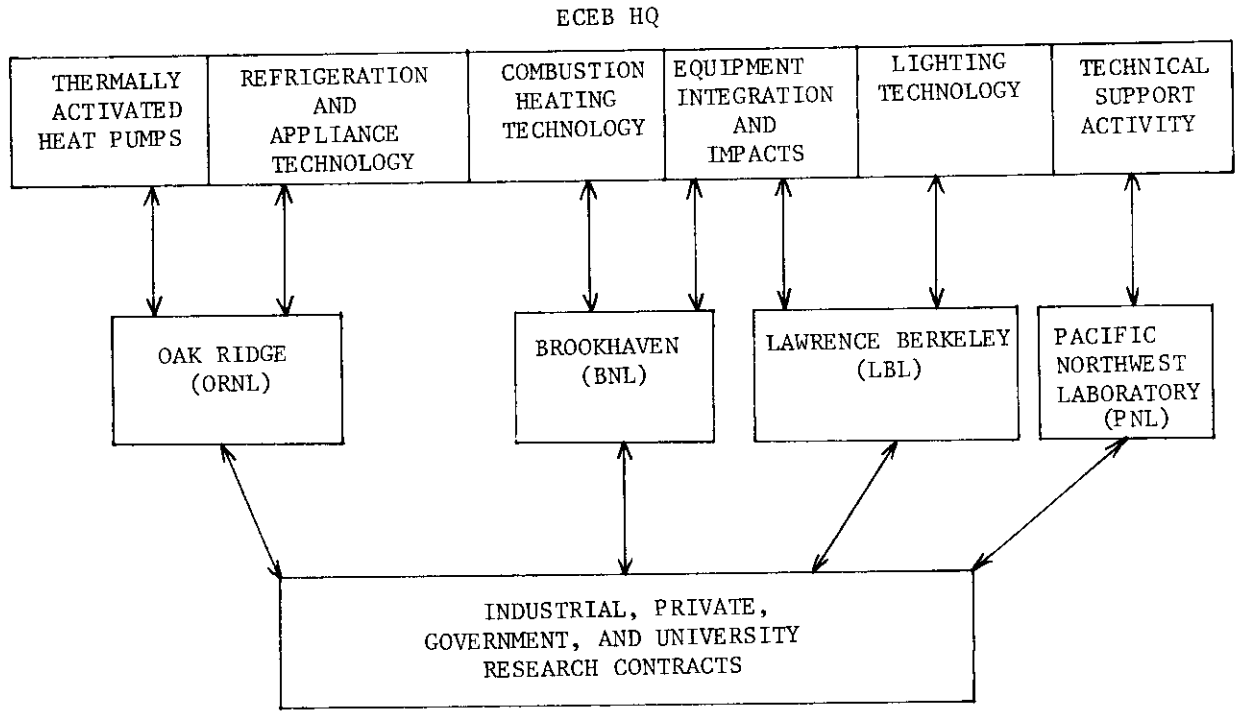
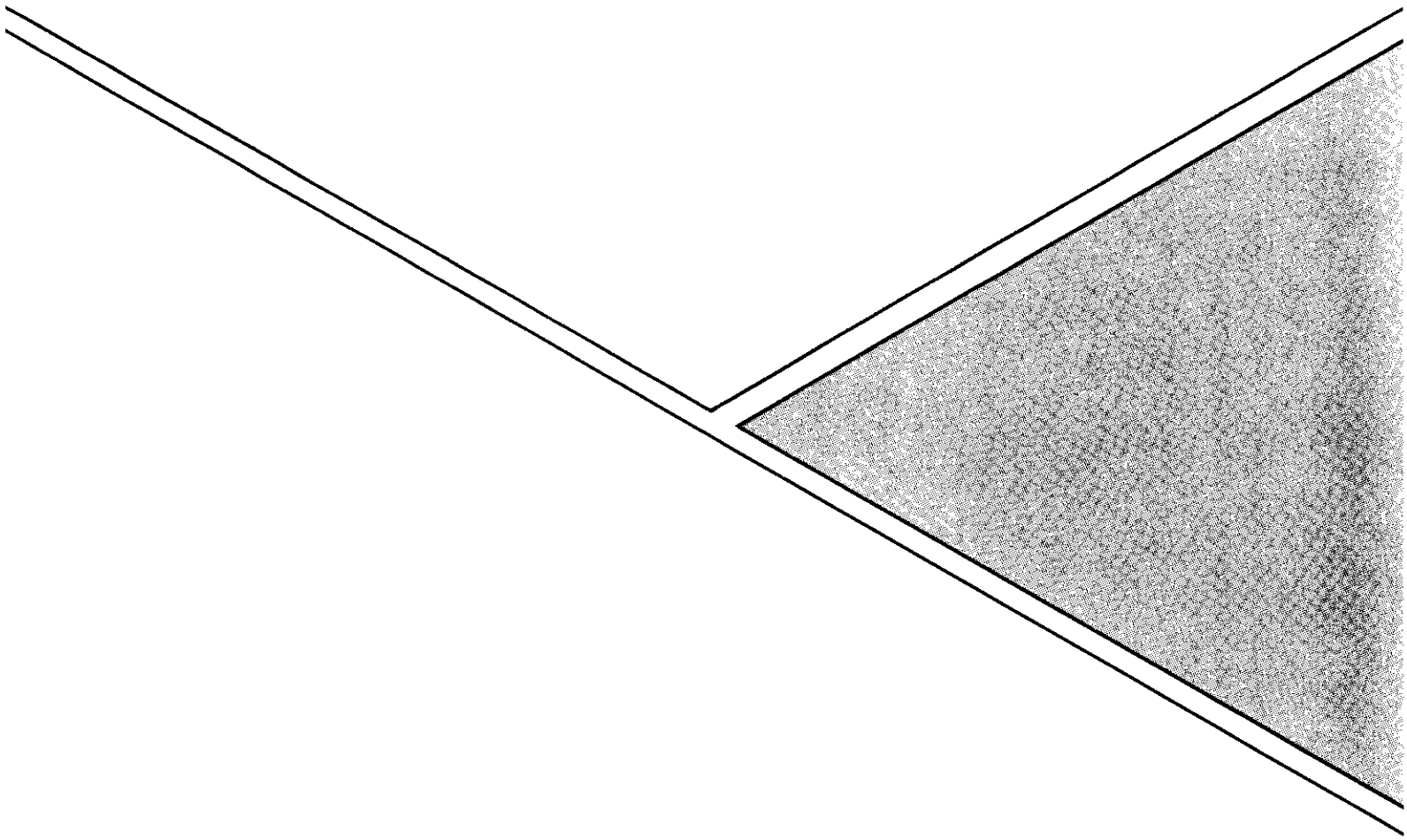


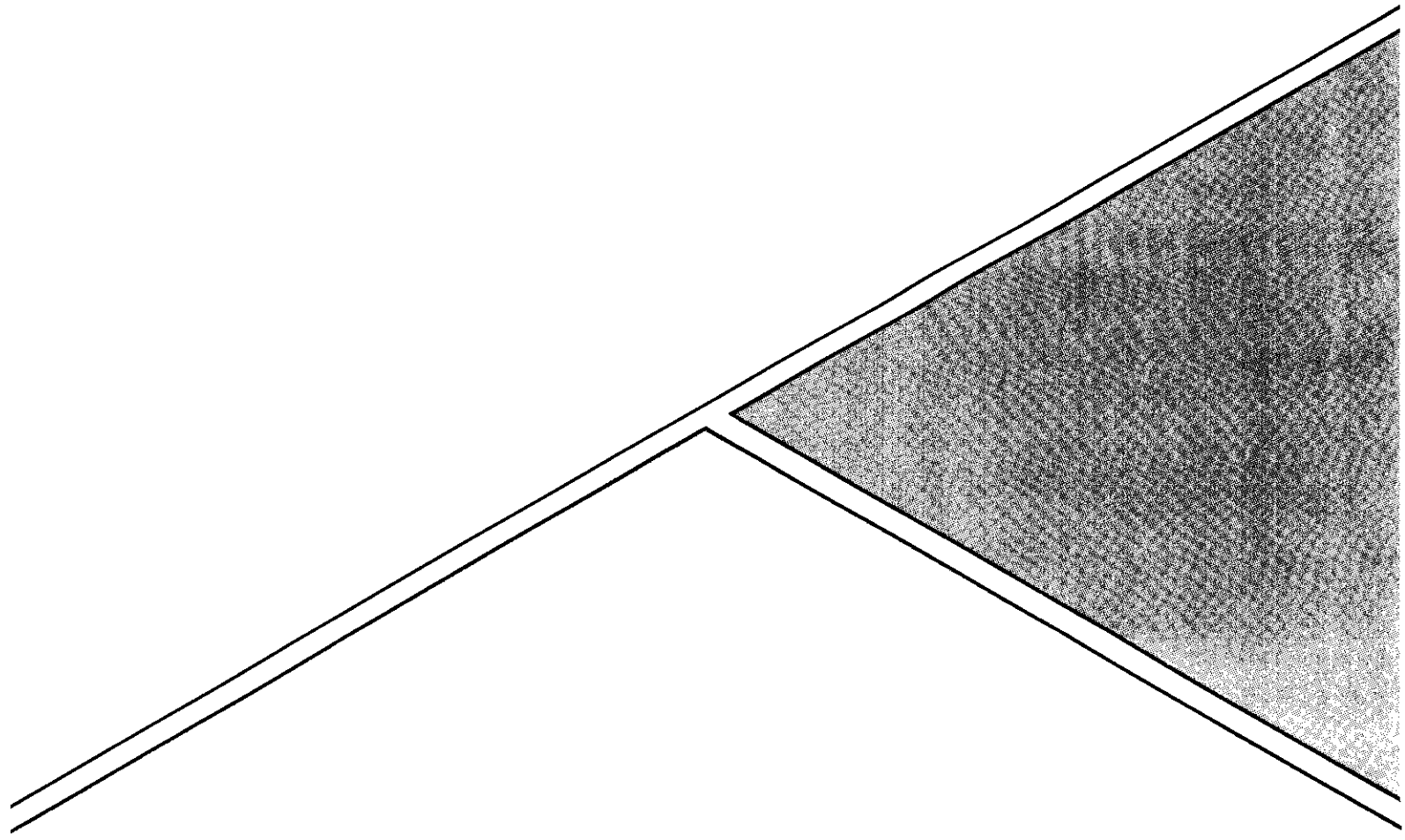
FIGURE 2: Organizational Structure Used by the Energy Conversion Equipment Branch

TABLE 1. ECEP's ANNUAL BUDGETS SINCE FORMATION OF DOE

FISCAL YEAR	1978	1979	1980	1981	1982	1983	1984	1985
BUDGET $MILLIONS	9.2	20.4	29.6	12.5	10.6	10.6	10.0	10.0



ELECTRIC HEAT PUMP TECHNOLOGY—I



EPRI Heat Pump R&D Overview

Arvo Lannus

Electric Power Research Institute

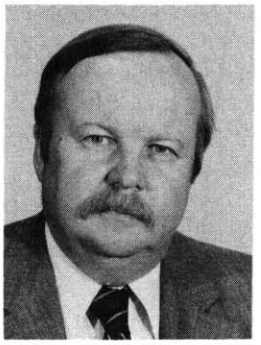
Palo Alto, California 94303

Arvo Lannus is manager of the Residential and Commercial Program, Energy Management and Utilization Division, at the Electric Power Research Institute (EPRI) in Palo Alto, California.

Before joining the Institute in 1980, he spent 6 years with Gordian Associates, an international consulting subsidiary of Pullman, Inc. As director of Advanced Technology, he was responsible for research on heat pumps and on the efficient use of energy in buildings and industrial centers. He also served as director of research for the Herculite Corporation. From 1969 to 1973, he was an assistant professor of chemical engineering at the Cooper Union School of Engineering, while also working as a consultant on energy and environmental problems.

Dr. Lannus holds B.S. (chemistry) and Ph.D. (chemical engineering) degrees from the Drexel Institute of Technology.

The author is a member of the American Institute of Chemical Engineers, the American Society of Heating, Refrigeration, and Air-Conditioning Engineers, and the International Solar Energy Society. He is the author of numerous papers on the subjects of heat pumps, energy conservation, and environment.



EPRI HEAT PUMP R&D OVERVIEW

Arvo Lannus
Electric Power Research Institute
Palo Alto, California

Introduction

Residential and commercial customers of electric utilities account for about 36% of the U.S. energy consumption, 60% of the electricity consumption and 99% of electric customers. Residential electric space heating, cooling and water heating and commercial/industrial space cooling, lighting and refrigeration represent the major uses of electricity in buildings. These customers are primarily responsible for the pronounced summer and winter peaks of electric utilities, yet they are least able to bear the cost of capital-intensive investments to save energy and reduce demand.

Electric utility interest in heat pumps stems from the potential benefits to customers of an energy-efficient, reliable and cost-competitive electric heating and cooling technology and from the need to maintain system annual load factors in the face of a steadily increasing market demand for comfort cooling. About 70% of the new single-family homes, 89% of the new multi-family residences and 80% of the new commercial floor space built in 1983 was equipped with air conditioning. Since most of these new buildings will need space and water heating as well, there is a significant potential for widespread acceptance of a reliable and cost-effective heating system - the electric heat pump. Indeed, nearly one new single-family home in three are equipped with a heat pump. The replacement and conversion (including add-on) markets, too, represent growth opportunities for the heat pump.

Clearly, much more is at stake for the electric utility industry than just the additional revenues from increased electric heating load. Consider that more than two-thirds of the utilities in the United States are summer-peaking. Air-conditioning demand growth has the effect of steadily eroding the annual system load factor (the ratio of average hourly demand to maximum demand) and, hence, the profitability because for many (if not most) utilities in this country the incremental cost of new capital investment to serve a kilowatthour of new load on peak is considerably higher than the average embedded cost of a kilowatthour.

Under these circumstances, an electric heating technology that can effectively compete for the heating load can significantly improve the utilization of utility plant — all the more so, because most areas of the country have two to five times as many heating hours as cooling hours.

But winter-peaking utilities, too, can benefit from the heat pump. High-efficiency equipment offers customers a means of keeping energy cost increases in check. And if gas, oil or propane is used for supplemental heat or if well-water or the earth itself is used as the heat source, eliminating electric resistance back-up, sharp heating peak demands are avoided.

The electric heat pump is currently only 15% of its ideal or Carnot efficiency; whereas, fossil fuel furnaces are almost at their maximum efficiency. The potential for efficiency

improvement and flexibility in utilizing a variety of heat sources offer a promising future for further development of electric heat pump technology.

EPRI Research Program in Heat Pumps

The EPRI research program is focused on developing broadly applicable heat pumps that combine improved seasonal performance and efficiency with more favorable utility load characteristics for both new and existing buildings. The principal goals for this subprogram are to:

- Develop a central heat pump with improved economics and efficiency, improved power demand characteristics in the heating and cooling modes, integrated water heating and greater flexibility for multizone and retrofit applications.
- Develop a unitary bivalent heat pump with high efficiency in cold climates, eliminating the use of supplemental resistance heat.
- Develop and test improved design and control procedures for water-loop heat pumps for commercial applications and improve the efficiency of equipment used in water-loop systems.
- Seek further increases in heat pump performance through development of advanced technology.

The nearer-term strategy of the subprogram is to continue improvement of vapor-compression-cycle air source heat pumps and to broaden applications of heat pumps capable of utilizing ground, well water, surface water and internal heat sources as well as waste or stored heat.

In the 1984-88 time period, major efforts will concentrate on developing central heat pumps for residential and small commercial buildings with warm air distribution systems as well as room (or single-zone) heat pumps for single-family home retrofit and multi-family apartment applications. The seasonal efficiency targets for central heat pump development are illustrated in Table I.

Table I

Overall Performance Goals for
Central Heat Pump Development

Heating efficiency	HSPF 10.4
Cooling efficiency	SEER 14 or better
Power Demand	Heating demand reduced by 2 kW
Noise	Reduced outdoor sound levels
Reliability	Improved defrost, reduced cycling
Controls	Set-back, load management capability
Zone control	Multi-zone (as option)
Water heating	Integrated into system

Supporting the equipment and systems development activities described above is a program of applications research to evaluate the performance of heat pump water heaters, develop and validate improved methods of system performance estimation, and field test equipment which is being commercially introduced and represents novel technical approaches to performance improvement. Examples are bivalent heat pumps for add-on to oil or gas furnaces, and dual-capacity modulating systems. Research and testing of large central-plant heat pump systems for commercial and industrial buildings will be initiated in this planning period, and community-wide applications of heat pumps using waste heat and other energy sources will be explored.

Advanced concepts are being investigated for their potential to achieve major improvements and broader applications of heat pumps in the longer term. Examples include non-vapor-compression cycles for speciality applications, nonazeotropic working fluids, hermetically sealed variable-speed motors for heat pumps, as well as other control, heat exchange and compressor component improvements. As these promising new approaches are evaluated and tested, heat pump systems projects will be undertaken to integrate, demonstrate and commercialize these advances.

Oak Ridge National Laboratory Electric Systems Overview

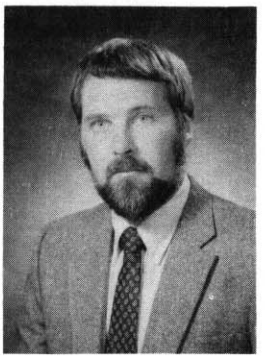
Phillip D. Fairchild

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

Phillip D. Fairchild is a group leader for heat pump research activities in the Energy Division at Oak Ridge National Laboratory (ORNL). He is presently in charge of the electric equipment subprogram and was previously in charge of the gas-fired heat pump subprogram. He began his employment with ORNL in 1978 as a project manager.

Fairchild previously worked for Ebasco Services, Inc., an architectural-engineering firm, where he served as principal engineer and senior engineer responsible for design engineering. He received the B.S. and M.S. degrees in mechanical engineering from Louisiana Tech University in 1965 and 1966, respectively. He is a registered professional engineer and is actively involved in the American Society of Heating, Refrigeration, and Air-Conditioning Engineers.



ORNL ELECTRIC SYSTEMS PROGRAM OVERVIEW*

P. D. Fairchild
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Introduction

Technical management of Department of Energy (DOE) research activities related to residential and commercial heat pump technology is assigned to ORNL. The history and objectives of the overall program were described in the previous opening remarks. As shown in Fig. 1, the major share (~85%) of the ORNL Building Equipment Research (BER) program resources is allocated to electric and thermally activated heat pump research. The remaining 15% is allocated to appliance-related research and program support activities (including administration, project evaluation, industry interaction, and foreign liaison). The activities comprising the electric heat pump systems portion (35%) of the program will be the focus of the two sessions today and the first session tomorrow morning. Most of our major ongoing research projects will be covered in some detail in these sessions.

In this overview I want to describe our overall program strategy, to outline the major areas of ongoing electric systems research, to mention some activities that will not be covered in the technical highlight talks, and to describe the new areas in which we plan major emphasis in the future.

*Research sponsored by the Building Equipment Research Division, U. S. Department of Energy under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

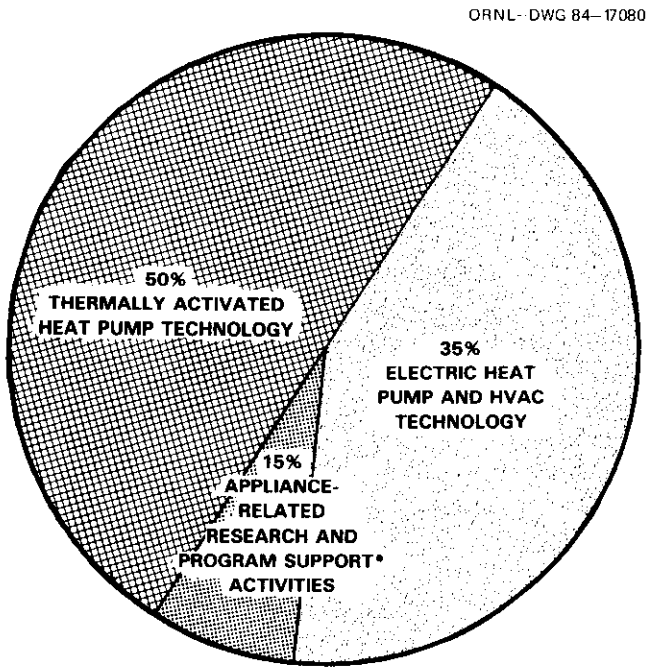


Fig. 1. Allocation of resources in the Building Equipment Research Program at ORNL.

Overall Program Strategy

Figure 2 is a diagram of the process we attempt to follow in formulating our research program and in transferring the resulting technology to the industry for the ultimate beneficiary, the consumer. The diagram illustrates the importance we attach to our relationship with the heating, ventilating, and air conditioning (HVAC) equipment manufacturers and the utilities [including their research organizations, the Electric Power Research Institute (EPRI) and the Gas Research Institute (GRI)] in the technology transfer process. However, the reluctance on the part of most domestic equipment manufacturers "to get involved with the government" has presented us with a major dilemma in implementing this strategy to the desired degree. Although we believe the situation is improving, partially in response to the pressure of foreign competition, we would like to have more industry participation in the program, not only in technology transfer and in conducting contracted R&D projects, but also in "technology evaluation" and defining industry's R&D needs.

One of the prime purposes of this conference is to provide an opportunity for feedback on the content and direction of the program from HVAC equipment manufacturers, utilities, universities, independent R&D organizations, and other research funding and management organizations—an opportunity not only to listen to our presentations but to speak out, to give us your criticisms and comments. Please let us hear from you, both during the conference and later, after you have returned home.

Ongoing Research Activities

Major areas of ongoing research associated with electric heat pump technology include the following:

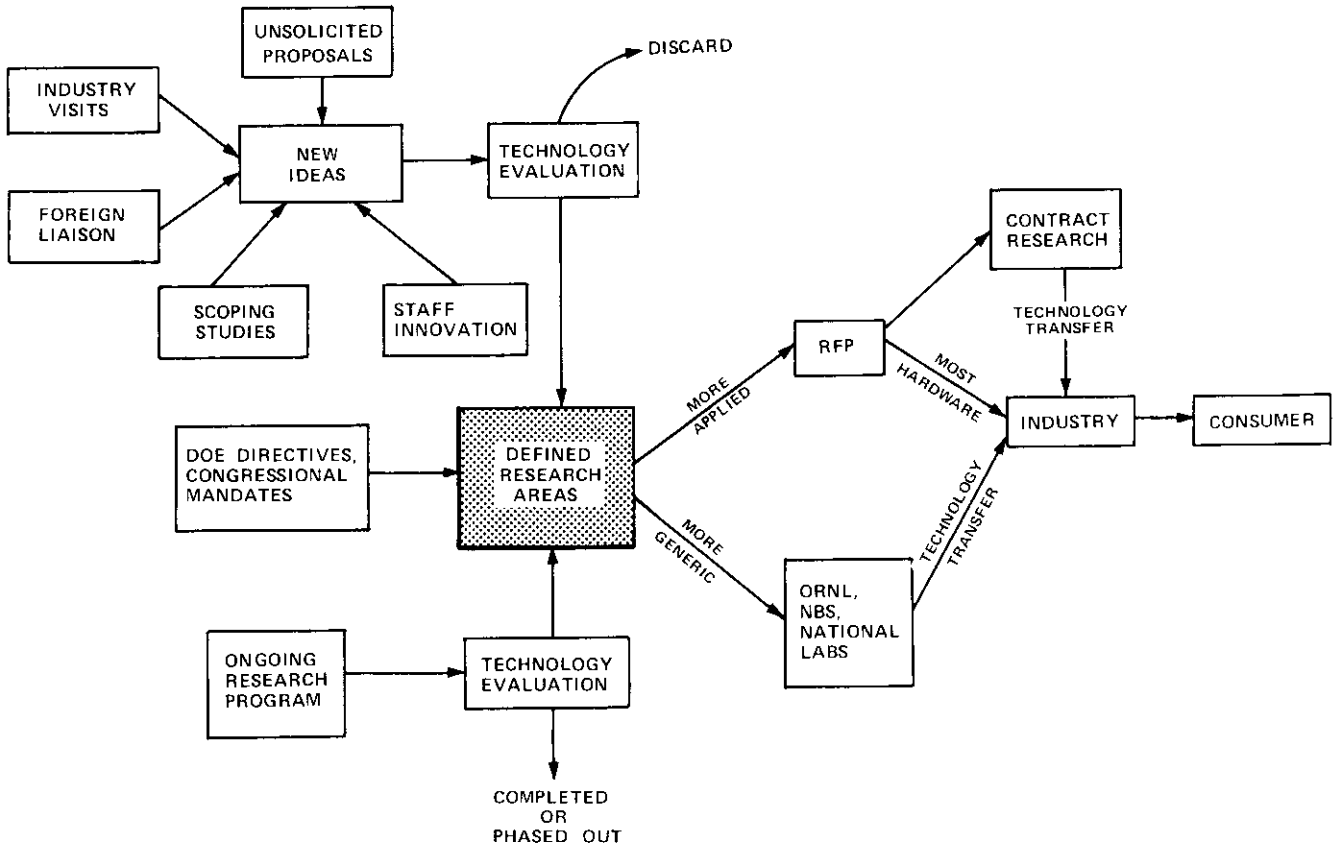


Fig. 2. Overall program strategy.

1. heat pump field performance,
2. dynamic loss phenomena,
3. heat pump design analysis software,
4. heat pump capacity modulation,
5. nonazeotropic refrigerant mixture research,
6. novel or unconventional refrigeration cycles, and
7. ground-coupled heat pump (GCHP) technology.

The first four areas listed are closely related and may be viewed as basic supporting research for all heat pump technology in the sense that they address the field performance of equipment under actual service conditions, characterizing and understanding dynamic loss phenomena, and establishing what effect these and other off-design losses could have on advanced technology equipment. No component or systems development per se is included in this research, although performance data from one prior equipment development project supported by DOE/ORNL are being utilized to help establish advanced technology equipment characteristics.

The objective of this work is to provide basic technical information and design data to enable the improvement of seasonal or "as operated" performance of heat pump cycle equipment. From understanding the underlying mechanisms of the losses that occur, techniques for eliminating or reducing them may be developed not only in state-of-the-art (SOA) equipment but in new technologically advanced equipment, possibly with significantly different configurations or design features.

Major projects in these areas include two field experiments, one laboratory study, one data survey, and an analytical effort.

Of the two field experiments, one is being conducted by ORNL at the Tennessee Energy Conservation in Housing (TECH) site near Knoxville and the other is being conducted by Westinghouse in a residence near Pittsburgh. The TECH site work has focused on SOA single-speed residential air-to-air heat pumps for the past few years, but is currently changing to a SOA two-speed unit.[1] The Westinghouse preprototype dual-stroke advanced heat pump developed under a previous DOE/ORNL cost-shared contract is being tested at Pittsburgh.[2] Both of these field experiment projects will be covered in more detail in the subsequent talks this morning.

The ORNL laboratory study is aimed at the detailed measurement of refrigerant migration with varying on-off cycles, frost formation with varying relative humidities, and other such factors that may help develop an understanding of the effect and cause of dynamic losses.[3] This investigation is showing some interesting results, which will be reviewed in the first presentation in the afternoon session. The study has to date focused on single-speed equipment, but will be extended to variable-capacity heat pump behavior in the coming months.

The field performance data survey is a subcontractor project with A. D. Little to determine the extent and quality of the data

that is available on heat pump field performance from manufacturers, utilities, and others. A broader data base on various equipment could help establish the validity of current conclusions (which are based on a detailed testing of a limited sampling of equipment) about the seasonal performance of heat pumps and about the total and relative magnitude of the identified dynamic losses. Further work is also planned on field experiment methodology, which will include documenting both successful experiences and the pitfalls of equipment performance monitoring.

To utilize the dynamic loss data, relate it to key design parameters and component features, and thereby effectively address performance improvement and optimization on a seasonal or other appropriate "as operated" basis, design analysis tools and techniques are needed. Work on advanced computer-aided design analysis tools and techniques for residential air-source heat pump and air-conditioning equipment has been conducted at ORNL for several years. Earlier work focused on the steady-state model for single-speed equipment and related design optimization. More recent work has emphasized dynamic loss effects and seasonal performance modeling. Computer programs have been developed, and current efforts are concerned with validation, technology transfer (to heat pump manufacturers and others), and documentation. Two of the presentations scheduled for the afternoon session will cover analytical studies conducted at ORNL, and will illustrate the usefulness of this design analysis software.[4,5]

Cooperative working arrangements with industry are also being implemented to provide ORNL with guidance for extending the usefulness of existing programs and with additional data for validation, in addition to facilitating transfer of this analysis technology to equipment manufacturers. Current analytical efforts focus on capacity modulation, since it is judged to be the most promising technique currently being considered or pursued by heat pump manufacturers for improving performance in advanced technology equipment.

Nonazeotropic Refrigerant Mixtures

In theory, the use of nonazeotropic refrigerant mixtures instead of a single refrigerant in vapor compression systems can lead to modest improvements in cycle efficiency, to capacity leveling, and potentially to active capacity modulation.

Activity to date in this area has focused on scouting experimental work with nonazeotropic mixtures in conventional equipment [refrigerator-freezer (R/F) and residential air-source heat pump], on basic supporting research, and on technology assessment and evaluation of R&D needs. Our evaluation of R&D opportunities has identified heat pump capacity modulation through mixture composition control as one of the most promising applications for the use of nonazeotropic refrigerant mixtures; therefore, it will be the main focus of our work in this area during the coming year. The first phase is an analytical study of advanced cycle concepts to achieve mixture composition (and system capacity) control.

Supporting research activities will continue on the heat transfer coefficients of mixtures and on developing a firm analytical base for system design, including reliable techniques for predicting mixture properties. A long-range objective is to identify and characterize new, improved nonazeotropic mixtures (e.g., ternary mixtures) that could represent technical breakthroughs in refrigerant technology.

One project completed in 1984 was an international SOA survey on mixed refrigerant technology[6], and another was the scouting experiment, in which an R-22/R-114 mixture was tested in the R/F application. The R/F tests showed no appreciable energy savings (vs R-12 baseline performance).[7] The other experimental project, involving the residential air-source heat pump application, will be reported on by the National Bureau of Standards (NBS) in the afternoon session.[8] The heat pump experimental project is complementary to work being supported by EPRI at NBS. DOE/ORNL work in this technical area is being coordinated with EPRI and NBS, who are working to develop a heat pump simulation model capable of handling binary nonazeotropic refrigerant mixtures. (NBS also has a related program involving physical measurements of mixture properties.) The basic supporting research project at the University of Illinois under Professor W. F. Stoecker involves condensing heat transfer studies of mixtures and generalized system simulation work.[9] It will also be covered in the session this afternoon. The new project on advanced adjustable capacity cycles is just getting underway.

Findings to date indicate that use of nonazeotropic refrigerant mixtures may lead to adverse refrigerant-side heat transfer effects. The net benefits, if any, of the use of mixtures has not been demonstrated experimentally, and the analytical tools are in an early stage of development. Even the basic properties data on these mixtures are very limited. Thus, more research results are needed to adequately evaluate the potential performance advantages of the use of nonazeotropic refrigerant mixtures and, if promising, to develop the needed technology for their successful application by equipment manufacturers.

Novel or Unconventional Refrigeration Cycles

There are a variety of refrigeration cycles or systems in existence that could theoretically substitute for the conventional vapor-compression system in building equipment. Examples are magnetic, thermoelectric, Brayton-cycle, and Stirling-cycle refrigerators. Some of these cycles are used in other fields, such as aircraft space conditioning or cryogenic refrigeration. One such unconventional cycle of particular interest is the Stirling cycle for air-source heat pump application. In contrast to the typical vapor-compression cycle, which loses capacity as the outside air temperature decreases, the capacity of the Stirling cycle remains relatively constant and is, therefore, potentially well suited to northern climate areas. The final paper in the afternoon session will describe our recent activities on this cycle concept.[10]

Although the conventional vapor compression system employing a single refrigerant has to date offered the best combination of cost, efficiency, and favorable operating characteristics for most heat pump or refrigeration applications, we believe it is nevertheless important to devote some effort to evaluating novel or unconventional system concepts, cycles, and fluids. If a breakthrough in the technology is achieved, it will likely come in these "high risk" research areas.

Ground-Coupled Heat Pump Technology

In GCHP technology, the objective of our work is to establish the technical limitations and preferred ground-coil design configurations for various soil and weather conditions. The performance of ground-coil heat exchangers (GCHXs) during cooling-mode operation, in particular, is not well understood for different soil types. Rejection of heat to the soil causes migration of moisture away from the coils, and this may reduce heat transfer drastically. There is not adequate data on this effect, nor is there a GCHX analytical method which accounts for these soil heat and mass transfer effects. A secondary objective is to develop a heat pump mechanical package design that is more suited to the ground-coupling application and to analyze cost/performance tradeoffs between GCHX sizing and heat pump efficiency levels.

The basic approach on GCHP technology is to develop reliable, analytically-based design methods for optimum coil configuration, and to demonstrate thermal performance in field experiments. Some laboratory experiments are also being used to provide detailed validation data. A series of technology assessment projects has also been conducted over the past few years under subcontract by Battelle. The series included a worldwide SOA survey of the technology, and technical and economic assessments of several different ground-coil configurations.[11, 12, 13]

The analytical models of the ground-coil heat transfer and related soil moisture effects are being developed at ORNL. GCHX model development and validation is proceeding progressively, starting with a heat transfer-only model for single coils and then adding capabilities that account for soil freezing, surface effects, moisture migration, and thermal interaction of adjacent coils.[14] This work will be described in more detail in a talk tomorrow morning (Session 3).

Program-supported field experiments have been conducted on two horizontal-coil and one vertical-coil GCHP installations. Results are summarized in Fig. 3. The horizontal-coil experiments

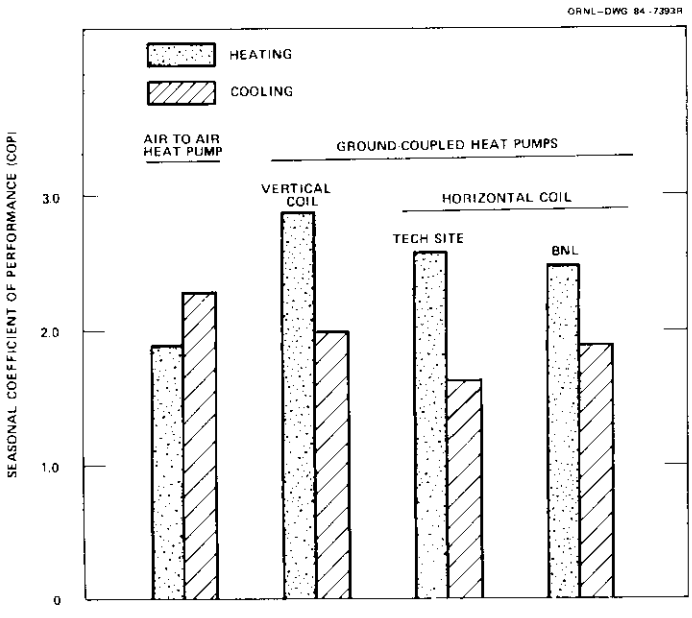


Fig. 3. Comparison of ground-coupled heat pump technology.

at the TECH site near Knoxville and at Brookhaven National Laboratory (BNL)[15] involved two different soil types, clay and sand. Heating mode results show a heating season performance factor about 30% better than an air-source heat pump (ASHP) system with equivalent technology. However, the experimentally observed cooling season performance was worse than the ASHP, particularly at the TECH site (clay soil). The TECH site horizontal-coil testing is being done by subcontract with the University of Tennessee, which will report in more detail on the results[16] in the GCHP session tomorrow morning. The superior performance observed for the vertical-coil installation is largely attributed to the coils, which extend well below the water table into saturated soil. Another observation of particular interest to electric utilities is that none of the installations required any supplemental resistance heat to meet the heating load.

In addition to these field experiments, a cooperative agreement has been implemented with Niagara Mohawk (a New York state utility) to provide additional experimental data for validation of the GCHX analytical models. Also, further laboratory experimental work to examine moisture migration and soil freezing effects in detail is being conducted by National Research Council–Canada under an International Energy Agency (IEA) cooperative project.

The remaining project is a system design effort and technology transfer activity related to the secondary objective of defining a more optimum heat pump/ground-coil "system" design. This joint effort of ORNL, BNL, and Friedrich has come up with some extremely interesting results,[17] which will be reviewed by BNL in the session tomorrow morning. The project was also conceived as a technology transfer activity in that it has given us an opportunity to work closely with a manufacturer in utilizing the heat pump model and other related analytical tools and techniques previously developed at ORNL. We also feel it has been a success in that regard.

Areas for Future Emphasis

Finally, I want to describe briefly the major new areas in which we plan expanded emphasis in the future, which are commercial-building HVAC equipment and occupant-equipment-building interaction.

In the first area, the focus is on systems that require applications engineering as opposed to the direct use of manufacturer-designed unitary equipment. The second area encompasses research on advanced HVAC distribution systems and controls (such as occupancy-following or zone-control systems), and on potential use of multiple comfort variables to provide equivalent occupant comfort with reduced energy input.

Commercial-Building HVAC Equipment

Past research activities in the BER Program have focused on residential unitary heat pump equipment, in part because this category represents the fastest-growing segment of the HVAC equipment market (and, as such, provides an attractive target for efficiency improvement), but also because it is a conveniently

sized vapor compression "system" suited to laboratory and field research investigations.

HVAC equipment for light commercial applications, say up to 15 tons cooling (180,000 Btu/h), is basically similar in concept to equipment for residential applications. However, for larger commercial buildings such as large office buildings, for example, there are substantial differences. Distribution systems and controls are more complicated. Concurrent heating and cooling are often required, and cooling loads tend to dominate because of large internal lighting and equipment loads.

In commercial-building HVAC systems, energy is often wasted by the mixing of hot and cold supply streams, nonoptimum use of outside air, excessive ventilation rates, load transfer between heated and cooled zones, oversized equipment, and the use of equipment when it is not needed. This illustrates that there are many factors beside component efficiency that affect *system* energy efficiency, and many of these factors form the basis for opportunities for R&D to develop technology for reducing energy consumption.

Our recent emphasis in this research area resulted, in part, from a Building Energy Equipment Workshop held in December 1983 in Airlie, Virginia. This event was organized by ASHRAE and the Committee on Science and Technology, U. S. House of Representatives. Two of the conclusions reached at this workshop were that there is a need for a federally funded R&D program on commercial-building equipment, and that an advisory panel with representatives of the various industry groups involved should be established to advise DOE on the projects that would have the most benefit to the nation. This advisory panel was organized by ORNL, and the first meeting was held in June 1984.

Some possible areas for federally funded R&D that have been identified are as follows:

1. generating field data on actual system performance as a function of system design variables;
2. developing improved analytical tools for optimizing system design, including electrical demand considerations;
3. developing advanced system concepts, including control schemes; and
4. working with the private sector in developing retrofit approaches and characterizing their benefits.

Coordination efforts have been initiated with other organizations working in this area, including other DOE laboratories (Lawrence Berkeley Laboratory, Pacific Northwest Laboratory, Argonne National Laboratory), the U. S. Army Construction Engineering Research Laboratory (CERL), NBS, EPRI, and GRI. ORNL will continue to work with these organizations, the advisory panel, and others to formulate a priority-ranked agenda for commercial-building systems research.

Building-Equipment-Occupant Interaction

Another perceived opportunity for reducing energy consumption for HVAC in both residential and commercial buildings lies in considering the interaction between the building envelope, equipment, the indoor thermal environment, and the occupants.

To date, most of the R&D for energy conservation in the buildings sector has been concerned with either improving the thermal envelope of the building structure or improving the efficiency of conventional types of space-conditioning equipment. An additional, broad, multifaceted area of research exists that deals with the characteristics of the indoor thermal environment provided, how the equipment is used (controlled), and how both of these factors affect the human occupants of the conditioned space.

Whereas HVAC systems are typically controlled by a single environmental variable—air temperature—the occupant's sensation of thermal comfort is affected by four environmental variables:

1. air temperature,
2. air humidity,
3. air movement, and
4. the mean radiant environment.

There has been an extensive body of human-factors research conducted that defines the combinations of these environmental factors that will be comfortable. There should be some combination of these factors that will produce equivalent comfort with reduced energy consumption. It appears that this opportunity has not yet received much attention.

Another opportunity is the possibility of reducing HVAC energy consumption by zone control or controls that follow occupancy patterns. While various experiments and studies in this research area have been conducted, it appears that a comprehensive exploration of possibilities and potential benefits is still needed.

Additionally, the relationship between building equipment and indoor air quality (IAQ) needs to be considered. While IAQ research is presently concerned primarily with indoor air pollutants such as NO_x, radon, and formaldehyde, the concept of IAQ needs to be broadened to include other factors such as air movement, stratification, infiltration, latent load, and pollutant dispersion. All of these factors are influenced by the characteristics and operation of the building equipment.

Our current activities in this new area of research are aimed at assessing the potential energy savings and other possible benefits of HVAC systems that control multiple comfort factors and/or that follow occupancy patterns. A subcontractor for this initial assessment study is being selected competitively through the request for proposals process (proposals are in the final stage of evaluation as of this writing). A SOA survey of existing knowledge in these areas is included in the scope as a first step in this assessment. Provided that a significant benefit appears attainable, it is planned that one or more preferred schemes will be defined and evaluated in laboratory or field experiments, or both. These experiments will provide information on the practicality of such systems, design-requirement information on system components, and data on energy savings and economics.

ORNL activities in this research area are being coordinated with related activities at BNL. BNL and ORNL cooperated in

conducting a Workshop on Thermal Comfort Management Technology in May 1984 in which research needs in these areas were discussed with authorities from the private sector.

Conclusions

We are in a period of declining budgets for DOE-supported research on electric heat pump and HVAC equipment technology. However, the outlook seems very promising for expanded use of heat pump equipment and for further improvements in equipment application and performance. Our challenge is to select the technology elements for our research that are needed to complement and enhance the R&D investment of the manufacturers, utilities, and other research organizations such as EPRI. We believe there are a number of remaining needs on the heat pump research agenda that are appropriate for DOE involvement. Therefore, we will continue to cooperate with you, the organizations represented here at this conference, to define appropriate DOE/ORNL research and to implement the program insofar as future funding and program philosophy permits.

References

1. V. D. Baxter, "ORNL Air-Source Heat Pump Field Experiments," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
2. S. E. Veyo, "Dual-Stroke Heat Pump Field Performance," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
3. W. A. Miller, "Laboratory Dynamic Loss Experiments," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
4. C. K. Rice and S. K. Fischer, "A Comparative Analysis of Single- and Continuously Variable-Capacity Heat Pump Concepts," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
5. J. C. Moyers and E. A. Nephew, "Analytical Study of the Benefits of Zone-Control Heat Pumps," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11- 12, 1984.
6. T. Berntsson, F. Moser, et al, "High Temperature Working Fluids and Nonazeotropic Mixtures in Compressor Driven Heat Pumps, A State-of-the-Art Report," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
7. *An Evaluation of a Two-Evaporator Refrigerator-Freezer Using Nonazeotropic Refrigerant Mixtures,* ORNL/Sub/82-47952/1, prepared for Oak Ridge National Laboratory by A. D. Little, Inc., November 1984.
8. D. A. Didion, "Laboratory Tests with a Heat Pump Using a Nonazeotropic Refrigerant Mixture," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
9. W. F. Stoecker, "Condensing Heat Transfer with Nonazeotropic Refrigerant Mixtures," (University of Illinois).
10. N. Domingo, "Analysis of Stirling Cycle Heat Pumps," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
11. D. A. Ball, et al, "Survey of Existing Knowledge for the Design of Ground-Source Heat Pumps," *ASHRAE Trans.* **89**(2B) (June 1983).
12. R. D. Fischer and G. H. Stickford, Jr., *Technical and Economic Feasibility of Horizontal, Multiple Shallow-Well, and Deep-Well Ground Coupling for Residential Heat Pump Applications–Final Report,* ORNL/Sub/80-7800/3&06, Battelle Columbus Laboratories, October 1983.
13. R. D. Fischer, G. H. Strickford, Jr. and F. E. Jakob, "An Assessment of Ground-Coil Heat Pump Technology," *Proceedings of the 7th Annual Heat Pump Technology Conference*, Oklahoma State University, Stillwater, October 1984.
14. V. C. Mei, "ORNL Ground Coil Analytical and Experimental Studies," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
15. P. D. Metz, "Ground-Coupled Heat Pump System Experimental Results," *ASHRAE Trans.* **89**(2B) (June 1983).
16. W. S. Johnson, "Ground-Coupled Heat Pump Research at the University of Tennessee," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
17. M. Catan, "Optimized Ground-Coupled Heat Pump Mechanical Package," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.

Dual-Stroke Heat Pump Field Performance

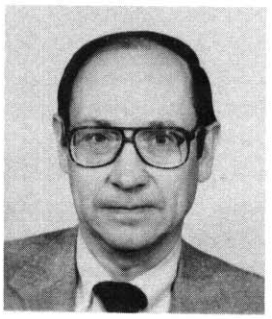
Stephen E. Veyo

Westinghouse Electric Corporation

Pittsburgh, Pennsylvania 15235

Stephen E. Veyo received his Ph.D. degree in mechanical engineering from Carnegie-Mellon University in Pittsburgh, Pennsylvania. Prior to joining the Westinghouse R&D Center, he served with the U.S. Army in South Vietnam as a Captain in the Corps of Engineers. As manager of Thermohydraulic Systems Research, he is responsible for guiding activities concerned with the thermodynamic heat transfer and fluid flow aspects of heat exchangers, refrigeration machines, power generation machinery, nuclear steam generators, thermodynamic instrumentation, boiling heat transfer, and two-phase flow phenomena.

Dr. Veyo is the current Westinghouse project manager for the U.S. Department of Energy-sponsored Advanced Electric Heat Pump project, having previously been the technology development leader for the same project. Since joining Westinghouse, his work has involved the development of mathematical models and their associated computer programs, and the design and execution of suitable verifying experiments. Dr. Veyo is the author of several articles concerned with heat pumps and is a member of the American Society of Mechanical Engineers and the American Society of Heating, Refrigeration, and Air-Conditioning Engineers.



DUAL-STROKE HEAT PUMP FIELD PERFORMANCE

S. E. Veyo, Manager
Thermohydraulic Systems
Research and Development Center
Westinghouse Electric Corporation
Pittsburgh, PA 15235

ABSTRACT

Westinghouse has developed an advanced electric heat-pump for the United States Department of Energy under contract 86X24712C with the Oak Ridge National Laboratory. Two nearly identical preprototype systems, each employing a unique dual-stroke compressor, were built and tested in the laboratory. One of these was installed in an occupied residence in Jeannette, Pa (near Pittsburgh) on November 24. 1982. It has provided the heating and cooling required from that time to the present. The system has functioned without failure of any prototypical advanced components, although early field experience did suffer from deficiencies in the software for the breadboard micro processor control system. Analysis of field performance data indicates a heating performance factor (HSPF) of 8.13 Btu/Wh, and a cooling energy efficiency (SEER) of 8.35 Btu/Wh for the year April 1, 1983 through March 31, 1984, with 450 hours missing because of malfunctions in the heat-pump, the data system, and the electrical service. Data indicate that the heat-pump is oversized for the test house since the observed lower balance point is 8°F whereas 17°F is optimum. Oversizing coupled with the use of resistance heat to maintain delivered air temperature warmer than 90°F results in the consumption of more resistance heat than expected, more unit cycling, and therefore lower than expected energy efficiency. Analysis indicates that if properly sized, the system could deliver an HSPF = 8.47 Btu/Wh, and an SEER=9.33 Btu/Wh for the observed weather profile at the field test site. Our analysis indicates that with optimal sizing the dual-stroke heat-pump will yield an HSPF 30% better than a single capacity heat-pump representative of high efficiency units in the market place today for the observed weather profile.

INTRODUCTION

The Westinghouse Electric Corporation has developed an advanced electric heat-pump for the United States Department of Energy under contract 86X-24712C with the Oak Ridge National Laboratory. The objective of the development program was a heat-pump that was at least 20% more efficient than the most efficient heat-pump in the marketplace at contract inception in 1979.

Two preprototypic models were produced, each utilizing a unique dual-stroke compressor, a unique refrigerant circuit, highly efficient air movers and electric motors, and a breadboard microprocessor control system. Details of the development effort may be found in references 1-6. The indoor unit is shown in Figure 1, and the outdoor unit in Figure 2. The indoor unit is of modular construction and is unique in that the lower module contains the compressor plus all other refrigerant handling components except the outdoor heat exchanger, the center module contains the indoor air mover and controls, and the upper module contains the resistance heat package and main fuses. The outdoor unit is also of unique design in that it contains only the outdoor fan and heat exchanger in a package that can be adjusted for elevation using integral legs. Frost melt is easily shed from the inclined face, horizontal tube heat exchanger elements. The elevation of the heat exchanger above the ground provides some protection against drifting snow.

Laboratory performance data, shown in Figures 3 and 4, when compared with catalog data or the ARI directory, shows that under steady conditions the efficiency of these preprototype units exceeds by more than 20% units available in 1979. In low capacity heating at 47° outdoor ambient, the system COP=3.83, (EER=13.1 Btu/Wh), and at 35°F, the COP=3.26 (EER=11.1 Btu/Wh). The system switches to high capacity mode heating for ambient temperatures less than 35°F. In high capacity mode at 47°F outdoor ambient, the COP=3.48 (EER=11.9 Btu/Wh) and at 17°F, the COP=2.71 (EER=9.25). In low capacity cooling mode, at 95°F outdoor

ambient, the COP=2.52 (EER=8.6 Btu/Wh), and at 82°F, COP=3.34 (EER=11.4 Btu/Wh). High capacity cooling is enabled at outdoor temperatures greater than 90°F and at 95°F, high capacity cooling COP=2.67 (EER=9.1 Btu/Wh). The system was designed to provide cooling with a sensible to total capacity ratio not greater than 0.7. This limits the achievable cooling efficiency because it prevents evaporator temperatures greater than about 45°F.

Defrosting should not be required in low capacity heating mode, since under frosting test conditions there was insufficient frost accumulation to degrade performance after 4.5 hours of continuous running. The defrosting penalty for high capacity operation was less than 5% at 35°F. The cyclic degradation coefficient for high capacity heating is C_d=0.04, for low capacity heating, C_d=0.18, for high capacity cooling C_d=0.29 and low capacity cooling, C_d=0.34. The rather high cyclic degradation coefficients in cooling mode, and low coefficients in heating mode can be attributed in large measure to the location of the oil cooler and compressor in the indoor air stream.

Optimally sized for residential application in a DOE Region IV northern climate, the heat-pump should balance the heating load in high capacity mode at about 17°F. In reference 1 we estimated that if optimally applied in the Pittsburgh, PA area we should achieve a heating season performance factor or HSPF=8.81 Btu/Wh, and a cooling season energy efficiency ratio or SEER=10.28. This efficiency estimate was based upon optimum heat-pump sizing for a house with level 2 insulation (see reference 7), in Cleveland weather. Predicted efficiency exceeds project goals.

After completion of the lab testing phase of our development project, one preprototype system was installed in an occupied residence in Jeannette, PA (near Pittsburgh) on November 24, 1982 for an extended engineering evaluation of field performance. The objectives of this field performance evaluation were threefold. First, the durability of the dual-stroke compressor was to be demonstrated as part of a host heat-pump system subjected to the trials of actual weather and residential application. Second, the thermodynamic performance of the system was to be monitored in detail so as to demonstrate that the system met its design goals for superior performance. Third, through detailed analysis of the field performance data, the merit of the advanced concepts embodied in the system was to be evaluated, and means identified to improve system performance.

TEST SITE

A northern climate site was desired for the dual-stroke heat-pump field evaluation test since the system was optimized for DOE region IV. The selection of a specific site for the field evaluation test was constrained by the fact that there are but two samples of the dual-stroke heat-pump in existence and no field personnel experienced with its installation and service. Since the professional and technician staff responsible for the development of the heat-pump system would have to service it in the field, a test site near Pittsburgh was highly advantageous. An engineer at the Westinghouse R&D Center not associated with this project expressed an interest in hosting the test in exchange for the installation of a state of the art unit at the completion of the program. His well insulated 2100 sq ft two story home, shared with a wife and two children, is located near Jeannette, PA, about 30 miles east of Pittsburgh. His home was then space conditioned by a seven year old three ton heat-pump. The building load was estimated from engineering drawings for the dwelling and the high capacity balance point for the preprototype dual stroke heat-pump was estimated at 18°F, very near the analytical optimum.

The heat-pump suitable for field installation was preprototype #2, which was cycled through the laboratory performance tests in October of 1982. It was installed in the field without exhaustive laboratory testing of the fault diagnosis and system trip software (ROM based) resident in the microprocessor in order to initiate the field evaluation phase of the research and development project immediately after hardware completion and laboratory testing, but before the onset of severe winter weather. Proof testing of the control system therefore took place in the field. The heat-pump was installed during the third week of November in a small utility room located in the basement, between an integral two-car garage and a finished family room. Except for cramped access, the installation was straightforward and the unit was pressed into service after installation.

INSTRUMENTATION

Site weather is monitored by a commercial roof mounted weather station consisting of an anemometer, wind direction indicator, temperature sensor, solar insolation indicator and a rain gauge. Installation of the weather station took place in early December 1982, shortly after the the equipment was delivered.

Thermodynamic instrumentation was installed to measure indoor return air flow, delivery and

return air temperature and relative humidity, and several temperatures and pressures within the refrigerant circuit, including compressor suction and discharge pressure and temperature. Barometric pressure in the utility room is measured using a pressure transducer. Condensate is collected and pumped to a self emptying reservoir. The water column height in the reservoir is monitored by the data system using a pressure transducer in order to estimate delivered latent cooling.

All motors, the entire heat-pump system, the resistance heat package and whole house energy consumption are monitored using electronic Watt/Watt-hour transducers. In addition, conventional rotating disc Watt-hour meters are used to measure the energy consumption of the hot water heater, the electric clothes dryer, an auxiliary game room heater, the heat-pump system and the resistance heaters. Line voltage is also monitored.

The control system contains 14 relays which are energized by the microprocessor's ROM based software to permit 20 distinct operating modes. The data system monitors the state of these control relays.

The data system, a Hewlett Packard HP3497A plus an HP85F computer, scans continuously, monitoring 84 channels of data (60 analog voltage channels, 14 relay states, and 10 digital counters). A full scan is recorded on an on-site Quantex dual cassette tape recorder only if it immediately precedes or follows an event or is triggered by a 15 minute timer. An event is defined as the change in state of any one of the fourteen relays in the heat-pump control system. A group of priority channels is recorded from every scan. There are 140 seconds between successive scans in order to accommodate statistical processing, thermocouple linearization and units conversion. Time resolution of an event is thus of the order of 140 seconds. The on site tape recorder is queried periodically over the telephone using an automatic answering modem and the recorded data transferred to the main frame computer at the R&D Center. Typically, the period is once each day.

Although the heat-pump system was installed in November, the installation of the data system and its field resident software was not completed until March 4, 1983. Heat pump performance could be monitored at the Westinghouse R&D Center, some 30 miles distant, using a computer terminal and the telephone system, starting in mid January 1983. This monitoring process was tedious since the available data were instantaneous in nature and for many channels the data were in voltage units. Proper unit function was relatively easy to determine, but efficiency was not. Data were archived on computer tape at the R&D Center, but those data prior to March 4, 1983 were judged to be too expensive to process because of frequent changes in the data logger software. Data have been archived continuously since April 1, 1983 and processed.

Data reduction software resident in the Univac 1100/81 computer at the R&D Center attributes each scan of the data to one of the operating modes, and to the appropriate ambient temperature bin. In order to determine operating efficiency, both the heating and cooling energy delivered and the respective energy consumed must be known. Energy consumption has been determined without any ambiguity for the heat-pump system and the constituent components. The heating energy delivered is estimated by integrating with respect to time the instantaneous readings for the change in air enthalpy across the indoor section of the heat-pump. The cooling energy delivered is estimated similarly using the amount of condensate collected as a measure of the latent heat. An effective condensate handling system was not in operation for the entire reporting period however, and some early data was corrected using the sensible to total ratio as a function of temperature that was measured in the latter portion of the cooling season. Laboratory measurements of transient heat-pump behavior are used in conjunction with models for the transient behavior of the sensors to interpolate between data samples, and to extrapolate to event initiation when performing the integration. Data reduction software at the R&D Center processes approximately 34,000 data points each day and prepares a summary of performance segregated by ambient temperature bin for each day. A summation program then operates on these daily records to provide summary reports for periods of arbitrary length, such as a week or month.

FIELD PERFORMANCE

Operating Status

The dual-stroke advanced heat-pump has supplied the heating and cooling required by an occupied residence from November 24, 1982 through October 1984 without failure of any of the advanced components in the system. In particular, the oil cooled dual-stroke compressor has performed reliably with no deterioration in performance. The only components to suffer failure were the compressor contactor, start relay and capacitor, all items of commercial hardware. The indoor blower has required adjustment on several occasions to restore critical clearance dimensions that had diminished with use.

The breadboard microprocessor control system and its resident software demonstrated that they were indeed breadboard in nature by virtue of the problems encountered. Upon installation, the

micro processor system proved to be vulnerable to line voltage fluctuation, most particularly dropouts, which occurred with regularity. A redesigned power supply cured these problems. The first year, and in particular the first quarter, of operation was marked by repeated nuisance trips of the system, caused not by refrigeration system malfunction, nor by electricity supply problems, but primarily by bugs in the microprocessor software. The system would mistakenly diagnose a variety of fatal ailments, low charge most frequently, and proceed to shut down the system. The first version of the resident software did not automatically engage emergency heat, with the result that the occupants of the residence were considerably annoyed and inconvenienced. Revised software eliminated most nuisance trips and ameliorated host dissatisfaction. Lastly, several bugs in the software were discovered, but since project resources were limited and since the bugs could be circumvented without deleterious impact upon system function, new ROM's were not prepared. System trips plus necessary down time for instrumentation, maintenance and system inspection resulted in heat-pump unavailability for 239 hours during the report period of 8784 hours. The data system was down for 198.5 hours, and the dwelling was without electrical service for 12.5 hours during the report period. System status is shown in Figure 5. The occupants of the residence have on many occasions reset the microprocessor in order to permit its continued function after a nuisance trip.

Operating History

The histogram of hours of temperature occurrence for the field test site is presented in Figure 6, along with the mode of heat-pump operation. The inferred building load is shown in Figure 7 along with heat-pump capacity. Note that although the zero load point is approximately 57°F, the hours in heating mode and in cooling mode diffuse across this temperature. Note also that the high capacity heating balance point is approximately 8°F, considerably lower than both the 18°F expected and the 17°F that is optimum. The low capacity heating balance point is approximately 27°F, considerably below the cutoff temperature for low capacity operation-roughly 32°F. Figure 7 indicates that high capacity cooling operation should not be required.

The dual-stroke advanced heat-pump must be configured for control purposes through the input of control parameters. Since the expected balance point was 18°F, the first stage of resistance heat was enabled at 19°F. The switchover temperature between low capacity and high capacity operation was set to occur at 39°F. The second stage of resistance heat was set to come on with the compressor upon a call for heat at ambient temperatures less than 8°F, the third stage at -3 °F and the fourth stage at -14°F. The resistance heat stages are nominally 4.8 kW each. Whenever resistance heat is energized, the control system locks it on until the system cycles off with a satisfied thermostat. High capacity cooling is enabled at temperatures greater than 90°F, if the load has not been met within 20 minutes of run time.

For the report period, system status is shown in Figure 5. Note that the occurrence of emergency mode does not necessarily mean that the system had a fault, but could also indicate that the system was parked in this mode while some maintenance was being performed on the system, its instrumentation, the data system or its power supply. Figure 8 shows that the unit ran in the cooling mode for 25% of the hours during which the unit was set to cool, with only 4% of the run time occurring in high capacity mode. Figure 9 shows that of the time spent in heating mode, the unit was off 73% of the time, running in low capacity for 6% and in high capacity for 22%. The distribution of duty hours as a function of ambient temperature is shown in Figure 10. The duty factor is lower than is optimum primarily because the unit is oversized. The distribution of running hours by operating mode is shown in Figure 11.

Thermal Energy Delivery

The profile of delivered space conditioning thermal energy is shown in Figure 12. Approximately 54.9 million Btu of heat was delivered and 21.1 million Btu of cooling. The distribution of delivered heat between high capacity mode, low capacity mode and resistance heat is shown in Figure 13 as a function of outdoor ambient temperature. As shown in Figure 14, 72% of the heat was provided by high capacity operation, 16% by low capacity, 9% by resistance heat and 4% was delivered under the emergency mode of operation.

Electrical Energy Consumption

The distribution of consumed electrical energy between the HVAC system, the uninterruptable power supply for the data system and the remainder of the building is shown in Figure 15 as a function of the outdoor ambient temperature. Electrical energy consumption totaled 33,200 kWh, with the HVAC system consuming 9280 kWh, emergency operation 630 kWh, and the UPS consuming 9160 kWh. The water heater consumed approximately 6250 kWh, the electric dryer approximately 930 kWh and the game room heater approximately 390 kWh. The lighting, cooking and miscellaneous electrical load then totals approximately 6500 kWh.

The distribution of the energy consumed by the HVAC system among the various components of the system is shown in Figure 16 as a function of the outdoor ambient temperature. Figure 17 shows that in the heating mode, the control system and the crankcase heater consume about 4% of the total energy consumed. The dual-stroke compressor uses an external belly band heater of the self regulating temperature type, rated at about 50 W maximum and connected directly across the power supply. (The compressor is well insulated in the preprototype units.)

Efficiency

The observed annual performance factor heating for the system during the report period was HSPF=8.13 Btu/Wh, excluding periods when the system was in the emergency heat mode. The observed annual energy efficiency ratio cooling was SEER=8.35 Btu/Wh. For the season May 1 through October 15, the SEER=8.36 with a sensible to total ratio equal to 0.64. Figures 18 and 19 show the distribution of energy efficiency with outdoor ambient temperature.

Because the dual-stroke heat-pump is oversized relative to the observed building load, excessive cycling is an expected contributor to less than expected efficiency. Figure 20 shows the distribution of the estimated on and off times as a function of outdoor ambient temperature. At about 27°F, the building load is roughly half the heat-pump high capacity. Note that below 17°F, the average length of a run cycle declines because the resistance heat is applied. Recall that the control system clamps resistance heat on once initiated, until the thermostat is satisfied. The amount of resistance heat used in the 17°F bin indicates that the second stage thermostat contacts closed quite regularly since they alone activate the first resistance heat stage.

The expected HSPF referenced earlier was estimated on the basis of optimum sizing (17°F high capacity heating balance point), a ten year average Cleveland weather profile and a simple resistance heat implementation strategy that assumed that only that amount of resistance heat was used as was necessary to supply the building heat load. The estimated HSPF based upon the observed weather profile (with emerging hours deleted) and the observed building load is 8.56. The estimated SEER=8.56 for this same application. Table 1 contrasts the measured and predicted performance of the dual-stroke heat-pump as applied with the performance under optimum sizing of the dual-stroke unit, a 1979 two-speed unit and a 1984 single speed unit. All but the first two cases use the full observed weather profile normalized to 8760 hours. Note that with optimum sizing and test site weather, the dual-stroke heat-pump should have an HSPF that is 29% better than a 1979 two-speed heat-pump and an SEER that is 17% better. Relative to a 1984 single-speed heat-pump, the dual-stroke HSPF is 31% superior while the SEER is the same. On an annual basis, the two-speed unit is estimated to consume 22% more electricity and the 1984 single-speed unit 26% more electricity than the dual-stroke unit. Project goals for efficiency have clearly been met.

ACCOMPLISHMENTS

The Westinghouse/DOE dual-stroke advanced electric heat-pump in preprototypic form has survived almost two years of use in an occupied residence providing the space conditioning function.

There have been no failures of any of the advanced preprototypic components: dual stroke compressor; dual strength compressor motor; high efficiency blower; fan; blower motor; or fan motor.

A low cost compressor capacity modulation concept, the dual stroke cam and two strength hermetic motor, has been demonstrated under field conditions.

An on line remote performance monitoring, data acquisition, data processing and data compilation system has been developed and used to process field data.

The dual-stroke advanced electric heat-pump has demonstrated that it can deliver superior performance.

The penalties associated with heat-pump oversizing and simple resistance heat management have been quantitatively demonstrated.

An extensive data base for monitored heat-pump performance has been assembled.

EXPECTED BENEFITS

The demonstration that there exists a low cost capacity modulation concept, and that this concept is reliable and efficient, will encourage further development and adoption of capacity modulation for northern climate heat-pumps.

ACKNOWLEDGEMENT

The development of the advanced dual-stroke heat-pump system was funded in part by the U.S Department of Energy through the Oak Ridge National Laboratory under subcontract 86X24712C. The continuation of this contract for the purposes of evaluating this hardware in the field and discovering means to improve performance was funded in its entirety by DOE. The guidance offered by Mr. George Privon and Mr. Phillip Fairchild of the Oak Ridge National Laboratory is gratefully acknowledged. The development of the advanced electric dual-stroke heat pump was a team effort and the

successful performance of this unit in the field is a tribute to the entire team. The data acquisition and processing software for this project was developed by Dr. R.A. Lucheta, and he has remained responsible for the maintenance of both the data software and hardware. System performance analysis and maintenance has been the responsibility of Mr T.J. Fagan, Jr., assisted by Mr. N.J. Wilds. The contribution of the homeowner who has participated in this project is also gratefully acknowledged, for he has suffered through the learning pains associated with early field test of laboratory hardware and software and has accommodated the intrusions due to visitors, maintenance and service personnel and a rack of instrumentation.

REFERENCES

1. Veyo, S. E., and Fagan, T. J., "Advanced Electric Heat Pump Dual- Stroke Compressor and System Development", ORNL/Sub/79-24712/3, Oak Ridge National Laboratory, December 1983.

2. Veyo, S. E., Advanced Electric Heat Pump Component Development and Evaluation - Interim Report", ORNL/Sub/79-2472/2, Oak Ridge National Laboratory, September, 1982.

3. Veyo, S. E., "An Optimized Two-Capacity Advanced Electric Heat Pump", ASHRAE Journal, November 1982, pp. 42-48.

4. Young, R. R., Veyo, S. E. and Kastovich, J. C., "A High Efficiency Dual Stroke Compressor for Heat Pumps", Symposium Paper AT 84-19, No. 1, to be published in ASHRAE Transactions.

5. Ariewitz, D., Lackey, R. S. and Veyo, S. E., "A High Efficiency Indoor Air Mover", ASHRAE Transactions, Vol. 89, Part 2, 1983, (Symposium Paper DC 83-13, No. 3).

6. Wright, T., Lackey, R. S. and Veyo, S. E., "A High Efficiency Heat Pump Fan", ASHRAE Transactions, Vol. 89, P. 2, 1983, (Symposium Paper DC 83-13, No. 2).

7. Kirschbaum, H. S. and Veyo, S. E., " An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate", EPRI EM-319, Project 544-1, January, 1977.

TABLE 1

HEAT PUMP PERFORMANCE COMPARISON (8760 hours except *)

	AS APPLIED			OPTIMUM SIZE		
	*FIELD TEST	*SIMPLE ESTIMATE	SIMPLE ESTIMATE	DUAL-STRK	2-SPEED 1979	1-SPEED 1984
HSPF	8.13	8.56	8.53	8.47	6.54	6.46
HEAT-HI	40874	34104	37685	35388	24045	43409
HEAT-LO	8969	16957	17290	17290	27511	0
HEAT-I2R	5031	1346	1631	3922	5050	13197
EMERGENCY	2169	0	0	0	0	0
HEAT DELIV	57043	52407	56606	56600	56606	56606
BAL-HI/LO	8F/26F	8F/26F	8F/26F	17F/31F	17F/27F	30F
COMP HRS HI	1086	614	718	1198	1136	1944
COMP HRS LO	293	1662	1714	1780	1500	0
SEER	8.35	8.56	8.56	9.33	7.96	9.35
COOL-HI	1392	0	0	413	0	23117
COOL-LO	19679	23117	23117	22704	23117	0
COOL DELIV	21071	23117	23117	23117	23117	23117
COMP HRS HI	37	0	0	16	0	1362
COMP HRS LO	804	948	940	1292	1148	0

* 8334 HOURS

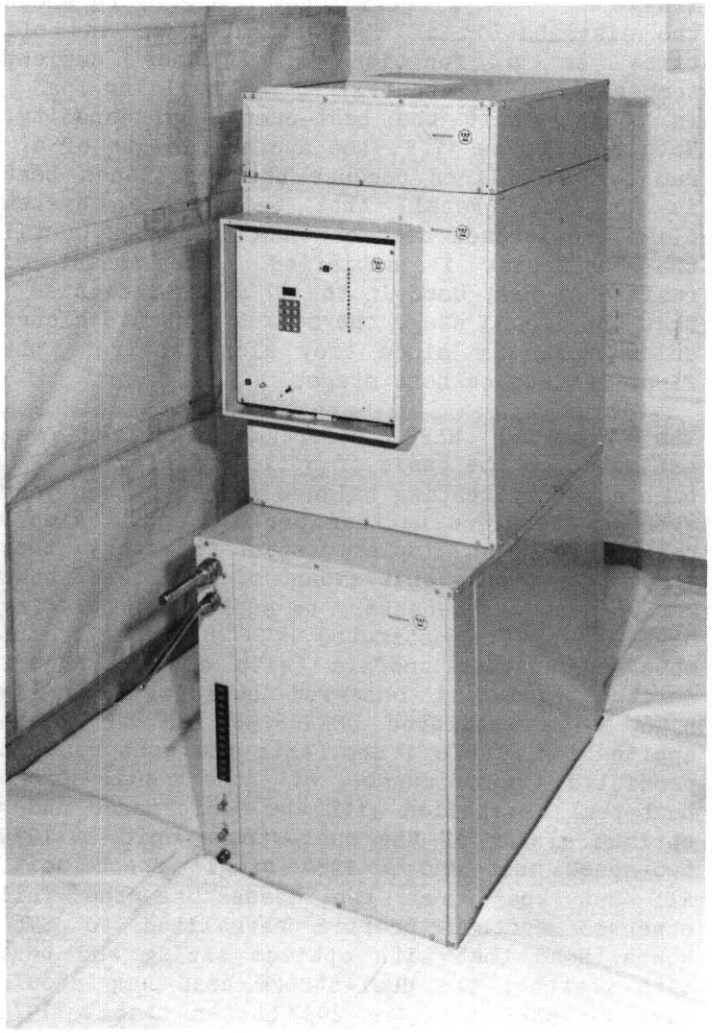


Fig. 1 - Preprototype heat pump indoor unit

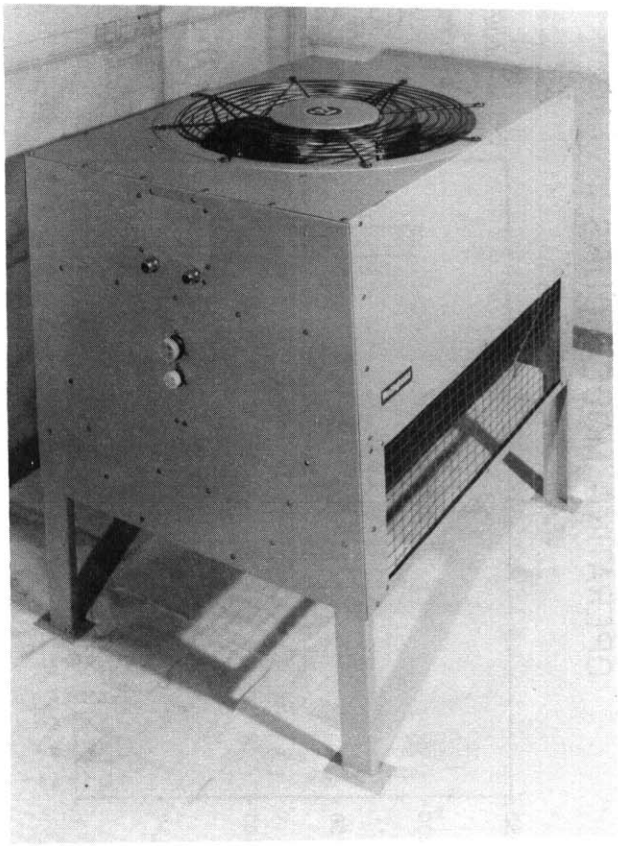


Fig 2 - Preprototype heat pump outdoor unit

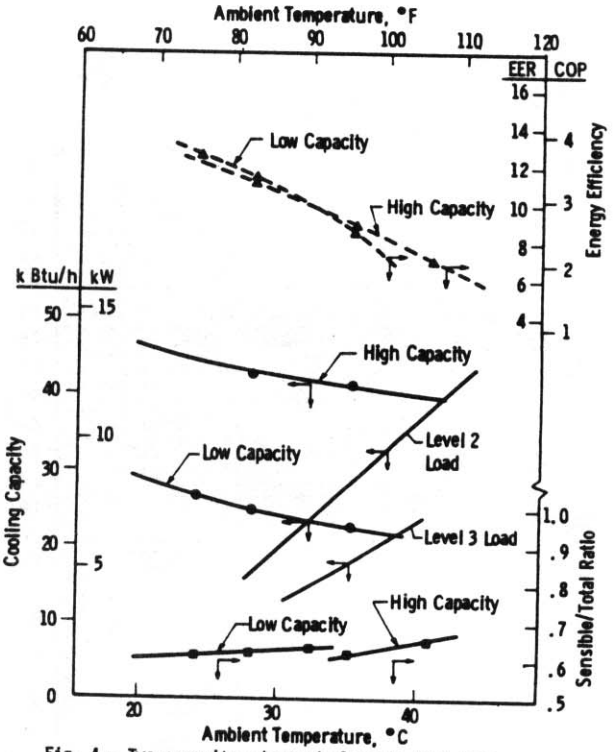


Fig. 4 - Two-capacity advanced electric heat pump measured cooling performance

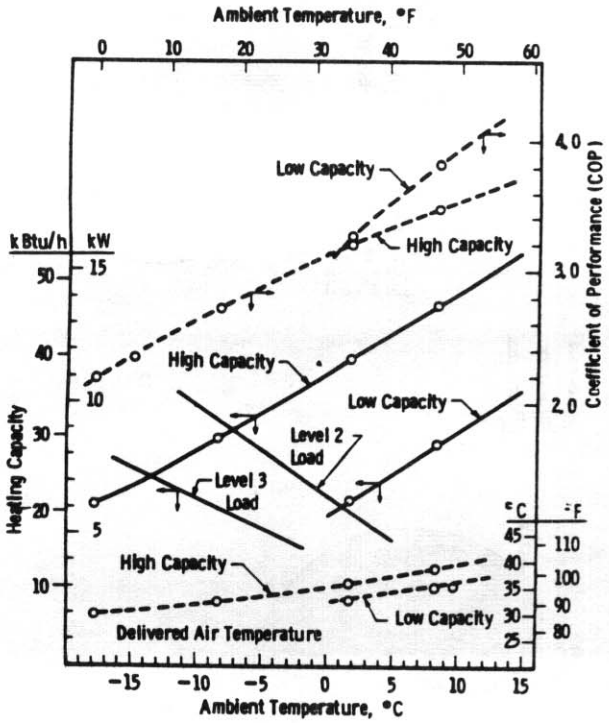


Fig. 3 - Two-capacity advanced electric heat pump measured heating performance

Fig. 5

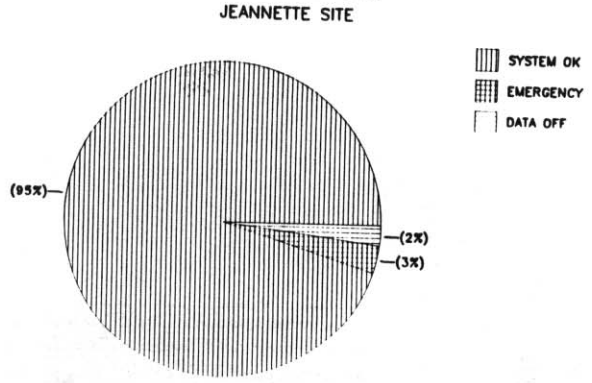


Fig. 6

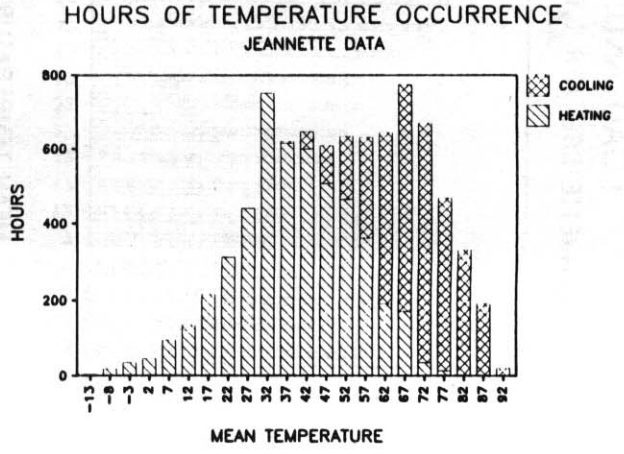


Fig. 7

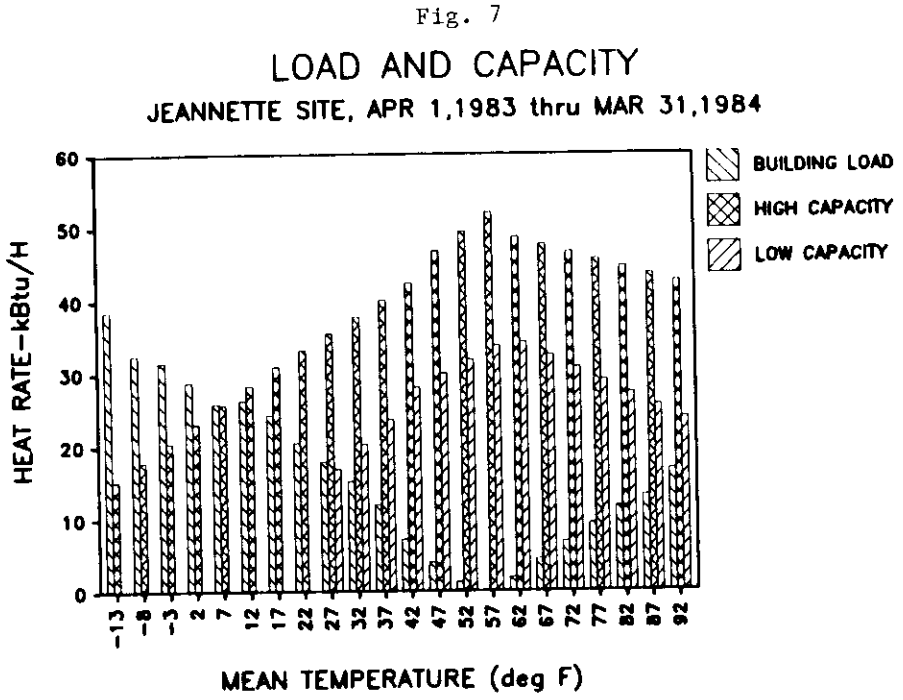


Fig. 8

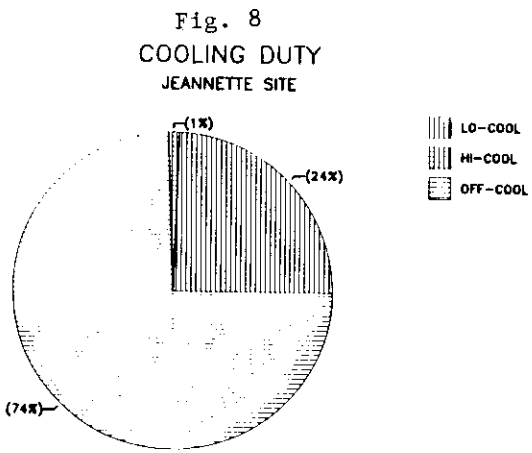


Fig. 9

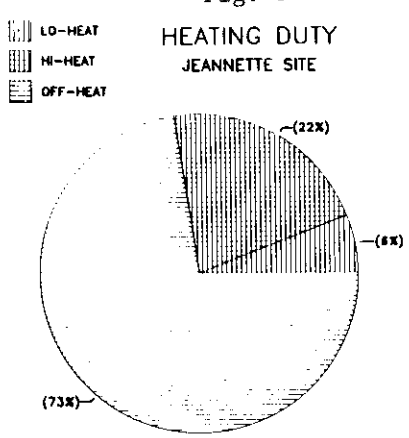


Fig. 10

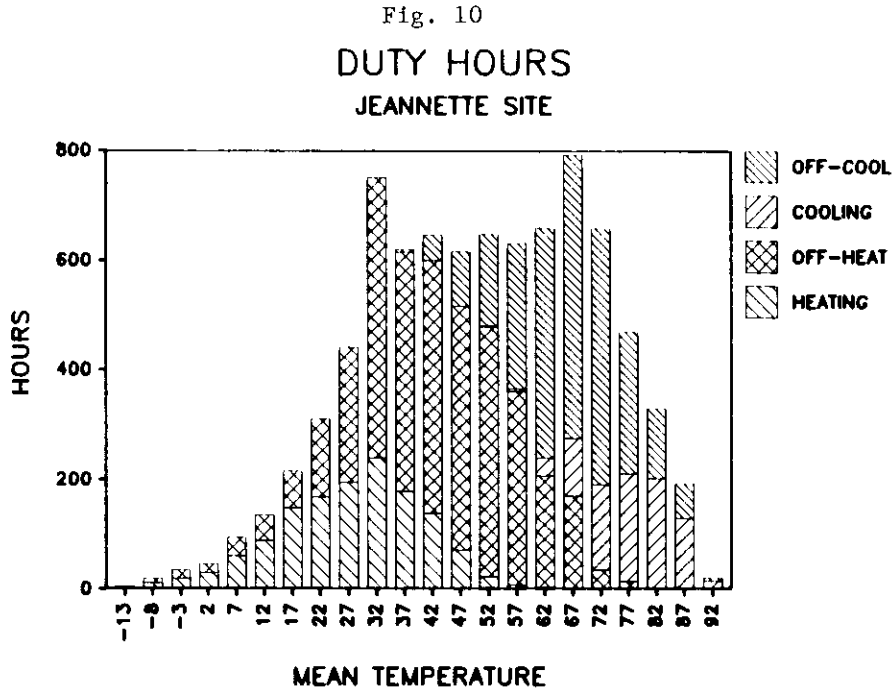


Fig. 11

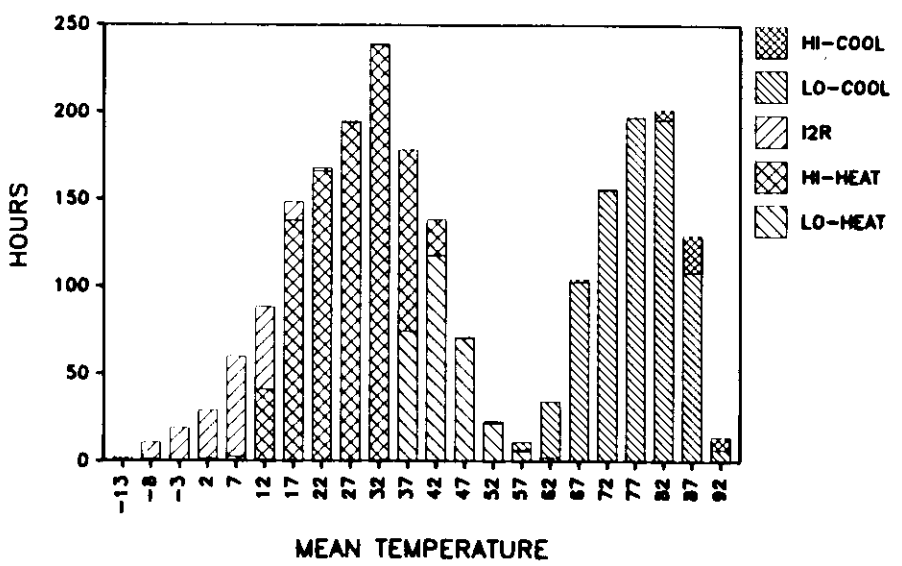


Fig. 12

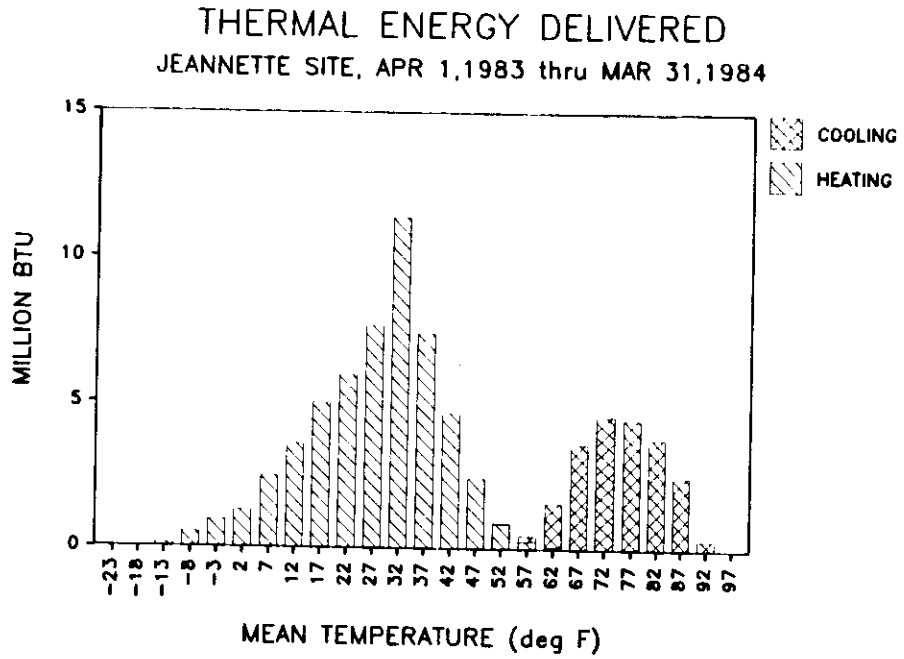


Fig. 13

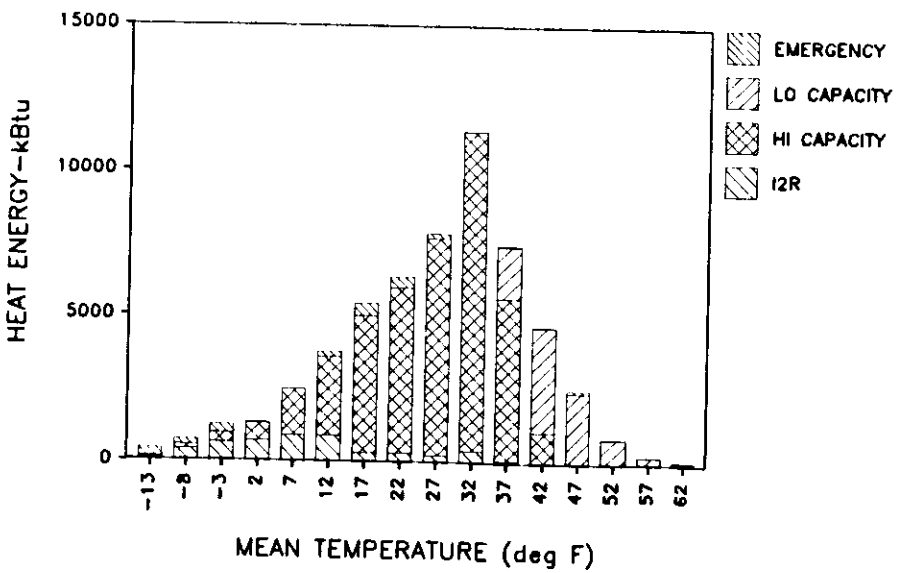


Fig. 14

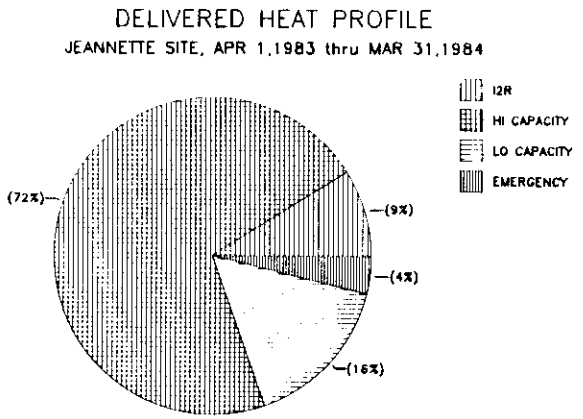


Fig. 15

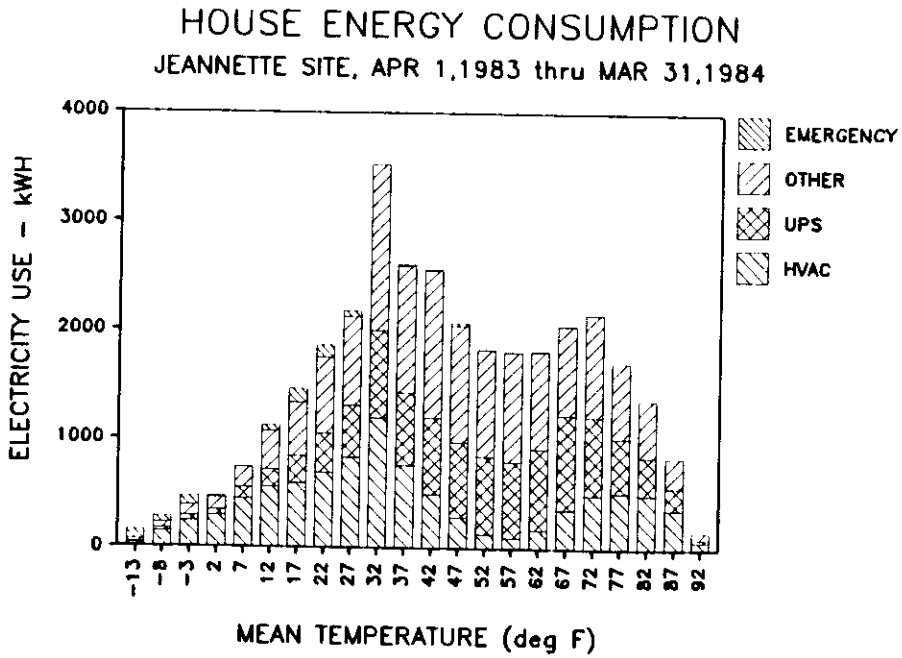


Fig. 16

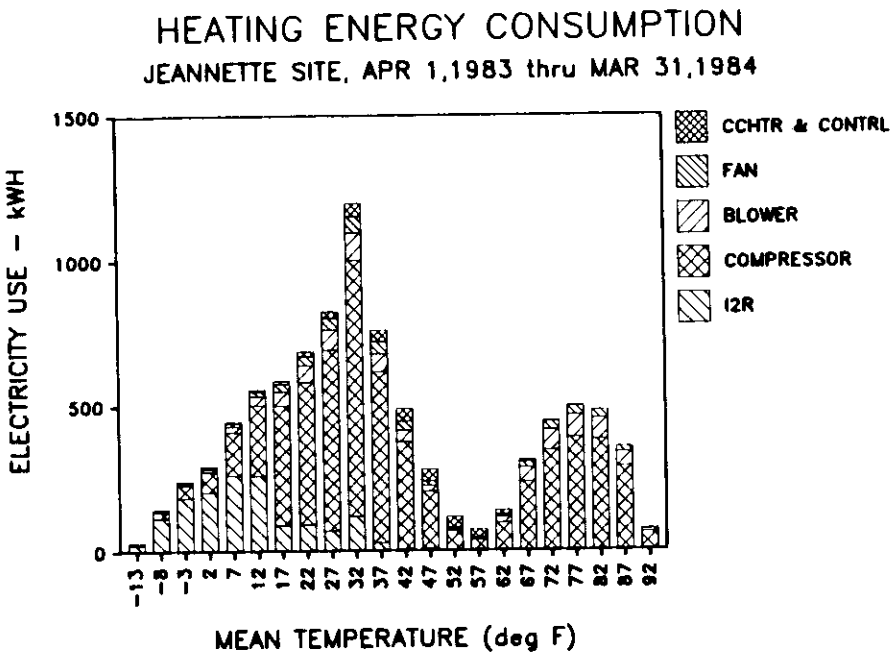


Fig. 17

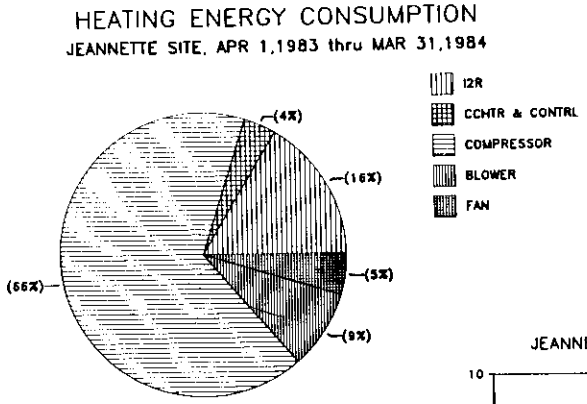


Fig. 18

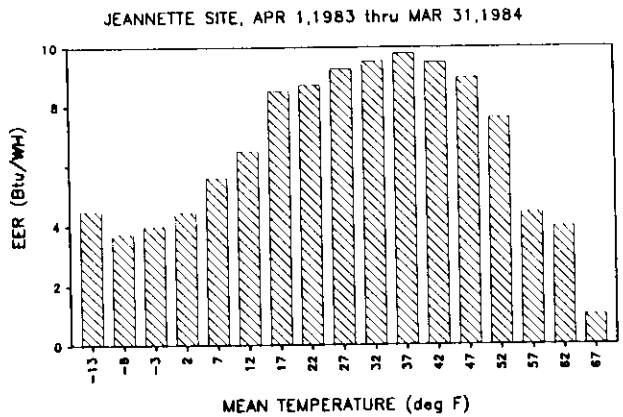


Fig. 19

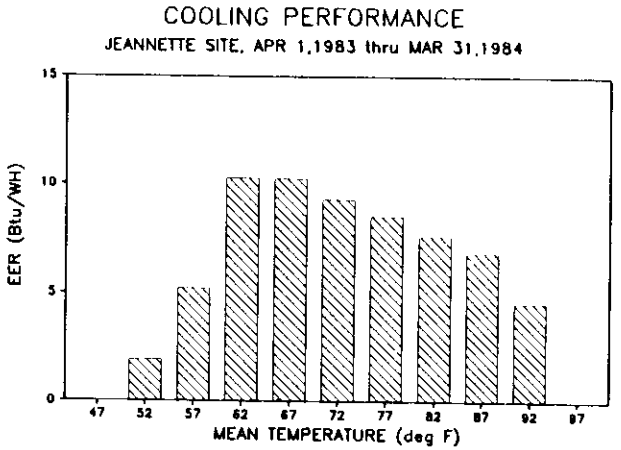
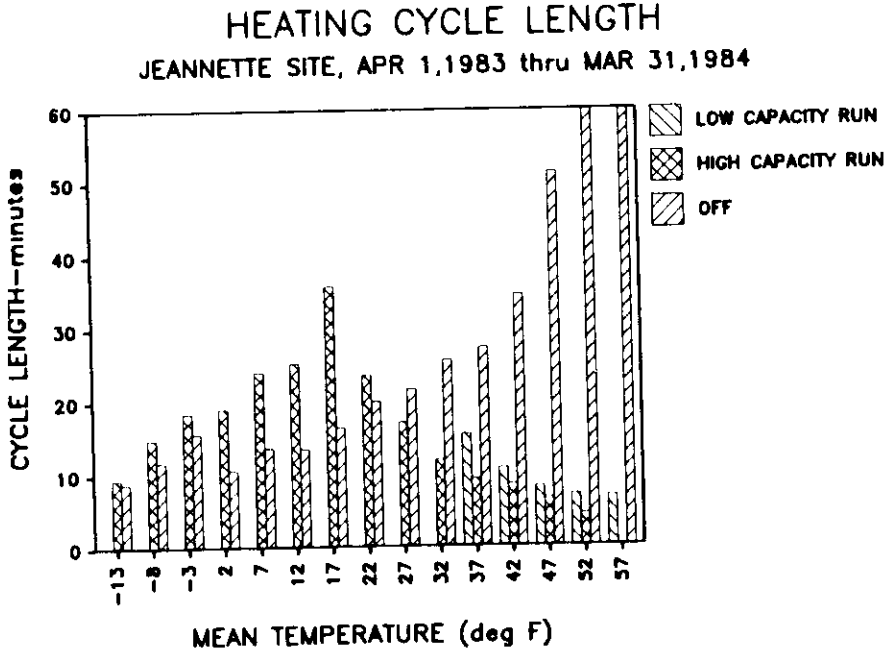


Fig. 20



ORNL Air-Source Heat Pump Field Experiments

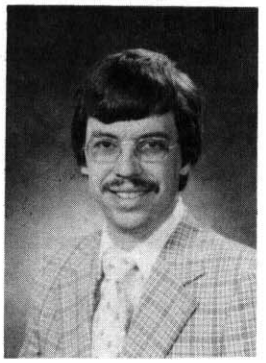
Van D. Baxter

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

Van D. Baxter received the B.S. and M.S. degrees in mechanical engineering from the University of Tennessee. Soon after receiving the M.S. degree in 1977, he joined Oak Ridge National Laboratory where he has done research on a number of innovative heat pump concepts, including the Annual Cycle Energy System and ice-maker heat pumps.

Since 1979 he has been responsible for a field test facility to investigate the field performance of air-source and ground-source heat pumps. He has written numerous papers for publication in *ASHRAE Transactions* and was the recipient of ASHRAE's 1982 Willis H. Carrier Award.



ORNL AIR-SOURCE HEAT PUMP FIELD EXPERIMENTS*

V. D. Baxter
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Objective

The objective of the air-source heat pump field tests at ORNL is and has been to develop a data base of field-measured heat pump performance. Information obtained includes seasonal performance factors; magnitudes of frosting, defrosting, and cycling losses; and an assessment of the effect of a desuperheater water heater on system performance.

Introduction

Before about 1981, heat pump performance ratings published by the Air-Conditioning and Refrigeration Institute (ARI) and others included only steady-state performance for heating (at −8.3°C and 8.3°C) and cooling (at 35°C). While such ratings give an indication of relative efficiencies of different units, they are not accurate indicators of seasonal performance since a heat pump's seasonal performance is degraded from steady-state levels.[1-3] In the heating season, degradation is caused by decreased efficiency at low outdoor temperatures, by frosting of the outdoor coil and subsequent defrosting, by duty cycling, and by resorting to electric resistance (I^2r) elements to supplement heating capacity at low outdoor temperatures. In the cooling season, the degradation results from duty cycling and decreased efficiency at high outdoor air temperatures. The National Bureau of Standards (NBS) has developed for the Department of Energy (DOE) methods for estimating the seasonal performance of heat pumps which attempt to take into account degradation due to cycling and supplemental heating.[4-6] Since 1981, ARI ratings have included estimates of heating and cooling seasonal performance factors (SPFs).

Except for some early tests on heat pumps near Washington, D.C.,[1] and in several northern cities[7] there was little documented field data on seasonal performance of heat pumps before the time of the studies reported herein. The purpose of this paper is to report on a series of well-instrumented air-source heat pump field tests in an unoccupied residence at the Tennessee Energy Conservation in Housing (TECH) complex near Knoxville, Tennessee, from 1978 through 1983.

From 1978 through 1980 two split-system heat pumps of different efficiency levels were tested in TECH house III to measure their SPFs and compare their field performance with their steady-state ratings.[8] The higher efficiency unit was tested with a desuperheater (DS) water heater during the 1981–82 test year to determine the improvement in SPF for space conditioning and water heating over that of the same heat pump with a standard I^2r water heater.[9] From 1981 through 1983 the high-efficiency heat pump was instrumented and tested to determine the magnitude of losses due to frosting, defrosting, and cycling.[10]

A description of the heat pumps used for these tests is given in Table 1. Indoor thermostat set points for the tests were 21°C for heating and 26°C for cooling. For the space-conditioning and water heating tests, a domestic water heating usage rate of 265 L/d was imposed on the system. Both heat pumps used a time/temperature type of defrost control set on a 90-min cycle.

Data from these experiments have been used as a basis for validating a computer code for calculating air-source heat pump annual performance factors (APFs).[11] House load data from the project are being used for validation of a loads program developed by North Carolina State University.[12]

Table 1. Description of heat pumps used in field tests

	Heat pump I 1978–79	Heat pump II 1979–83
Indoor coil	4-row, plate-fin	3-row, plate-fin, "A" type
Blower capacity, L/s	570	570
Outdoor coil	1-row, spine-fin	1-row, spine-fin
ARI cooling/heating capacity, kW	8.8/9.1	9.8/10.0
Refrigerant charge, kg	3.2	4.6
Outdoor thermostat setting, °C	0	0
Strip heaters, kW	12	9
Indoor thermostat settings, °C		
Heating	21.1	21.1
Cooling	25.6	25.6

*Research sponsored by the Building Equipment Research Division, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

Summary

Two air-source heat pumps from the same manufacturer were tested at TECH house III from 1978 through 1980. ARI-rated coefficients of performance (COPs) for the heat pumps were 2.46 (at 8.3°C) for the first and 3.11 (at 8.3°C) for the second. Measured SPFs were 1.58 and 1.99, respectively, for heating and 1.64 and 2.27, respectively, for cooling. Degradation of field performance from ARI-rated steady-state performance ranged from around 10% at low outdoor temperatures (2°C) to over 40% at high temperatures (8.5°C). The high-efficiency heat pump had a somewhat greater efficiency advantage over the standard unit under field conditions than indicated by the steady-state ratings.

From November 1981 through October 1982 the high-efficiency heat pump was tested with a DS water heater and with a standard I^2r water heater on an alternating monthly basis. Heating SPFs for space and water heating were nearly equal for both systems because reduced electric water heating requirements for the DS system were counterbalanced by increased I^2r supplemental space heating requirements. Cooling SPF for the DS system was about 30% better than for the I^2r water heating system. Overall APF for space conditioning and water heating was 1.96 for the DS water heating system vs 1.72 for the I^2r water heating system for a 14% improvement.

As a follow on to the tests of 1978–80, the high-efficiency heat pump was more extensively instrumented and tested to measure the magnitudes of cycling, frosting, and defrosting losses. These tests were conducted from December 1981 through August 1983. Test results showed that defrost losses accounted for 10.1% of the heating season energy consumption, frosting losses for 3.6%, on-off cycling for 8.4%, and off-cycle parasitics for 4.3%. Definitive evaluation of cycling losses during the cooling season was made difficult by a degradation of the unit's steady-state cooling capacity and COP. Best estimates of the apparent steady-state performance indicate that cycling losses accounted for about 16% of the cooling season energy use.

Technical Accomplishments

Characterized Field Performance of Two Split-System, Air-Source Heat Pumps as Compared with Their Rated Performance

ARI ratings and on-site steady-state performance test results for test heat pumps I and II are given in Table 2. Deviations between field-measured and rated COPs are small, indicating that the equipment was properly installed and performed as expected. Table 3 gives average SPFs and operating temperatures for both heat pumps. According to ARI ratings, heat pump II is 24% more efficient than heat pump I for cooling and 22–26% more efficient for heating. The field test results indicate that heat pump II was 38% more efficient for cooling (despite operating at higher average temperatures) and 26% more efficient for heating. This is encouraging in that customers who pay a premium for high efficiency do indeed reap a benefit under field conditions.

Table 2. ARI ratings and steady-state performance test results for the test heat pumps

	Heat pump I		Heat pump II		
	On-site test	ARI	On-site test	On-site test	ARI
Heating mode					
Outdoor temperature, °C	11.1	8.3	9.7	8.0	8.3
Heat output, kW	10,856	9,085	10,892	10,328	9,965
Energy input, W	4,210	3,700	3,542	3,475	3,200
COP	2.58	2.46	3.08	2.97	3.11
Cooling mode					
Outdoor temperature, °C	31.3	35.0	35.2	26.5	35.0
Cooling capacity, W	7,500	8,795	9,379	10,424	9,818
Energy input, W	4,075	4,300	3,816	3,441	3,900
COP	1.86	2.04	2.45	3.03	2.52

Table 3. Average SPFs, temperatures, and total loads for heat pumps I and II (1978–80)

	Average outdoor temperature (°C)		House loads (GJ)	Electric energy used (kWh)	SPF
	During season	During hours of operation			
Heating					
Heat pump I	6.2	0.6	32.85	5787	1.58
Heat pump II	6.4	0.9	34.18	4765	1.99
Cooling					
Heat pump I	22.6	26.1	23.90	4051	1.64
Heat pump II	25.2	28.3	22.70	2775	2.27

Compared Space-Conditioning and Water Heating Performance of Heat Pump II with a DS Water Heater to That with an I^2r Water Heater

Average weekly COPs for space conditioning and water heating as a function of average temperature during heat pump operation are plotted in Fig. 1. For temperatures between 3 and 7°C, the DS system had a slightly better COP than did the I^2r water heating system. At higher and lower temperatures, both systems had nearly the same COP. At lower temperatures, I^2r heat saved in water heating by the DS system was negated by additional I^2r supplemental space heating requirements. This was caused by a reduction in the heat pump's space heating capacity when using a DS water heater as shown in Fig. 2. At higher temperatures, little space heating was needed, and nearly all water heating was done by I^2r elements for both systems. Thus, both system COPs approached 1.00. During the summer, the DS system weekly performance was 30–45% better than that of the I^2r water heater system except during periods of low average operating temperature (<23°C).

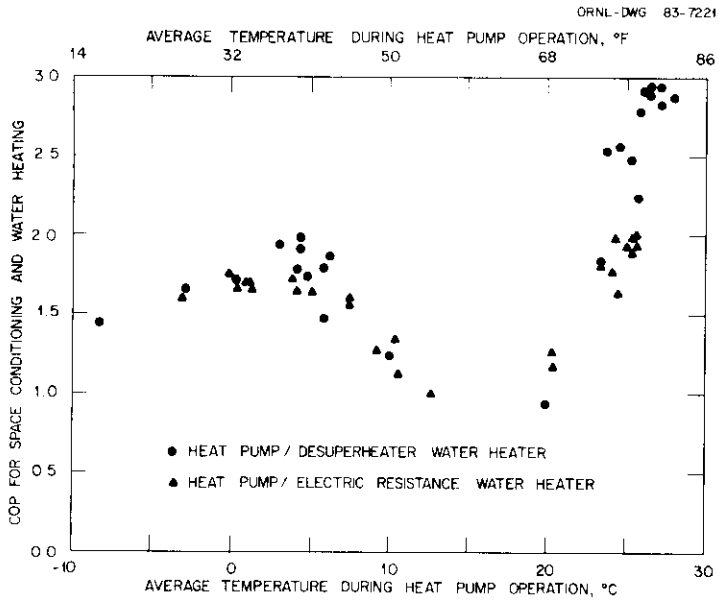


Fig. 1. Weekly average COPs for space conditioning and water heating vs weekly average outdoor temperature during heat pump operation.

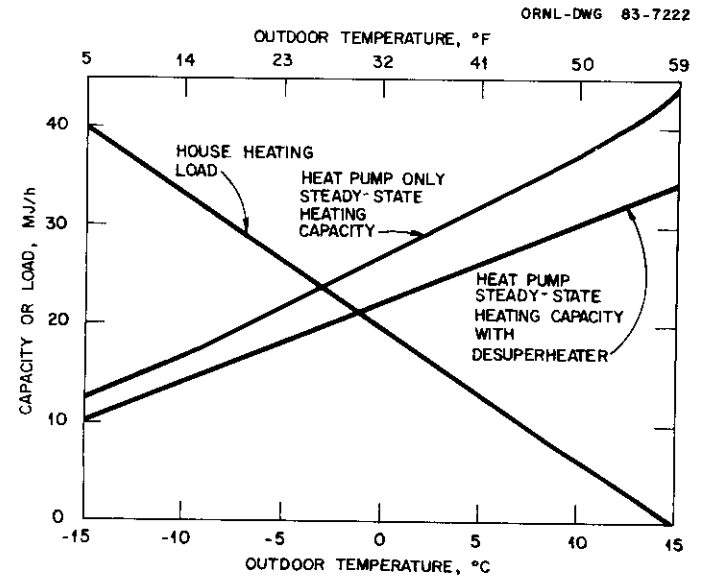


Fig. 2. TECH house III heating load and steady-state heating capacity of heat pump II with and without desuperheater vs outdoor temperature.

Table 4 contains measured SPFs and APFs for both systems. The heating SPFs for the two systems were within 3% of each other for the reasons discussed above. Cooling SPF improvement due to the DS was 30% while APF improvement amounted to 14%, principally because of the recovery of normally wasted heat during air-conditioning operation.

Table 4. Seasonal and annual performance factors (for space conditioning and water heating) for heat pump II with a desuperheater water heater vs heat pump II with an electric resistance water heater

	Electric resistance water heater system	Desuperheater water heater system
SPF, heating	1.63	1.68
SPF, cooling	1.90	2.50
APF	1.72	1.96

Quantified Field Performance Degradations of Heat Pump II due to Frosting, Defrosting, and Cycling

Table 5 gives cumulative energy consumption penalties for the various losses measured over parts of the 1981–82 and 1982–83 heating seasons. The steady-state performance used as a base was taken from laboratory measurements made by Miller on an identical heat pump.[13] Losses due to defrosting accounted for 10.1% of the total energy use, frosting losses for 3.6%, start-up transients for 8.4%, and off-cycle parasitics for 4.3%. Tempering I^2r during defrost accounted for about 60% of the defrost losses. Table 6 shows the effect of these losses on heating SPF. Excluding

Table 5. Energy consumption penalties due to dynamic losses for heat pump II: 1981–82 and 1982–83 heating seasons

Energy use due to	Energy use (kWh)	Percentage of total energy use
Off-cycle parasitics	232.0	4.3
Start-up transients	456.6	8.4
Defrosting	552.2	10.1
Frosting	198.9	3.6
Steady-state[a]	4002.8	73.6
Total energy use[b]	5442.5	100.0

[a]Energy use required to meet the load if the heat pump always operated at steady-state efficiency.
[b]Excluding supplemental I^2r heat.

Table 6. Effect of dynamic losses on heating SPF

(1) Measured SPF	1.937
(2) Measured SPF without supplemental resistance heat	1.943
(3) SPF (2) without off-cycle parasitic losses	2.030
(4) SPF (2) without start-up transient losses	2.121
(5) SPF (2) without defrost losses	2.163
(6) SPF (2) without frosting losses	2.017
(7) SPF (2) with no losses	2.642
(8) SPF (7) with supplemental resistance heat	2.627

supplemental I^2r heating requirements, there was an overall degradation in heating SPF of 26.5% (from 2.642 to 1.943). About 52% of this degradation was due to frosting/defrosting losses and the remainder to cycling losses (off-cycle parasitics and start-up transients).

Figure 3 illustrates graphically the effects of cycling, frosting, and defrosting on the COP of heat pump II. COPs plotted are taken from data for December 1981 and November 1983 and are average values for 5°C temperature bins. Data points are eliminated from the figure for clarity. At low temperatures, defrosting was the major cause of performance degradation. Cycling losses began to be noticeable at around 0°C and became predominant at higher temperatures. Frosting caused a maximum degradation from steady-state COP of about 9% at 0°C; the degradation became smaller at higher and lower temperatures.

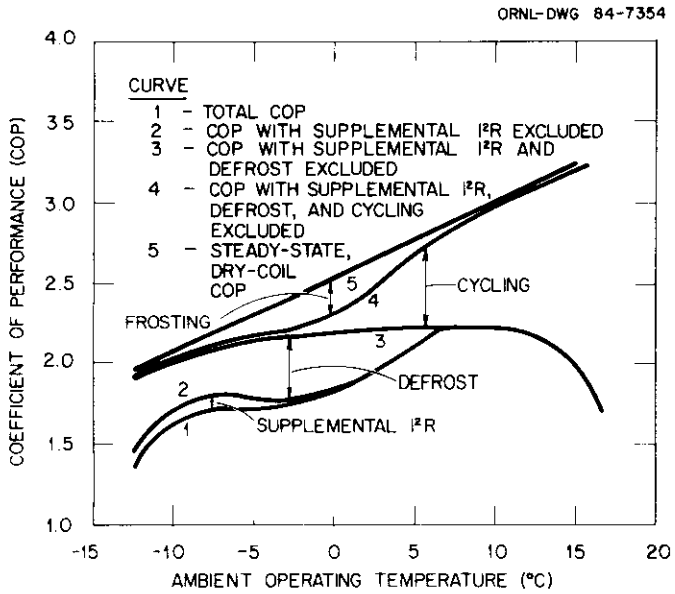


Fig. 3. Effects of cycling, frosting, and defrosting on heating COP of heat pump II.

During the 1982 and 1983 cooling seasons, heat pump II performance degradation due to cycling was measured. Because the steady-state cooling performance was degraded from the manufacturer's ratings and from 1980 performance levels (illustrated in Figs. 4 and 5), actual steady-state performance levels for 1982 and 1983 had to be deduced from the field data. The relationship of the steady-state capacity and COP thus derived to on-site performance tests made during 1982 and 1983 is shown in Fig. 6.

Table 7 gives cumulative energy consumption penalties, based on the data-derived steady-state performance, due to start-up transients and off-cycle parasitics for the 1982 and 1983 cooling seasons. Cycling losses reduced the cooling SPF by 20.2% from 2.82 (with no losses) to 2.34 (measured).

Table 8 gives the total measured heating and cooling loads and SPFs for October 1982 through September 1983 along with rated SPFs from the ARI directory.[14] These ratings are not for the same model as tested (since its production has been discontinued) but are for models from the same manufacturer with similar steady-state COP and capacity ratings as those of the test unit. For heating, the rated SPF is nearly the same as that measured.

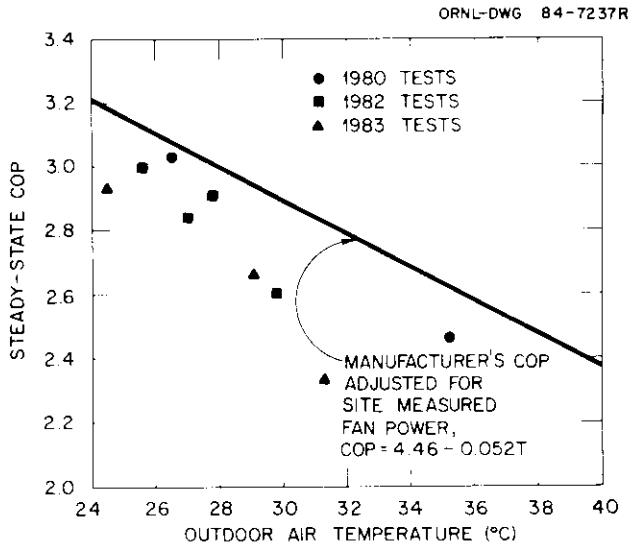


Fig. 4. Steady-state cooling COP vs outdoor air temperature.

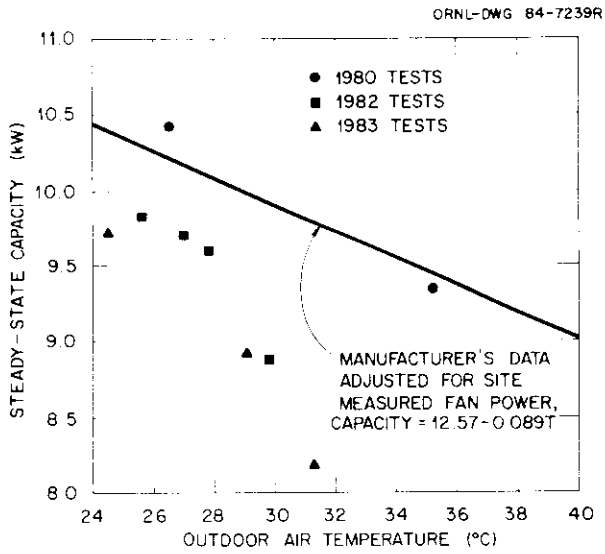


Fig. 5. Steady-state cooling capacity vs outdoor air temperature.

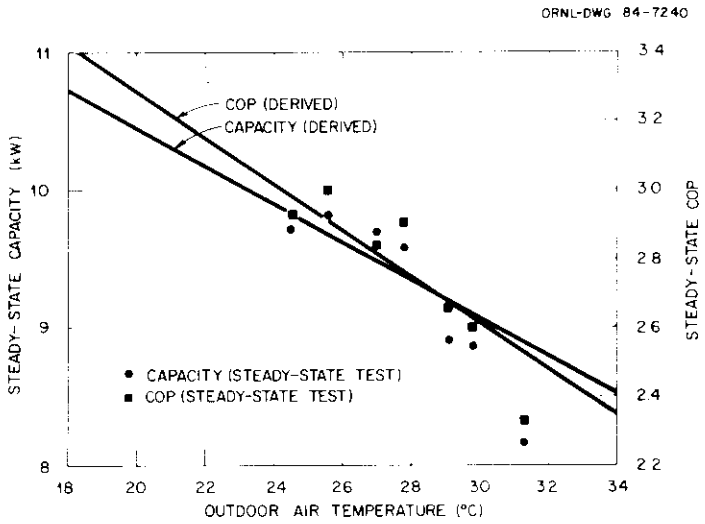


Fig. 6. Derived steady-state COP and capacity vs outdoor air temperature, with on-site test results.

Table 7. Energy consumption penalties due to cycling losses for heat pump II: 1982 and 1983 cooling seasons

Energy use due to	Energy use (kWh)	Percentage of total energy use
Off-cycle parasitics	185.0	5.1
Start-up transients	422.3	11.7
Steady-state[a]	3006.2	83.2
Total energy use	3613.5	100.0

[a]Energy use required to meet load if heat pump always ran at steady-state efficiency.

Table 8. Rated vs measured seasonal performance factors

Season	Load (GJ)	Seasonal performance factor	
		Measured	Rated[a]
Heating	35.53	1.96	1.99
Cooling	24.26	2.21	2.96

[a]For models with steady-state ratings similar to those of heat pump II.

The rated cooling SPF is 34% higher than the field-measured value. If the test unit had performed up to the manufacturer's ratings, it is estimated that the cooling SPF would have been about 2.44 or 18% lower than the rated value.

Future Work

A two-speed compressor heat pump is being installed in TECH house III at this time. This unit will be tested over a heating and cooling season to compare its performance with and without the two-speed feature. The unit will be tested in both the single- and two-speed modes on a weekly changeover schedule.

References

1. G. E. Kelly and J. Bean, *Dynamic Performance of a Residential Air-to-Air Heat Pump,* NBS BSS-93, National Bureau of Standards, U.S. Department of Commerce, Washington, D.C., March 1977.
2. W. H. Parken, Jr., et al., "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump," *ASHRAE Trans.* **83** (Pt. 1),839–49 (1977).
3. C. E. Bullock and W. R. Reedy, "Heat Pump Cyclic Performance and Its Influence on Seasonal Operation," pp. 7-1 and 7-19 in *Proceedings of the Third Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, April 10–11, 1978* Oklahoma State University, Stillwater, 1978.
4. G. E. Kelly and W. H. Parken, Jr., *Method of Testing, Rating, and Estimating the Seasonal Performance of Central Air-Conditioners and Heat Pumps Operating in the Cooling Mode,* NBSIR 77-1271, National Bureau of Standards, April 1978.
5. W. H. Parken, Jr., et al., *Method of Testing, Rating, and Estimating the Heating Seasonal Performance of Heat Pumps,* NBSIR 80-2002, National Bureau of Standards, April 1980.
6. D. A. Didion and G. E. Kelly, "New Testing and Rating Procedure for Seasonal Performance of Heat Pumps", *ASHRAE J.,* **21**(9), 40–44 (September 1979).
7. G. C. Groff, "Heat Pump Performance Improvements for Northern Climate Applications," pp. 838–46 in *Proceedings of the 13th Intersociety Energy Conversion Engineering Conference, August 20–25, 1978, San Diego, California,* vol.1, Society of Automotive Engineers, Warrendale, Penn., 1978.
8. V. D. Baxter, L. A. Abbatiello, and R. E. Minturn, "Comparison of Field Performance to Steady State Performance for Two Dealer Installed Air-to-Air Heat Pumps", *ASHRAE Trans.* **88** (Pt. 2), 941–53 (1982).
9. V. D. Baxter, "Comparison of Field Performance of a High-Efficiency Heat Pump with and without a Desuperheater Water Heater," *ASHRAE Trans.* **90** (Pt. 1B),180–90 (1984).
10. V. D. Baxter and J. C. Moyers, *Air-Source Heat Pump: Field Measurement of Cycling, Frosting, and Defrosting Losses, 1981–83,* ORNL/CON-150, Oak Ridge National Laboratory, November 1984.
11. C. K. Rice, Oak Ridge National Laboratory, Oak Ridge, Tennessee, personal communication with V. D. Baxter, Oak Ridge National Laboratory, Oak Ridge, Tennessee, December 1984.
12. F. Y. Sorrell and D. K. Hollingsworth, *Validation Project of the Performance Evaluation Program,* Interim Report, North Carolina Alternative Energy Corporation, Research Triangle Park, N.C., November 1983.
13. W. A. Miller, *Laboratory Evaluation of the Heating Capacity and Efficiency of a High-Efficiency Air-to-Air Heat Pump with Emphasis on Frosting/Defrosting Operation,* ORNL/CON-69, Oak Ridge National Laboratory, December 1982.
14. *Directory of Certified Unitary Air Conditioners, Unitary Air Source Heat Pumps, Sound Rated Outdoor Unitary Equipment, Solar Collectors, July 1–December 31, 1983,* Air-Conditioning and Refrigeration Institute, Arlington, Va., 1983.

ELECTRIC HEAT PUMP TECHNOLOGY—II

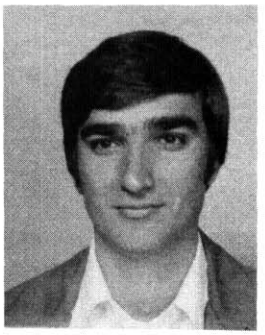
Laboratory Experiments of Heat Pump Dynamic Losses

William A. Miller

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

William A. Miller received B.S. degrees in agronomy and mechanical engineering from the University of Tennessee. He further obtained the M.S. degree in mechanical engineering while also working part-time at Oak Ridge National Laboratory. A licensed professional engineer, Miller is a member of Pi Tau Sigma honor society, the American Society of Heating, Refrigeration, and Air-Conditioning Engineers, and the Refrigeration Service Engineering Society. Much of his recent work at Oak Ridge has been concerned with experimental studies addressing dynamic losses of air-to-air heat pumps.



LABORATORY EXPERIMENTS OF HEAT PUMP DYNAMIC LOSSES*

W. A. Miller
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Objective

Air-source heat pump experiments at ORNL (1) provide detailed system and component performance data usable in understanding dynamic loss phenomena and (2) enable formulation and evaluation of methods that could reduce dynamic losses.

Introduction

Results presented herein are from a series of laboratory experiments performed at ORNL[1,2,3] aimed at providing detailed characterization of frosting losses, cycling losses, and defrosting losses as well as steady-state performance data. Continuous modulation heat pump experiments to observe dynamic loss trends under part-load heat pump operation will follow.

Summary of Work

Frosting Experiments on a One-Row, Spine-Fin Outdoor Coil

Previous experiments addressed the frosting and defrosting operation of an air-to-air heat pump having a one-row, spine-fin outdoor heat exchanger. The frosting and defrosting study was performed in bootstrap air loops constructed of glass-fiber duct board backed by aluminum. Detailed descriptions of the test facility and salient features of the heat pump are presented in ref. 1. Briefly, the heat pump has no suction line accumulator, and liquid refrigerant is throttled in the heating mode by a combination of a thermostatic expansion valve and distributor tubes.

The spine-fin frosting study encompassed an evaluation of system and component performance for ambient temperature levels of 8.3, 4.4, 1.7, −1.1, and −8.3°C (47, 40, 35, 30, and 17°F) and for discrete humidity levels ranging from 50 to 90%.

Frosting tests conducted at 1.7°C (35°F) outdoor temperature display a time-dependent degradation of coefficient of performance (COP) with increasing outdoor relative humidity (Fig. 1). The rate of performance degradation due to frost formation increases with increasing relative humidity because of the increased rate of frost accumulation on the outdoor heat exchanger. At 70% relative humidity and 1.7°C (35°F) outdoor ambient conditions, frosting caused COP and capacity reductions of 10 and 15% of steady-state performance. The observed nominal 10% reduction of COP and 15% reduction of heating capacity occurred within 60 min, at a point when the outdoor heat exchanger was nearly blocked with frost.

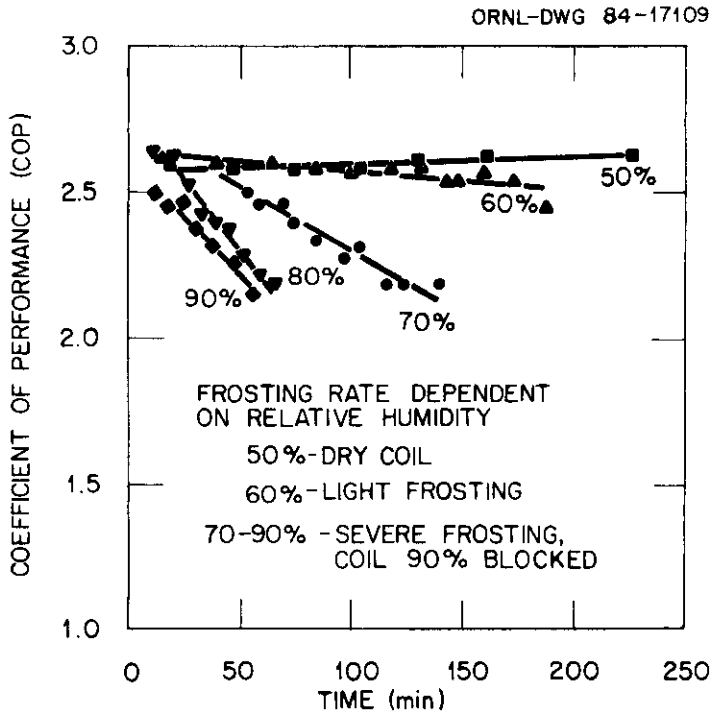


Fig. 1. Average COP observed at 1.6°C (35°F) outdoor temperature.

The heat pump having a one-row, spine-fin outdoor heat exchanger was observed to be susceptible to frosting degradations in COP and capacity; however, yearly analysis of heat pump performance (presented in the section on "Technical Accomplishments") is required for determining the magnitude and significance of frosting and defrosting losses.

Cycling Experiments Conducted on an Air-to-Air Heat Pump

The most recent laboratory work was conducted in environmental chambers having automatic temperature and

*Research sponsored by the U.S. Department of Energy under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

humidity control. Dry-coil cycling tests were performed on a split-system, air-to-air heat pump which has continuously modulating speed control capability. Both compressor and indoor blower motor speeds are variable by frequency change of three-phase input power. However, for the cycling studies both motors were supplied with a fixed 60-Hz, three-phase power source.

The heat pump refrigerant circuit is similar to conventional split-system, air-to-air, single-speed heat pump design. Liquid refrigerant is throttled by capillary tubes in both cooling and heating modes. Both indoor and outdoor heat exchangers are of tube and plate-fin configuration. A liquid-line to-suction-line heat exchanger heats the refrigerant before it enters the compressor housing, and the heat pump has a suction line accumulator to protect the compressor.

Calculations of the heat pump's steady-state COP and capacity based on air-side measurements were within 3% of those based on refrigerant-side measurements. Refrigerant flow rate in the liquid line was measured by a gyroscopic mass flowmeter. Refrigerant temperatures and pressures were measured with fast-response thermal-ribbon resistance temperature detectors (RTDs) and pressure transducers connected at strategic locations in the refrigerant circuit. Air-side indoor heat exchanger capacity was measured with a 30-junction thermopile grid (series combination of 24 AWG copper constantan wire) and a duct airflow measuring device (multipoint pitot tube averaging traverse).

Laboratory cycling work investigated the effect of on-time, off-time, and outdoor temperature on heat pump cycling transients. Cycling tests were also conducted with various control strategies so that improvements in cycling COP and capacity could be observed. Cycling tests were conducted at 10 and −1.1°C (50 and 30°F) outdoor temperatures with a moisture content that ensured dry outdoor coil heat transfer. A broad range of cycling rates were selected, including those that simulate cycling rates observed by Baxter.[4]

Cycling tests conducted at 10°C (50°F) outdoor temperature display reduction of COP as percent on-time decreases (Fig. 2). The performance curve in Fig. 2 was developed, using techniques in ref. 5, by relating house load to cycling rate through use of a thermostat model. The solid curve, representing part-load COP as a function of load and cycling rate, displays COP and capacity degradations of 10.6 and 6.6% of steady state at 90% on-time. Results at 20% on-time reveal a 34.5% reduction in capacity and a 29.6% reduction in steady-state COP. As the percent on-time decreases (decreasing on-time, increasing off-time), the amount of refrigerant in the condenser available for latent heat transfer decreases, yielding reductions in capacity and COP.

The underlying assumptions for this plot, which is comparable to the Parken and Kelly plots,[5] are that cycling performance is dominated by thermostat dynamics and is independent of outdoor temperature. Data to be presented later in this paper indicate that the cycling performance is dependent on the outdoor temperature.

Technical Accomplishments

Determined Magnitude of Frosting Losses on a Seasonal Basis

The heat pump having a one-row, spine-fin outdoor heat exchanger was observed to be susceptible to frosting degradations

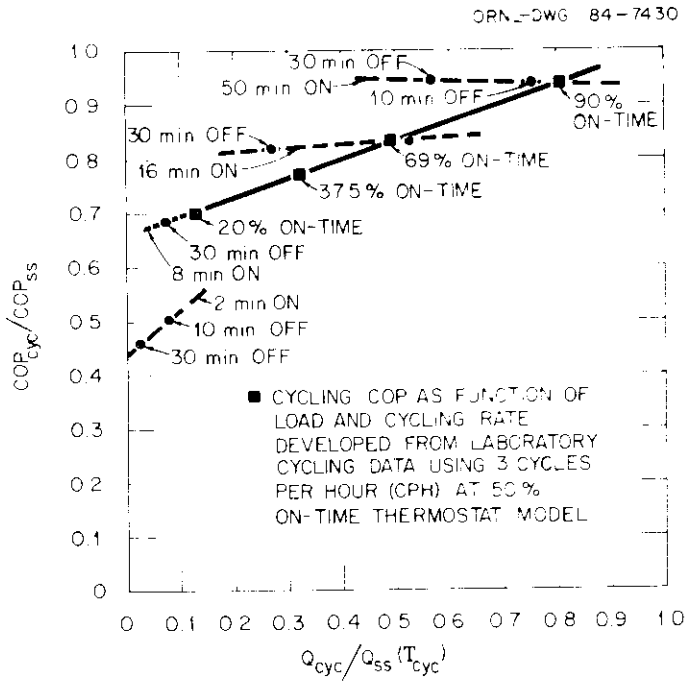


Fig. 2. Heat pump cycling performance observed at 10°C (50°F) outdoor temperature.

in COP and heating capacity at outdoor ambient temperatures between 4.4 and −8.3°C (40 and 17°F) with relative humidities greater than 70%.

Seasonal analysis was conducted with frosting loss coefficients, developed from a laboratory data base, that express the percentage loss in cumulative steady-state heat output and COP. The loss coefficients were coupled to a quadratic interpolation routine and used in a seasonal performance computer code developed by Rice, Emerson, and Fischer.[6]

Dynamic frosting operation for heat pumps with a one-row, spine-fin outdoor coil, as applied to various climatic regions in the United States (Table 1), results in a nominal 2 to 5% increase in yearly energy consumption. The heating seasonal performance factors (HSPF) for these climatic regions are also listed in Table 1. Dynamic frosting operation degraded HSPF nominally 5% of HSPF that includes losses from part-load heat pump operation.

Identified Majority of Frosting/Defrosting Degradations as due to Defrosting of the Outdoor Heat Exchanger

The frosting/defrosting losses for a heat pump having a one-row, spine-fin outdoor heat exchanger were calculated in a seasonal analysis program that incorporated loads data from the University of Tennessee Annual Cycle Energy System house, Knoxville, Tennessee. Defrosting yielded increases in yearly energy consumption three times the magnitude of energy increases due to the frosting of the outdoor coil. Yearly energy consumption was increased 2.9% by frosting, while defrosting further increased yearly energy consumption by 9.7%. These calculations are in good agreement with measured field energy consumptions observed by Baxter.[4] Baxter observed a 2.7% increase in energy consumption due to frosting and a 6.6% further energy increase

Table 1. Seasonal analysis of dynamic frosting operation for heat pumps with a one-row, spine-fin outdoor coil

Location	Annual load		HSPF[c]	Δ HSPF[d] (% FR)	Δ Energy year[e] (% FR)
	HDD[a]	CDD[b]			
Cheyenne, Wyo.	<7000	<2000	1.81	5.4	4.7
Chicago, Ill.	5500–7000	<2000	1.84	4.1	3.8
Washington, D.C.	4000–5499	<2000	1.99	5.5	4.2
Knoxville, Tenn.	<4000	<2000	1.97	4.3	2.8
Fort Worth, Tex.	<4000	<2000	1.99	4.4	1.8

[a]HDD = heating degree days.
[b]CDD = cooling degree days.
[c]HSPF accounts for cycling losses.
[d]HSPF degradation due to frosting of outdoor coil.
[e]Increase in yearly energy consumption due to frosting.

due to defrosting (a total frosting/defrosting energy increase of 9.3%).†

Percentage breakdowns of frosting/defrosting losses shown in Table 2 reveal that auxiliary heat usage during defrost operation accounts for roughly half of the losses due to the defrost operation. HSPF frosting/defrosting degradation was 16.0%, with defrosting being 11.7% of the total degradation. The increase in yearly energy consumption was 12.6%, with defrosting being 4.9% due to reverse cycle defrost and 4.8% due to auxiliary heat usage.

Table 2. Seasonal analysis of frosting and defrosting losses for Knoxville, Tennessee

Mode of operation	Δ HSPF[a] (%)	Δ Yearly energy[a] (%)
Frosting	4.3	2.87
Defrosting		
Reverse cycle	7.2	4.9
Auxiliary heat	4.5	4.78
Frost/defrost	16.0	12.55

[a]Percentage calculated using performance base that accounts for energy losses due to cycling operation.

†Increase in energy consumption based on energy consumption that includes cycling.

‡Heat pump cycling would not be expected at −1.1°C (30°F) outdoor temperature, which is below house balance point. However, a broad temperature range was selected to observe the effect on cycling transients.

**Time required to reach equilibrium observed by allowing system to cycle on to equilibrium after three previous 8-min on/30-min off cycles.

Experimentally Observed that Ratio of Cycling COP to Steady-State COP Is Dependent on Outdoor Ambient Temperature

The COP and heating capacity of a heat pump are directly affected by outdoor heat exchanger capacity, which in turn is directly affected by outdoor temperature. Steady-state and dynamic performance decreases as outdoor temperature decreases; however, the ratio of part-load COP to steady-state COP was observed to increase with decreasing outdoor temperature, as shown in Fig. 3. The inverse relationship of cycling COP ratio and outdoor temperature is the result of refrigerant charge distribution as affected by temperature and is also the result of off-cycle refrigerant migration.

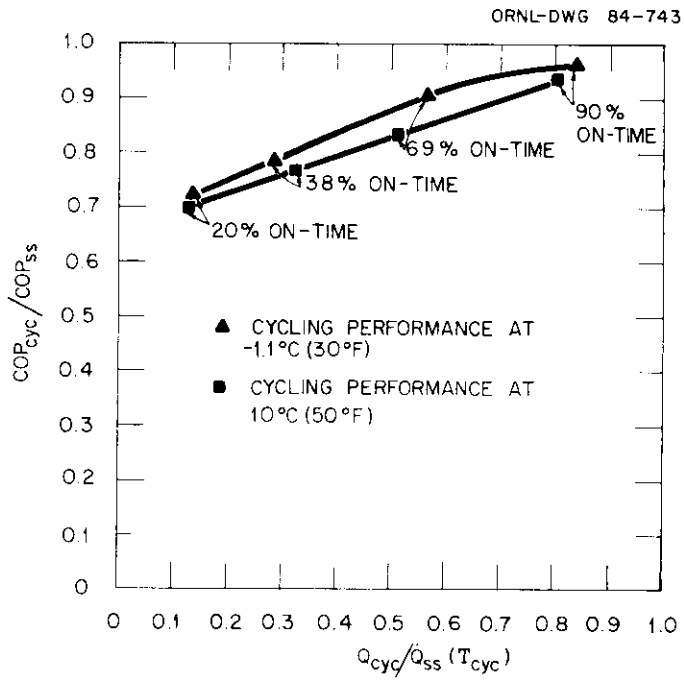


Fig. 3. Effect of outdoor temperature on heat pump cycling dynamics.

The inverse relationship of cycling COP ratio and outdoor temperature was analyzed with tests conducted at 10 and −1.1°C (50 and 30°F)‡ outdoor temperature having identical cycling rates of 8-min on/30-min off (21% on-time). The capacity loss per cycle for the −1.1°C (30°F) outdoor temperature cycling test was only 60% of the capacity loss per cycle observed for the 10°C (50°F) outdoor temperature cycling test. Refrigerant weight, measured in the outdoor unit, vs compressor run time in Fig. 4 shows that 3.4 kg (7.4 lb) of refrigerant was pumped to the indoor coil and liquid line during the 8-min on period of the 10°C (50°F), 21% on-time cycling test. At −1.1°C (30°F) outdoor temperature, 2.3 kg (5 lb) of refrigerant was pumped from the outdoor unit during the 8-min on period. The 1.1-kg (2.4-lb) differential of refrigerant movement as a function of outdoor temperature yields a greater time lag in achieving indoor heat exchanger steady-state condensing temperature at the higher outdoor ambient temperature of 10°C (50°F). The increased refrigerant mass within the indoor coil caused a time lag of 12 min** in

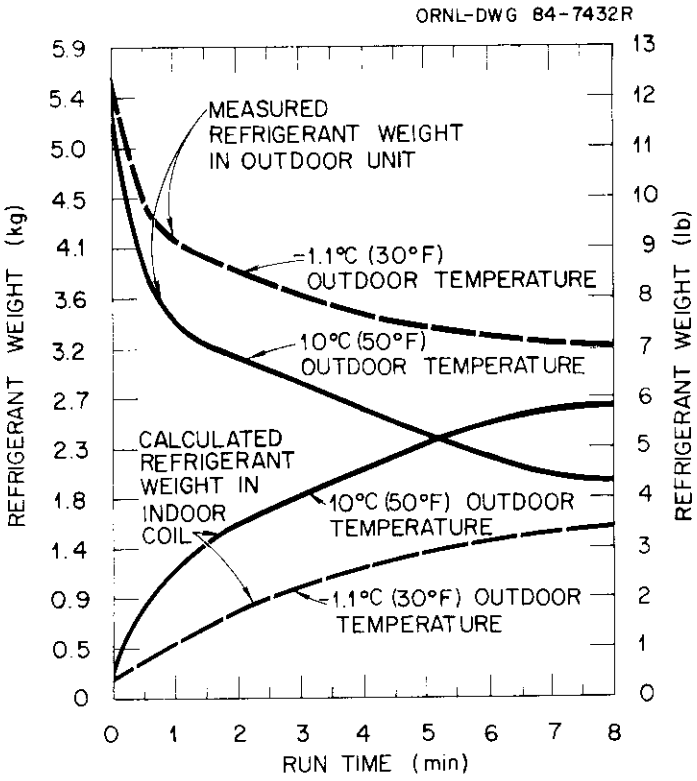


Fig. 4. Refrigerant charge distribution as affected by outdoor temperature for 8-min on/30-min off cycle.

establishing near steady-state condensing and evaporating temperature and pressure for the 10°C (50°F), 21% on-time cycling test. However, only 6 min was required to establish quasi-steady-state condensing and evaporating conditions for the −1.1°C (30°F) cycling test.

Calculations of refrigerant charge distribution showed the indoor heat exchanger to contain 48% of total charge during steady-state operation at 10°C (50°F) outdoor temperature. At the lower outdoor temperature of −1.1°C (30°F), the indoor unit held 29% of total charge during steady-state operation. The reduction of outdoor temperature decreases both refrigerant-to-air temperature difference within the outdoor coil and evaporating pressure. These reductions in turn decrease mass flow rate pumped by the constant-volume reciprocating compressor, resulting in an increase of refrigerant stored in the accumulator. The reduction of suction line refrigerant density and the associated reduction in "steady-flow" refrigerant inventory as outdoor temperature drops decreases the severity of start-up refrigerant dynamics, resulting in less severe cycling losses. As outdoor temperature increases, refrigerant density at the compressor inlet increases, and refrigerant flooding the compressor at start-up will yield considerable delays in achieving steady-state performance.

Improved Heat Pump Cycling Performance

A series of heating mode cycling tests were conducted at 10°C (50°F) outdoor temperature to observe possible system improvements with the following control strategies:

1. continuous indoor blower operation with compressor and outdoor fan cycling on/off;
2. compressor, indoor blower, and outdoor fan cycling on/off (normal cycling operation);
3. indoor blower operation extended into off cycle with compressor and outdoor fan cycling on/off;
4. refrigerant isolated in indoor coil during off cycle with system operating in normal cycling mode; and
5. refrigerant isolated in indoor coil coupled with extended indoor blower operation during off cycle.

The best cycling performance, displayed in Fig. 5, resulted from control of off-cycle refrigerant migration coupled with 2 min of extended indoor blower operation. At 68% on-time, COP and capacity degradations observed for off-cycle refrigerant flow control and extended blower operation were 3.9 and 9.6% of steady state as compared with normal-mode cycling COP and capacity degradations of 17 and 28%. At 68% on-time, controlling off-cycle refrigerant migration also resulted in performance improvements above that observed with extended indoor blower operation. However, at 20% on-time, cycling performance for off-cycle refrigerant migration control drops below the performance observed with 2 min of extended indoor blower operation. Apparently the heat regained during the 2 min of extended indoor blower operation compensates for cycling losses incurred by refrigerant flooding the compressor during the start of an on cycle.

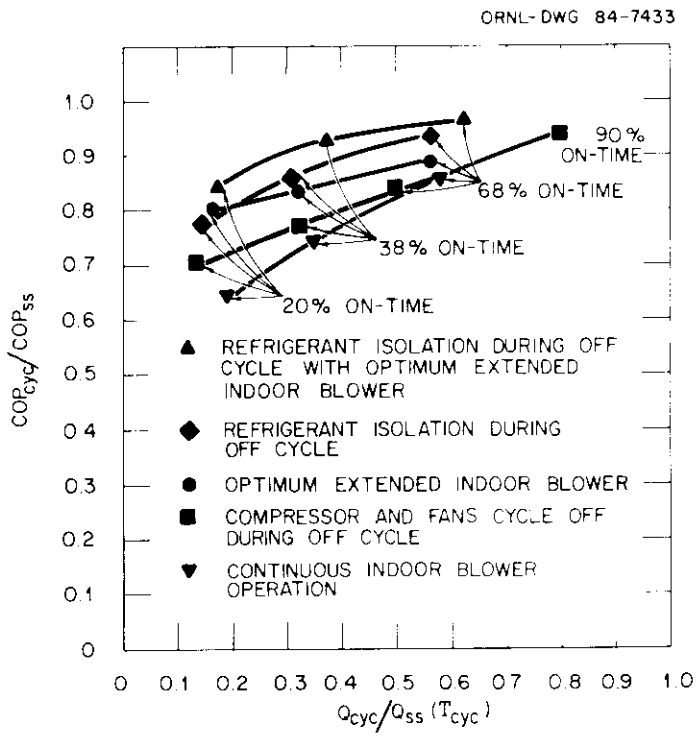


Fig. 5. Cycling performance comparison using various control strategies for tests conducted at 10°C (50°F) outdoor temperature.

Continuous operation of the indoor blower increased cycling degradations as compared with normal cycling mode operation for on-times less than 68%. At 20% on-time, heat delivered during the off cycle resulted in only a 21% degradation in steady-state capacity (34% for normal mode); however, a 36% degradation in COP (16.9% for normal mode) was attributed to increased power consumption during the off-cycle.

Showed by Analysis That Refrigerant Migration Is Predominant Cycling Loss

A breakdown of losses per cycle for cycling tests conducted at 10°C (50°F) outdoor temperature with an 8-min on/30-min off cycling rate is presented in Table 3 for two of the previously discussed control strategies.

Refrigerant migration coupled with start-up refrigerant floodback to the compressor, identified as refrigerant dynamics in Table 3, accounted for a 19.3% capacity loss in steady-state heat output under normal-mode cycling operation. Refrigerant migration control during the off cycle reduced this loss to 5.1% of steady-state heat output. The migration of roughly 3.2 kg (7 lb) of refrigerant during the off-cycle of normal-mode cycling operation caused severe compressor transients, as seen by measurement of compressor housing temperature. Housing temperature measured at compressor bottom showed a 9°C (17°F) temperature drop within 1 min after system start-up for normal-mode cycling operation. The severe temperature drop implies refrigerant flooding and boiling within the compressor during start-up. These compressor transients, as caused by start-up refrigerant dynamics, cause time delays in achieving steady-state capacity output and result in the majority of the cycling loss. No drop in the bottom of the compressor housing temperature was observed during start-up for the refrigerant migration control cycling test presented in Table 3, and as a result capacity output increased more rapidly to steady-state level than did capacity output observed for normal-mode operation.

Internal energy losses from the indoor heat exchanger (ΔU_{HX}) increased slightly with off-cycle refrigerant isolation simply because hot refrigerant was isolated in the indoor coil during the off cycle. However, the net benefit of the off-cycle isolation of refrigerant in the indoor coil is still substantial, resulting in an 11.3% net improvement in cycling loss magnitude. The internal energy loss (ΔU_{HX}) can be further utilized by extending the indoor blower operation as previously discussed.

Analysis for the listed control strategies in Table 3 reveals the effect of refrigerant migration on system cycling performance. Minimizing migration via refrigerant isolation during the off cycle yielded the least losses in achieving steady-state operation; thus, refrigerant migration is the predominant mechanism causing the cycling loss. These cycling test results will be covered in more detail in an ASHRAE paper that is being reviewed for presentation and publication in 1985.

Future Activities

Presently the laboratory facility is being modified for continuous modulation studies. Motor-generator sets will supply power to the compressor motor and indoor blower motor. The primary objectives of the study are as follows:

1. determine efficiency of variable-capacity systems at different speeds and ambients,
2. identify effect of variable capacity on cycling losses,
3. typify frosting/defrosting losses for modulating systems, and
4. verify improvements in seasonal performance through use of continuously modulating systems.

References

1. W. A. Miller, *Laboratory Evaluation of the Heating Capacity and Efficiency of a High-Efficiency, Air-to-Air Heat Pump with Emphasis on Frosting/Defrosting Operation,* ORNL/CON-69, Oak Ridge National Laboratory, December 1982.
2. W. A. Miller, "Frosting Experiments for a Heat Pump Having a One-Row Spine-Fin Outdoor Coil," *ASHRAE Trans.* **90**(1) (1984).
3. W. A. Miller, unpublished data, December 1984.

Table 3. Loss analysis for 8 min on/30 min off cycling rate observed at 10°C (50°F) outdoor temperature

	$Q_{LOSS/CYC}$[a]		ΔU_{HX}		$Q_{REF.DYN}$[d]	Total
	Wh	Btu	Q_{MASS}[b] [% of $(Q_{ss})(T_{on})$]	Q_{REF}[c] [% of $(Q_{ss})(T_{on})$]	[% of $(Q_{ss})(T_{on})$]	
Compressor and fans cycle on/off	367	1254	5.5	4.05	19.3	28.9
Refrigerant isolated[e] in indoor heat exchanger during off cycle	237	809.5	6.3	6.3	5.1	17.6

[a] Loss calculated over 8-min on period based on air-side measurements.
[b] Thermal capacity heat loss of indoor heat exchanger metal is the product of heat exchanger metal thermal mass and coil temperature gain during on cycle.
[c] Indoor heat exchanger refrigerant energy loss during off cycle using first-law analysis.
[d] Refrigerant heat loss due to refrigerant migration and its effect on system start-up due to compressor floodback.
[e] Isolation valves in liquid and vapor lines leaked 0.8 kg (1.8 lb) of refrigerant during the 30-min off cycle.

4. V. D. Baxter and J. C. Moyers, *Air Source Heat Pump: Field Measurement of Cycling, Frosting, and Defrosting Losses, 1981–1983,* ORNL/CON-150, Oak Ridge National Laboratory, December 1984.

5. George E. Kelly and Walter H. Parken, "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump," *ASHRAE Trans.* **583**(1), 839–49 (1977).

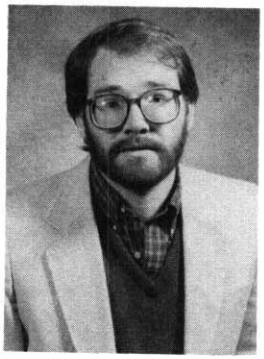
6. C. K. Rice, C. J. Emerson, and S. K. Fischer, Oak Ridge National Laboratory, Oak Ridge, Tenn., personal communication to W. A. Miller, December 1984.

A Comparative Analysis of Single- and Continuously Variable-Capacity Heat Pump Concepts

C. Keith Rice and Steven K. Fischer

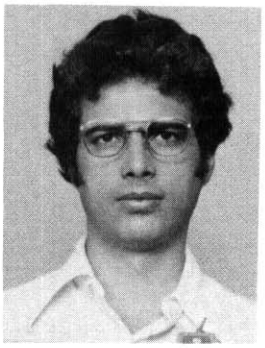
Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831



C. Keith Rice received B.S. (physics, 1974) M.S. (nuclear engineering, 1976), and Ph.D. (mechanical engineering, 1978) degrees from Virginia Polytechnic Institute and State University. His master's and doctoral research specializations were in the areas of pulsed-reflected reactor and active solar collector analysis, respectively.

Dr. Rice has been a research engineer at Oak Ridge National Laboratory for 7 years. His primary responsibilities are in the Building Equipment Research Section of the Energy Division. He has had a leading role in the development of the Oak Ridge heat pump models and has been equally active in the application of these models to system design optimization.



Steven K. Fischer received B.S. and M.S. degrees in mathematics from the College of William and Mary (1971) and the University of Kentucky (1974), respectively. He has worked at Oak Ridge National Laboratory since 1974 in the Computer Sciences and Energy divisions. His work has included data analysis for the fusion energy research program and the simulation of the dispersion of chemical effluents from power plants in rivers and estuaries. Most recently, he has been involved in the modeling of steady-state and seasonal performance of heat pumps and the optimization of heat pump design.

A COMPARATIVE ANALYSIS OF SINGLE- AND CONTINUOUSLY VARIABLE-CAPACITY HEAT PUMP CONCEPTS*

C. K. Rice and S. K. Fischer
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Objective

This work is an initial assessment of the potential benefits of continuous-capacity-modulation in electric-driven, air-to-air heat pumps for residential application. The purpose of the project was to provide a quantitative estimate of the possible annual performance gains of advanced continuously modulating heat pumps relative to single-speed designs at comparable levels of development.

Background

Previous analytical design work in this area at ORNL dealt with single-design-point, heating-mode optimization of single-speed heat pumps.[1] For that work the ORNL Steady State Heat Pump Design Model[2] was connected to a constrained numerical optimization code.[3]

The present work represents an extension of the earlier work in two directions. First, seasonal (heating and cooling) and annual performance factor (APF) analysis capability was added to allow direct evaluation of annual energy use from heat pump performance data generated by the ORNL heat pump model. Secondly, a modulating version of the heat pump model was developed to provide a means for simulating the steady state performance of continuously variable-speed (CVS) systems. With these tools, the APFs of both single- and continuously variable-capacity (CVC) concepts could be studied as basic heat pump design variables were varied.

Approach

These ORNL steady state heat pump models and the APF model were used with numerical optimization routines to find design configurations of currently marketed heat pump components that give best annual performance for some typical house loads. In this way, the annual performance of single-capacity and CVC heat pump concepts were each optimized (maximized) to evaluate the energy-saving potential of likely competing candidates of the two types.

The use of an optimization program ensured that the maximum potential of each operating concept (given certain assumptions)† would be found that was consistent with the applied constraints. For this analysis, engineering constraints reflecting current practice were applied to:

- total heat exchanger size,
- compressor and fan performance, and
- design cooling capacity.

The first two types of constraints were imposed directly within the program, while the cooling capacity constraints were externally maintained by the optimizer. The design variables that were optimized for the best performance subject to the constraints included heat exchanger frontal area, tube rows, circuits, and air flow rates; compressor displacement; and condenser subcooling.

Assumptions

Overview Of Cases

Four primary cases were considered to provide values for comparison of the efficiency and energy-savings potential of optimized CVS vs single-speed heat pumps. These cases were:

1. single-speed systems with regular‡ cycling and frosting/defrosting losses,[4]

*Research sponsored by the Building Equipment Research Division, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

†E.g., it was assumed that speed modulation had the highest efficiency potential for the continuous-capacity-modulation concept.

‡The cycling loss factors assumed[4] were larger than the alternative DOE default C_D factors[5] of 0.25 (ranging in equivalent values from 0.25 to 0.35).

2. single-speed systems with reduced** cycling losses (through reduced loss factors),

3. CVS systems with advanced inverter-motor combinations, regular‡ cycling losses, and estimated frosting/defrosting losses, and

4. CVS systems with advanced inverter-motor combinations and reduced** cycling losses (through reduced loss factors).

Case 4 was not reoptimized, but was obtained from case 3 by simply applying the different loss assumptions to that optimized configuration. The effect of this shortcut was estimated to be minimal, based on the results of case 1 and 2 optimizations.

Care was taken to make consistent assumptions regarding component performance between the two operating concepts. The only component efficiency advantages allowed the CVS systems were (1) those due to inherent flexibility and (2) those resulting in significant performance gains unique to that concept. These allowed advantages were:

1. use of three-phase compressor and indoor fan motors for the CVS systems because the inverters convert residential service to three-phase power as an added benefit, and

2. use of advanced synchronous, permanent-magnet, electronically commutated, motor-inverter combinations (PM-ECMs) for the CVS systems because they have unique application advantages over conventional induction motor, inverter-driven systems—by eliminating compressor motor slip losses (which increase significantly at lower speeds), by reducing motor harmonic losses, and by better maintaining fan efficiency at reduced fan speeds.

Speed-Change Ranges

For the CVS systems, we assumed the speed ranges used in the York ENMOD CVS heat pump,[6] which give a continuous modulation of:

- the compressor over the speed ranges of

  6 to 1 in heating mode (15 to 90 Hz) and
  4 to 1 in cooling mode (15 to 60 Hz), and

- the indoor fan over a speed range of

  2.5 to 1 in either mode (24 to 60 Hz).

The outdoor fan speed was held fixed based on earlier unpublished work which indicated that any gains resulting from modulation of this fan were minimal.

Inverter-Driven Induction Motors Vs PM-ECMs

During the course of the study, we concluded that the present commercially available inverter-motor combinations do not provide the technology to closely approach the potential of continuous-capacity modulation. This is because conventional three-phase induction motors have:

- motor slip losses that increase substantially (reducing the motor efficiency from about 4 to 16 percentage points) as the operating frequency is reduced (from 60 to 15 Hz) under constant-torque conditions typical of compressor applications, and

- motor losses due to choppy inverter waveforms that have been measured at 2 to 5 percentage points (depending on frequency and inverter type).[7]

These losses can be greatly reduced though the use of a PM-ECM because:

- all motor slip losses are eliminated because slip is not used to generate torque—elimination of the rotor and other losses associated with slip results in a 4 to 5 percentage point motor efficiency boost at rated torque at all speeds, and

- losses due to the choppy (nonsinusoidal) inverter waveforms are significantly reduced (estimated at 1.5 percentage points) with the use of PM motors.

Because we were interested primarily in the potential of the CVC concept and it appears highly probable that the cumulative efficiency gain of the PM motors would more than offset any reasonable cost premium, we assumed PM-ECMs in the CVS systems.

The inverter performance was modeled as that of an efficient, present-day inverter[8] as shown in Fig. 1 with a peak efficiency of 95% at rated power and with a reduction of performance at reduced output power.

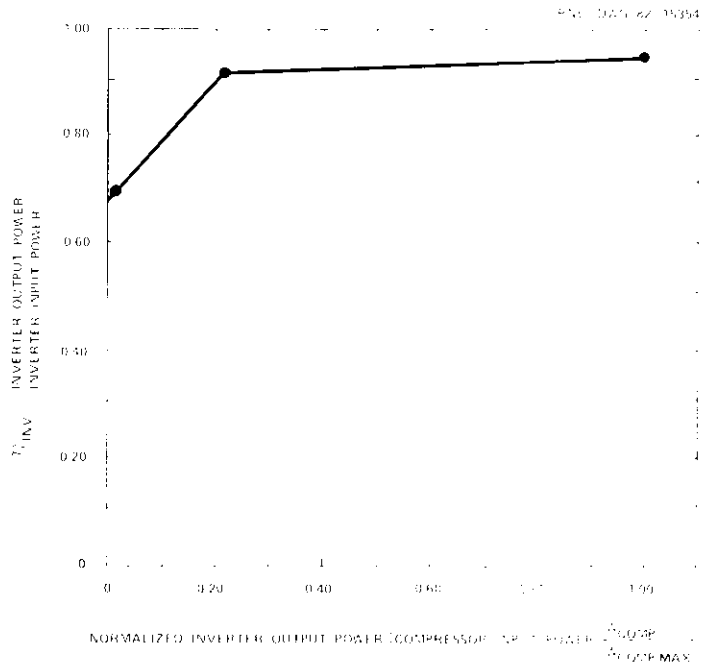


Fig. 1. Assumed frequency-inverter efficiency.

Compression Efficiency Change With Speed

A residential-size, high-efficiency reciprocating compressor was assumed for the analysis. We also included in the CVS compressor

**C_D factors of 0.05 were assumed in both heating and cooling modes.

model limited data[8] which indicated that, at the lower pressure ratios which would exist across a compressor at lower speed operation, the basic compressor efficiency (exclusive of the motor) could increase by as much as 15%. This is presumably due to a reduction in flow losses. Note that other types of modulating compressors may not provide this same net benefit at lower capacity operation (e.g., stroke-change schemes or rotary compressors). No data were available for the overspeed operation range (60 to 90 Hz); in this region 60 Hz compressor efficiencies were assumed.

Results

The design configurations for the four systems were computer optimized, using these assumptions, in order to maximize the APF of each system. Comparative results for the optimum configurations and system performance are summarized as follows:

Optimum Configurations

The optimized single-speed configurations with high and low cycling loss factors were nearly identical, suggesting that cycling-loss reduction techniques could be retrofitted to existing designs with little loss in performance potential.

Optimum heat pump configurations for the single- vs variable-speed concepts showed the following notable configuration changes for the variable-speed case:

- a 20% larger compressor displacement that, with overspeed operation, reduced the heating balance point from 25°F to 11°F with the unit still capable of precisely tracking the cooling load down to 82°F and the heating load up to 42°F;
- a 25% smaller outdoor air flow rate;
- lower indoor air flow rates in the cooling mode; and
- half as many refrigerant circuits in the indoor coil.

The first result indicates that, even with overspeed capability, the variable-speed systems can use a larger compressor to good advantage in a moderate climate (60% heating / 40% cooling loads) to minimize backup heat use. Apparently, the unit can slow down sufficiently to meet the lighter loads at milder ambients without significant cycling loss penalties.

In Fig.2, the load matching characteristics and balance points for optimized cases of the two operating concepts (case 1 and case 3) are shown with frosting/defrosting effects included. This figure shows how the CVS system dramatically improves the matching between heat pump capacity and house load requirements over the range of heating and cooling mode ambients.

Figure 3 shows the optimum speed variations vs ambient for the indoor fan and compressor over the range of heating and cooling loads shown in Fig. 2. Superimposed in Fig. 3 are the normalized annual heating and cooling energy outputs required of the heat pumps in each ambient temperature bin—outputs which are the product of the load line values in Fig. 2 multiplied by the hours of temperature occurrence in each bin. Comparison of the required output distribution against the modulation range shows that the unit modulation range is shifted off center of the output

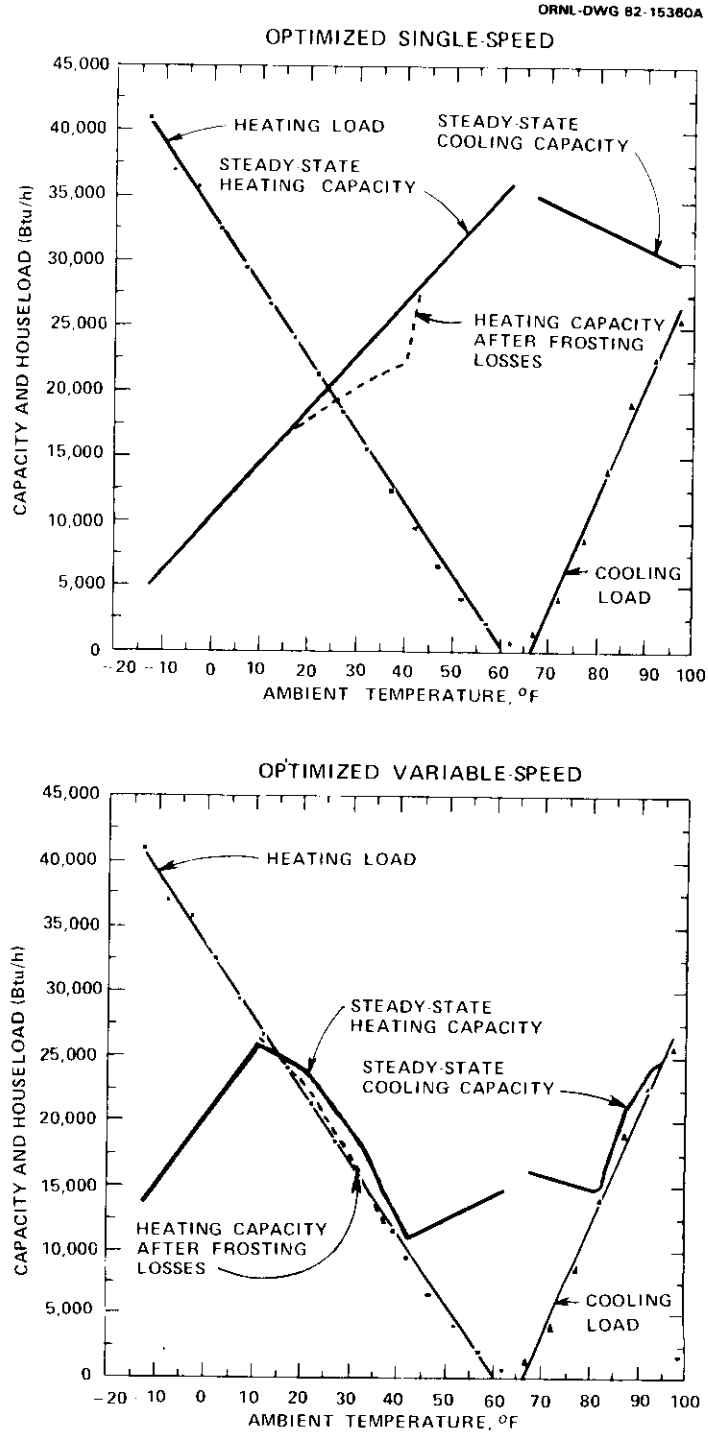


Fig. 2. Load matching characteristics of optimized single- vs variable-speed heat pumps.

distribution to obtain the benefits of reduced backup resistance heat use at the colder ambients. This was found to maximize the annual performance even though it results in more cyclic operation at the milder ambients in both heating and cooling modes.

Maintenance of Comfort Conditions

With regard to comfort criteria, the optimized single-speed cases had better than acceptable indoor-air delivery temperatures (a minimum of 89°F at the single-speed balance point) and indoor air dehumidification rates (sensible-to-total cooling capacity ratios

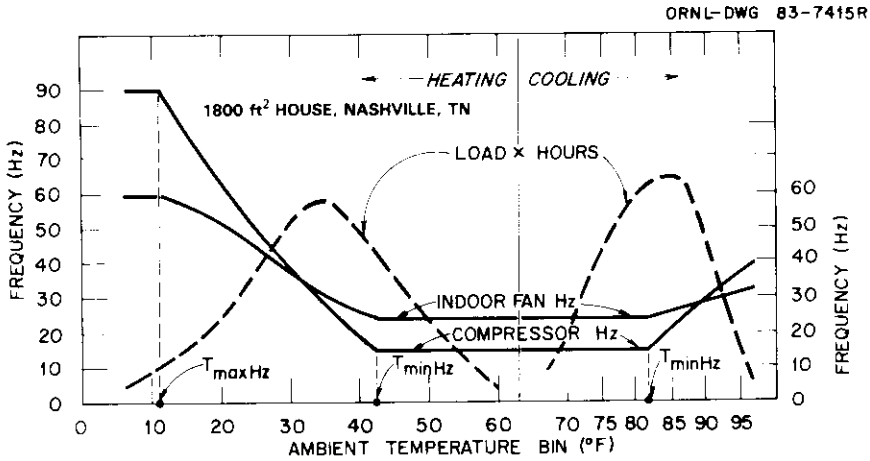


Fig. 3. Optimal inverter frequency vs load-hour distribution and ambient temperature.

of 0.69 to 0.71). These results call into question the notion that high-efficiency, single-speed heat pumps must sacrifice comfort.

The variable-speed cases had dehumidification rates that were low by conventional measures (sensible-to-total ratios of 0.72 to 0.82) but that we judged acceptable for the chosen house and location because of the nearly continuous operation of the unit. Continuous, or nearly continuous operation, should:

- achieve the actual steady-state, sensible-to-total cooling ratio instead of a higher net value that would be expected with a cycling single-speed unit;
- allow a higher tolerable humidity level because air is in continual circulation and temperature and humidity swings are minimized; and
- minimize re-evaporation of water from the coil because of cyclic operations—especially with continuous fan operation.

Delivery air temperatures for the optimized CVS configuration were, however, too low in the heating mode (remaining between 85 to 86°F for all ambients). This indicates the need for an additional comfort constraint in future analyses. Preliminary analysis indicates that such a constraint will significantly reduce the maximum indoor flow rate in heating mode but will not have that large an effect on the APF values.

Comparison of Maximum Performance Predictions

All the performance and energy savings values which follow were predicted for an 1800-ft² house with HUD minimum levels of insulation in Nashville, Tennessee. The first sets of comparisons will be made between cases 1 and 3—single- vs variable-speed systems with regular dynamic losses—to present the general performance differences between the two basic concepts. Following this, cross-comparisons between all four cases will be made.

The net coefficients of performance (COPs) vs ambient for cases 1 and 3 are illustrated in Fig. 4. The net COPs include the effects of backup resistance heat, frosting/defrosting, and cycling. Significant performance differences are evident throughout the heating and cooling ambients, with the largest percentage differences occurring at the lower ambients in the cooling mode. In

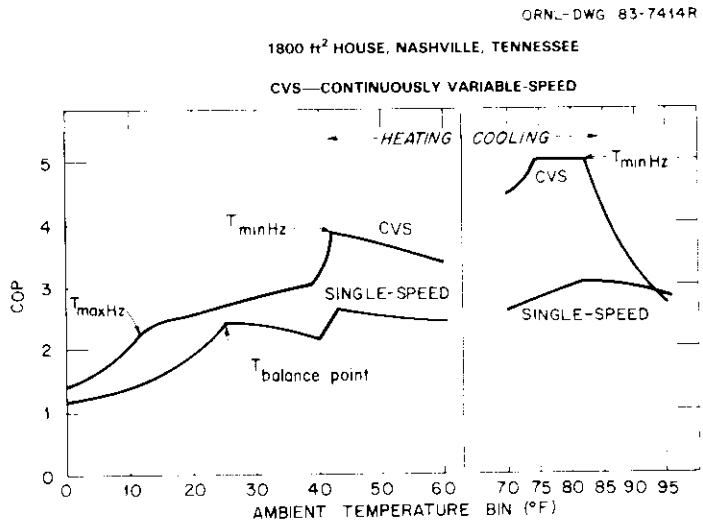


Fig. 4. Net COP comparison between single-speed and CVS systems—regular dynamic losses.

heating mode, only near the single-speed balance point do the two approach each other. Here both units are supplying the same instantaneous output and there are no heat exchanger unloading nor resistance heat displacement advantages for the CVS case; the slight COP advantage remaining in the absence of these benefits is because of the higher efficiency three-phase PM motors which could be used (and possibly also because of the slightly lower compressor speed with the larger displacement). In cooling, the CVS case has much better performance except near the cooling design point, where apparently a poorer performance configuration was acceptable to gain the higher performance at the milder ambients.

The resultant effect of these COP differences on the energy use between the two systems is given in Fig. 5. Here the distance between the two curves at each ambient gives the energy savings per 1F° temperature bin from the CVS system performance advantage. In the heating mode, the absolute savings are greatest around the peaks of the required system output curves of Fig. 3 and next largest at the milder ambients in both heating and cooling. Only at the higher ambients in cooling are there no savings with the modulating systems.

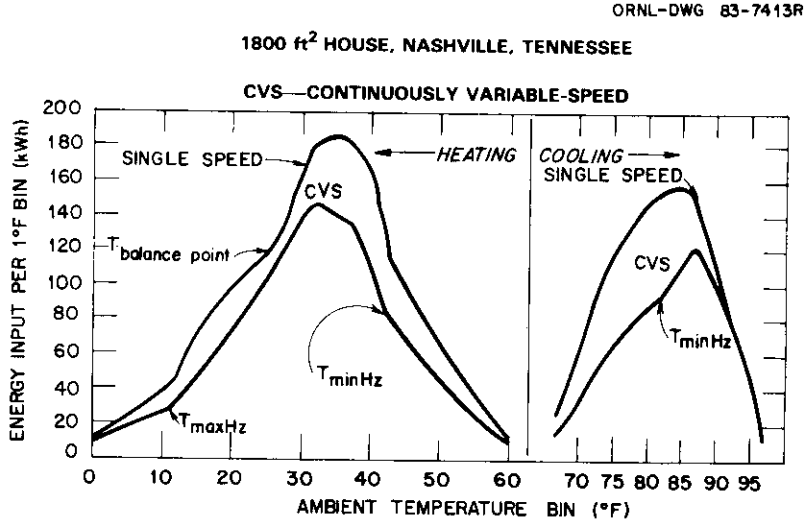


Fig. 5. Net COP comparison between single-speed and CVS systems—regular dynamic losses.

The summation of the differences between the curves in Fig. 5 gives the net annual energy savings of the modulating system. This difference is shown in Fig. 6 as a portion of the total energy use for the nonmodulating case, which has been subdivided into various steady state and dynamic loss components. The energy use (26.7% less) and breakdown for the CVS case is given in the remaining portion of the pie. The savings are shown as subdivided into four categories. These are as follows:

- reduced cycling losses because of better load following (reduced 68% from the nonmodulating case),
- heat exchanger unloading because of lower speed operation (reduced compressor and fan consumption by 11%),
- reduced backup heat use because of compressor oversizing and overspeed operation (reduced by 82%), and
- reduced frosting/defrosting losses (estimated) resulting from heat exchanger unloading (reduced by 51%).

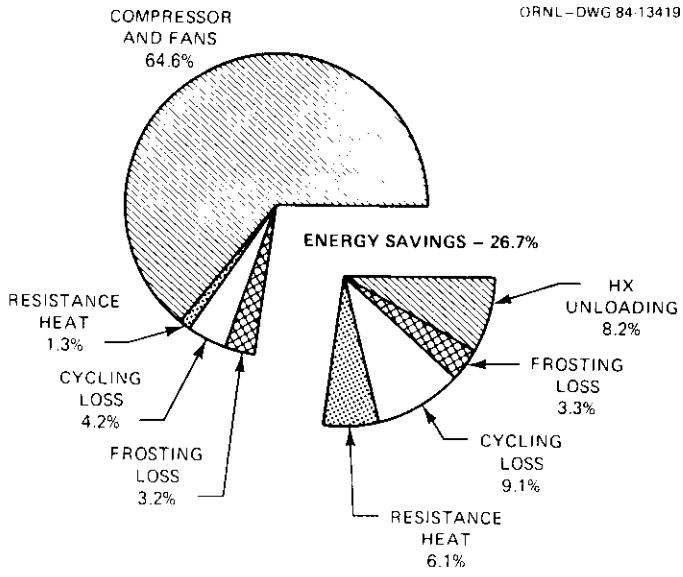


Fig. 6. Energy use and savings for a continuously modulating heat pump.

The relative contributions of the four factors to the 26.7% total energy savings are indicated in Fig. 7. While reduced cycling losses are the largest single percentage of savings, they represent only one-third of the total savings, with primary and secondary benefits of heat exchanger unloading (amounting to 43.3%) and reduced backup heat use (saving 22.8%) providing the remaining two-thirds.

This completes the more detailed comparative analysis between the nonmodulating and modulating systems with regular dynamic losses. The comparative results between all four cases are summarized in Figs. 8 and 9 where the potential for energy savings and efficiency improvement by each path is given. These results are again for the house and location with a 60/40 split between heating and cooling loads.

In Fig. 8, the 27% (26.7%) energy savings of the CVS system with relatively large cyclic loss factors (case 3) are compared to an 11% savings for the single-speed system with cycling losses decreased by assuming greatly reduced cyclic loss factors (case 2). When continuous modulation is combined with reduced cyclic loss coefficients (case 4), the net effect is a 30% savings over the optimized base case (case 1). The individual savings (from cases

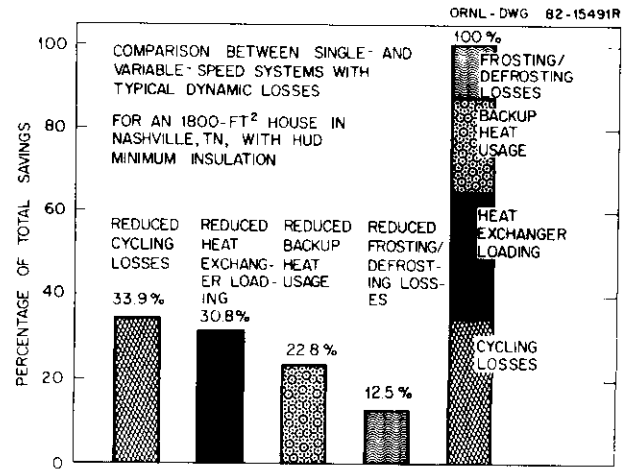


Fig. 7. Comparison of relative contributions to the total energy savings with a CVS system.

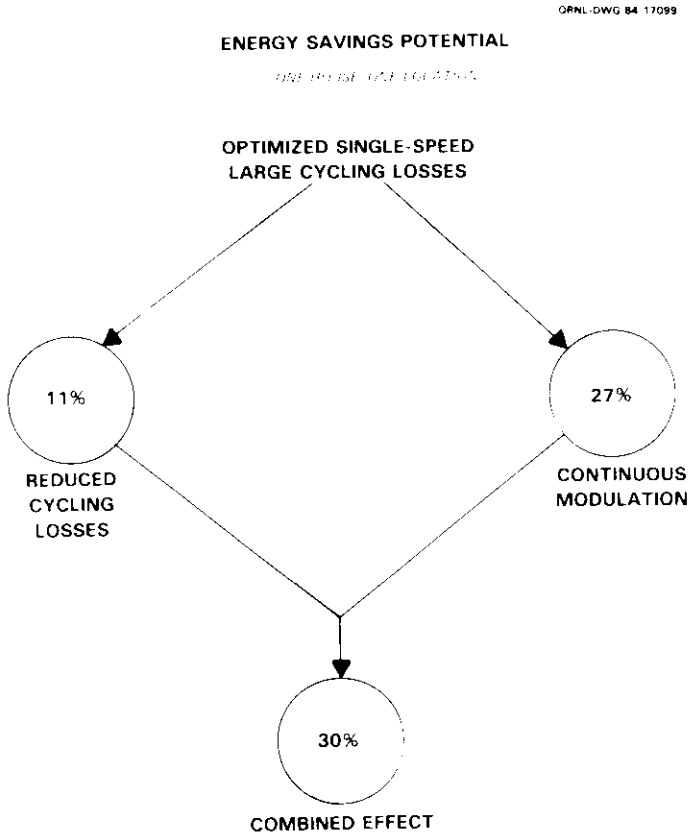


Fig. 8. Energy savings potential relative to optimized reference system.

2 and 3) are not additive because the cycling losses are reduced significantly by either path; thus these reductions cannot be counted twice.

The efficiency improvement potential for the various cases is similarly shown in Fig. 9. By definition, the efficiency improvement numbers are always higher than the energy savings percentages. Performance improvements of 36 to 43% are thus predicted for the continuous modulation cases when compared to the optimized "regular" dynamic loss, single-speed system.

Allowable Consumer Cost Premiums

Based on the energy savings computed for the various cases for the average-insulated 1800-ft^2 house in Nashville, maximum-allowable, consumer first-cost premiums to obtain 3- and 5-year simple paybacks were computed. These are shown in Fig. 10, where the values within the circles are again relative to the optimized base case. An electricity cost of 5 cents/kWh was assumed. For a 3-year simple payback, a consumer could pay $145 more for a nonmodulating heat pump with reduced cycling losses or $386 more for a similar modulating system. Allowable premiums for 5-year paybacks are proportionally increased (by 5/3 to 67% larger than the 3-year values).

The values in Fig. 10 are absolute numbers as opposed to the percentages given in Figs. 8 and 9. As such, they will have a much wider variation for different houses, climates, and electricity costs than the percent energy savings and performance improvements. Nevertheless, the numbers in Fig. 10 should provide ballpark

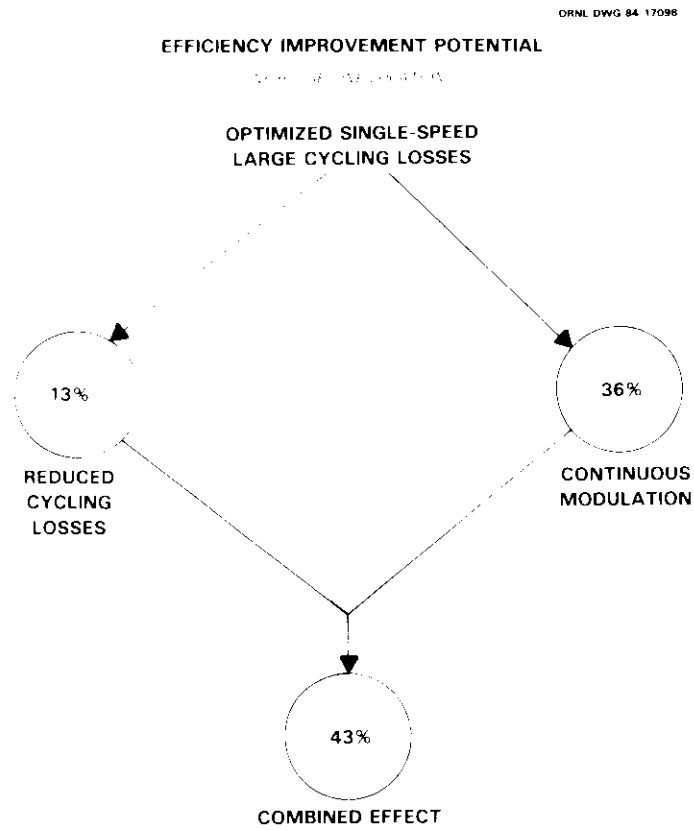


Fig. 9. Efficiency improvement potential relative to optimized reference system.

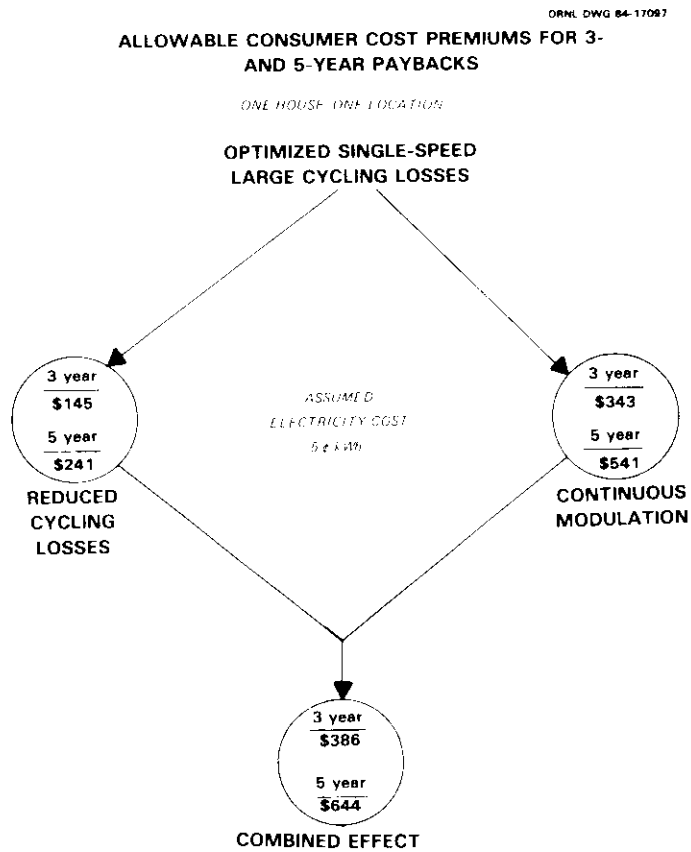


Fig. 10. Allowable consumer cost premiums for 3- and 5-year paybacks relative to optimized reference system.

values for the prices that the speed control and/or the cycling loss control devices must approach for short paybacks unless other concomitant benefits are considered.

Additional Advantages Of Continuous-Modulation-Systems

Continuous-capacity-modulation does offer other benefits that either are difficult to quantify with specific dollar figures, or relate to comfort and noise (which are somewhat subjective), or both. These include the potential for:

- improved compressor reliability since there would be far fewer on/off cycles and a much smaller inrush of motor current during startup;
- improved indoor temperature and humidity control because of the almost continuous operation of the unit when heating or cooling is required;
- multipurpose use of required microprocessor-based thermostats for setback, smart setup, and humidity control;
- reduced outdoor noise because of smaller required air flow rates;
- oversizing on cooling in northern climates to further reduce the backup heat needs while still providing adequate dehumidification in cooling, and
- significant additional operating cost savings through zone control (possibly by using a microprocessor thermostat to automatically adjust the capacity of the central unit by monitoring the average rate of temperature rise in the conditioned rooms).

Continuously modulating heat pumps would be uniquely well suited to zoning applications. For those whose lifestyles would allow them to reduce their annual house loads by zoning by 25%, the allowable first cost for case 3 would rise 69% to $580 for a 3-year payback and 73% to $936 for a 5-year payback (assuming the same annual performance factors apply for a zoned vs a non-zoned application[9]). This zoning application has been studied specifically for various possible lifestyles and is reported in the following paper.[10]

Benefits of System Optimization

In Fig. 11, the performance of available nonmodulating and modulating equipment is shown alongside comparable optimized results. It is estimated that all the cases have roughly equivalent levels of component efficiency. The solid dots in Fig. 11 show performance results for specific manufacturers' models of the various modulation types.

For all cases in Fig. 11, an ORNL binned APF model[11] was applied to either the manufacturers' or the heat pump model-generated steady state data. Dynamic loss factors (cycling and frosting/defrosting factors) for the commercial models were obtained directly from the manufacturers whenever available. For cases where data were unavailable or incomplete, consistent assumptions were made between the commercial and optimized

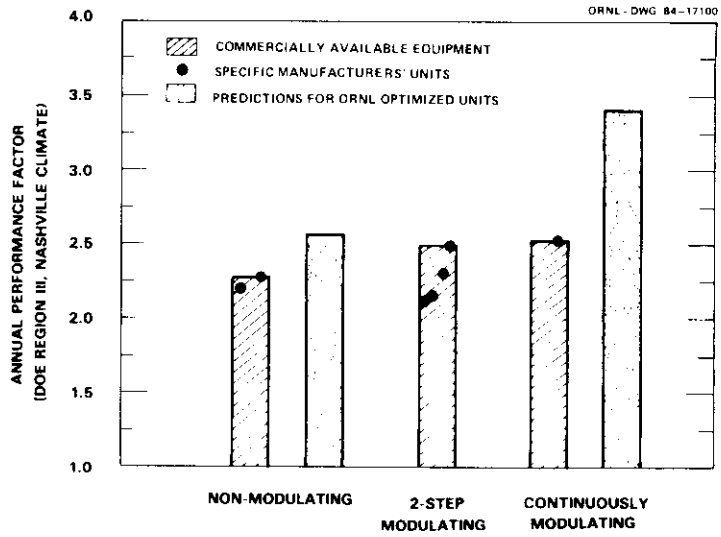


Fig. 11. Relative performance of current product lines vs optimized configurations—typical high-efficiency components.

cases. Default cyclic degradation coefficients (C_Ds) of 0.25[5] were used for the optimized cases (for Fig. 11 only) and where manufacturers' values were not available.

From these comparisons, it is seen that there is a relatively small performance difference between the "optimized" and available nonmodulating equipment (13%) as compared to the much larger (35%) difference between the optimized and currently available CVS systems. Clearly, heat pump manufacturers are much closer to optimum designs (as would be expected) for the existing single-speed technology—where the manufacturers have had many years of design experience and product evolution. But for the newer modulating technology, our results using computer-aided design optimization indicate that substantial further benefits are obtainable. Such design tools become more important in the modulating heat pumps because these systems are:

- more complex—more design variables to integrate, more design flexibility to capitalize on;
- in higher performance areas—where design integration is more important in obtaining maximum performance; and
- under a more restrictive cost-vs-benefit situation—where designers must get the most efficiency out of the system to justify the cost of the improved components.

This seems to hold true even when higher-efficiency components are used and dynamic losses are reduced. In Fig. 12, the relative performances of the most advanced single-speed and step-modulating equipment are shown†† relative to comparable optimized CVS systems. In both CVS systems, compressor and indoor fan efficiency levels comparable to the Westinghouse/DOE heat pump[12] were assumed. In one case, dynamic loss factors equivalent to those measured for the Westinghouse/DOE unit were used. For the other CVS case, the lowest-loss combination of cycling and frosting/defrosting factors from the two advanced products was

††The performance of the step-modulating unit relative to the nonmodulating unit would be better in a more northern climate for which the unit was primarily intended.[12]

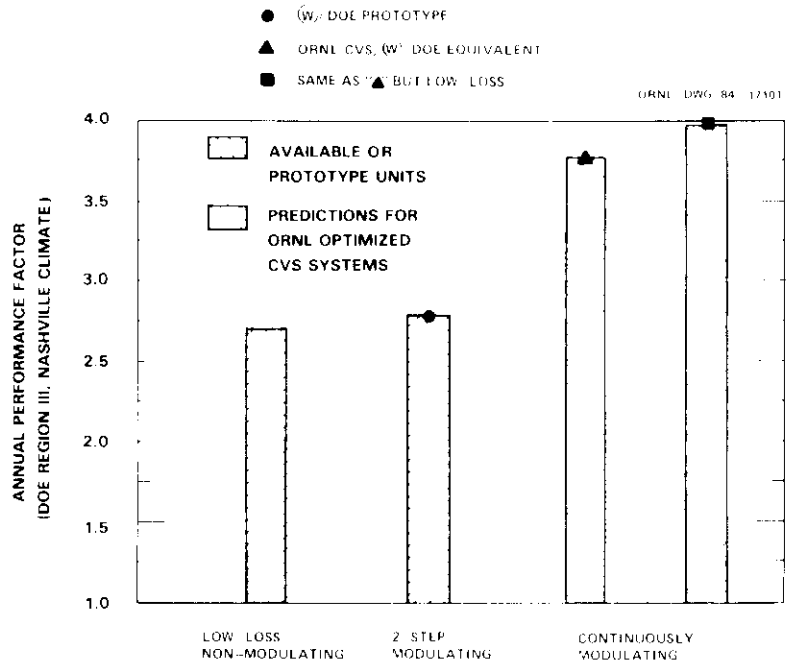


Fig. 12. Relative performance of advanced products vs optimized CVS configurations—best high-efficiency components available.

used. The Westinghouse/DOE-comparable CVS case shows a 35% performance advantage and the best CVS system has a 47% higher performance than the best nonmodulating system.‡‡ These results suggest that system optimization benefits are equally available at both moderate and high component efficiency levels.

Recommendations

Based on this initial evaluation of CVS systems and considering the potential for electronics costs to further decrease as electricity prices rise, we see such advanced CVS systems as a strong future competitor to single-speed systems and as decidedly superior in energy conservation potential.

To achieve this position, it seems especially important that such systems be computer optimized to take full advantage of the increased design flexibility available. Further, development of PM-ECMs or equivalent technology must continue to the point of providing speed controllers of

- moderate to low cost,
- moderate to high performance, and
- high reliability

for successful penetration of this technology into the marketplace.

Status of Related Work and Future Activities

Since this initial assessment work was completed, efforts have been largely devoted to APF modeling, system design validation work, and collection of information to complete a more refined evaluation of CVS system benefits.

‡‡Note that no judgements are being made here regarding the relative cost-effectiveness of the various advanced designs.

The APF model used in the initial capacity-modulation study has been greatly broadened to include step-modulation cases (as those shown in Figs. 11 and 12); a more general treatment of continuous-capacity modulation; a variety of frosting, defrosting, and cycling loss schemes; and thermally activated heat pumps. This program has been combined with a binned loads program[13] and a 117-city weather data base[14] to provide a convenient tool for scoping and optimization studies and a common comparison basis, including loss breakdowns, for a variety of heat pump types. Validation and documentation of the APF/Loads program has just recently been completed.[11] Documentation of a set of computer design tools for use with these models is well underway.[15]

The system design validation work includes further heat pump model validation against laboratory data on high-efficiency single-speed heat pumps and assessment of the effect of charge inventory assumptions and of optimized capillary tube sizing relative to more idealized refrigerant flow control schemes.[16] Some further laboratory validation tests of the modulating heat pump model are planned as well.

Efforts have begun to more thoroughly evaluate the potential of continuous-capacity-modulation. More current and complete data are being obtained for the performance of PM-ECMs and reciprocating compressors as a function of operating speed and torque. Current plans are to optimize a PM-ECM-driven, reciprocating compressor system with this new data for a broader range of climates—with sufficient constraints added to ensure acceptable comfort levels. This analysis should (1) provide a good estimate of the national energy savings and application potential of one type of advanced modulating system, and (2) provide a credible demonstration of the use of a set of computer-aided design tools on a problem of interest to the heat pump industry as a whole.

Concurrent with these activities are efforts to transfer the programs and methodology to potential users in industry. Cooperative

formal and informal working arrangements are being actively pursued with individual manufacturers of heat pumps and compressors.

References

1. C. K. Rice, et al., *Design Optimization and the Limits of Steady-State Heating Efficiency for Conventional Single-Speed Air-Source Heat Pumps*, ORNL/CON-63, Oak Ridge National Laboratory, October 1981.
2. S. K. Fischer and C. K. Rice, *The Oak Ridge Heat Pump Models: I. A Steady-State Computer Design Model for Air-to-Air Heat Pumps*, ORNL/CON-80/R1, Oak Ridge National Laboratory, August 1983.
3. Numerical Analysis Group, *NAG Fortran Library, Mark 7*, Atomic Energy Establishment, Harwell, England (1978).
4. W. H. Parken, et al., "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump," *ASHRAE Trans.* **83**(1), 839-850 (1977).
5. Department of Energy, "Test Procedures for Central Air Conditioners, Including Heat Pumps," 10 CFR Part 430, *Federal Register*, Vol. 44, No. 249, pp. 76700-76723, December 1979.
6. "ENMOD Split System Heat Pump," *York Application Data*, Form 515.50-AD1 (380).
7. J. D. Lloyd, "Variable-Speed Compressor Motors Operated on Inverters," *ASHRAE Trans.* **88**(1) (1982).
8. Ken Cooper, York Division, Borg-Warner, York, PA., personal communication with C. K. Rice, Oak Ridge National Laboratory, 1982.
9. J. C. Moyers and E. A. Nephew, *Scoping Evaluation of Potential Benefits of Zoning with Residential Space-Conditioning Applications*, ORNL/CON-167, Oak Ridge National Laboratory (in press).
10. J. C. Moyers and E. A. Nephew, "Analytical Study of the Benefits of Zone-Control Heat Pumps," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11–12, 1984.
11. C. K. Rice, et al., *The Oak Ridge Heat Pump Models: II. An Annual Performance Factor/Loads Model for Residential Air-Source Heat Pumps*, ORNL/CON-160, Oak Ridge National Laboratory (to be published).
12. S. E. Veyo and T. J. Fagan, *Advanced Electric Heat Pump Dual-Stroke Compressor and System Development, Final Report*, ORNL/Sub/79-24712/3, December 1983.
13. M. L. Ballou, et al., *MAD: A Computer Program for ACES Design Using Monthly Thermal Loads*, ORNL/CON-51, Oak Ridge National Laboratory, March 1981.
14. U. S. Air Force, *Engineering Weather Data*, AFM-88-29, July 1978.
15. S. K. Fischer, et al., *The Oak Ridge Heat Pump Models: III. Design Tools for Heat Pump Development*, ORNL/CON-175, Oak Ridge National Laboratory (to be published).
16. S. K. Fischer and C. K. Rice, "System Design Optimization and Validation for Single-Speed Heat Pumps," submitted for presentation at the ASHRAE 1985 Summer Meeting.

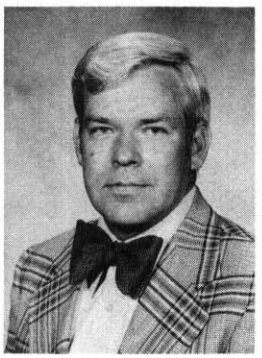
Scoping Evaluation of Potential Benefits of Zoning with Residential Space-Conditioning Systems

Edmund A. Nephew and John C. Moyers

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

Edmund A. Nephew received the B.S. degree in physics from Youngstown State University and the M.S. degree (physics) from Ohio State University. He is presently a senior staff member in the Energy Division of Oak Ridge National Laboratory. He has been a member of the Laboratory staff for the past 25 years and has been responsible for systems analysis and performance evaluations of home heating and cooling equipment for the past 8 years. He is an active member of the Deutsches Atomforum and the American Planning Association.



John C. Moyers received B.S. and M.S. degrees in mechanical engineering from the University of Tennessee and Columbia University, respectively. He is a registered professional engineer in Tennessee.

Moyers has been employed at the Laboratory for 28 years, with research, development, and managerial responsibilities in the areas of nuclear energy, desalination, fossil energy, and energy conservation. He is presently a senior development staff member in the Engineering Technology Division.

SCOPING EVALUATION OF POTENTIAL BENEFITS OF ZONING WITH RESIDENTIAL SPACE-CONDITIONING SYSTEMS*

E. A. Nephew
Energy Division

J. C. Moyers
Engineering Technology Division

Oak Ridge National Laboratory
Oak Ridge, Tennessee 37831

Abstract

The potential for load reduction through zoning and for the reduction of national energy use deriving from application of variable-capacity heat pumps in zoned residential space conditioning was evaluated. Load reductions of 30–50% of full-time, whole-house loads were estimated to be attainable through combined zoning and temporal temperature setback that created no loss of comfort in the occupied zones. Load reductions of up to 60% were attainable when a portion of the house was unoccupied and received only minimal space conditioning. Maximum potential national energy savings, achievable through complete market penetration, of 1.7 and 2.1 quads/year were estimated for the fully occupied and partially occupied residences, respectively.

Introduction

In the quest for means for conserving energy used for residential space conditioning, both improvements in equipment efficiency and modifications in temperature control strategy to reduce the space-conditioning load imposed on the equipment warrant investigation. The development of high-efficiency oil and gas furnaces, air conditioners, and heat pumps has been pursued vigorously by manufacturers and research and development organizations for the past several years, providing marked improvements in equipment performance. Temperature setback (setting the thermostat at a lower temperature during unoccupied or sleeping hours) is an example of a temperature control strategy modification that can appreciably reduce space-conditioning loads and energy consumption.

A second temperature control strategy modification that offers possible reductions in space-conditioning loads and energy consumption—zoning—takes advantage of the fact that different functional areas of the residence have different temperature requirements at various times of day. For example, lower temperatures are acceptable or even preferred in the bedrooms, compared with temperatures required for comfort in the family living areas. Unfortunately, standard single-capacity heat pumps may not be amenable to zoning. Reduction of air flow across the indoor heat exchanger as a result of a lower demand for heating or cooling in one of the zones lowers the heat pump's efficiency in both heating and cooling operations. In the heating mode, the compressor discharge pressure and temperature would be elevated, possibly shortening compressor life. In the cooling mode, the compressor suction pressure and temperature would be lowered, reducing capacity and possibly causing excessive dehumidification as the sensible-to-total cooling ratio decreases.

A brief evaluation of the potential for load reduction through zoning and for the reduction of national energy use deriving from application of variable-capacity heat pumps in zoned residential space conditioning has been completed. The evaluation methodology and results are presented herein.

Temperature Control Strategies

Six feasible temperature control strategies are described in Table 1. Temperature control strategy 1 provides full-comfort conditions (temperatures between 70 and 78°F at all times). This is the base, energy-affluent temperature control strategy. Temperature control strategy 2 is a somewhat frugal scheme, with zoning but without time-of-day temperature setback. In the living area, a slight sacrifice in summertime comfort accompanies raising the air-conditioning control temperature to 82°F. The bedroom control temperature is decreased to 60°F in the winter but is maintained at 78°F in the summer, providing near-ideal sleeping comfort year round. This temperature control strategy was also evaluated with insulated partitions between the two zones (referred to in later discussions as temperature control strategy 2A). Temperature control strategy 3 is more austere in that the summer and winter control temperatures in the living area are 85 and 68°F, respectively. The bedroom area is maintained at the near-ideal sleeping comfort range. There is no time-of-day temperature setback.

Temperature control strategy 4 involves both zoning and time-of-day temperature setback. Each of the two zones is maintained

*Research sponsored by the Building Energy Research Division, U.S. Department of Energy, under Contract No. DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

Table 1. Description of temperature control strategies

Temperature control strategy TCS	Minimum winter/maximum summer control temperatures (°F)					
	(Living area) Zone 1			(Sleeping area) Zone 2		
	12 m.–8 a.m.	8 a.m.–4 p.m.	4 p.m.–12 m.	12 m.–8 a.m.	8 a.m.–4 p.m.	4 p.m.–12 m.
1	70/78	70/78	70/78	70/78	70/78	70/78
2	70/82	70/82	70/82	60/78	60/78	60/78
3	68/85	68/85	68/85	60/78	60/78	60/78
4	60/85	70/78	70/78	60/78	60/85	60/78
5	60/78	70/78	70/78	60/78	70/78	70/78
6*	60/78	70/78	70/78	50/85	50/85	50/85

*Allocation of space between the two zones for TCS-6 is as follows: zone 1 is the kitchen, family room and one bedroom, while zone 2 is the rest of the house and is essentially unoccupied.

at a comfortable temperature level during the zone's occupied period but is allowed to float within rather wide limits during the unoccupied period. Temperature control strategy 5 includes temperature setback during the sleeping hours but has no zoning. This energy-conserving temperature control strategy can be implemented with a single-capacity heat pump, but it cannot provide the flexibility included in temperature control strategy 4 (which requires a capacity-modulating heat pump to accommodate zoning).

Temperature control strategy 6 represents a departure from full occupancy of the house and might be appropriate for an older couple whose children are no longer living in the family home. Zone 1, the principal occupied zone, includes a reduced living area and one bedroom; this zone is maintained at a comfortable temperature appropriate to the time of day. The remainder of the house, zone 2, is allowed to float between temperature limits outside the comfort range. There would be periods when zone 2 would be brought to temperatures within the comfort range, but these would constitute only a small fraction of the total seasonal time and were not considered in estimates of annual loads and energy consumption.

The floor plan of the dwelling chosen for zoning evaluation is shown in Fig. 1. The dwelling is an 1800-ft^2, three-bedroom, single-story building, with R-11 insulation in the walls, R-19 in the attic, and R-9 in the floor. The windows are single-glazed. This house has essentially HUD minimum insulation characteristics and represents existing housing stock rather than new houses being built today.

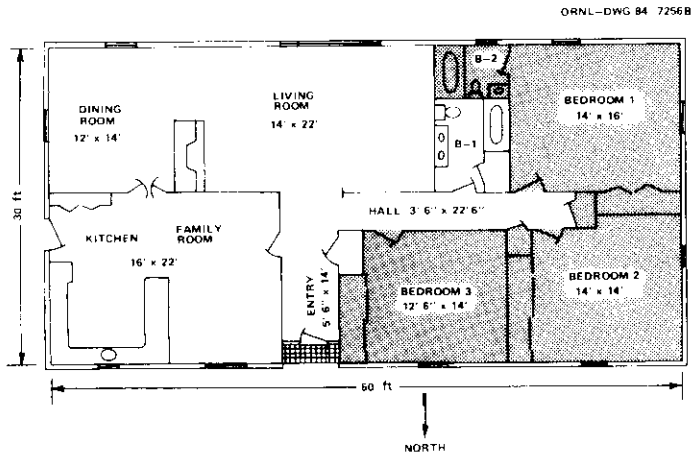


Fig. 1. Residence floor plan.

The six temperature control strategies described above were implemented by dividing the house into two zones and imposing the hypothesized schedule of control temperatures. Each zone may have separate temperature control for each of three 8-h time periods within the day. Generally, the partitions separating the two zones were not insulated but consisted of 2 × 4 studs with 1/2-in. gypsum board on each side. As a check on the sensitivity of zoned performance to partition insulation, one temperature control strategy analysis was repeated with R-11 insulation in the partition.

Allocation of space between the two zones for five of the six temperature control strategies was as shown in Fig. 1, where zone 2 is represented by the shaded area. In this arrangement, zone 1 contains the areas normally occupied during day and evening for food preparation, dining, and other nonsleeping activities. Zone 2 includes the three bedrooms and the second bath. This zonal arrangement appeared to offer the opportunity for implementing temperature control strategies representing differing levels of comfort conditioning for a family of four.

Internal sources of heat due to occupancy, lighting, and appliance use were hypothesized for each zone and each of the three 8-h periods of the day. For temperature control strategies 1 through 5, a family of four was assumed to occupy the residence, with three members absent from 8 a.m. to 4 p.m. and 3 person-hours of absence from 4 p.m. to midnight. Lighting and appliances contributed an average of 932 W of heat, distributed appropriately among the two zones and three daily time periods. Temperature control strategy 6 included occupancy by two persons, with one absent during the day and 5 person-hours of absence during the evening. Average contribution of heat by lighting and appliances was 806 W, again distributed appropriately by zone and time.

Computed Building Loads and HVAC Systems Energy Consumption

Building heating and cooling loads were computed by use of a modified version of the MAD computer program[1] which is capable of treating two zones and three distinct 8-h time periods during the day. The computation is for steady-state conditions and uses house description and bin weather data as input.

Building heating and cooling loads can be calculated more precisely by use of one of several large computer programs that account for the dynamic effects of heat storage and discharge in the mass of the structure during temperature transients. Previous undocumented comparative evaluations by the authors indicated only minor differences in loads as calculated by steady-state and dynamic models for light residential construction.

It should be noted that the calculational procedure used to estimate energy consumption did not include any loss in efficiency during recovery from temperature setback periods. For the usual single-capacity heat pump system, recovery is accompanied by heavier-than-normal use of the electric resistance supplemental heaters. Thus the energy saved during the setback period is partially expended during recovery. It appears that this energy use penalty may be smaller, or even nonexistent, for variable-capacity systems, especially in zoned applications where the heat pump may have adequate capacity to accomplish the recovery without assistance from the resistance heaters.

Discussion of Results

Lowering the control temperature for heating in a given zone while maintaining the other zone at a constant temperature significantly lowers the heating load for the changed zone but increases the heating load for the unchanged zone. This effect is due to the change in heat flow through the partition separating the two zones. For example, in going from temperature control strategy 1 to temperature control strategy 2 (zone 2 control temperature changes from 70 to 60°F while zone 1 remains at 70°F) in Knoxville reduces the heating load for zone 2 by approximately 34% of the two-zone total for temperature control strategy 1 (see Table 2). Concomitantly, the heating load for zone 1 is increased by approximately 12% of the two-zone total, so that the net change in total heating load is about 22%. Addition of R-11 insulation to the partition, as in temperature control strategy 2A (with the same control temperatures as in temperature control strategy 2), changes the zone 2 reduction to 27%, the increase in zone 1 to 4%, and the net change in total heating load to 23%.

Reducing the heating control temperature for only part of the 24-h day, either in a given zone or in the entire house, also provides significant reductions in heating load. For example, comparing temperature control strategies 2 and 4 (where the zone 1 heating control temperature is reduced from 70 to 60°F during the unoccupied period from midnight to 8 a.m.) for

Table 2. Annual loads and energy consumption for Knoxville

	Temperature control strategy						
	TCS-1	TCS-2	TCS-2A	TCS-3	TCS-4	TCS-5	TCS-6
Annual heating load (MBtu)							
Zone 1	23.378	28.615	24.989	24.408	19.761	16.872	16.712
Zone 2	21.305	6.238	9.165	7.001	8.156	16.622	3.310
Total	44.141	34.468	33.818	31.158	27.657	33.134	19.850
Annual cooling load (MBtu)							
Zone 1	19.117	13.188	14.449	9.424	20.037	19.117	17.435
Zone 2	8.745	10.490	9.165	11.524	5.143	8.745	4.216
Total	27.619	23.591	23.524	20.884	24.955	27.619	21.382
Annual electricity use (kWh)							
Advanced-component, single-capacity							
Heating	4,967					3,755	
Cooling	2,843					2,843	
Total	7,810					6,598	
Advanced-component, dual-capacity							
Heating	4,662	3,707	3,652	3,331	3,078	3,626	2,207
Cooling	2,630	2,277	2,272	2,051	2,373	2,584	2,087
Total	7,292	5,984	5,924	5,382	5,451	6,210	4,294
Optimized standard-component, single-capacity							
Heating	5,523					4,293	
Cooling	2,691					2,660	
Total	8,214					6,953	
Optimized standard-component, continuously variable-capacity							
Heating	4,071	3,197	3,152	2,922	2,652	3,119	1,944
Cooling	1,689	1,445	1,442	1,302	1,536	1,689	1,326
Total	5,760	4,642	4,594	4,224	4,188	4,808	3,270

Knoxville shows a reduction in total heating load of almost 20%. Similarly, comparing temperature control strategies 1 and 5 (where the entire-house heating control temperature is reduced from 70 to 60°F during the same 8-h period) shows a total heating load reduction of almost 25%.

Temperature control strategy 5, which provides the same comfort level in each of the zones during their respective occupied hours as in temperature control strategy 4 but which does not benefit from zoning, has a total heating requirement in Knoxville which is almost 20% higher than that for temperature control strategy 4. Comparison of these two temperature control strategies provides the clearest indication of the potential load reduction benefits derivable from zoning, with no sacrifice in comfort but with temperature setback.

Temperature control strategy 6 provides the largest reduction in heating load, a reduction of 55% in comparison with temperature control strategy 1 or 28% in comparison with temperature control strategy 4. The reduced load results from a greatly reduced comfort area (zone 1) and the very low heating control temperature (50°F) for the rest of the house.

Heating loads for each of the temperature control strategies in Knoxville are shown in Fig. 2 as percentages of the heating load for temperature control strategy 1. Loads for the two other cities, Chicago and Phoenix, are indicated by range bars for each temperature control strategy.

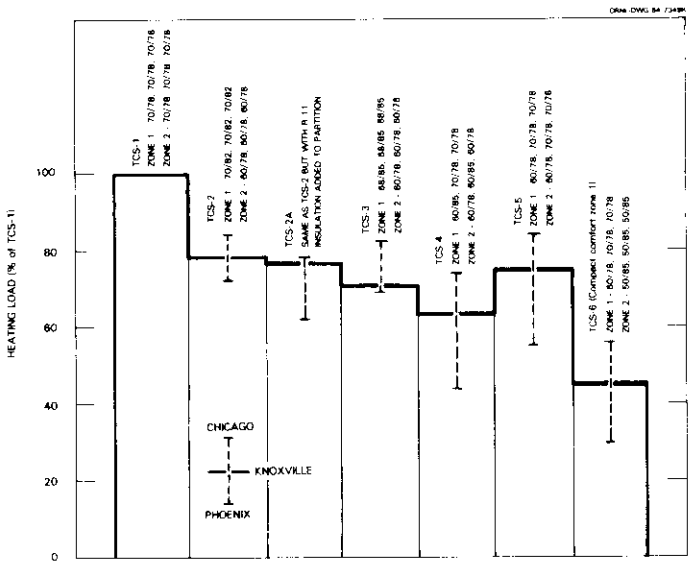


Fig. 2. Relative heating loads for various temperature control strategies. Numbers listed after zones represent allowable temperature ranges during the three 8-h periods of the day.

Cooling load trends with changes in cooling control temperature are similar to those for heating but, of course, are of reverse sign. Raising the cooling control temperature in one zone while maintaining that of the other zone constant reduces the cooling load in the altered zone but increases the load in the other.

Reductions in cooling load offered by the various temperature control strategies, compared with temperature control strategy 1, are small relative to heating reductions. As shown in Fig. 3, temperature control strategies 2 and 3 in Knoxville provide load reductions of 15 and 24% because of increasing the control

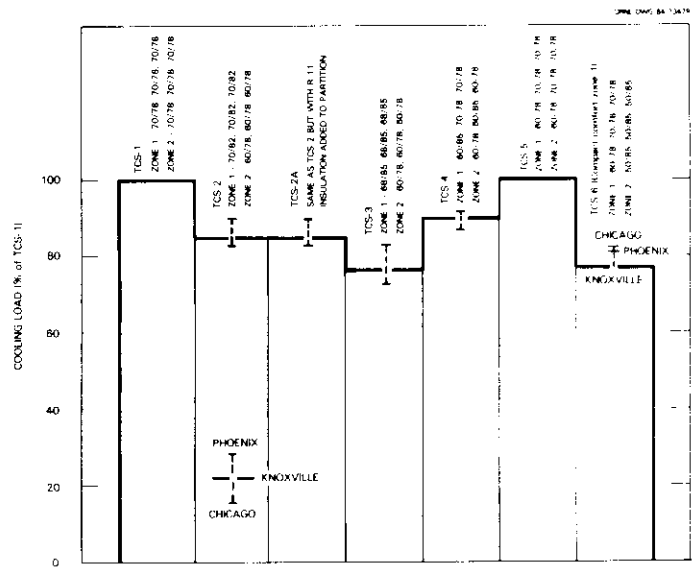


Fig. 3. Relative cooling loads for various temperature control strategies. Numbers listed after zones represent allowable temperature ranges during the three 8-h periods of the day.

temperature for zone 1 by 4 and 7°F, respectively. Adding R-11 insulation to the partition has no significant effect on cooling load, as seen from comparing temperature control strategies 2 and 2A. Temperature control strategy 4 provides a modest 10% reduction in cooling load, at no sacrifice in comfort within any occupied zone. Temperature control strategy 5 has the same cooling load as temperature control strategy 1 since, because it has no zoning, the entire house is controlled at 78°F at all times. Temperature control strategy 6, with its compact comfort zone, has approximately the same cooling load as temperature control strategy 3.

Energy consumption for each of the temperature control strategies generally tracks the building loads. In Fig. 4, annual energy consumption in Knoxville for each temperature control strategy is presented as a percentage of that for temperature

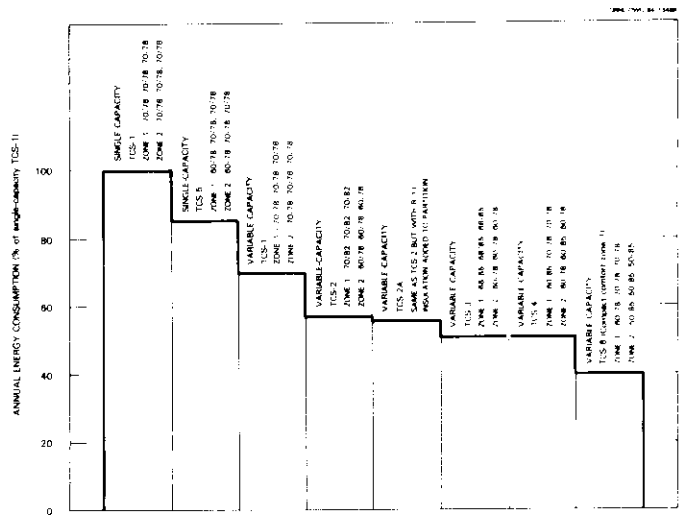


Fig. 4. Relative annual energy consumption with optimized standard-component systems for various temperature control strategies—Knoxville. Numbers listed after zones represent allowable temperature ranges during the three 8-h periods of the day.

control strategy 1, using either optimized single-capacity or optimized variable-capacity systems as appropriate. Night setback without zoning, as in temperature control strategy 5, provides a saving in consumption of 9 to 15% without the need for a variable-capacity system. The more efficient variable-capacity system, when applied to temperature control strategy 1, provides a 26 to 30% reduction in consumption.

Temperature control strategy 2, with cooler zone 2 winter temperature and warmer zone 1 summer temperature, requires a variable-capacity system and provides a 37 to 43% reduction in consumption. Adding insulation to the partition separating the two zones (temperature control strategy 2A) has only a trivial effect on consumption. Temperature control strategy 3, with wider deviation from comfort in zone 1 for both summer and winter, provides savings of 42 to 49%. Temperature control strategy 4 provides similar savings but at no sacrifice in comfort. Temperature control strategy 6 provides the greatest savings, ranging from 47 to 60%.

Conclusions

Appreciable savings in consumption of primary energy can be obtained by developing and encouraging the use of residential HVAC systems that can provide heating and cooling on demand to selected zones of a house. The results of this study have shown that use of variable-capacity HVAC systems having controlled distribution and temperature setback capability can, with relatively minor changes in temperature control strategy, produce annual savings of more than 1.5 quads nationally. Greater savings of 2.0 quads or more can be achieved with the same systems by means of more significant changes in strategy.

The temperature control strategies evaluated in this study differ in the application of night setback, capacity modulation, and zoning. The potential energy savings estimated for each of these three departures from the full-time, full-comfort strategy are as follows: night setback, 14–15%; capacity modulation, 7–30%; and zoning, 12–32%. Thus, zoning provides an additional incentive and justification for the development of capacity-modulated equipment.

A key feature in assuring the acceptability of zoning is to provide the homeowner with complete flexibility in selecting the particular temperature control strategy that best meets his needs at any given time. This means that the HVAC system must be sized large enough to provide whole-house heating and cooling when required, as for a family or when house guests are present. On the other hand, the homeowner will be adequately served by a smaller comfort-conditioned zone when the family size is smaller. The HVAC system selected should be able to accommodate these limiting use patterns as well as a large variety of intermediate situations.

References

1. M. L. Ballou, E. A. Nephew, and L. A. Abbatiello, *MAD: A Computer Program for ACES Design Using Monthly Thermal Loads,* ORNL/CON-51, Oak Ridge National Laboratory, March 1981.

Condensing Coefficients When Using Refrigerant Mixtures

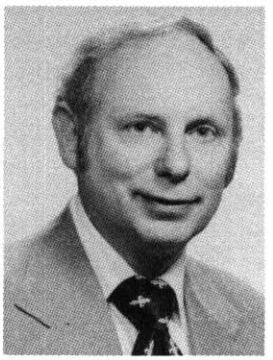
Wilbert F. Stoecker

University of Illinois at Urbana-Champaign

Urbana, Illinois 61801

Wilbert F. Stoecker is a professor of mechanical engineering at the University of Illinois at Urbana-Champaign where he conducts research in refrigerant mixtures and computer control of thermal systems. His teaching responsibilities include courses in the simulation, optimization, and computer control of thermal systems, including refrigeration and air-conditioning facilities. He is the author of two textbooks, *Refrigeration and Air Conditioning* and *Design of Thermal Systems.*

Professor Stoecker is a fellow in the American Society of Heating, Refrigeration, and Air-Conditioning Engineers and was granted the F. Paul Anderson Award from that organization. He is a member of the International Institute of Refrigeration and serves on the Scientific Advisory Council of the Refrigeration Research Foundation.



CONDENSING COEFFICIENTS WHEN USING REFRIGERANT MIXTURES

W. F. Stoecker
Department of Mechanical and Industrial Engineering
University of Illinois at Urbana-Champaign
1206 West Green Street
Urbana, IL 61801

Importance of Heat-Transfer Coefficients

There is now considerable research activity both in the United States and abroad in the field of refrigerant mixtures. Along with some of the special reasons for considering mixtures, such as tailoring the pressures for the required evaporating and condensing temperatures, the principal motivation for the interest is for energy conservation. The potential for energy conservation appears for two different reasons. One reason is a thermodynamic one where in certain situations it may be possible to take advantage of the "sliding temperatures" experienced when a mixture boils or condenses at a constant pressure. The second reason which is of special interest in heat pumps is to be able to adjust heat pumping capacity by changing the concentration of the mixture in the active refrigerant circuit.

Whatever the motivation for considering the use of refrigerant mixtures, maintaining high heat transfer coefficients is necessary in order to preserve whatever improvements may derive from the use of the mixture. Several investigators have reported a reduction in heat transfer coefficients of mixtures in comparison to either of the individual components when operating singly. Such has also been the experience of the author's organization in research on a refrigerant system that uses a mixture of refrigerants.

Condensing Coefficients

At least one other research organization in the United States is concentrating on refrigerant mixture heat transfer coefficients during boiling, so it was the decision of our university and Oak Ridge National Laboratory (ORNL) to concentrate on condensing coefficients. Associated with the studies on a two condenser, two evaporator refrigeration system, some condensing heat transfer coefficients were measured. The two condensers are shown schematically in Fig. 1 and consist of tube in tube heat exchangers with refrigerant flowing in the inner tube and cooling water in the annular space. The water flow was counter to that of the refrigerant. Each condenser consisted of three passes.

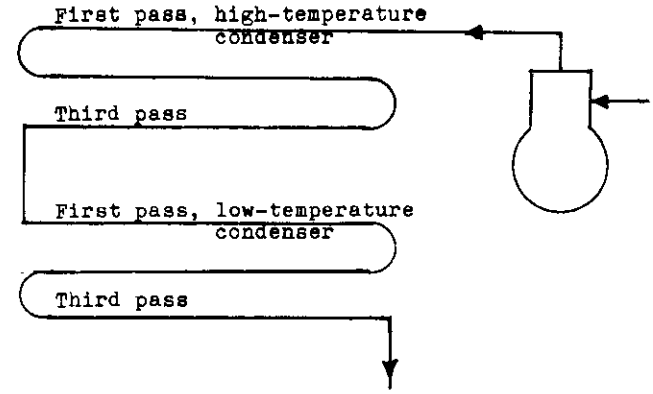


Fig. 1 The three passes on the high-temperature and low-temperature condensers.

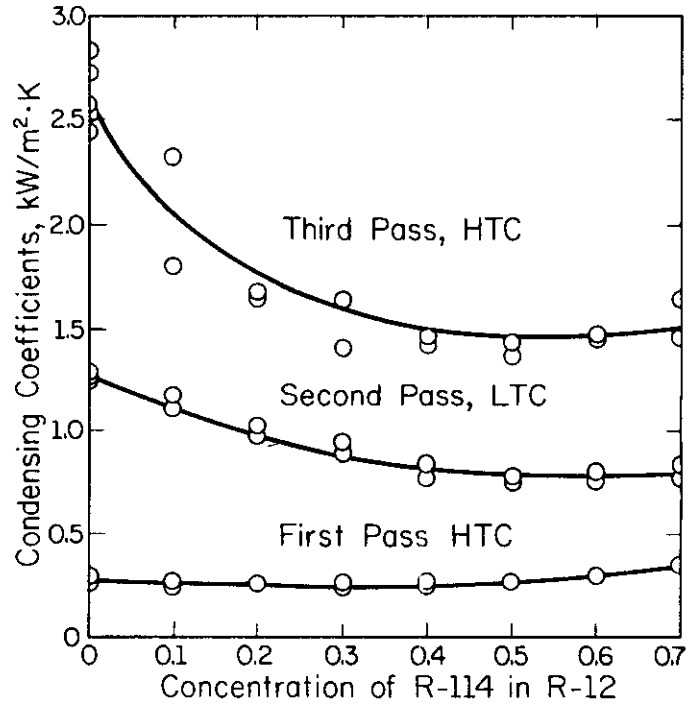


Fig. 2 Condensing heat-transfer cofficients as a function of the concentration. The flow rate is approximately 0.022 kg/s.

As a byproduct of studies of the system efficiency, the condensing heat transfer coefficients were measured (Ref. 1) and are shown in Fig. 2. After observing results, we conducted some further tests (Ref. 2) that were run at about 20 to 30 percent higher refrigerant flow rates (Fig. 3). These tests showed that in the

first pass of the high temperature condenser (where desuperheating was also occurring) that the magnitude of the concentration had no effect on the coefficient. The most dominant change occurred in the third pass of the high temperature condenser (which would be near the mid-range of the condenser). Here the coefficient was highest of any of the sections, and also dropped the most when the R114 concentration was increased.

Condenser Test Facility

It appeared valuable to study the condensing heat transfer coefficients further with two objectives: learn more of the mechanism of condensation of mixtures and especially to forestall, if possible, any reduction in magnitudes of the coefficients. In focusing on the condensing heat transfer coefficients, a two-part plan was adopted in which the first phase would attempt to learn more of the mechanism of condensation of mixtures and the second phase a further study of numerical values that could be correlated to the regimes established in the first phase. This report covers Phase I which was designed around a glass tube condenser that was cooled by water flowing in an annulus formed by transparent plastic pipe surrounding the glass condenser tube. Figure 4 is a photograph of the instrument panel with manifolds for measuring the pressure at various locations

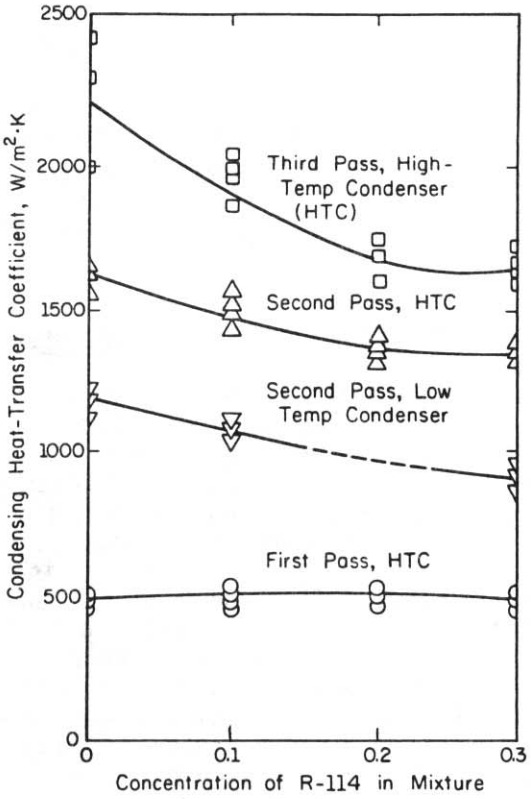


Fig. 3 Condensing heat-transfer coefficients as a function of the concentration. The flow rate is approximately 0.024 to 0.027 kg/s.

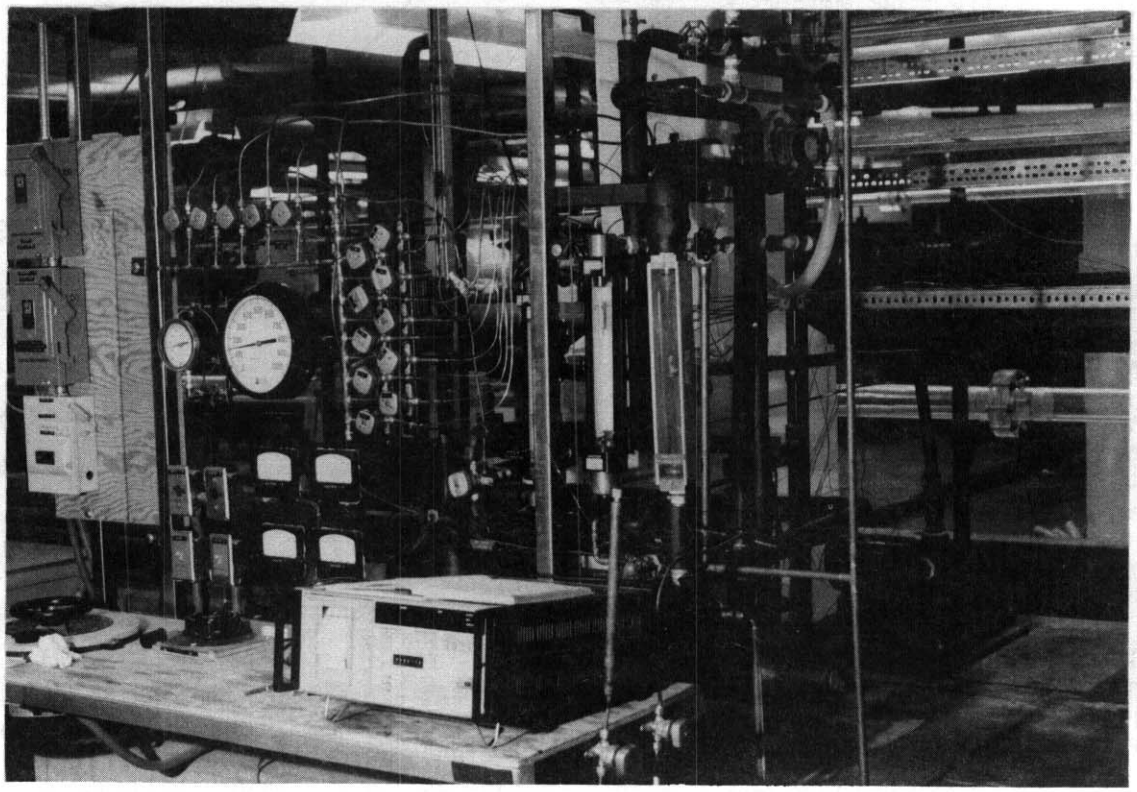


Fig. 4 Photograph of the condenser test facility with the transparent condenser in the upper right-hand corner.

along the tube, flow meters, and a data acquisition system, for reading thermocouples. Figure 5 is a flow diagram of the overall assembly indicating that the condensate from the condenser was subcooled further, flowed to a pump and then to a boiler which transferred heat from water to vaporize the refrigerant.

The condenser consisted of twelve 1.5 m long beaded process glass tubes, 13 mm internal diameter, and 19 mm outside diameter. The glass tubes were pressure tested to 1,000 kPa. Between each glass section was mounted a brass test section, shown in Fig. 6, which provided access for a thermocouple probe, pressure tap, and at certain of the sections, a sample-removal port.

Test Results

A number of tests were conducted at the conditions summarized in Fig. 7. Some tests were conducted with 100 percent R12, others with 100 percent R114, and others at various mixtures of the two refrigerants. In most cases, the refrigerant flow rate was approximately 0.023 kg/s which corresponded to the flow rates experienced in the system studies from which the data in Figs. 2 and 3 were obtained.

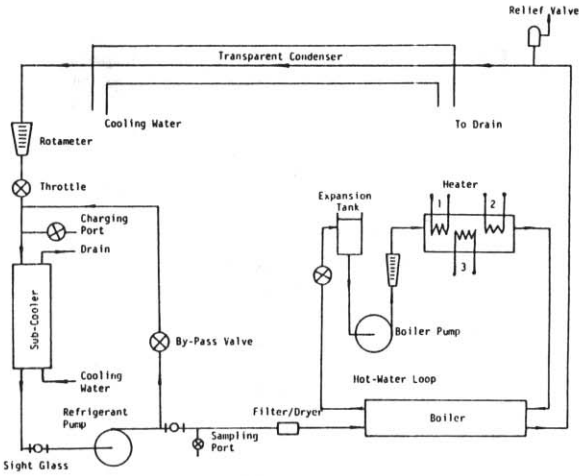


Fig. 5 Schematic diagram of the condenser test loop.

Figure 8 shows temperature profiles along the tube from the temperature of entering superheated refrigerant of approximately 45°C and leaving as subcooled liquid in the 17 to 20°C range. The expected constant temperature profile for either R12 or R114 alone essentially prevailed, although there was a slight drop in temperature of condensing R114 because of the higher drop in pressure (as Fig. 9 shows) and the greater change in saturation temperature for a given change in pressure. Figure 8 shows the progressive drop in temperature expected during the condensation of mixtures with the greatest drop occurring with the 47 percent R114 test.

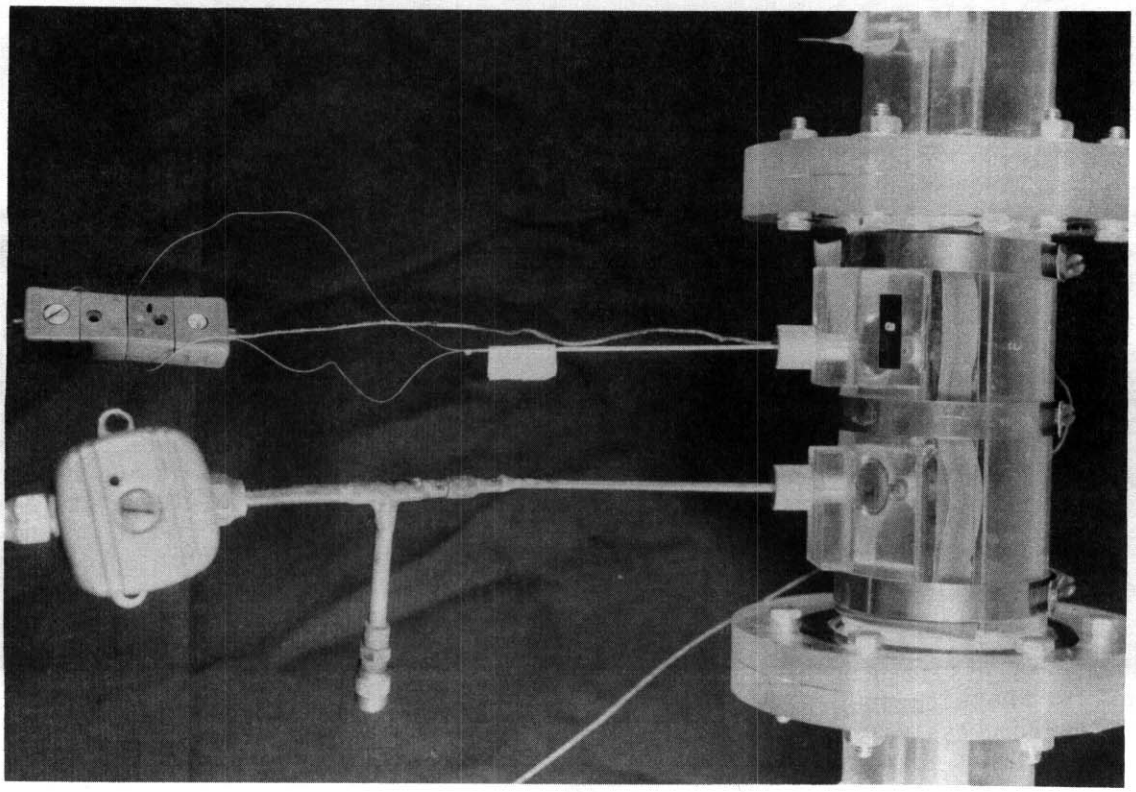


Fig. 6 Test section within the plastic outer tube. The section includes a pressure tap, thermocouple probe, and sampling tap.

	Test No.	% R-114	Refrig. Flow Rate (Kg/s)	Average Heat Flux $\frac{kW}{m^2}$	Condenser Entering Refrig. temperature (°C)	Condenser Entering Pressure (kPa Gauge)	Cooling Water Entering Temp. (°C)	Boiler Flow Rate (kg/s)	Condenser Flow Rate (kg/s)
ries 1	3	46.7	0.0228	4.22	45.1	478	14.9	0.238	0.295
	4	38.4	0.0233	4.37	45.1	503	15.1	0.245	0.297
	5	29	0.0227	4.24	45.1	543	14.6	0.243	0.296
	6	20	0.0232	4.33	44.8	579	14.8	0.238	0.299
	7	10	0.0227	4.25	45.0	610	14.4	0.241	0.295
	8	100	0.0238	4.14	45.0	208	14.3	0.236	0.297
	9	0	0.0230	4.32	45.1	639	14.5	0.236	0.299
	10	50	0.0239	4.44	45.1	420	13.9	0.235	0.300
	11	40	0.0237	4.44	45.1	475	14.0	0.234	0.299
	12	30	0.0236	4.45	45.0	537	14.3	0.232	0.299
ries 2	13	20	0.0227	4.25	45.0	575	14.5	0.233	0.292
	14	10	0.0229	4.29	45.0	593	14.0	0.233	0.302
	15	100	0.0243	4.23	44.9	193	13.9	0.233	0.296
	16	0	0.0231	4.32	45.0	638	13.8	0.232	0.299
	17	48.4	0.0241	4.50	44.9	465	14.5	0.233	0.299
ries 3	18	29	0.0234	4.50	44.9		13.6	0.233	0.298
	19	10	0.0234	4.44	45.1	592	13.3	0.232	0.294
	20	0	0.0230	4.35	45.1	609	13.0	0.233	0.300

Fig. 7 Summary of test conditions.

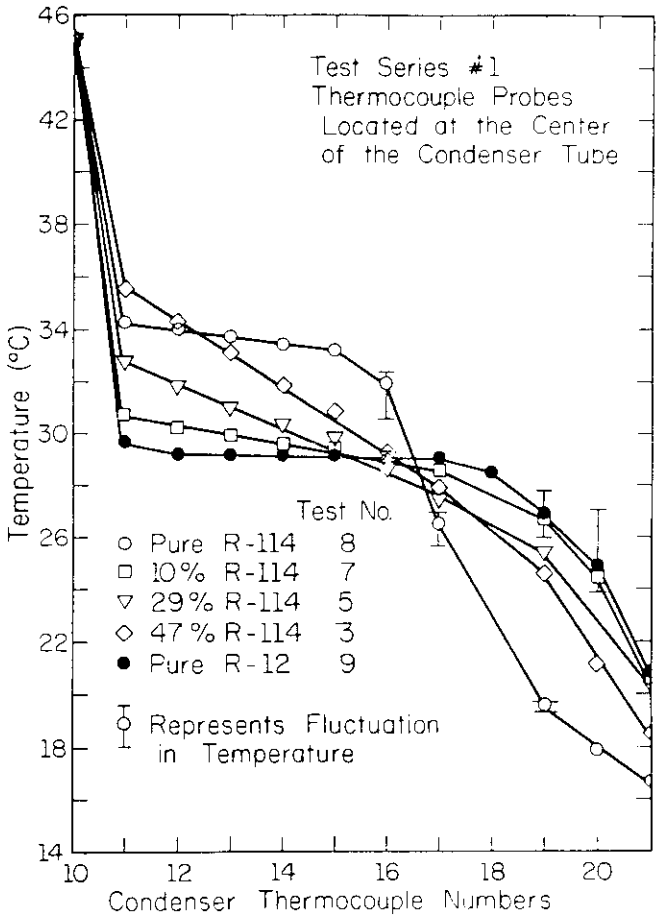


Fig. 8 Temperature profiles along the condenser (from selected tests in Series 1).

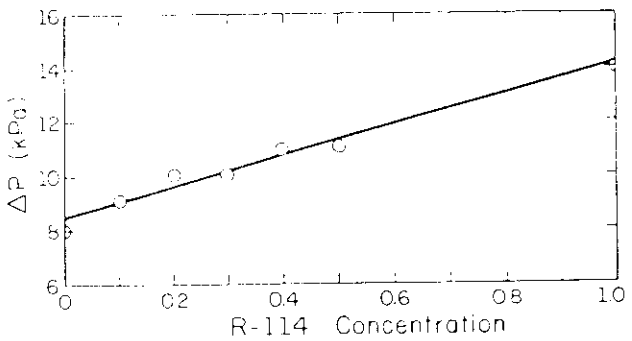


Fig. 9 Pressure drop through the condenser as a function of concentration.

Figure 9 shows a pressure drop of 8.5 kPa with 100% R12 and 14 kPa with 100% R114. The major reason for the progressively higher pressure drop (with essentially identical mass flow rates) lies in the higher specific volume of R114 than of R12, as Fig. 10 shows.

Flow Patterns shown by Photographs

One of the objectives of this phase of the project was to determine the regimes of condensation at various positions along the condenser tube and in a later phase correlate these regimes with magnitudes of the condensing coefficients which would have to be determined in a metal tube condenser where it would be possible to measure the surface temperature of the tube. Another potential benefit of knowing the flow patterns in a heat exchanger condensing refrigerant mixtures is that some ideas might emerge on how to avoid the reduction in condensing coefficients that occurs for the mixture.

property	R-12	R-114	% difference
sat. liquid density @ 37°C (kg/m^3)	1266	1417	11.9
specific vapor volume @37°C (m^3/kg)	0.020	0.044	120.0
surface tension @ 26.7°C (mN/m^2)	8.9	12.0	34.8
Viscosity, sat. liquid @ 37°C (Pa • s)	199	302	51.8
conductivity, sat. liquid @ 37°C (mW/m•K)	65.0	61.4	-5.5

Fig. 10 Comparative properties of R-12 and R-114.

Two sets of photographs are shown: one set is for 100 percent R12 in Figs. 11, 12, and 13, and the other set is for a 50 percent mixture shown in Figs. 14, 15, and 16. In each set of figures, the first group shows the early stages of condensation, the next the middle stages, and the third the final stages. The patterns change progressively from annular flow in which a thin film of liquid first clings to the inside surface around the entire circumference with the vapor flowing rapidly through the central

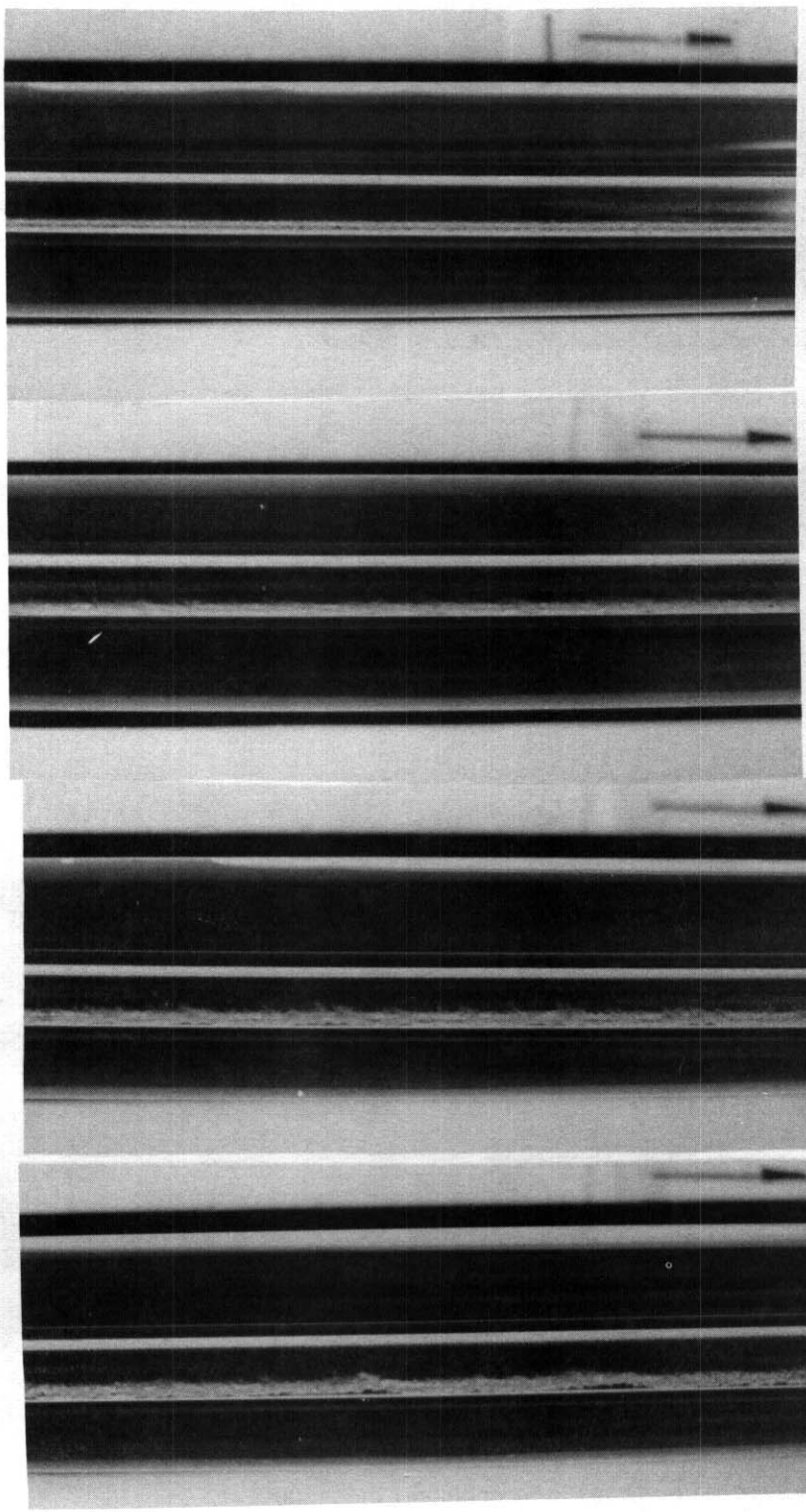


Fig. 11 Flow regimes in early stages of condensation, 100% R-12.

core. As more vapor condenses to liquid, the gravity effect carries more liquid along the bottom of the tube. Near the end of the condenser slug flow develops with liquid bridging the entire cross section trapping vapor between. Near the end of the condenser, the tube is almost completely filled with liquid and only occasional vapor bubbles move slowly at the top of the tube. It is possible to watch the size of these bubbles shrink as they move toward the end of the condenser.

In combination with temperature measurements, the photographs reinforce the fact of non-equilibrium of temperatures within a cross section of the tube. Refrigerant enters the condenser superheated, yet there is immediate condensation on the inside of the tube. Near the outlet of the condenser, the liquid is subcooled, yet exists simultaneously with vapor bubbles.

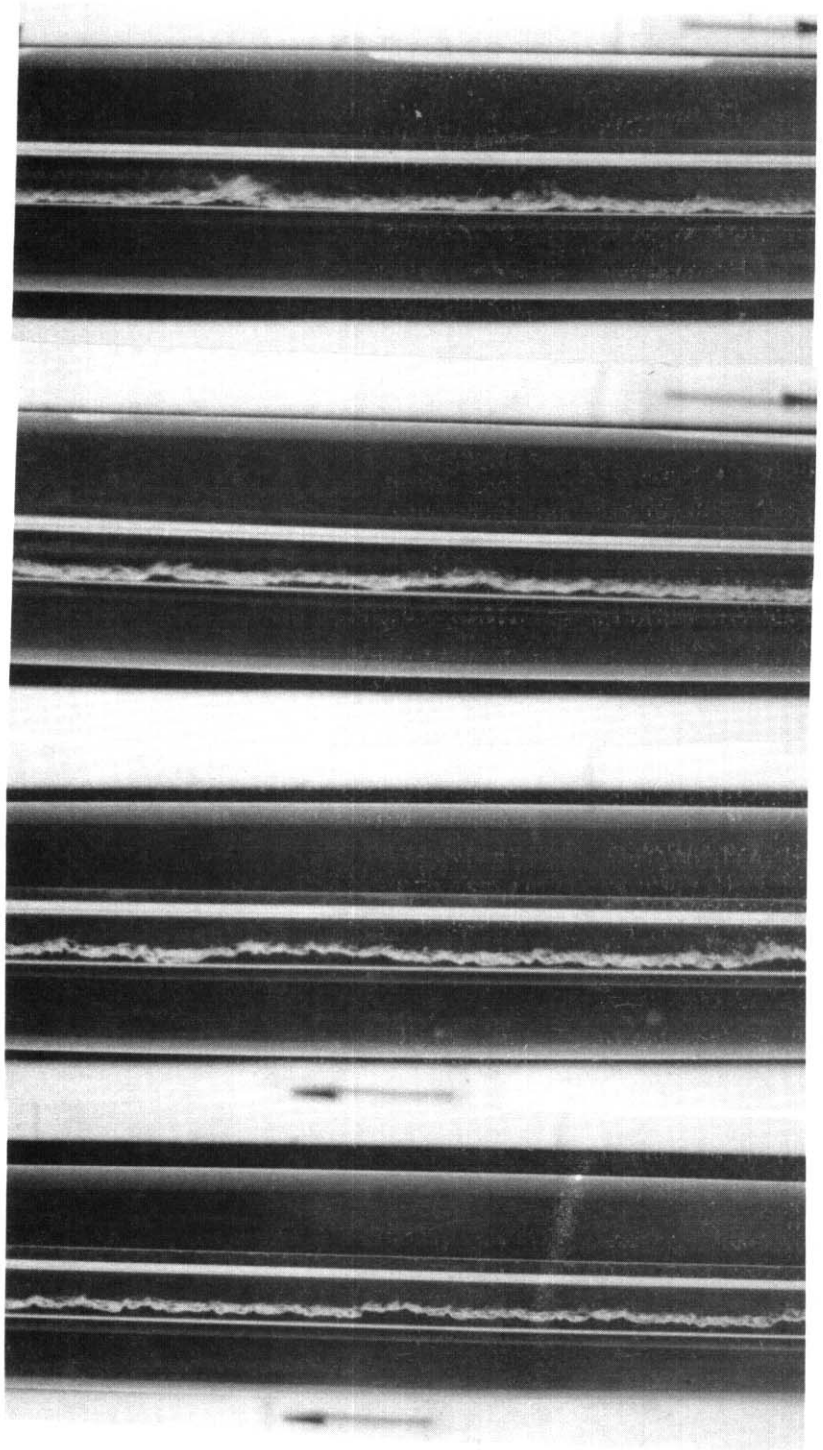


Fig. 12 Flow regimes in mid-stages of condensation, 100 R-12.

Temperature Fluctuations at a Given Position

Figure 17, 18, and 19 show temperatures measured by rapidly responding thermocouples placed in the refrigerant flow stream. Figure 17 applies to 100 percent R12, Fig. 18 to a 50 percent mixture, and Fig. 19 to 100 percent R114. A notion of where these thermocouples are located along the tube is available from Fig. 8 which indicates that they are near the end of the condenser. Thermocouples upstream in the condenser were not capable of sensing temperature fluctuations. The temperature fluctuations shown in Figs. 17, 18, and 19 occur because of the alternate passage of liquid and vapor. When vapor passes by the thermocouple, it first must dry off the thermocouple and then bring it to the vapor temperature. Because this process was necessary, it cannot be concluded that the thermocouple ever showed for the vapor a temperature that was truly as high as

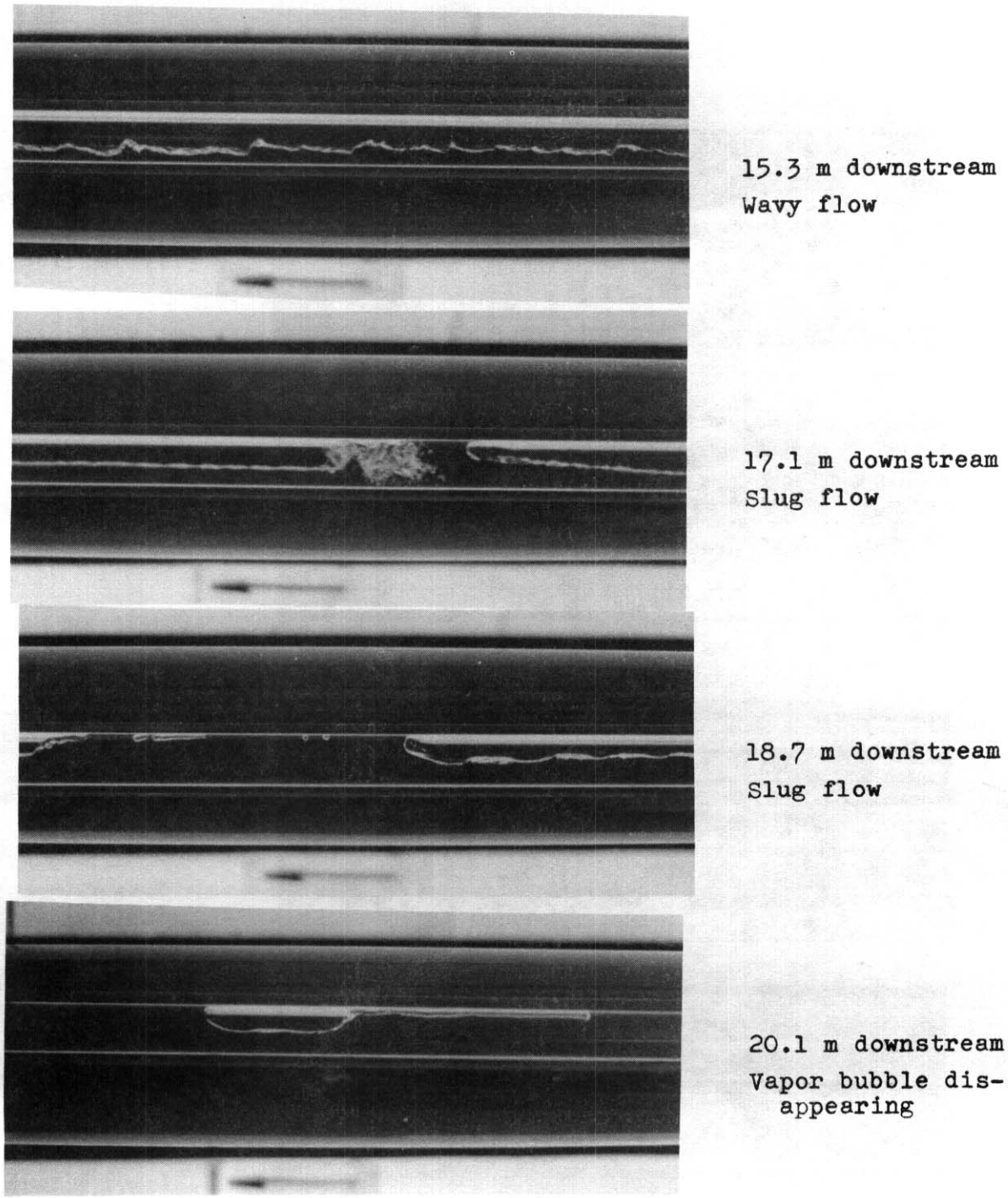


Fig. 13 Flow regimes in final stages of condensation, 100 R-12.

actual. Figure 17, for example, shows only minor variations in temperature at position No. 18 but what really occurred was a rapid alternation of liquid and vapor. The saturation temperature during this test was probably about 25°C as indicated by the low temperature shown by thermocouple No. 20 and verified by a saturation temperature of 24.2°C based on the pressure of 639 kPa. A mean temperature of vapor and liquid at position No. 18 is approximately 28.5°C. When the duration of the vapor periods and the liquid periods was longer, the difference in vapor and liquid temperature shows up more pronounced.

The largest fluctuations in temperature between vapor and liquid occurred with the 100 percent concentrations and the 50 percent mixture showed lower magnitudes of fluctuations.

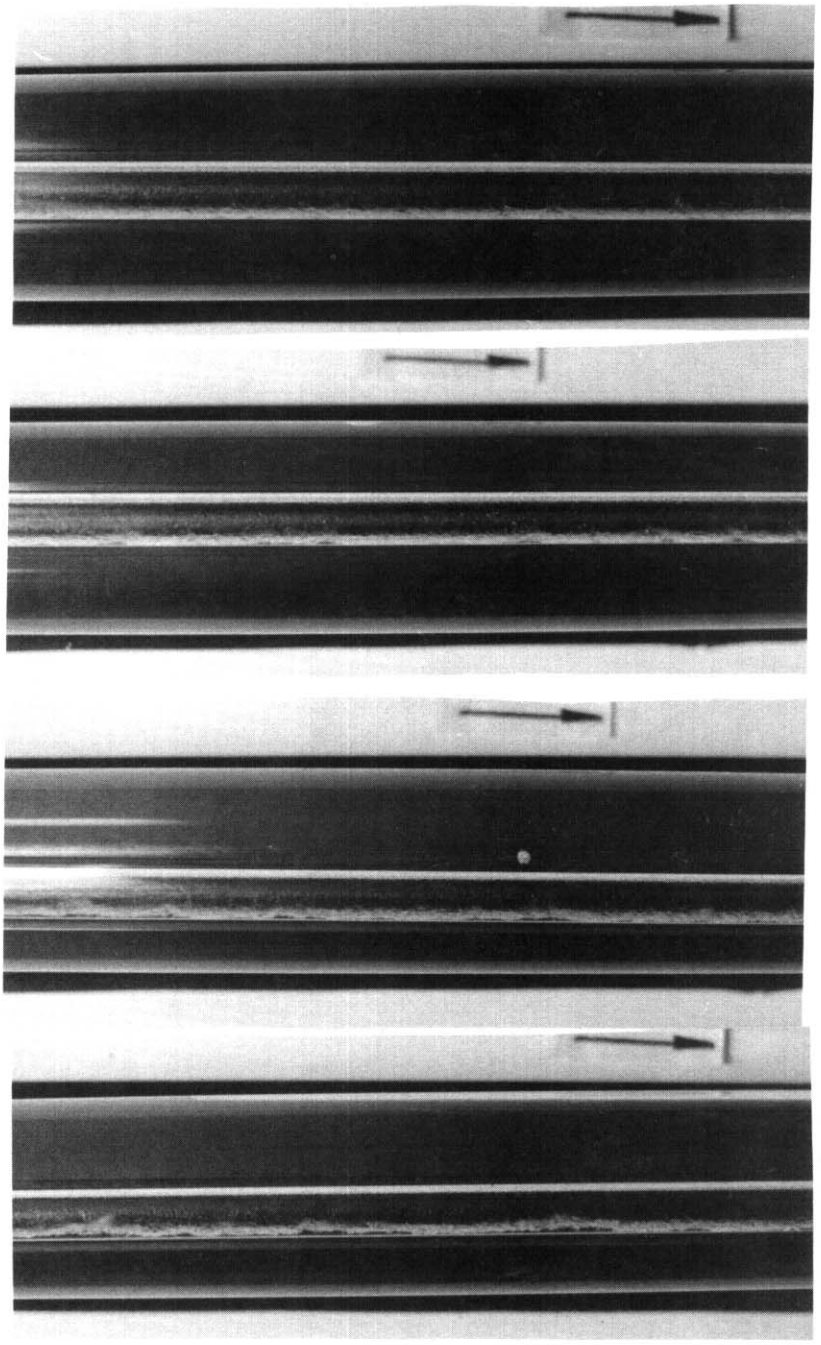


Fig. 14 Flow regimes in early stages of condensation, 50% R-12/R-114.

Chromatograph Analyses

Figure 20 and 21 refer to gas-chromatography results of several samples of liquid taken during tests. Figure 20 is the typical pattern of data coming from the graphical output of the instrument. The area under the curves (R12 on the left and R114 on the right) indicates the concentrations of the two constituents. The upper graphs in Fig. 20 are for a standard sample and the lower one from a sample of the condenser. The areas must be corrected for densities of the constituents.

Figure 21 shows a few of the results of the chromatograph tests and for reference the concentration as measured from the mass charge for that particular test. The samples were of liquid at the bottom of the condenser tubes at several locations--one just past the midpoint of the condenser (position 6) and positions 8 and 9 further downstream. In tests 10 and 17, there does seem to be a change in concentration between the several sampling positions and it is in the direction to be expected--the concentration of R12 in the liquid progressively in-

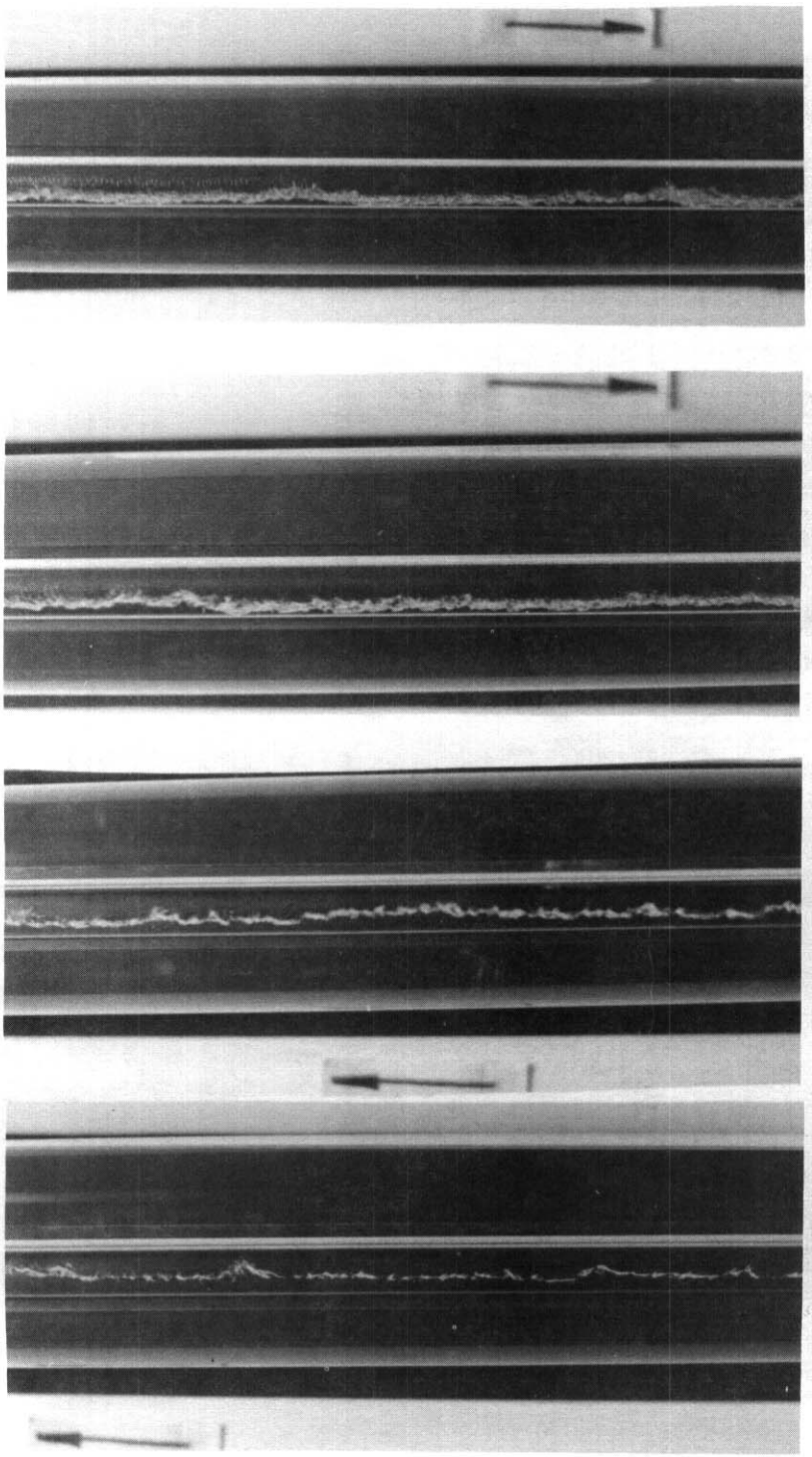
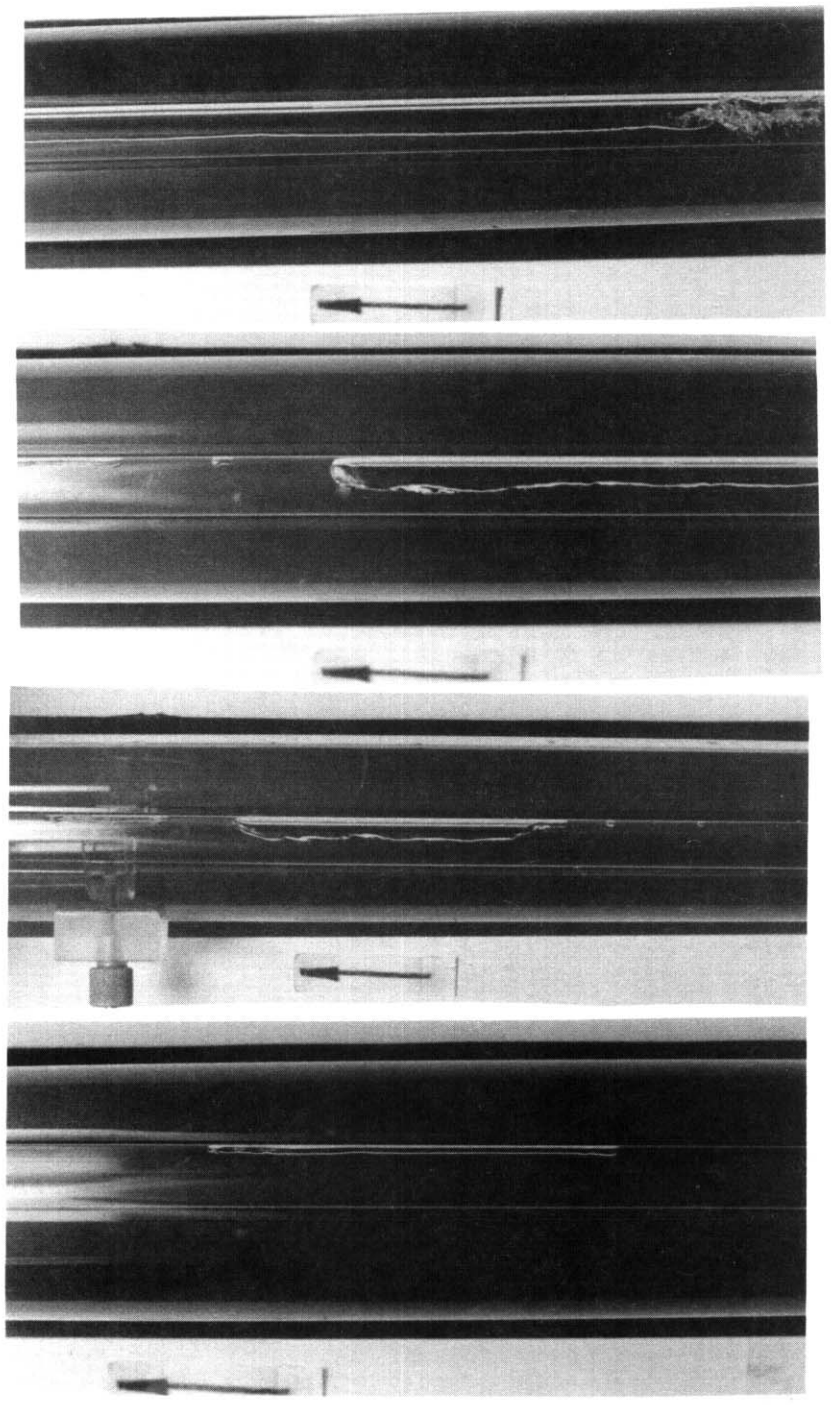


Fig. 15 Flow regimes in mid-stages of condensation, 50% R-12/R-114.

creases toward the overall mixture concentration as the condensation process continues. The experimental technique of extracting the sample was improved with experience, but a more refined technique with extremely small samples still is needed. Also, it would be valuable to be able to extract vapor-only samples early in the condenser.

Conclusions

The work so far provides a base for the planned continuation of measuring condensing coefficients with the same mass flow rates, heat fluxes, and temperature levels as used in the glass tubes. The photographic indication of regimes will assist in correlating the data.



15.3 m downstream
Stratified/slug flow

17.1 m downstream
Slug flow

18.7 m downstream
Slug flow

20.1 m downstream
Small vapor bubble

Fig. 16 Flow regimes in final stages of condensation, 50% R-12/R-114.

Even the work thus far gives some insight into the flow regime where the mixture causes the greatest degradation of coefficients. The regime occurs in the mid-range of the condenser where a distinct separation of vapor and liquid occurs. In this region, it is possible for the vapor to move on ahead (slip) such that additional non-equilibrium of concentration compounds the normally occurring gradients of temperature. The ultimate objective is to take measures to prevent the drop in heat-transfer coefficient of a mixture compared to a single substance and one approach would appear to maintain higher turbulence in the mid-range of condensation.

References

1. W. F. Stoecker and C. I. McCarthy, "The Simulation and Performance of a System using an R-12/R-114 Refrigerant Mixture," ORNL, 1984.

2. E. Kornota, "Flow Regimes of Refrigerant Mixtures Condensing Inside Tubes," M.S. Thesis, University of Ill. at U-C., 1984.

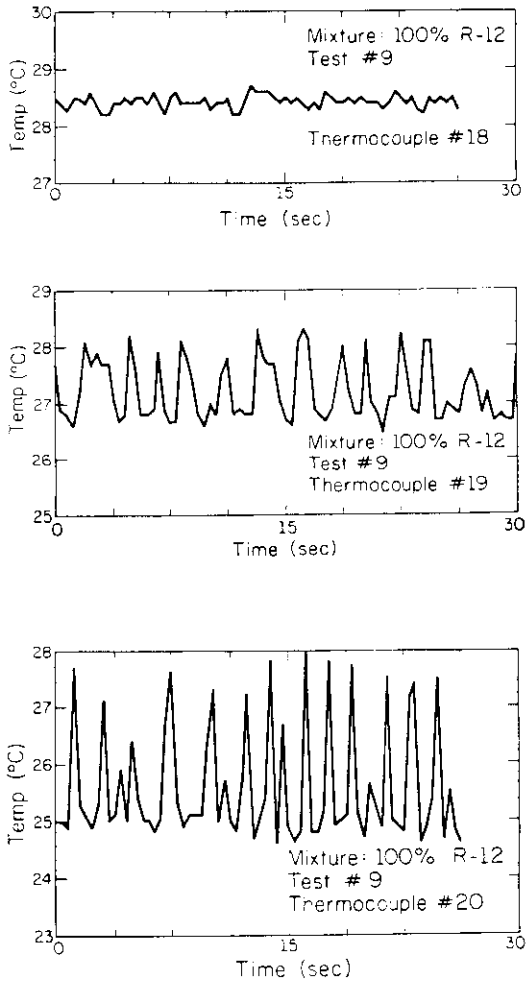


Fig. 17 Temperatures indicated by fine-gauge thermocouple, final stages of condensation, 100% R-12.

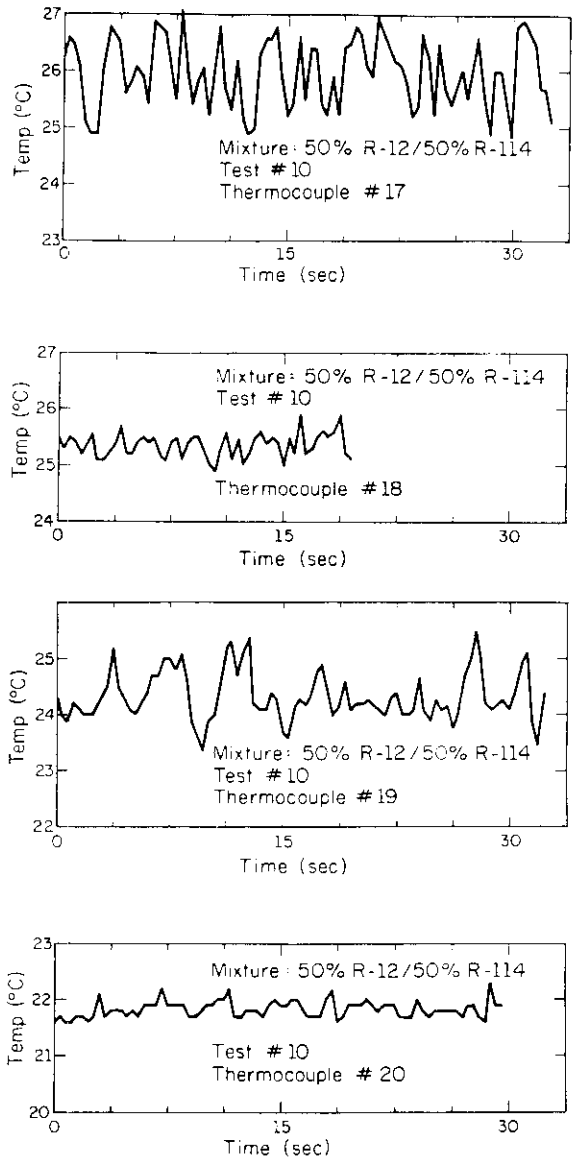
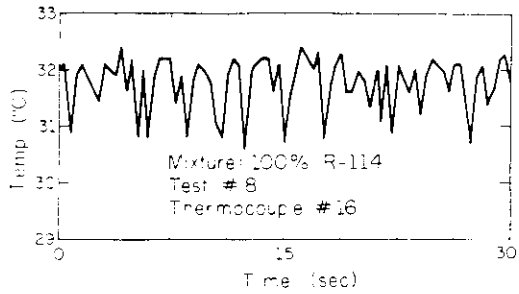
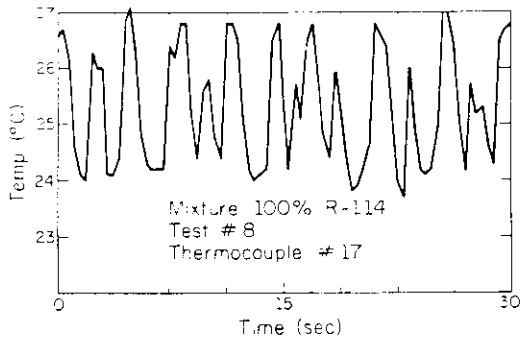
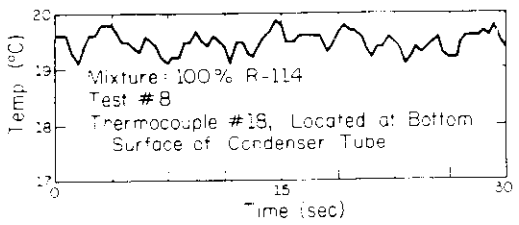


Fig. 18 Temperatures indicated by fine-gauge thermocouple, final stages of condensation, 50% R-12/R-114.







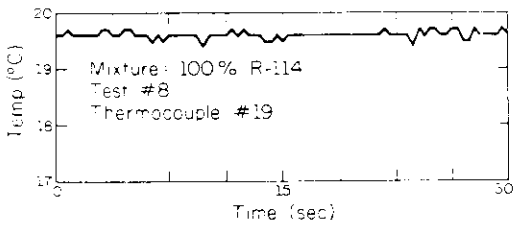
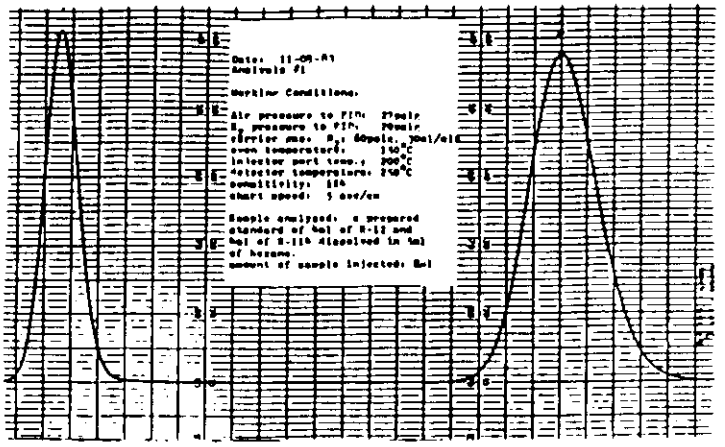
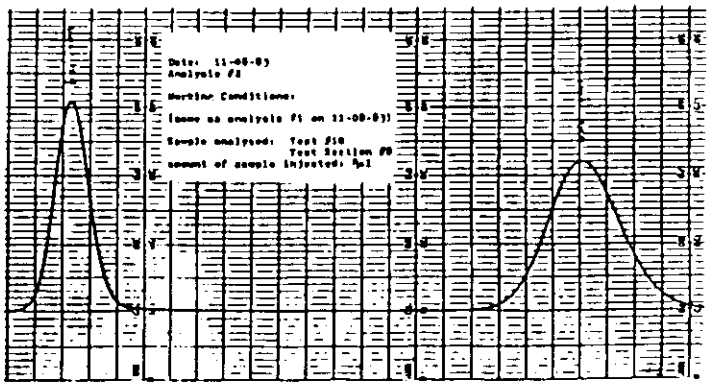


Fig. 19 Temperatures indicated by fine-gauge thermocouple, final stages of condensation, 100% R-114.



Analysis #1 Standard Prepared with Equal Volumes of R-12 and R-114



Analysis #2 Test #10, Test Section #8
Result: 47.5% R-12

Fig. 20 Typical chromatography trace of a standard sample and a test sample.

Date of Analysis	Test Number	% R-12 Charged	Sampling Location	Result (% R-12) A	B	Difference B-A
10-28-83	7	90	pump	91.5	1	
			Test Sec. #8	90.8		
11-08/09-83	10	50	pump	44.7	1	
			Test Sec. #6	37.8		
			Test Sec. #6S	37.4		
			Test Sec. #8	47.5		
			Test Sec. #9	55.3		
			Test Sec. #9S	55.1		
12-01/02-83	17	51.6	pump	57.6	61.7	4.1
			pump C	52.6	56.8	4.2
			Test Sec. #6	43.2	47.4	4.2
			Test Sec. #6C	41.1	45.3	4.2
			Test Sec. #9	48.7	53.0	4.3
12-07/08-83	19	90	pump #1	90.3	92.4	2.1
			pump #2[2]	90.7	92.8	2.1
			pump #2C	90.3	92.4	2.1
			pump #3	90.7	92.7	2.0
			pump #3C	90.7	92.6	1.9
			pump #4	90.5	92.6	2.1
			pump #4C	90.4	92.4	2.0

1 Not available.
2 The sample was prepared a day before the analysis.

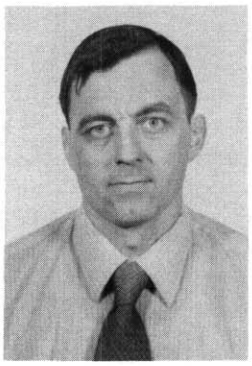
Fig. 21 Summary of chromatographic measurements.

The Performance of a Residential Heat Pump Operating with a Non-Azeotropic Binary Refrigerant Mixture—An Interim Report

David A. Didion and William J. Mulroy

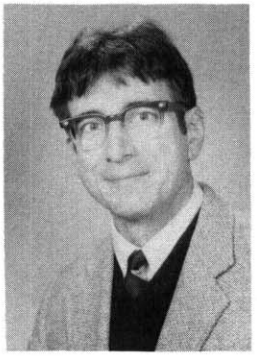
National Bureau of Standards

Gaithersburg, Maryland 20899



David A. Didion rejoined the National Bureau of Standards in 1971 as a project engineer in the Thermal Engineering Systems Section. He served as analyst for energy programs in the Office of the Director during 1972-73 as part of a work-training program. In December 1974, he was appointed chief of the newly formed Mechanical Systems Section. Since its inception, this group's activity has been dominated by energy performance analyses and standards development for building equipment and heating, ventilation, and air-conditioning (HVAC) systems.

By 1980, the group's activities had expanded to the point that the HVAC systems and controls work had to be reorganized. This allowed Dr. Didion to concentrate on the thermal performance of machinery and components. This work has now developed to include a major interest in the use of non-azeotropic mixtures as a working fluid for heat pumps. In conjunction with the mixtures work, an emphasis on the measurement of two-phase flow heat-transfer coefficients has occurred, and a new laboratory has been developed. In support of this development, Didion spent the summer of 1983 on a work assignment at the National Engineering Laboratory (Heat Transfer Centre) in Scotland.



William J. Mulroy has been a mechanical engineer with the National Bureau of Standards (Center for Building Technology) for 20 years. Most of his experience has been with refrigeration systems; however, for 4 years he worked in performance evaluation of furnaces and boilers. He is currently assigned to the heat pump evaluation procedure development project for the U.S. Department of Energy where he is responsible for evaluating the "off the shelf" heat pump using a non-azeotropic mixture as a working fluid. He is also participating in the revision of the Tri-Services heat pump purchase specification and is performing laboratory tests in support of the mixed/matched heat pump rating procedure.

THE PERFORMANCE OF A RESIDENTIAL HEAT PUMP OPERATING WITH A NON-AZEOTROPIC BINARY REFRIGERANT MIXTURE - AN INTERIM REPORT

David Didion
William Mulroy
Building Equipment Division
Center for Building Technology
National Bureau of Standards
Gaithersburg, Maryland 20899

Abstract

This report presents results of laboratory measurement of the performance change of a substantially unmodified residential heat pump designed for R22 when charged with a non-azeotropic, binary mixture of R13B1 and R152a. Results are presented for various sizes of fixed expansion devices. The effect of gliding temperature in the saturation zone was found to be small. The effect of composition shift by flash distillation in the accumulator was found to measurably improve low temperature heating performance. It was further observed that some system modification (such as the addition of a receiver) could have further enhanced this low temperature heating performance improvement.

Introduction

The original goal of this study was to determine the energy performance benefits of a non-azeotropic binary refrigerant mixture in an unmodified residential heat pump. Since the "benefits" would necessarily entail a comparison against a reference, the single component refrigerant, R-22, was selected as a basis. The mixture used in this study, R13B1/R152a, was chosen primarily because of convenience in that it was one of several touted in the literature[1,2,3] as having potential for improving performance and it was readily available in unlimited quantities from the manufacturer at no cost. Thus it is to be considered as an example of any of a number of mixture possibilities and the quantitative values of this study are significant only in that they indicate the relative merits of the different characteristics a mixture may offer (i.e., variable temperature phase change, variable capacity, bilevel temperature condensation/evaporation, etc.) Other mixtures may offer more or less quantitative values depending on their properties. Which mixture is the optimum for heat pumps is the subject of another study.

As this study progressed it became apparent that a simple substitution of a refrigerant mixture for R22 was not a fair evaluation of the relative benefits of the mixture. After all, the machinery components (i.e., heat exchangers, expansion device, compressor, etc.) had been sized originally with a particular quantity and type of refrigerant in mind. Substituting another quantity or type of refrigerant would necessarily cause the same machinery to perform differently not only due to the thermodynamic properties differences, but also due to the fact that the machinery has probably become over or under sized. In fact, the first question that one is faced with in any refrigerant substitution process is what is the quantity of the new charge. As a result of these phenomena the first part of this study focused on charge optimization and expansion device size determination.

The performance criteria of prime consideration are capacity and efficiency. Of the two, capacity is often the more important for heat pumps in that for many applications (i.e., northern climates), an increased capacity at lower temperatures will reduce the

resistance heat demand such that the seasonal efficiency will be significantly improved. Other performance criteria relating to reliability were also felt to be important and thus discharge pressure and temperature, suction pressure, pressure ratio, and pressure difference were measured and reported as well.

Background

Non-azeotropic mixtures are those which exhibit a temperature change as they either condense or evaporate due to the fact that they are comprised of components with different normal boiling points. Binary mixtures are those of two components. This report deals exclusively with a binary, non-azeotropic mixture of refrigerants 13B1 and 152a as applied to a residential heat pump.

Several advantages have been claimed for non-azeotropic mixtures in refrigeration systems. Among these are improved thermodynamic efficiency (as a result of gliding temperatures in the saturation zone, i.e., Lorenz vs. Carnot cycle), multi-temperature level evaporators or condensers (useful for frozen and produce compartments in refrigerators and for domestic hot water in heat pumps and air conditioners), and refrigerant composition change which alters the working media's capacity to suit load requirements. The last of these is the principal concern of this study.

The refrigerant mixture used in this study is one that has been recommended by W. D. Cooper of the Du Pont Freon® Products Laboratory for heat pump applications[1,2,3]. In comparison to refrigerant 22, refrigerant 13B1 is more volatile operating at higher pressures for equivalent saturation temperatures, produces more capacity in a typical constant volumetric flow type refrigeration system as a result of having a vapor density great enough to more than compensate for a smaller latent enthalpy change, and, in a typical cycle,[4] is less efficient. Refrigerant 152a, in comparison, is of lower volatility, has a lower operating pressure, and is a lower capacity refrigerant with a relatively large latent enthalpy and good efficiency at higher evaporator temperatures. The mixture of the two, 65 percent by weight of R13B1, was chosen by DuPont to obtain the best possible capacity and efficiency during air conditioning operation.

The greater the volatility difference between the two refrigerants, the easier it is to shift the composition of the circulating refrigerant from its initial value. This shifting may be effected passively by flash distillation in a conventional accumulator or it may be done actively with an added distillation system. The shifting process for the simple heat pump of this report was a result of liquid flooding into the accumulator during low ambient temperature of the heating mode only.

When evaporation or condensation of a non-azeotrope takes place, the concentrations of the two components is different in the vapor and liquid phases. In particular, the vapor is always richer in the more volatile component and the liquid is always richer in the less volatile component in comparison to the initial composition. The extreme composition differences occur at the beginning and end of the state change process. For example, as shown in figure 1, a liquid mixture of 65% 13B1/35% 152a by weight ($x_{\ell,a}$ in figure 1) at 90 psia, when it first reaches its bubble temperature, will be in equilibrium with vapor at an approximate composition of 82% 13B1/18% 152a ($x_{v,a}$). An intermediate point approximately half way through the two phase region is shown in figure 1 at b to result in equilibrium between liquid 50% 13B1/50% 152a ($x_{\ell,b}$) and vapor 74% 13B1/26% 152a ($x_{v,b}$). The final example of figure 1 shows nearly completed evaporation at the dew line, c, where a liquid of 36% 13B1/64% 152a ($x_{\ell,c}$) is in equilibrium with vapor at the original 65% 13B1/35% 152a ($x_{v,c}$) composition.

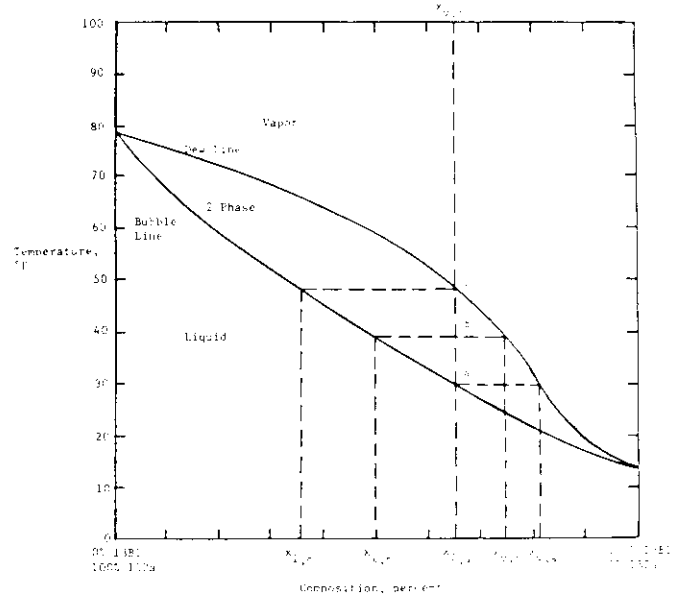


Figure 1. Dew and bubble temperatures for mixtures of R13B1 and R152a at a pressure of 90 psia.

When this process occurs in the evaporator of a heat pump whose outdoor temperature is dropping, a transient condition of flooding through to the accumulator will occur, as it would with any refrigerant, until a lower circulating mass flow is established. However, for the mixture, the refrigerant liquid in the accumulator will grow richer in the least volatile component causing the refrigerant remaining in the system to be enriched in the more volatile refrigerant. In the case of the mixture used in these tests, flooding into the accumulator would increase the percentage of R13B1 circulating in the system and consequently increase low temperature heating capacity resulting in a substantial system

efficiency increase by the reduction of need for electric resistance heat as is shown in figure 2.

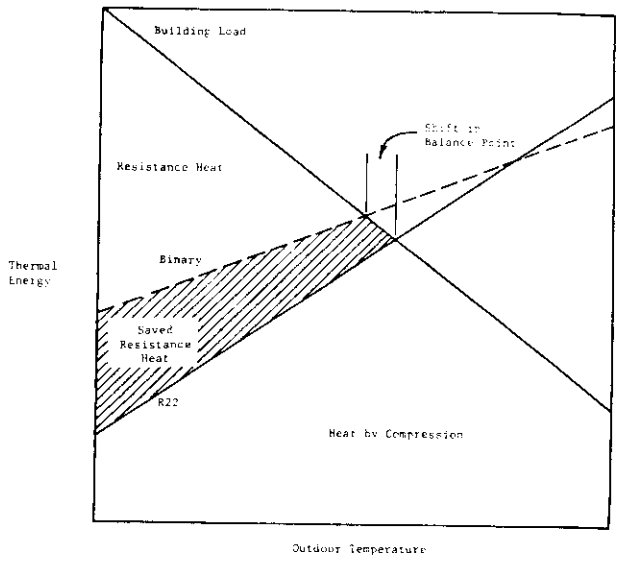


Figure 2. Energy savings as a result of increased low temperature compression system capacity reducing the need for "make-up" resistance heat.

Cooling Mode Test Results

The capacity and efficiency test results at the DoE "A" and "B" rating conditions are summarized in figure 3. The NBS steady-state heat pump simulation model, HPSIM[7] was used to verify the shape of these curves for the R22 tests and did confirm the observation that as the expansion device becomes overly restrictive the 82°F and 95°F capacity approach each other and eventually cross. The shape of these graphs for the binary refrigerant are quite similar showing the above described narrowing of the 95°F and 82°F capacity difference for small orifices* and showing simultaneous peaks in capacity and efficiency curves.

The orifice giving the highest capacity at 95°F has a 30% smaller area for the R22 tests and a 6% greater area for the binary tests than that giving the greatest capacity at 82°F. For general comparison, the optimum expansion device and charge were chosen as those giving the best efficiency at the 82°F rating point. These orifices were the 0.0781 in I.D. for R22 and the 0.0820 in I.D. for the binary mixture.

*The heat pump tested was originally equipped with capillary tubes. For the tests described in this report it was modified to use interchangeable orificew elements 1/2 in length. The charging criterion employed was 10'F of superheat leaving the evaporator at the DoE "A" test condition.

For both refrigerants, best performance occurs with minimal high-side pressure while still maintaining some subcooling entering the expansion device.

That the percent latent cooling is unaffected by the different refrigerants and their associated different temperature patterns through the evaporator are shown in figure 4. This suggests that the binary mixture's

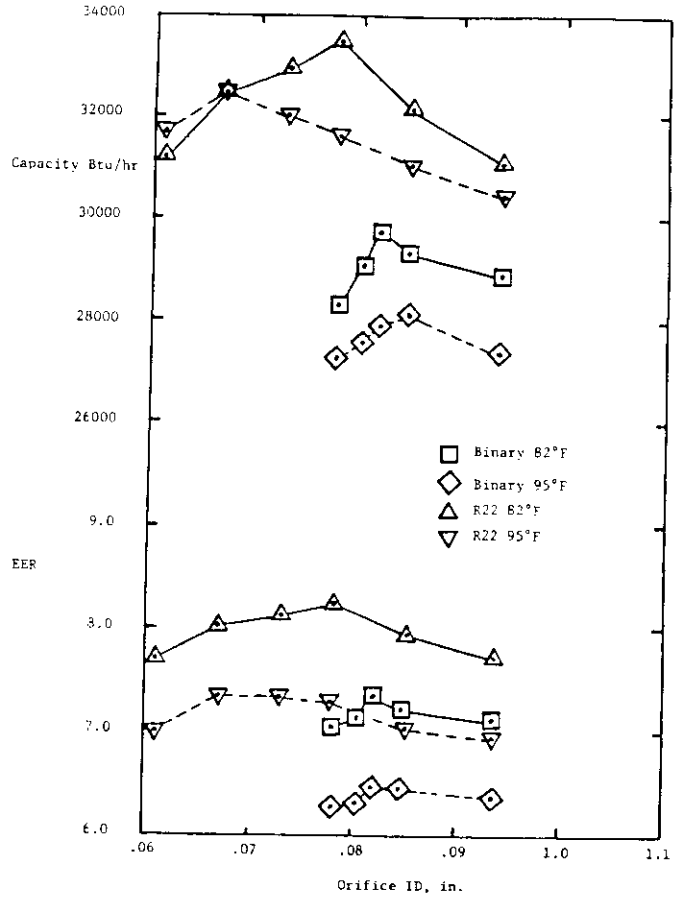


Figure 3. Summarized cooling mode test results.

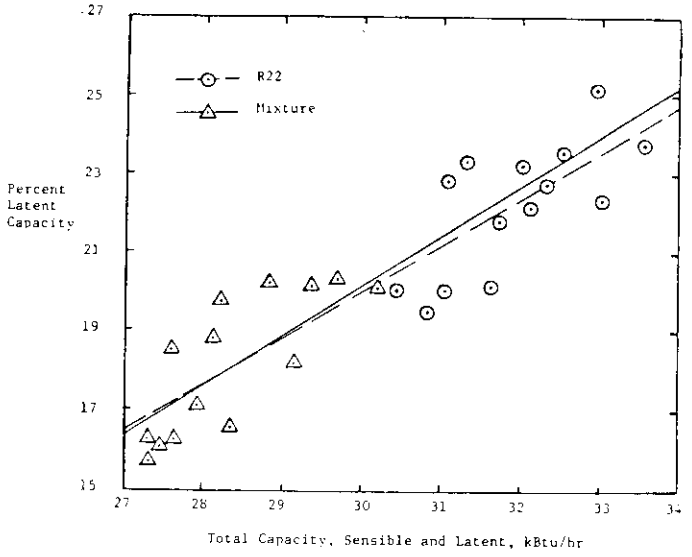


Figure 4. Comparison of latent capacity as a function of total capacity.

decrease in total capacity also included a decrease in latent capacity due to an average increase in evaporator temperature. The apparent continuity of this plot for the two refrigerants as shown by the near coincidence of their respective least square linear regressions indicates that there is no other substantial difference between them in latent cooling. This latent capacity difference is not surprising since the evaporator was sized for R-22 and is thus oversized for the mixture.

Plots of return bend temperatures through the evaporator for a typical test are shown in figure 5. The saturation, bubble and dew lines shown are calculated from pressures at the evaporator inlet and outlet, assuming a

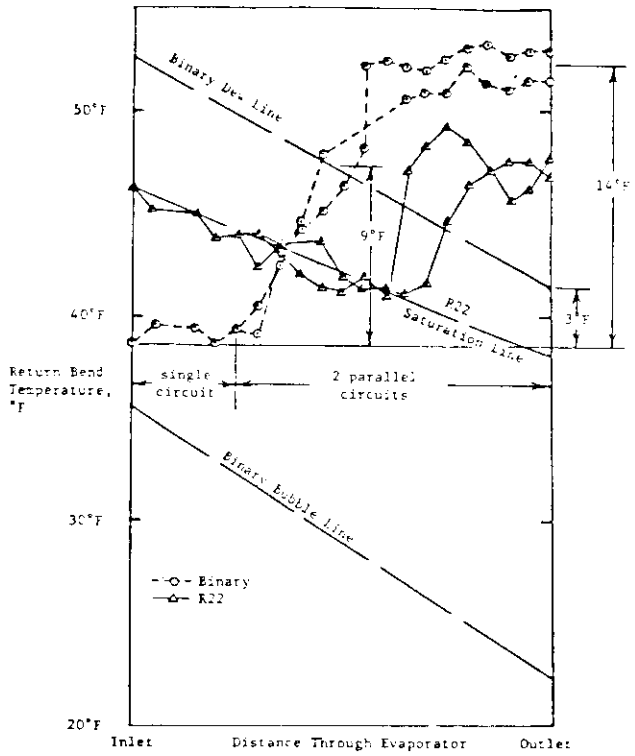


Figure 5. Evaporator (indoor coil) return bend temperatures during a typical 95°F, DoE "A", cooling test.

straight line gradient through the coil. The refrigerant 22 temperatures can be seen to follow the saturation temperature line predicted from these pressures until the refrigerant reaches a quality of one at which point it rapidly reaches the imposed conditions of 10°F of superheat. The binary refrigerant temperature gradient results from a tradeoff between the rate of heat gain and pressure drop. The most rapid period of temperature rise occurs when the coil circuitry splits in half reducing the mass flow accordingly and simultaneously is routed to the front of the coil where the highest heat gain rate occurs. The temperature rise through the binary evaporator, is not as great as would be expected from comparing the dew temperature to the bubble temperature at this pressure (18°F). This is due to the fact that while the changing liquid composition is causing an increase in fluid temperature, the flow pressure drop is causing a decrease in saturation temperature at the fluid. The actual temperature rise through the evaporator for the binary refrigerant was approximately 14°F and that of the R22 approximately 1°F. This is, however, a result of the 10°F superheat criteria reinforcing the temperature gradient for the binary refrigerant and cancelling that of the R22. If the refrigerant left the evaporator as saturated vapor, the temperature gradient, inlet-to-outlet, for the binary refrigerant would be only 3°F and that of the R22 would be -8°F. The actual temperature gradient for the binary refrigerant while it is in saturation is indicated in figure 5 as 9°F. The temperature gradient of this mixture through the evaporator tends to be of the same order of magnitude as that for R22. In general the value for a binary mixture would be greater or less than that for R22 depending on the temperature difference between its bubble and dew lines, pressure gradient, and the amount of leaving superheat specified. Return bend temperatures are shown and discussed for a typical condensing operation and for a flooding evaporator in the following section on heating test results.

A principal reason for performing these expansion device optimization tests was the concern about the high pressures observed during earlier binary refrigerant tests which had used the capillary tube sized for R22. The improvement in observed pressure at the most severe of the cooling test conditions, 95°F, is summarized in Table 1.

Table 1. Comparison of System Pressures at the DoE 95°F, "A", Test Condition

Refrigerant	R22	R22	Binary	Binary
Expansion Device	Cap	0.78	Cap	.082
Discharge Pressure, psia	278	279	317	297
Discharge Temp., °F	232	242	221	208
Suction Pressure, psia	82.9	91.1	82.8	90.9
Pressure Ratio	3.35	3.06	3.83	3.27
Pressure Difference, psi	195	188	234	206

The increased suction pressure between the capillary tube and the orifice tests for

R22 is a reflection primarily of the change in test criteria to a lower value for the refrigerant superheat leaving the evaporator since the expansion device was essentially optimized in both cases. Optimization of the expansion device did reduce the binary pressure ratio to a value between the listed R22 values. The pressure difference remains slightly higher than for R22. Compressor discharge temperature, also listed in Table 1, is seen to be consistently better (i.e., lower) for the binary mixture than for R22.

Heating Mode Test Results and Discussion

Heating mode test results are summarized in figure 6 for those tests performed at the 47 °F DoE rating condition and in figure 7 for those tests performed at the 17 °F DoE rating condition. The refrigerant charge used for these tests was that which had produced the best efficiency at the 82°F cooling rating condition as shown in figure 3. For the binary tests this resulted in a charge of 9.5 lbs with the 0.0820 in. I.D. orifice. For the R22 tests a charge of 7.17 lbs was used with the 0.0780 in. I.D. orifice.

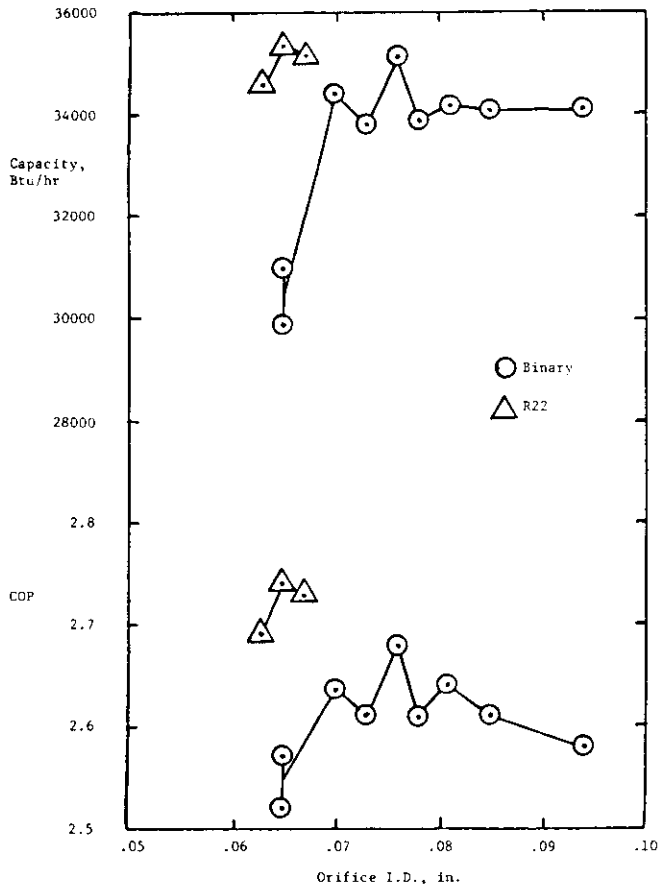


Figure 6. Summarized 47 °F heating mode test results.

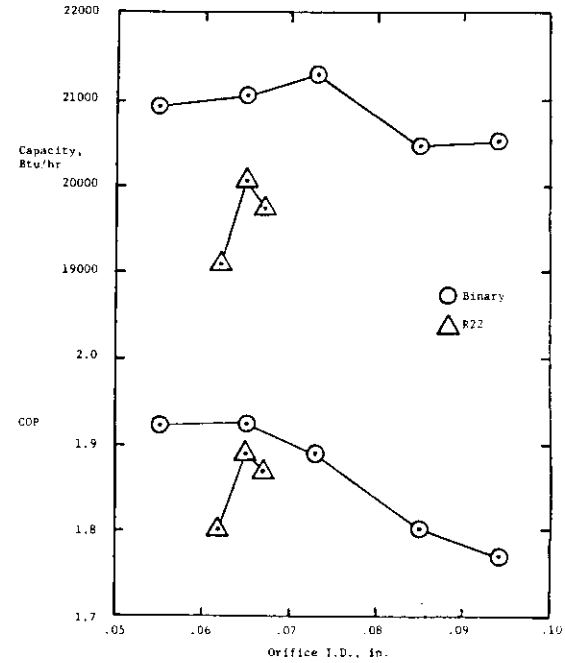


Figure 7. Summarized 17 °F heating mode test results.

The outstanding observation from these figures is the insensitivity of binary capacity to orifice size. An exception was the smallest orifice at 47°F which presumably failed to completely flood the evaporator. It appears that the capacity gain from concentration shift is being offset by the capacity loss resulting from the loss of subcooling (incomplete condensation) leaving the condenser. This point is illustrated in figure 8 in which the enthalpy change through the condenser is compared to the theoretial value that would have resulted had complete condensation taken place.

The enthalpy used in the above analysis and the tabulated mass flow for most of these tests was calculated by iterating the temperature at the turbine flowmeter to give a quality for the measured pressure which was consistent with the volumetric flow measured by the turbine meter and with the enthalpy determined from air side capacity.

Although the capacity was insensitive to orifice size the efficiency showed a clear peak indicating that the shift to greater concentrations of R13B1, offset the capacity loss but not the efficiency loss.

As was observed in the cooling mode tests, the binary discharge pressure is higher and the discharge temperature lower than would be the case for R22.

In figure 9 and 10, return bend temperatures through the evaporator and condenser for

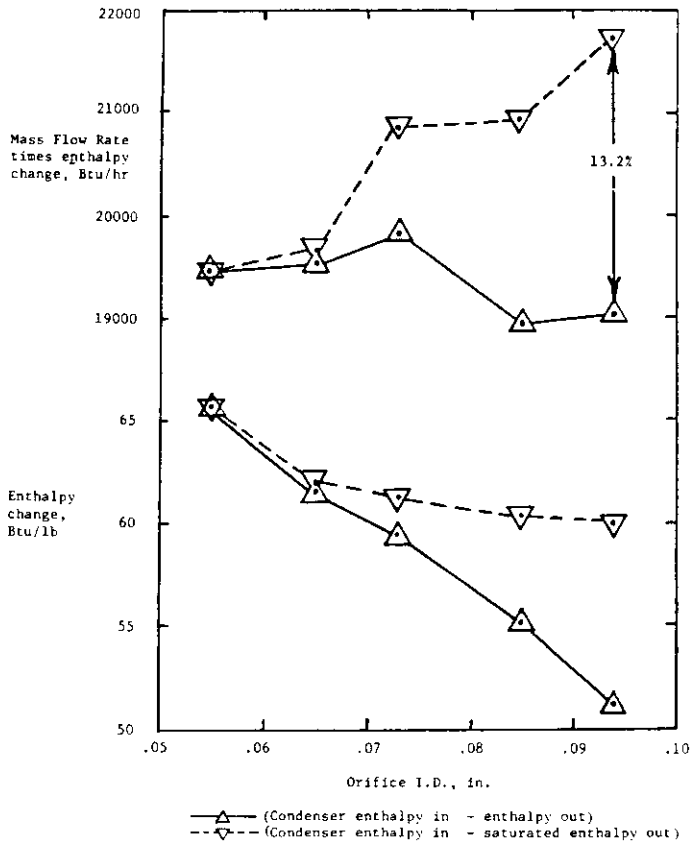


Figure 8. 17 °F heating condenser enthalpy change of binary mixture.

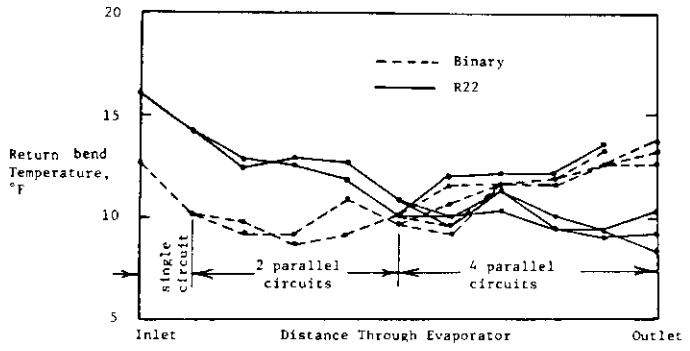


Figure 9. Evaporator (outdoor coil) return bend temperatures during a typical 17 °F heating test.

R22 and the mixture are compared for 17°F flooded evaporator operation. In this case the refrigerant mixture is seen to give temperatures within the evaporator (figure 9) that rise or fall depending on the relative effects of pressure drop and heat transfer with a leaving temperature approximately 1°F higher than its entering value, whereas the R22 temperatures steadily fall leaving the evaporator 5°F lower. The heat transfer rate to this coil was more uniform than the indoor coil in that its circuitry ran constantly zig-zagged from front to back instead of proceeding row-by-row. Thus at each branch point, the mass flow through a circuit is halved resulting in an approximate halving of the flow pressure drop rate and doubling of the heat transfer rate per pound.

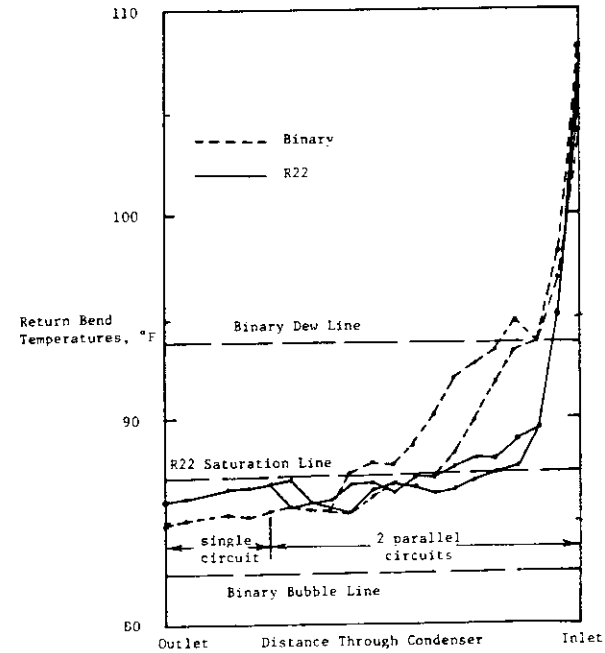


Figure 10. Condenser (indoor coil) return bend temperatures during a typical 17 °F heating test.

In the condenser (figure 10) where the pressure drop and the saturation temperature gradient reinforce, the binary mixture can be seen to go through approximately a 9 °F change in the two-phase zone in comparison to an approximately 3°F phase temperature gradient due to pressure drop alone for R22. It should be noted that in a condenser in comparison to an evaporator, although the temperature and pressure gradients reinforce, the pressure gradient is less because of the lower average void fraction of the refrigerant and the .pa temperature gradient will also, as a result of material properties (i.e., dew temperature closer to bubble temperature) be less.

Conclusions

The most highly publicized attribute of non-azeotropic mixtures, that of variable temperature phase change and its associated minimal heat transfer irreversibilities in a counter flow heat exchanger with a sensible heat sink or source, seems to have rather limited application for the air-to-air heat pump. Test results in this study confirmed our model predictions that the saturation temperature decrease due to the refrigerant flow's static pressure decrease virtually offsets the increase in composition shift temperature in the evaporator. Thus, the bulk temperature profile of the mixture in the evaporator is in fact more constant than the R22 which has only the saturation temperature decrease phenomenon occuring. Of course, in the condenser the mixture's composition temperature and satura-

tion temperature shifts are additive and a rather significant refrigerant phase change temperature decrease results. However, current outdoor coil designs are typically single row wraparound resulting in an almost purely cross-flow heat exchange. Furthermore, the reversing action of the heating/cooling heat pump confounds the problem of redesigning the heat exchangers to be of marginal cost effectiveness.

Mixtures, in general, should be evaluated on a reliability basis as well as energy performance. For the vapor compression cycle this usually means consideration of the discharge pressure and temperature levels as well as the pressure difference and ratio. For a heat pump the high stress region resulting from high pressures and temperatures will occur under steady state conditions during the highest condenser operating conditions.

In Table 1 some system temperatures and pressures for the cooling mode are summarized. The 65%/35% unshifted mixture can be seen to produce suction pressures substantially the same as R22. For this unshifted mixture the discharge pressure is approximately 18 psia higher for the mixture than R22 (and consequently the suction to discharge pressure ratios are different). The pressure ratio was, on average, 7 percent higher than R22. The discharge temperature was, on average, 19°F lower than with R22. Thus, when using a constant superheat as a charging criteria, this mixture does not produce pressures significantly higher than R22 and its lower discharge temperatures would likely more than compensate for the slightly higher pressures in affecting system life.

The energy performance of the 65%/35% (R13B1/R152a) mixture, charged in the cooling mode to give a similar superheat as R22, was evaluated. The system then operated such that the accumulator was dry throughout the cooling mode which meant no shift in composition was occuring.

In the heating mode, it was observed that flooding into the accumulator caused about a 10 percentage point increase in 13B1 but, as a part of normal fixed orifice system operation, the subcooling was decreased until completely lost thus offsetting most of the potential gain in capacity and efficiency at lower temperatures. A possible solution for this loss in subcooling is an additional volume (e.g., a receiver) on the high pressure side which could transfer refrigerant to the accumulator without starving the condenser.

The mixture's performance (i.e., capacity and COP) are shown in figure 11 relative to R22. All of the tests had an initial charge composition of 65% 13B1/35% R152a which was the operating composition at 47°F and which shifted to 76% 13B1/24% 152a at 17°F. This composition shift caused a cross-over in part-load capacity performance with R22; that is, the lower the evaporator temperature requirements the more available mixture capacity there is, relative to R22. The empirical results also suggest a cross-over in mixture efficiency performance relative to R22. These "cross-overs" are understandable when the basic properties of the refrigerants are considered.

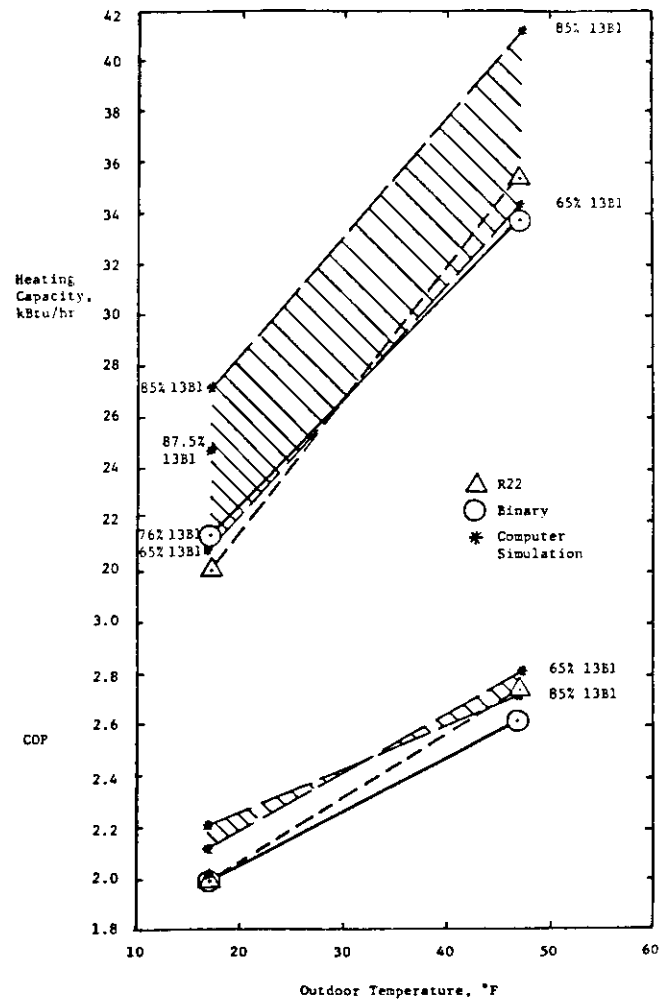


Figure 11. Comparison of binary refrigerant energy performance to R22.

A review of the p-h diagram[9] for the mixture indicates that while the vapor dome spread (i.e., the potential latent heat) decreases with increasing percentage of 13B1, the suction density of the mixture increases enough to more than compensate due to the weight of the 13B1 molecule. Thus, an increase in 13B1 should result in an increase in the product of ($\rho\Delta h$) to which capacity is proportional. Also, in going from an outdoor condition of 47°F to 17°F, the suction pressure does not have to decrease as much to attain the necessary evaporator temperature when the composition increases in 13B1 as it would if it were not shifting. The discharge pressure also increases with shifting composition but at a

slower rate than the suction pressure so that the (P_d/P_s) term, which work is proportional to, does not increase as much per unit capacity when composition shift occurs; thus efficiency also begins to "cross-over".

Also illustrated in figure 11 are the results of the NBS mixtures/heat pump simulation model. The crosshatch band represents the region of likely operation for this mixture. The capacity region is bounded on the bottom by a 65% 13B1/35% 152a circulating composition with a 0°F superheat and a 2°F subcooled condition for both the 47°F and 17°F. The top bound is an 85% 13B1/15% 152a composition with the same operating conditions. These bounds represent the optimum capacity this heat pump could do with the circulating compositions shown. In order to proceed through this region on any path it is necessary to have full control of the expansion device, the composition and the charge. For example, the path between the 85% 13B1 point and the 87.5% 13B1 point represents the fixed orifice and charge case; that is, the case that represents the current heat pump operation. This means where these parameters (i.e., expansion device and composition) are not fully controlled and thus, the resulting part load performance is degraded from the optimum; because, for this case, there is an associated loss of subcooling when liquid storage occurs in the accumulator. This loss of subcooling might well be mitigated with an increase in initial charge.

In summary then, there appears to be an inherent trade-off when operating an unmodified heat pump with mixtures so as to increase capacity at lower temperatures. This trade-off is that as the evaporator pressure decreases and refrigerant begins to flood through to the accumulator and and thus allowing the composition shift to occur, the subcooling on the high pressure side also decreases causing a decrease in the available latent enthalpy difference. A possible solution to this problem might be to increase the amount of initial charge. This would cause an increase in head pressures which could be a problem in the cooling mode. A possible solution to this excessive head pressure problem would be to incorporate a receiver in the system in the branch line just ahead of the expansion device for the outdoor coil. Thus, when this coil is acting as an evaporator (i.e., heating mode) the stored refrigerant would be released into the circulating system allowing for more composition shift to occur. Such a solution will be pursued as one of the items of study during the next phase of this project.

References

1. Cooper, W. D., and Borchardt, H. J., "The Use of Refrigerant Mixtures in Air-to-Air Heat Pumps," XVth International Congress of Refrigeration, Proceedings, Vol. IV, 1979.

2. Cooper, W. D., "The Use of Mixed Refrigerants in Air-to-Air Heat Pumps," ASHRAE Transactions, V. 88, Pt. 1, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1982.

3. Freon Products Division, Freon®, 13B1/152a Refrigerant Mixture for Heat Pumps, RT-74, E. I. DuPont De Nemours and Co., Inc., Wilmington, Delaware.

4. "ASHRAE Handbook of Fundamentals", American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1981.

5. Parken, W. H., Beausoliel, R. W., and Kelly, G. E., "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump," ASHRAE Transactions, Vol. 83, Part 1, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1977.

6. "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment, ASHRAE Standard 37-69, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1969.

7. Domanski, P. and Didion, D., "Computer Modeling of the Vapor Compression Cycle with Constant Flow Area Expansion Device," NBS BSS 155, National Bureau of Standards, Gaithersburg, Maryland, 1983.

8. Domanski, P., "Modeling of a Heat Pump Charged with a Non-Azeotropic Mixture Refrigerant," NBSIR 84-, National Bureau of Standards, Gaithersburg, Maryland, 1984 (to be published).

9. Domanski, P. and Didion, D., "Equation of State Based Thermodynamic Charts for Non-Azeotropic Refrigerant Mixtures," American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, Georgia, (to be published January 1985).

Stirling Cycle Heat Pump Studies

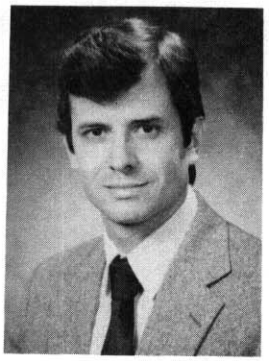
Norberto Domingo

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

Norberto Domingo is a research engineer in the Energy Division of Oak Ridge National Laboratory. Since joining the Laboratory in 1978, he has been involved in the development of advanced heat exchangers for power plant and process applications, in particular geothermal and ocean thermal energy conversion. His recent areas of special research interest include regenerative-cycle heat pumps and diesel and gas turbine heat recovery.

Before joining the Laboratory, Domingo worked several years for Pratt & Whitney Aircraft as a test engineer on experimental turbofan engines. He holds a B.S. degree in aerospace engineering and an M.S. degree in mechanical engineering from the Georgia Institute of Technology. He has published a number of papers on the enhancement of refrigerant condensation.



STIRLING CYCLE HEAT PUMP STUDIES*

N. Domingo
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Objective

The objective of this project is to evaluate a Stirling cycle heat pump (not a Stirling-engine-driven heat pump) with characteristics that are well suited for residential space conditioning applications.

Background

The Stirling cycle heat pump (SCHP) is well suited to high temperature lifts, as evidenced by its common use for cryogenic applications. However, its potential applicability in the residential space conditioning field has only recently been explored. Unlike the conventional vapor-compression heat pump cycle, the capacity of the SCHP is insensitive to source temperature levels. Also, the SCHP uses a non-chlorofluorocarbon (CFC) working fluid. A Stirling heat pump offers simple cycle reversal and defrost, capacity modulation by pressure control, and potential durability and reliability. While conventional thinking is that the vapor-compression system is cheaper and more efficient than the Stirling for moderate temperature lifts, there is no evidence that a serious effort to develop a competitive Stirling has ever been made.

Summary

This project is organized into two parts: analytical activities and subcontract activities. Major accomplishments for the analytical activities include (1) a computer simulation of a kinematic SCHP to determine the effects of source temperature, gas pressure, drive speed, dead volume, and working-space isothermalization on performance[1] and (2) a comparative analysis of SCHP performance and loss terms obtained with two different computer models.[2] Simulation results indicate some performance trends and trade-off relationships that may be general enough to cover a wide range of machine configurations. Results from the comparative analysis show that overall performance agreement between the models is fairly good, but for the individual losses, agreement is very poor.

Three subcontracts were awarded during the past year: a subcontract with Energy Research and Generation, Inc., (ERG) to verify experimentally the effects of isothermalizers on Stirling heat pump performance; a subcontract with Sunpower, Inc., to provide Oak Ridge National Laboratory (ORNL) with technical information and assistance in the feasibility analysis of the SCHP; and a subcontract with the University of Tennessee to modify and apply a near-isothermalized, second-order computer code to analyze the SCHP process of compression and expansion between the two limiting cases of isothermal and adiabatic processes.

Technical Accomplishments

Investigation Of Performance Trends And Trade-Off Relationships

Computer simulations were performed on a kinematic SCHP (modified from the GPU-3 heat engine mode) using the National Aeronautics and Space Administration (NASA) Lewis Research Center (Le-Rc) third-order code,[3] which employs nodal analysis to solve numerically the governing differential equations.

Parametric analyses determined the effect of changes in working gas, gas pressure, drive speed, and heat transfer fluid inlet temperature on heating coefficient of performance (COP) and heat capacity. Sensitivity of heat exchanger dead volume and isothermalization of the working spaces were also investigated.

Figure 1 shows that indicated heating COP (ex-mechanical losses) increased moderately with increasing fluid inlet temperature, but heat output remained fairly constant. Also, with hydrogen, COP increased, but heat capacity decreased.

Figure 2 shows that increasing gas pressure or drive speed decreased the indicated COP but increased heat output. Doubling gas pressure decreased the COP by 8% and increased heat output by 100%. The degradations in COP are the result of increased pressure-drop losses in the machine. Improvements in heat output are the result of the increased mass of gas.

Increases in heat exchanger dead volume, which affects gas compression ratio and flow friction, increased COP but decreased heat output. The effect of isothermalization is evident from Fig. 3. The indicated COP increased 19 to 28% above the adiabatic case when the calculated gas-side heat transfer conductance was increased by 1000 in both the expansion and compression spaces. Although the results from this study were obtained for a fixed-machine configuration (the GPU-3), the trends found may be general enough to cover other configurations.

*Research sponsored by the Building Equipment Research Division, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

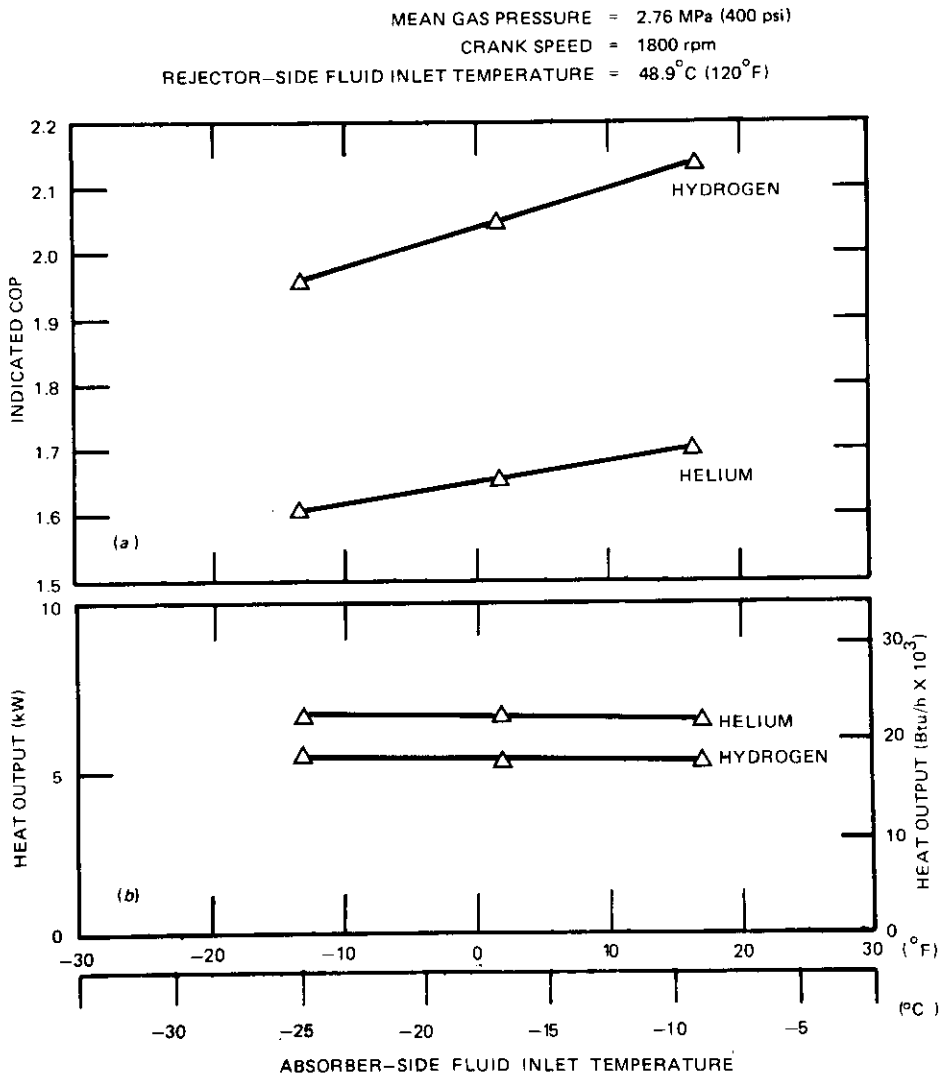


Fig. 1. Stirling heat pump performance for helium and hydrogen.

Comparative Analysis Using A Second-Order And A Third-Order Computer Model

To date, there is no strong evidence that third-order analyses (like the NASA code) are superior in accuracy to elaborate second-order analyses, which decouple losses. To determine the level of coherence between results predicted by the two types of simulations, the performance and loss terms of a SCHP were calculated with the NASA third-order code and compared with those obtained with the Martini ISO82 code.[4]

Parametric analyses on drive speed and gas pressure were carried out with both codes for fixed-source and sink-metal temperatures and working gas. Because one code accounted for more losses than the other and some losses were part of the basic heat calculations, it was not possible to make a one-to-one comparison of all performance and loss terms calculated for each model. However, much effort was made to provide consistency for the variety of terms compared.

Tables 1 and 2 from ref. 2 compare loss and performance terms obtained by the two methods of computation. Agreement between the overall predictions (Table 1) was -2 to 13% for indicated power (no mechanical losses), 18 to 29% for net heat output (includes conduction losses), and 10 to 20% for indicated heating COP (net heat output/indicated power), with the NASA predictions yielding the higher values. Poor agreement was found among the calculated pressure drop, mechanical friction, and conduction losses (Table 2). The NASA code yielded 53 to 69% higher values for pressure-drop losses and 23 to 111% higher values for mechanical losses. However, for the total conduction losses, the Martini code yielded values that were 260 to 440% times higher than the NASA predictions because the conduction paths considered by the two models were somewhat different, with the Martini model accounting for more conduction losses.

Figures 4 and 5 show a diagram of heat and power flows calculated with the NASA and Martini codes, respectively, for the same Stirling heat pump dimensions and operating conditions. Note that fair agreement is obtained between the overall performance predictions despite the poor agreement between the

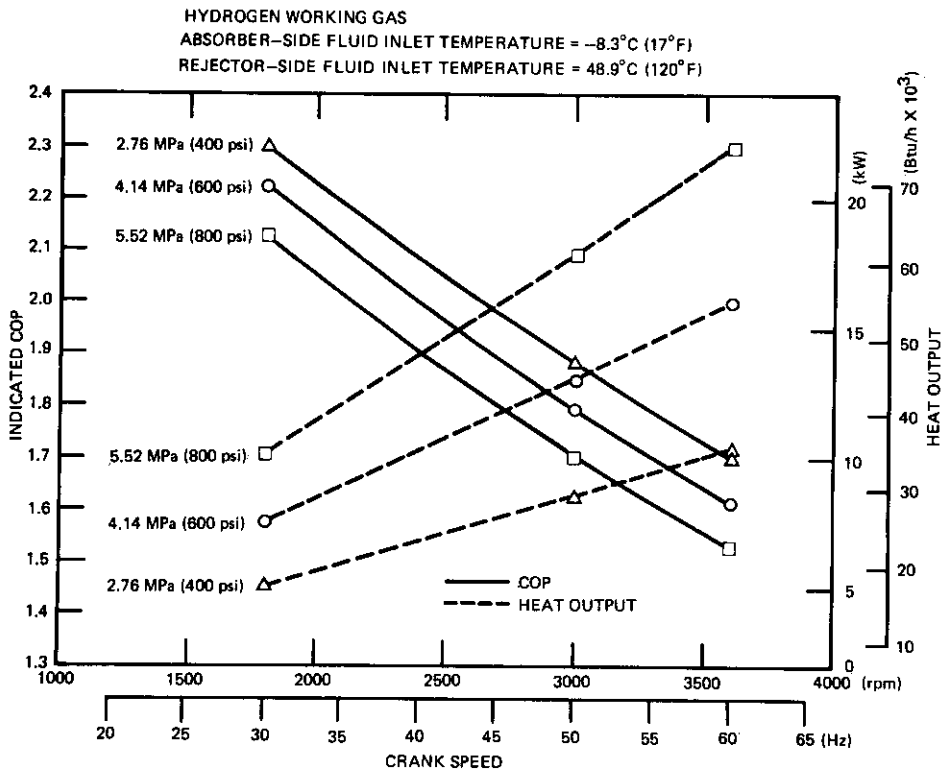


Fig. 2. Stirling heat pump performance as a function of crank speed and compression-space pressure.

loss predictions. Results from this study provide evidence that existing analytical tools need improvement and experimental verification in the area of loss mechanisms.

Subcontract Activities

There are three subcontracts managed by ORNL in support of this program: a component test project, a feasibility assessment project, and a computer model modification project. These individual subcontracts, along with the main features and accomplishments/status of each, are identified below.

Contractor: Energy Research and Generation, Inc.

Project: To verify experimentally the performance advantage of

1. isothermalized variable volume chambers and
2. a high performance regenerator matrix on a kinematic Stirling heat pump.

Task Activities: An experimental Stirling heat pump, utilizing isothermalizer (Thermizer) heat exchangers and a high performance (silicon carbide) regenerator, is to be tested over a range of heat input and output temperatures, gas pressures, pressure ratios, drive speeds, and phase angles for nitrogen and helium as the working gas. Next, heat pump tests will be conducted after the Thermizer heat exchangers are modified to operate adiabatically.

Accomplishments: Various mechanical and electrical problems have delayed start of Phase I tests. Extensive modifications to the heat rejection cooling loop have been made to correct heat transfer fluid flow instabilities through the heat rejection Thermizer.

Future Plans: Complete all testing, analyze data, and prepare final report.

Contractor: Sunpower, Inc.

Project: Provide technical input and assist ORNL in the feasibility assessment of the Stirling heat pump.

Task Activities:

1. Provide a description of Sunpower's heat pump design methodologies.
2. Identify and define technical problem areas requiring further R&D.
3. Describe improvements and/or new design concepts evolving from earlier proof of concept work.
4. Map heat pump performance for a system utilizing such improvements.

Accomplishments: Draft report in progress. Highlights from text include

1. Problem areas needing further R&D effort. These areas include heat exchangers, starting and load control systems, burner, and seals.
2. COP effects as function of burner, engine, and heat pump efficiency; fluid outlet temperature; and temperature drops across heat exchangers.

Contractor: University of Tennessee

Project: Modify the Stirling cycle, second-order computer model developed by I. Urieli of Sunpower, Inc., to account for finite heat transfer in the working spaces and analyze the cycle between the two limiting cases of isothermal and adiabatic conditions.

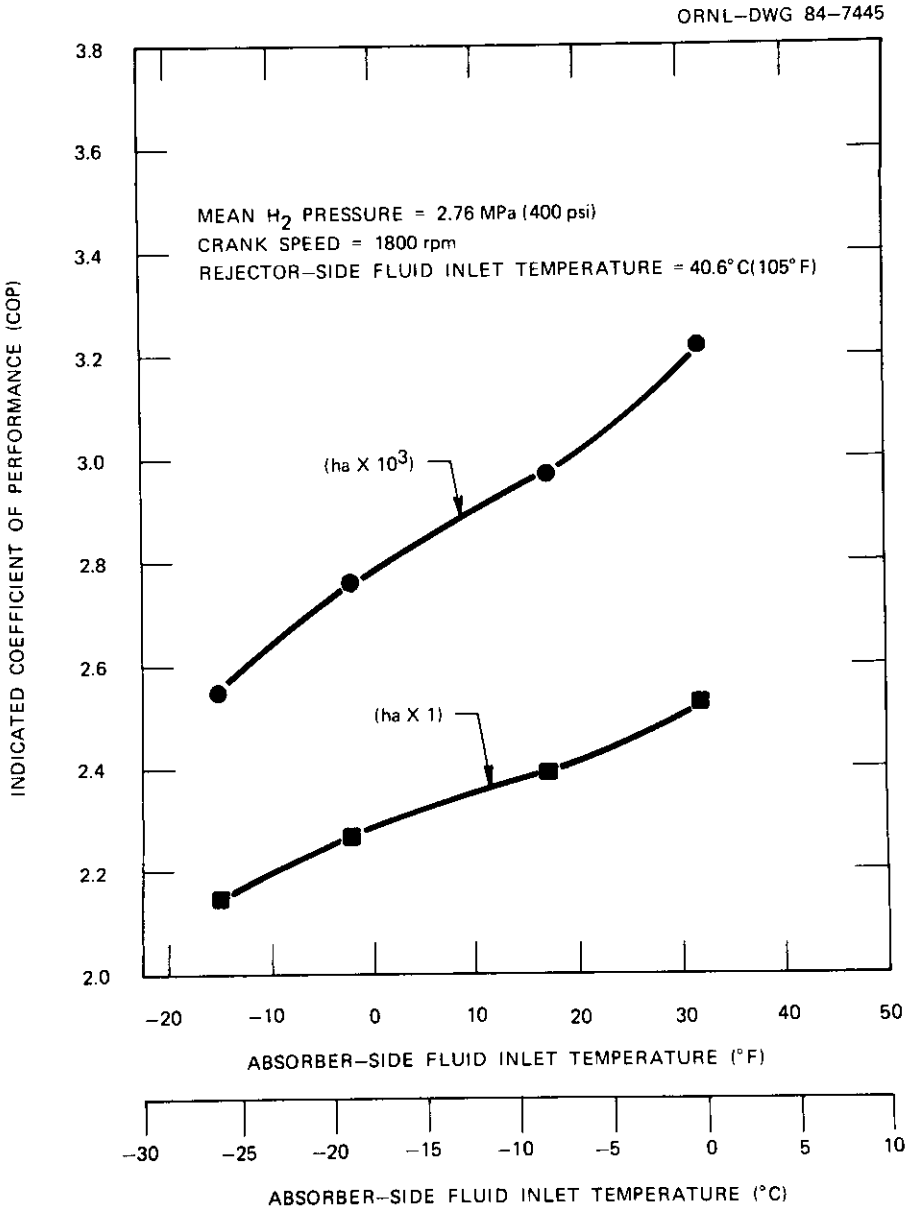


Fig. 3. Effect of isothermalization on heating COP.

Accomplishments: Computer code modification was completed. A sensitivity analysis between heat transfer coefficient and polytropic exponent for the general path equation $Pv^n = C$ has been carried out. Results indicate that the polytropic number for the working spaces approaches unity (n = 1 for isothermal process) when heat transfer coefficient is increased.

Future Plans: Validate isothermalized results obtained with the newly developed code against ERG's test results obtained with an isothermalized Stirling heat pump.

Future Activities

The major thrust of this project in the future will be aimed at completing a feasibility analysis with the assistance of Sunpower, Inc. If feasibility is established, a decision will be made to issue a request for proposal for hardware development.

References

1. N. Domingo, W. L. Jackson, and F. C. Chen, "A Theoretical Analysis of a Kinematic Stirling-Cycle Heat Pump for Space Conditioning Applications," ORNL/CON-149, Oak Ridge National Laboratory, March 1984.

2. N. Domingo, "Comparative Analysis of a Stirling Heat Pump Using a Second-Order and a Third-Order Computer Model," ORNL/CON-162, Oak Ridge National Laboratory, publication pending.

3. R. Tew, K. Jefferies, and D. Miao, "A Stirling Engine Computer Model for Performance Calculations," DOE/NASA/1011-78/24, NASA TM-78884, U.S. Department of Energy, July 1978.

4. W. R. Martini, "ISO82 A Fast Design Program for Stirling Cycle Heat Engines and Heat Pumps," final report for Oak Ridge National Laboratory, April 1983.

Table 1. Comparison of performance terms calculated by the NASA and Martini models

	Test point								
	1	2	3	4	5	6	7	8	9
Crank speed									
rpm	1,800	3,000	3,600	1,800	3,000	3,600	1,800	3,000	3,600
Hz	30	50	60	30	50	60	30	50	60
Mean pressure									
MPa	2.76	2.76	2.76	4.14	4.14	4.14	5.52	5.52	5.52
psia	400	400	400	600	600	600	800	800	800
Basic power, W									
NASA	1,934.2	3,558.8	4,491.0	3,081.1	5,937.1	7,625.1	4,367.0	8,707.1	11,359.6
ISO82 (adiabatic)	2,051.1	3,675.3	4,550.4	3,264.8	5,996.4	7,511.8	4,598.2	8,625.3	10,918.8
Error, %	−5.7	−3.2	−1.3	−5.6	−1.0	1.5	−5.0	0.9	3.7
Indicated power, W									
NASA	2,178.4	4,527.2	6,107.0	3,400.1	7,285.9	9,928.2	4,762.9	10,458.5	14,386.5
ISO82 (adiabatic)	2,199.5	4,292.9	5,576.3	3,470.5	6,853.9	8,935.8	4,857.7	9,708.4	12,714.6
Error, %	−1.0	5.5	9.5	−2.0	6.3	11.1	−1.6	7.7	13.1
Brake power, W									
NASA	3,164.8	6,249.7	8,202.0	4,646.5	9,405.5	12,495.7	6,274.3	12,993.1	17,454.6
ISO82 (adiabatic)	2,666.1	5,203.5	6,759.1	4,206.6	8,307.8	10,831.2	5,888.1	11,767.7	15,412.1
Error, %	18.7	20.1	21.4	10.5	13.2	15.4	6.6	10.4	13.5
Basic heat in, W									
NASA	3,095.9	5,004.9	5,925.3	4,480.4	7,154.6	8,433.1	5,773.9	9,126	10,710.2
ISO82 (adiabatic)	2,586.7	4,189.7	4,962.1	3,791.6	6,061.6	7,130.8	4,945.7	7,798	9,107.1
Error, %	19.7	19.5	19.4	18.2	18.0	18.3	16.8	17.0	17.6
Net heat in, W									
NASA	2,883.3	4,448.1	5,059.8	4,233.7	6,414.6	7,238.4	5,488.7	8,192.8	9,171.6
ISO82 (adiabatic)	2,059.6	3,092.7	3,457.1	2,995.1	4,293.4	4,678.9	3,822.0	5,204.5	5,490.8
Error, %	40.0	43.8	46.4	41.3	49.4	54.7	43.6	57.4	67.0
Basic heat out, W									
NASA	5,014.2	8,547.3	10,401.1	7,545.2	13,076.3	16,042.6	10,121.9	17,814.9	22,047.2
ISO82 (adiabatic)	4,637.8	7,865.0	9,512.5	7,056.4	12,058.0	14,642.6	9,543.9	16,423.3	20,025.9
Error, %	8.1	8.7	9.3	6.9	8.4	9.6	6.1	8.5	10.1
Net heat out, W									
NASA	5,046.7	8,959.1	11,151.5	7,616.5	13,684.9	17,151.1	10,232.6	18,633.2	23,535.4
ISO82 (adiabatic)	4,259.1	7,385.6	9,033.4	6,465.6	11,147.3	13,614.6	8,679.7	14,912.8	18,205.8
Error, %	18.5	21.3	23.4	17.8	22.8	26.0	17.9	24.9	29.3
Indicated COP									
NASA	2.32	1.98	1.83	2.24	1.88	1.73	2.15	1.78	1.64
ISO82 (adiabatic)	1.94	1.72	1.62	1.86	1.63	1.52	1.79	1.54	1.43
Error, %	19.6	15.1	13.0	20.4	15.3	13.8	20.1	15.6	14.7
Net COP/CARNOT									
NASA	35.7	36.3	36.5	40.4	41.8	42.2	43.6	45.8	46.7
ISO82 (adiabatic)	39.8	40.5	40.5	42.2	43.9	44.1	44.1	46.2	47.1
Error, %	−10.3	−10.4	−9.9	−4.3	−4.8	−4.3	−1.1	−0.9	−0.8

Table 2. Comparison of loss terms calculated by the NASA and Martini models

	Test point								
	1	2	3	4	5	6	7	8	9
Crank speed									
rpm	1800	3000	3600	1800	3000	3600	1800	3000	3600
Hz	30	50	60	30	50	60	30	50	60
Mean pressure									
MPa	2.76	2.76	2.76	4.14	4.14	4.14	5.52	5.52	5.52
psia	400	400	400	600	600	600	800	800	800
Heat absorber ΔP, W									
NASA	17.9	74.4	124.1	24.5	102.2	170.4	30.6	128.0	214.1
ISO82	42.2	185.7	313.2	61.3	266.1	445.1	79.5	339.9	563.3
Error, %	−57.6	−59.9	−60.4	−60.0	−61.6	−61.7	−61.5	−62.3	−62.0
Regenerator ΔP, W									
NASA	142.6	510.7	829.3	171.7	677.2	1143.7	203.7	866.8	1492.6
ISO82	74.0	286.6	463.8	96.8	374.7	606.0	117.2	453.2	732.3
Error, %	92.7	78.2	78.8	77.4	80.7	88.7	73.8	91.3	103.8
Heat rejector ΔP, W									
NASA	34.9	156.9	270.2	50.3	230.4	399.8	65.4	304.4	531.1
ISO82	32.2	145.3	249.0	47.5	216.7	373.0	62.9	289.8	500.5
Error, %	8.4	8.0	8.5	5.9	6.3	7.2	4.0	5.0	6.1
Total ΔP, W (with end effects)									
NASA	244.2	968.4	1616.0	319.0	1348.8	2303.1	395.9	1751.4	3026.9
ISO82	148.4	617.6	1026.0	205.6	857.5	1424.1	259.6	1082.9	1796.1
Error, %	64.6	56.8	57.5	55.2	57.3	61.7	52.5	61.7	68.5
Mechanical friction, W									
NASA	985.8	1721.8	2094.0	1245.3	2117.8	2565.2	1510.8	2532.4	3064.8
ISO82	466.6	910.6	1182.8	736.2	1453.9	1895.5	1030.4	2059.3	2697.1
Error, %	111.3	89.0	77.0	69.2	45.7	35.3	46.6	23.0	35.9
Shuttle loss, W									
NASA	28.2	28.2	28.2	28.2	28.2	28.2	28.2	28.2	28.2
ISO82	42.4	49.1	52.2	47.4	57.2	62.1	52.2	65.2	72.0
Error, %	−33.5	−42.6	−46.0	−40.5	−50.7	−54.5	−46.0	−56.8	−60.8
Cylinder wall conduction, W									
NASA	28.8	28.8	28.8	28.8	28.8	28.8	28.8	28.8	28.8
ISO82	37.8	43.6	46.3	42.3	50.7	54.9	46.5	57.7	63.4
Error, %	−23.8	−33.9	−37.8	−31.9	−43.2	−47.5	−38.1	−50.1	−54.6
Displacer wall conduction, W									
NASA	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
ISO82	12.1	13.9	14.8	13.5	16.2	17.5	14.9	18.4	20.3
Error, %									
Regenerator wall conduction, W									
NASA	17.7	17.8	17.7	17.7	17.7	17.8	17.7	17.7	17.7
ISO82	119.4	137.6	146.3	133.5	160.1	173.3	146.9	182.1	200.3
Error, %	−85.2	−87.1	−87.9	−86.7	−88.9	−89.8	−88.0	−90.2	−91.2

	Test point								
	1	2	3	4	5	6	7	8	9
Total conduction loss, W									
NASA[a]	67.2	67.3	67.2	67.2	67.3	67.2	67.2	67.3	67.2
ISO82[b]	174.7	201.3	214.0	195.3	234.3	253.5	214.9	266.4	293.0
Error, %	−61.5	−66.6	−68.6	−65.6	−71.3	−73.5	−68.7	−74.7	−77.1
Expasion space friction heat, W									
NASA	116.1	461.3	770.1	152.3	644.7	1099.2	189.8	837.7	1443.2
ISO82	79.2	329.0	524.1	109.7	453.5	748.1	138.1	566.5	929.5
Error, %	46.6	40.2	46.9	38.8	42.2	46.9	37.4	47.9	55.3

[a]Includes conduction through cylinder wall, displacer wall, regenerator wall, and insulation (Table 1, ref. 4).

[b]Includes conduction through cylinder wall, displacer wall, regenerator wall, regenerator matrix, and displacer-to-wall clearance (Table 2, ref. 4).

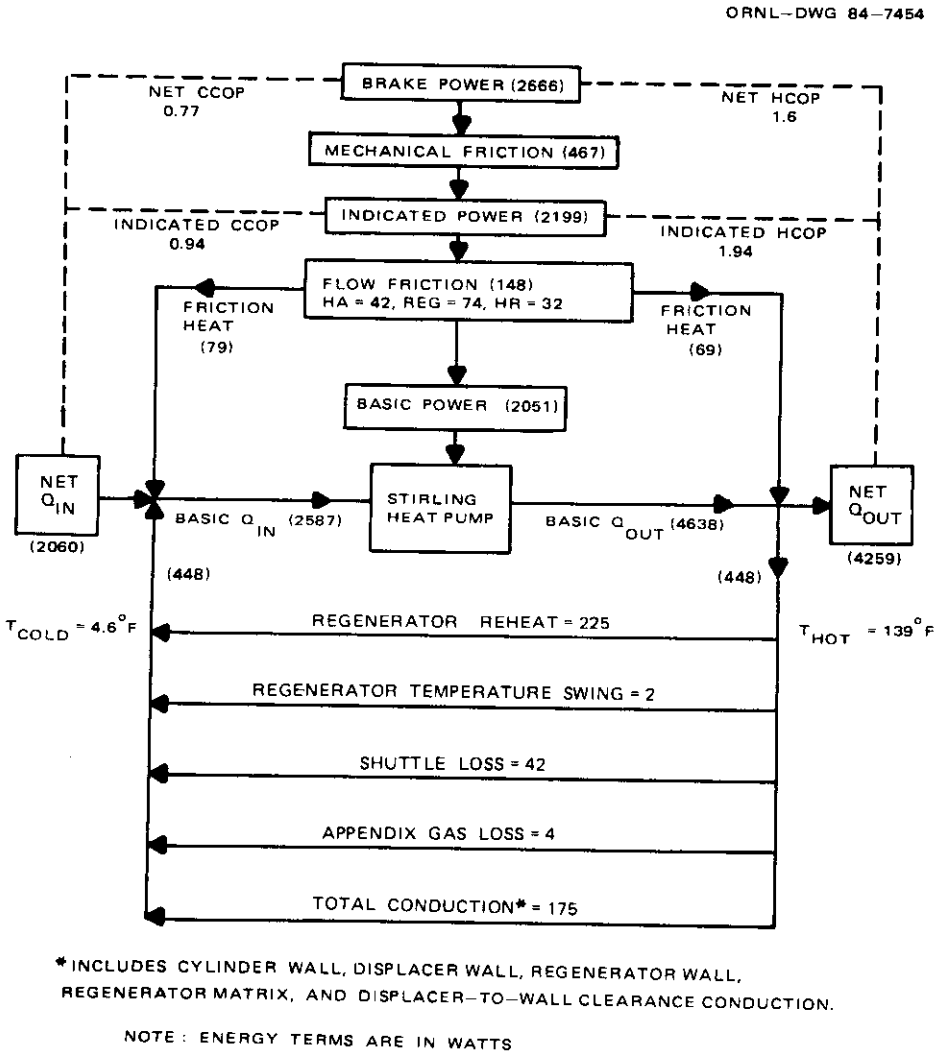


Fig. 4. Stirling heat pump energy diagram from NASA code data.

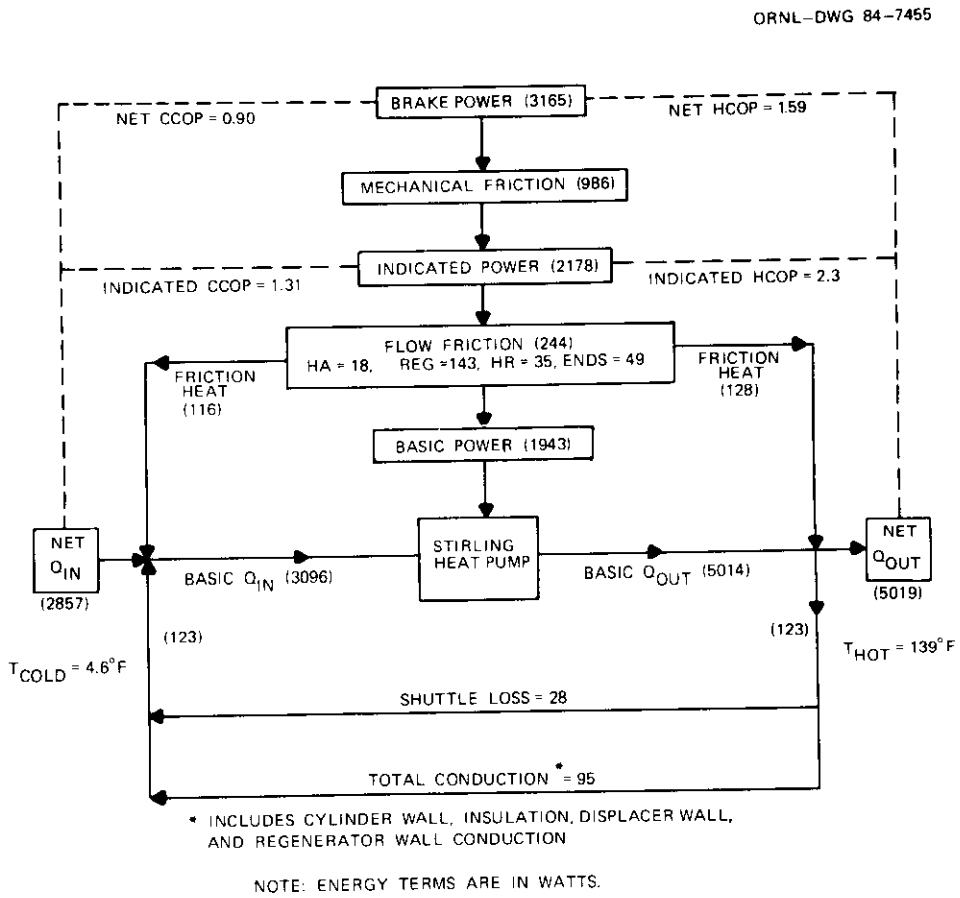
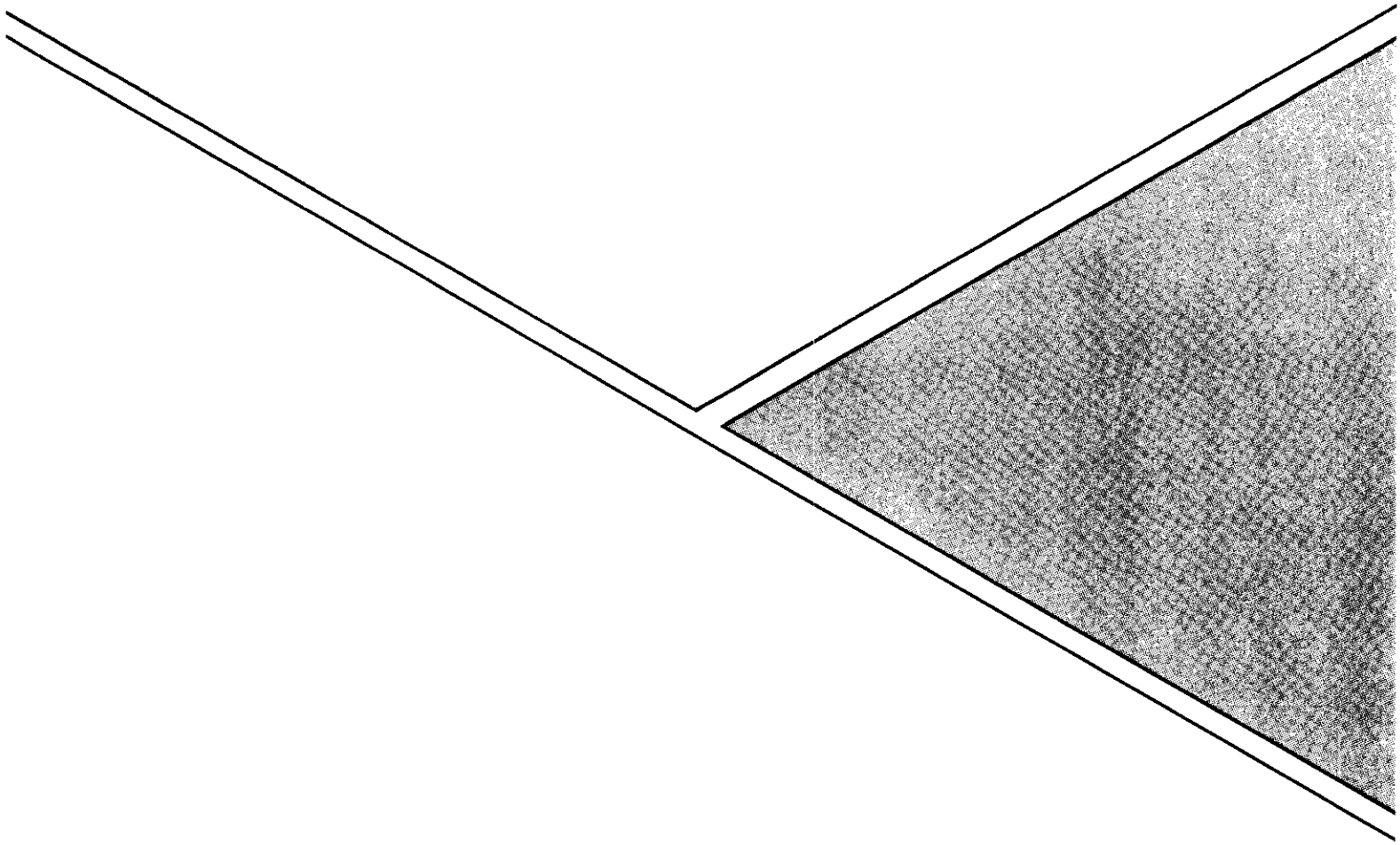
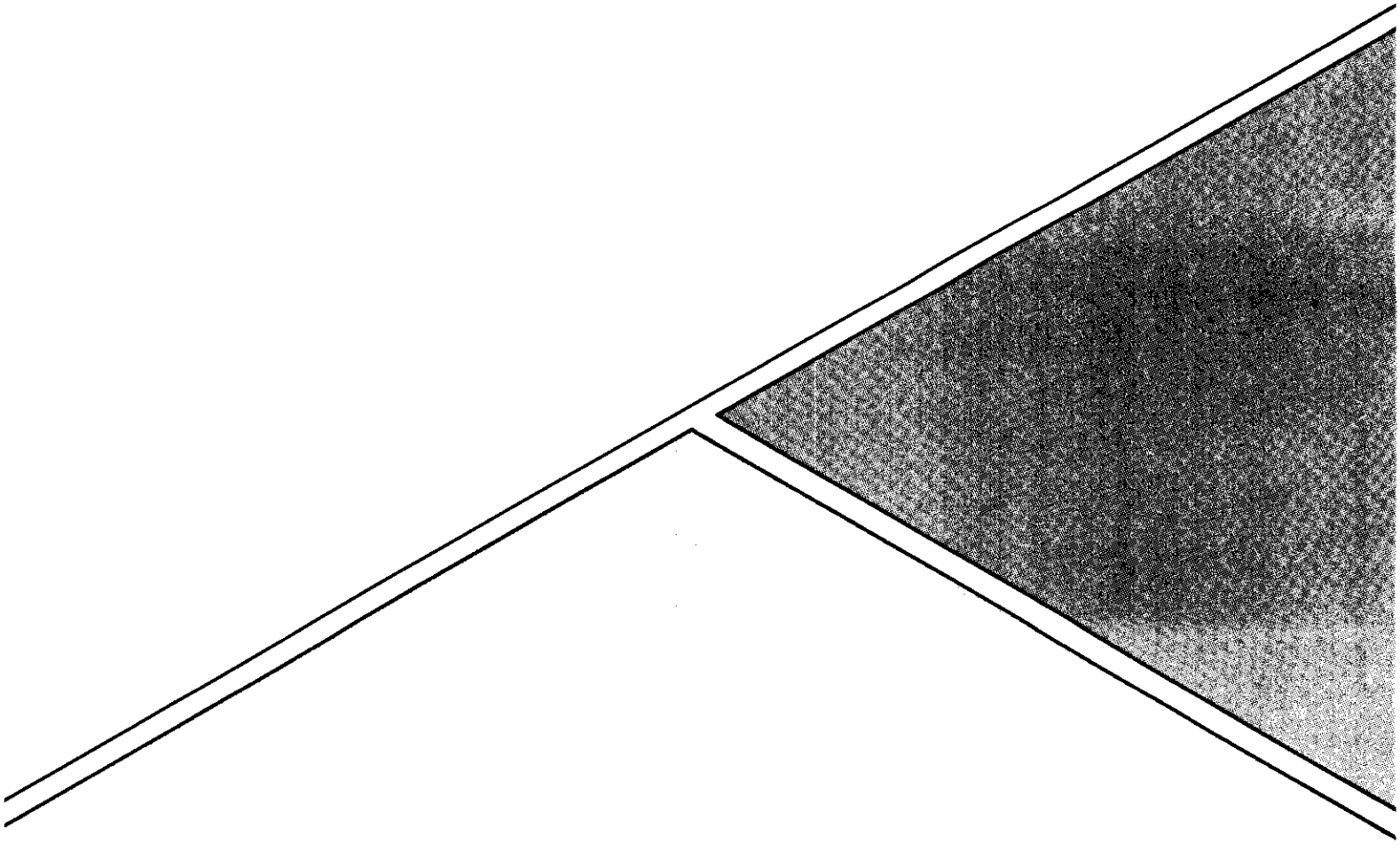


Fig. 5. Stirling heat pump energy diagram from Martini code data.



GROUND-COUPLED HEAT PUMP TECHNOLOGY



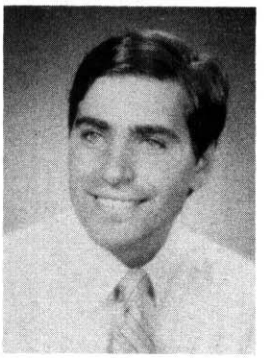
Optimized Ground-Coupled Heat Pump Mechanical Package

Mark A. Catan

Brookhaven National Laboratory

Upton, New York 11973

Mark Catan received the B.A. degree in physics at Grinnell College in 1978. He is currently pursuing the M.S. degree in mechanical engineering at the State University of New York at Stony Brook. Since 1980 he has been responsible for several analytical and experimental projects at Brookhaven National Laboratory (BNL), dealing primarily with ground-coupled and solar-assisted heat pump systems. Prior to his association with BNL, he administered field experiments on prototype electrostatic precipitators for United McGill Corporation in Columbus, Ohio.



OPTIMIZED GROUND-COUPLED HEAT PUMP MECHANICAL PACKAGE

Mark A. Catan
Solar and Thermal Systems Division
Building 701
Brookhaven National Laboratory
Upton, NY 11973

Introduction

This project addresses the question of how well a ground coupled heat pump system could perform with a heat pump which was designed specifically for such systems operating in a northern climate. Conventionally, systems are designed around water source heat pumps which are not designed for ground coupled heat pump application. The objective of the project is to minimize the life cycle cost for a ground coupled system given the freedom to design the heat pump and the ground coil in concert. In order to achieve this objective a number of modeling tools were developed which will likely be of interest in their own right.

The system which was optimized was a horizontal earth coil in a given house[1] in the Pittsburgh area. The types of components used in the heat pump are essentially conventional. Subsequent work will consider other climatic regions and ground coupling devices. The method of this study is derived from one used to optimize an air-source heat pump for operation at standard steady state conditions.[2] A computer model, created at Oak Ridge National Laboratory (ORNL), is used to predict heat pump performance for specified configuration and operating conditions. A pre-determined set of design parameters is automatically altered by a constrained minimization program (a Numerical Algorithms Group library routine, hereafter referred to as the optimizer)[3], to find the combination for which the highest COP is predicted by the heat pump model (See Figure 1).

In the present study, the set of parameters meters to be varied include some pertaining to the ground coil as well as some pertaining to the heat pump. The optimizer, in this case, pursues a minimum life cycle cost which is calculated using the heat pump model, a ground coil model (called FTECM for Fast Transient Earth Coil model), and a parameterized cost model. The flow chart in Figure 2 shows how the optimizer interacts with these programs to minimize life cycle cost.

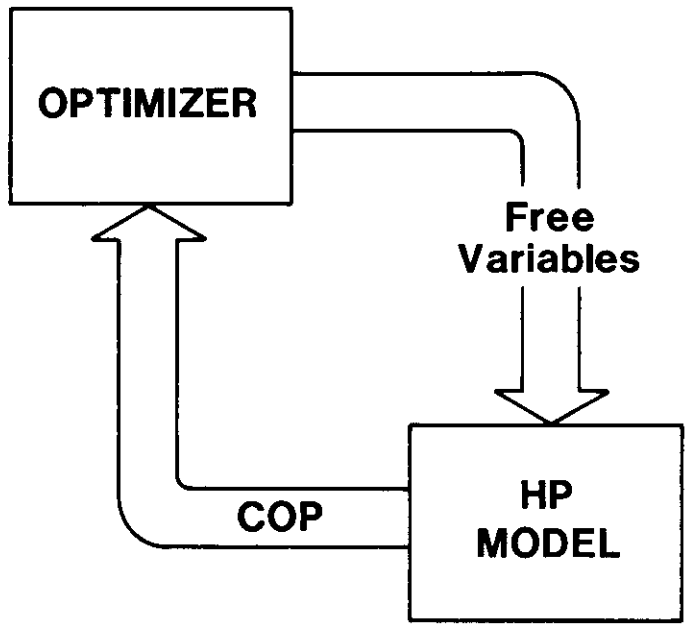


Figure 1. Simplified flow chart of steady state optimization of heat pump.

Life Cycle Cost Optimization

Initially the objective of this project was to optimize a heat pump for a predetermined ground coupling device and climate. The plan was to develop a bin representation of the time a ground coupled heat pump spends satisfying a given load at a given source temperature. This type of bin model can be created by way of a seasonal simulation but it presupposes that both the ground coil and heat pump are completely characterized. Since our purpose is to use this bin representation to determine the heat pump design, we would have to guess at the performance of the heat pump and use the

Research sponsored by the Building Equipment Research Div., U.S. Department of Energy (DOE) under Contract No. DE-AC05-840R21400 with Martin Marietta Energy Systems, Inc.
Work performed under the auspices of DOE under Contract No. DE-AC02-76CH00016.

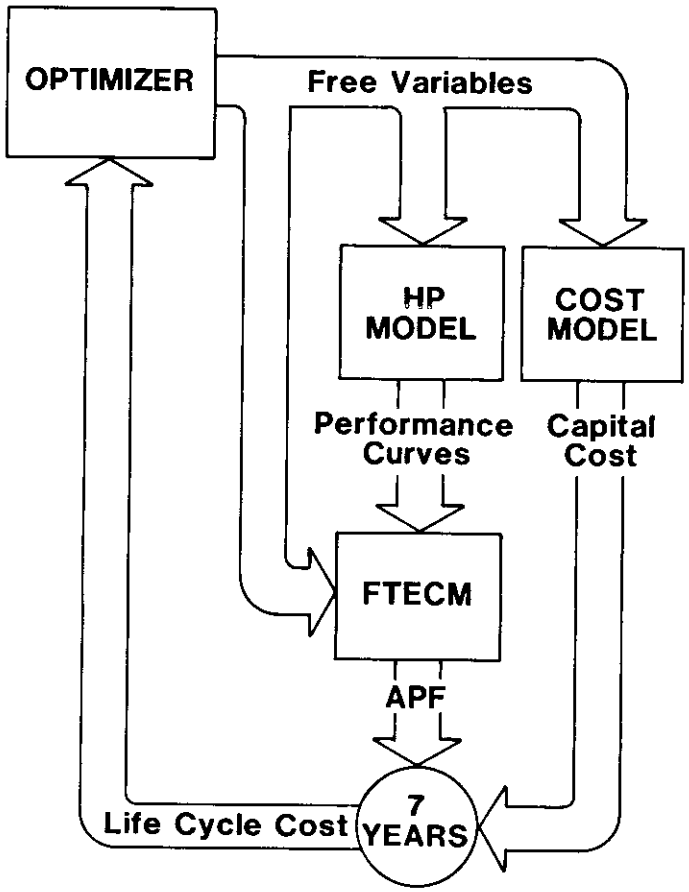


Figure 2. Flow chart for life-cycle cost optimization of ground coupled heat pump system.

resulting optimized heat pump design iteratively, to generate a new bin model[4]. Since these optimizations represent a substantial computer-time burden, an alter-native procedure was devised which, we believe, is more informative and requires less computer time.

The present objective is to optimize the entire system by minimizing its life cycle cost. No initial presuppositions concerning the performance of any components need be made but it is necessary to simulate the entire system for each set of design parameters. For each set of parameters generated by the optimizer algorithm (refer to Figure 2) the heat pump model is called upon to generate curves representing the heat pump capacity and COP vs source temperature for the heating and cooling modes. This requires several runs of the heat pump model. Then the annual performance of the system is calculated via a seasonal simulation of the system performed by the ground coil model. The cost model calculates the first-cost of the heat pump and ground coil and, finally, the life cycle cost is calculated from the annual performance factor (APF) and the first cost. The optimizer routine looks at this life cycle cost, generates a new set of design parameters, and the process is repeated until it finds the combination for which the lowest life-cycle cost was obtained.

Analysis Tools Development

A number of Fortran computer programs were created for this project. Their development represents the major portion of the work done to date. The programs can stand alone or operate in concert. The following is a discussion of three major modeling tools developed for this project.

Water-refrigerant Subroutines

The ORNL Air-to-Air Heat Pump Model was readily adaptable to the modeling of water source heat pumps with the addition of water refrigerant heat exchanger subroutines. A substantial amount of effort was devoted to the creation and installation of these subroutines in the air-to-air model for several reasons. For one thing, the types of heat exchangers which work best (i.e., highest performance for the dollar) in heat pumps in the size range of interest happen to be the most difficult to model, as will be explained further on. Secondly, the subroutines must be very accurate since the water-refrigerant heat exhanger is a very cost/performance sensitive component of a water source heat pump. Thirdly, the subroutines had to be fast to save computer time.[5]

Models were developed for two generic types of heat exchangers: shell-and-tube with a baffled shell side, and coaxial tube-in-tube. Most heat pumps in the size range of interest use the latter type of heat exchanger because they are generally cheaper, lighter, and more compact than the former. For this reason, only tube-in-tube heat exchangers are being used in the optimization work.

All of the tube-in-tube heat exchangers we have considered modeling employ some type of heat transfer augmentation via specially designed surfaces. Within the competitive group of heat exchangers commonly used in water source heat pumps are two which exhibit superior evaporating perfomance: one with helical convolutions in the inner tube, which affect the flow of both the refrigerant in the annulus and the water in the tube, and another which has a roughened surface on the outside of the inner tube, around which the water flows, and low profile helical fins on the inside where refrigerant flows. Both heat exchangers are characterized by relatively high pressure drops, in both evaporating and condensing modes, as applied in water source heat pumps.

A literature search was conducted to identify correlations and experimental data for use in the subroutines. The information available on two-phase heat transfer and pressure drop for augmented tubes was not found to be

adequate for the creation of the required models. On the other hand, there were a number of papers which demonstrated methods of correlating heat transfer and pressure drop data for augmented surfaces using modified smooth tube correlations. We therefore decided to test the heat exchangers we wanted to model and fit existing correlations to our data.

A heat exchanger test facility at BNL was used to obtain the necessary experimental data. Two sizes of each type of heat exchanger were tested. The heat exchangers were tested in pairs with one operating as a condenser and one operating as an evaporator of a laboratory heat pump, or Subcomponent Test Stand (STS; see Figure 3) which is a water-to-water heat pump with a continuously variable speed compressor and a dummy evaporator. These features allow the refrigerant mass flow and the evaporator outlet enthalpy to be varied independent of the evaporating and condensing temperatures. The experimental facility has a computer control system which was used to run the heat pump through a four-dimensional matrix of source temperatures, coolant temperatures, compressor speeds, and evaporator outlet enthalpies. The compressor speed and evaporator outlet enthalpy were varied in steps while the condenser and evaporator heatant/coolant were varied slowly enough to insure quasi-steady-state operation at all times. A separate battery of tests was performed with single phase fluid on both sides of the heat exchanger so that the Wilson Plot method could be used to find the single phase heat transfer coefficients on the water side.

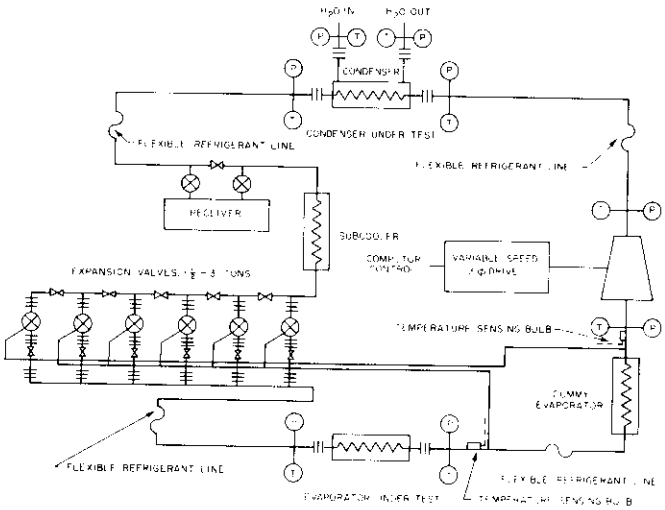


Figure 3. Subcomponent Test Stand.

The subroutines were written with the intention of testing a variety of correlations out on the experimental data. The method of fitting the free parameters in the heat transfer correlations to experimental data is shown graphically in Figure 4. For each set of pressure drop and heat transfer correlations to

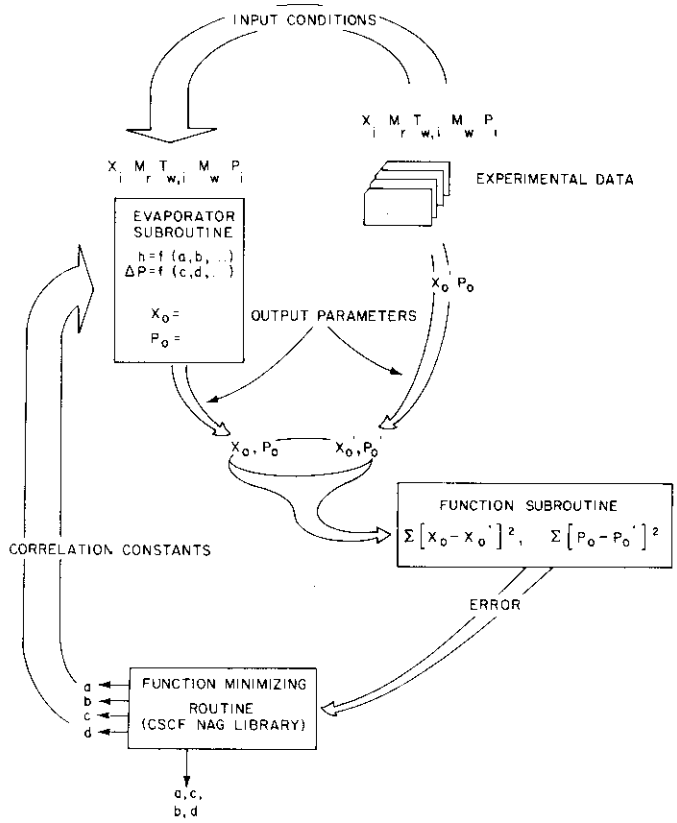


Figure 4. Method of fitting heat exchanger correlations to test data.

be tried, an optimizer routine would find the free constants which minimized the error between the experimental data and the subroutine predictions for all the test conditions. In this way the pressure drop correlations and the heat transfer correlations were fitted simultaneously.

Ground Coil Model. In general the simulation of ground coupling devices using a three dimensional finite element approach takes a great deal of computer time making shortcuts desirable. The approach used by the model GROCS is a good basis for the ground coil simulation routine required in present work since it makes use of large timesteps and a small number of discrete elements. A ground coil model which could account for heat pump cycling, but which did not require shorter timesteps or a larger number of earth elements than GROCS, was desired. The final version of the program has these features and is described briefly below and more completely in references 4 and 5.

The program GROCS was written in 1978 to model three dimensional underground heat flow in the vicinity of buried tanks. Later it was integrated with TRNSYS with a subprogram which allowed it to simulate buried pipe heat exchangers. As part of the heat pump optimization study the GROCS/TRNSYS program was improved with the addition of a device to handle cycling of the heat pump without requiring greater time and/or memory than the

original program. One other feature has been added which creates the earth block model required by GROCS, for horizontal earth coils. The set of routines have been excised from TRNSYS with the addition of a simple simulation driver and the resulting autonomous entity dubbed FTECM (Fast Transient Earth Coil Model). At the time of this writing the capability for modeling soil moisture freezing was under development for FTECM.

The FTECM program has been run using data from an actual ground coupled heat pump installation at Brookhaven National Laboratory to determine the accuracy of the model's predictions. The error given by Eq. (1) was found to be slightly more than 2% for the heating season and approximately 6% for the cooling season.

$$\frac{\sum_i \left| COP_{c,i} - COP_{m,i} \right| Qi}{\sum_i Q_i} \qquad (1)$$

Where the subscript i refers to the hour of the year, $COP_{c,i}$ is the COP calculated by the model, $COP_{m,i}$ is the COP measured, and Qi is the heating or cooling load.

A lower soil conductivity was used during the cooling season part of the simulation as recommended in reference 6. One serious concern is that the model has not been compared to a system for which significant amounts of soil freezing have occurred.

Parameterized Cost Model. The proposed life cycle cost optimization of the heat pump system requires a computer program for calculating the first cost of the heat pump options being explored by the optimizer. A set of equations relating the component sizes to the first cost of the heat pump was developed by Friedrich. This Cost Model was created using design and cost data from Friedrich's line of water source heat pumps in the two-to-ten ton nominal capacity range. The inputs to the cost model are physical parameters such as air coil face area and water coil length. The output of the model is the first cost of the heat pump to the consumer. The physical description of the heat pump does not have to be entered into the model in exhaustive detail because the minor component costs are correlated to the gross indicator parameters: ARI cooling capacity and cabinet area (the area of smallest parallelipiped which the heat pump unit could fit into). The cabinet size is slaved to air coil face area. All of the major component costs, which account for 80% of the heat pump cost, are correlated against independent descriptive parameters with one exception, the compressor cost, which is correlated against the compressor displacement. Labor rates, and some material prices are included explicitly. Retail price is obtained from the manufacturer's cost by multiplying the final cost by a factor of two which accounts for administrative costs and profits for the manufacturer and dealers, etc. A list of the required inputs to the cost model is given below:

Cost model inputs:

- compressor displacement
- nominal cooling capacity
- air coil face area
- air coil tube rows
- air coil tube diameter
- air flowrate
- external static pressure
- water coil length
- water coil shell outer diameter
- labor cost
- overhead rate
- cost of aluminum
- cost of copper
- air coil fin pitch

Using empirical straight line fits, the cost model relates these parameters to component and construction costs, and ultimately, to the consumer cost of the heat pump. The cost model and the methodology of its derivation are discussed in detail in reference 7.

System Sensitivity Studies

The high cost of the ground coil relative to the cost of the heat pump led us to construct a simplified scheme to compare gross system parameters prior to getting into a full scale system optimization. In particular we looked at the annual performance factors of systems defined by permutations of three independent parameters; two pertaining to the heat pump and one to the ground coil. The APF for each system was determined by simulation using FTECM. All of the systems had horizontal ground coils with 5 lengths of serially connected 1.5 inch nominal polyethylene pipe buried at a depth of 4 feet. The length of the ground coil was one of the independent parameters studied. Each system simulated used a total length of 750, 1000, 1500, or 2000 feet. The cost of the installed ground coil was taken to be $\$1.50/ft^2$. The cost and performance data for the heat pumps were obtained by first performing a steady state optimization for each of 4 different cost constraints: \$1200, \$1300, \$1400 and \$1500. The heat pump configurations resulting from the steady state optimizations were modified in such a way as to maintain their optimized COP while changing their capacities by a scale factor. The cost model was, of course, used to obtain a new cost for the scaled heat pumps. Thus we had two independent parameters for the heat pump: the capacity scale factor, and the

cost constraint used to obtain the configuration of the unscaled unit, or base case. If the scale factor is increased, the heat pump's relative size is increased. If the base case optimized cost is increased the relative efficiency is increased by whatever the cost increment could afford in the base unit. These two parameters, in a sense, substitute for the numerous individual heat pump parameters which were optimized for steady state performance.

Tables 2, 3, 4 and 5 give the simulation results for the 1200, 1300, 1400, and 1500 dollar optimized base cases respectively.

Table 2. System Performance and Economics for Heat Pump Optimized with a Cost Constraint of $1200*

Capacity Scale Factor	Heat Pump Cost ($)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Life Cycle Cost (dollars)	Simple Payback Relative to Air Source System (years)
1.0	1216	750	2814	10533	1464	2.31	3.1	2.4	9155	1.87
		1000	2204	9927	1388	2.45	3.26	2.55	9142	2.77
		1500	1621	9349	1327	2.60	3.41	2.70	9530	4.50
		2000	1340	9086	1304	2.67	3.48	2.77	10117	6.22
1.1	1293	750	2201	10246	1472	2.37	3.08	2.46	9073	2.00
		1000	1652	9666	1396	2.51	3.25	2.61	9076	2.84
		1500	1118	9113	1335	2.67	3.39	2.76	9477	4.48
		2000	882	8883	1312	2.73	3.45	2.83	10083	6.17
1.2	1381	750	1723	10049	1480	2.42	3.06	2.5	9054	2.20
		1000	1212	9482	1403	2.56	3.23	2.65	9063	2.98
		1500	760	8979	1342	2.71	3.38	2.79	9493	4.59
		2000	570	8782	1320	2.77	3.43	2.88	10118	6.27
1.3	1485	750	1330	9907	1487	2.45	3.05	2.53	9081	2.47
		1000	881	9373	1410	2.59	3.21	2.67	9109	3.21
		1500	507	8916	1350	2.72	3.36	2.81	9566	4.81
		2000	306	8750	1328	2.78	3.41	2.86	10209	6.50

* Cost constraint dollar figure has been rounded off to the nearest hundred.

Table 3. System Performance and Economics for Heat Pump Optimized with a Cost Constraint of $1300*

Capacity Scale Factor	Heat Pump Cost ($)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Life Cycle Cost (dollars)	Simple Payback Relative to Air Source System (years)
1.0	1300	750	2240	9384	1395	2.59	3.25	2.67	8549	1.59
		1000	1730	8693	1325	2.79	3.42	2.88	8492	2.27
		1500	1234	8112	1268	2.99	3.57	3.07	8879	3.66
		2000	1006	7887	1244	3.08	3.64	3.16	9488	5.08
1.1	1400	750	1700	9168	1404	2.65	3.23	2.75	8529	1.79
		1000	1228	8458	1333	2.87	3.40	2.94	8461	2.41
		1500	808	7906	1275	3.07	3.55	3.14	8864	3.75
		2000	629	7713	1252	3.15	3.62	3.22	9492	5.15
1.2	1517	750	1267	9026	1411	2.69	3.21	2.76	8570	2.05
		1000	857	8314	1340	2.92	3.38	2.99	8500	2.62
		1500	520	7799	1282	3.11	3.53	3.17	8925	3.93
		2000	381	7631	1259	3.18	3.60	3.24	9566	5.32
1.3	1654	750	935	8951	1419	2.71	3.19	2.78	8666	2.38
		1000	595	8243	1347	2.95	3.36	3.01	8601	2.90
		1500	324	7755	1289	3.13	3.52	3.19	9041	4.19
		2000	222	7613	1266	3.19	3.58	3.25	9697	5.59

* Cost constraint dollar figure has been rounded off to the nearest hundred.

Table 4. System Performance and Economics for Heat Pump Optimized with a Cost Constraint of $1400*

Capacity Scale Factor	Heat Pump Cost ($)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Life Cycle Cost (dollars)	Simple Payback Relative to Air Source System (years)
0.8	1206	750	4085	10275	1213	2.36	3.73	2.51	8856	1.57
		1000	3458	9703	1156	2.50	3.92	2.65	8873	2.44
		1500	2806	9113	1109	2.67	4.09	2.67	9262	4.05
		2000	2480	8829	1089	2.75	4.16	2.91	9839	5.63
0.9	1293	750	3120	9696	1223	2.51	3.70	2.64	8620	1.61
		1000	2539	9145	1165	2.66	3.89	2.80	8649	2.39
		1500	1965	8608	1117	2.82	4.06	2.96	9066	3.88
		2000	1683	8355	1098	2.91	4.13	3.05	9662	5.35
1.0	1394	750	2375	9281	1232	2.62	3.68	2.74	8490	1.75
		1000	1865	8770	1173	2.77	3.86	2.90	8541	2.46
		1500	1348	8268	1125	2.94	4.03	3.07	8975	3.87
		2000	1099	8040	1106	3.02	4.10	3.15	9588	5.29
1.1	1515	750	1811	8995	1240	2.70	3.66	2.82	8454	1.96
		1000	1343	8506	1180	2.86	3.84	2.98	8516	2.63
		1500	897	8049	1132	3.02	4.00	3.14	8979	3.99
		2000	700	7862	1114	3.09	4.07	3.21	9613	5.39
1.2	1659	750	1357	8787	1247	2.76	3.63	2.87	8483	2.24
		1000	947	8330	1187	2.92	3.82	3.03	8564	2.87
		1500	589	7934	1139	3.06	3.98	3.18	9062	4.22
		2000	434	7780	1122	3.12	4.04	3.24	9715	5.62

* Cost constraint dollar figure has been rounded off to the nearest hundred.

Table 5. System Performance and Economics for Heat Pump Optimized with a Cost Constraint of $1500*

Capacity Scale Factor	Heat Pump Cost ($)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Life Cycle Cost (dollars)	Simple Payback Relative to Air Source System (years)
0.8	1282	750	4068	10103	1206	2.40	3.76	2.55	8830	1.74
		1000	3415	9508	1147	2.55	3.95	2.70	8834	2.54
		1500	2751	8902	1098	2.73	4.13	2.88	9212	4.06
		2000	2426	8612	1078	2.82	4.20	2.97	9785	5.56
0.9	1378	750	3117	9525	1215	2.55	3.73	2.68	8603	1.79
		1000	2507	8949	1156	2.71	3.92	2.85	8614	2.50
		1500	1921	8393	1106	2.89	4.10	3.03	9023	3.91
		2000	1637	8133	1086	2.99	4.17	3.13	9614	5.32
1.0	1489	750	2383	9111	1224	2.67	3.70	2.79	8484	1.93
		1000	1842	8573	1163	2.83	3.89	2.96	8519	2.59
		1500	1311	8048	1114	3.02	4.07	3.15	8943	3.92
		2000	1062	7813	1094	3.11	4.14	3.24	9548	5.28
1.1	1620	750	1827	8821	1231	2.75	3.68	2.87	8454	2.15
		1000	1327	8304	1170	2.93	3.87	3.04	8501	2.76
		1500	869	7826	1120	3.10	4.04	3.22	8951	4.05
		2000	673	7629	1101	3.18	4.12	3.30	9578	5.39
1.2	1773	750	1381	8610	1238	2.82	3.66	2.93	8491	2.43
		1000	938	8124	1177	2.99	3.85	3.10	8556	3.01
		1500	567	7706	1127	3.15	4.02	3.26	9040	4.28
		2000	413	7541	1108	3.22	4.09	3.33	9685	5.62

* Cost constraint dollar figure has been rounded off to the nearest hundred.

Tables 6 and 7 give the heat pump performance data used for the simulations.

Table 6. Heating Performance for Optimized Heat Pumps used in Simulation

$C_{d,h}$ = (0.3022 - 0.0235 EWT)

Cost Constraint	EWT	COP_H	Capacity
1200	20	2.76	20864
	30	3.08	25422
	50	3.59	35407
1300	20	2.98	22417
	30	3.28	26704
	50	3.81	36500
1400	20	3.22	22268
	30	3.50	26353
	50	4.01	36732
1500	20	3.29	22219
	30	3.58	26493
	50	4.09	36566

Table 7. Cooling Performance for Optimized Heat Pumps used in Simulation

$C_{d,c}$ = (0.077 + 0.00121 EWT)

Cost Constraint	EWT	COP_c	Capacity
1216	70	3.80	39017
	85	3.19	35634
1300	70	3.94	41230
	85	3.32	37410
1400	70	4.47	41701
	85	3.77	38147
1500	70	4.48	41984
	85	3.76	38370

Each line in Tables 2, 3, 4, and 5 represents a different system defined by a unique combination of the independent parameters of capacity scale factor and ground coil length. The energy consumption of the systems in satisfying the heating and cooling loads are given along with the heating and cooling seasonal, and annual performance factors. The last column in each table gives, what we refer to as, the simple payback (SPB) relative to an air-to-air system. This is the difference in the capital cost of the system minus that of the air-to-air system divided by the difference in their annual operating costs. The alternative system, for which the simple payback is given, is air source system with a heating seasonal performance factor (SPF) of 1.8, a cooling SPF of 2.28, and an uninstalled first cost of 1880 dollars.

For each system one can see approximately how long it would take to recoup the additional expense of installing that system instead of the alternative. If the system in question has a small payback period relative to an alternative it means the system will save a great deal of energy relative to the difference in cost. But the system with the smallest simple payback is not necessarily the best choice, if one's objective is to save money, since the total energy savings need not be large for the SPB to be small. The SPB will not reveal, by itself, the best financial move to make but it will tell immediately whether a given system is an economically viable alternative to a competing system.

The best way to determine the best economic choice is to look at the life cycle cost (LCC) for the system which, in this case, is the cost of buying and operating the system for a period of time, called the economic life, which is on the order of, but always less than, the expected lifespan of the system. The values in the LCC column in the System Performance and Economics Tables were calculated using an economic life of seven years. They are simply the capital cost of the system plus the annual cost of electricity multiplied by 7, all in 1984 dollars (electricity cost assumed to rise at 1 plus the discount rate with no additional operating and maintenance costs). The way to determine the best system to buy is to look for the lowest LCC, which in this case is that for the 1500 dollar optimized base case, a capacity scale factor of 1.1, and a ground coil length of 750 feet. This system would have a simple payback relative to the air-source heat pump of only 2.2 years.

The above analyses suggests an important conclusion which is: to improve the performance of ground coupled heat pump systems one should improve the heat pump and not the ground coil. It is evident from Tables 2, 3, 4, and 5 that the cost of increases in the ground coil size are difficult to recoup in a reasonable period. On the other hand, a better heat pump buys a good deal of performance improvement for the investment. This is shown graphically in Figure 5 where each intersecting pair of lines indicates the net present value (for our purposes the negative of the life cycle cost) for each combination of optimized COP and ground coil length.

The direction to ground coupled heat pump system improvement suggested by the above results makes the case for ground coupled space conditioning in northern climates quite strong. The value of system optimization is very evident. The need for improved heat pumps for ground coupled heat pump systems can be argued rather simply by stating that the cost of the ground coils is so high relative to the cost of heat pumps that it makes more sense to add value to the heat pump than to the ground coil if both options offer a similar degree of benefit.

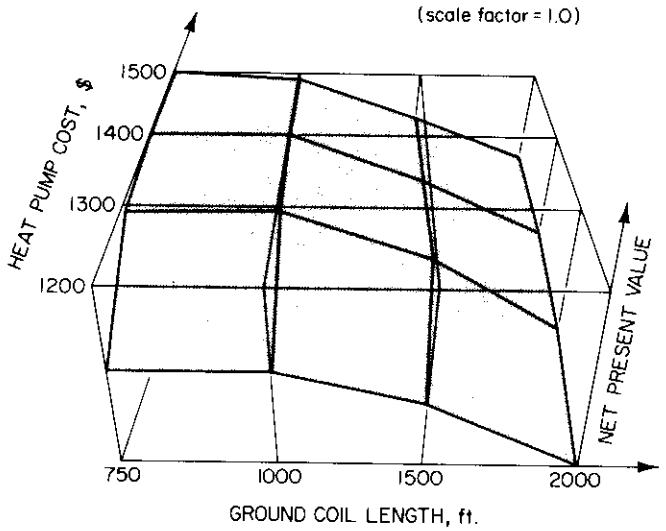


Figure 5. Net present value versus optimized COP and ground coil length for scale factor = 1.0

It is hoped that these results and the modeling tools developed for this project will be of value to people involved in ground coupled system design.

References

1. The house used in this study is an 1800 ft^2 moderately well insulated ranch built over a crawl space. Details on the construction of the house and the derivation of the space heating and cooling loads are available upon request from BNL or ORNL.

2. Rice, C.K., Fischer, S.K., Jackson, W.L., and Ellison, R.D. Design Optimization and the Limits to Steady-State Heating Efficiency for Conventional Single-Speed Air Source Heat Pumps. Report No. ORNL CON/63, Oak Ridge Natl. Lab., Oak Ridge, TN, Oct. 1981.

3. Numerical Algorithms Group, NAG Fortran Library, Mark X. Atomic Energy Estab., Harwell, England, 1983.

4. Andrews, J.W., Metz, P.D., and Saunders, J.H. A Refined Computer Program for the Transient Simulation of Ground Coupled Heat Pump Systems. BNL 34818, April 1984.

5. Andrews, J.W., Catan, M.A., LeDoux, P., Metz, P.D., and Saunders, J.H. Optimal Ground Coupled Heat Pump Project Phase I Final Report, in progress.

6. Metz, P.D. The Use of Serpentine Earth Coils in Ground Coupled Heat Pump Systems. BNL 33915, April 1983.

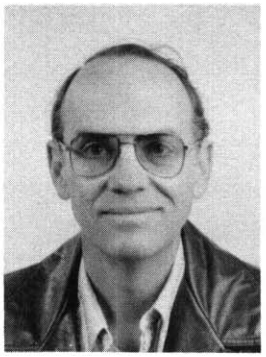
7. Hughes, M. A Parameterized Cost Model for Ground Coupled Heat Pumps, in press.

Ground-Coupled Heat Pump Research at the University of Tennessee

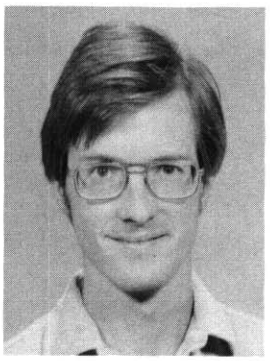
William S. Johnson and Bernie A. McGraw
Robert N. Baugh and William A. Griffith

University of Tennessee

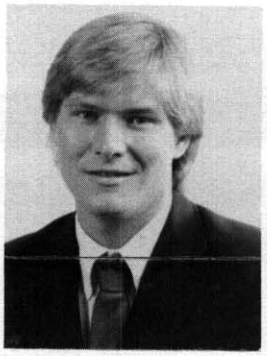
Knoxville, Tennessee 37996



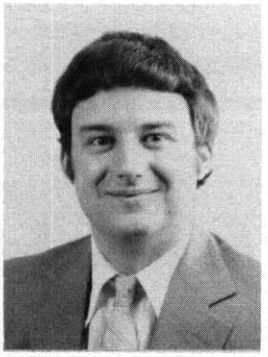
William S. Johnson is a professor of mechanical engineering at the University of Tennessee where he has been on the faculty for 16 years. He received the B.S. degree from the University of Tennessee in 1961 and the M.S. and Ph.D. degrees from Clemson University in 1965 and 1967, respectively. He has industrial experience with Pratt & Whitney Aircraft, Tennessee Valley Authority, and Oak Ridge National Laboratory. His major areas of research have involved unsteady fluid dynamics, boundary layer control, energy conservation in commercial buildings, and ground-coupled heat pump systems. Dr. Johnson teaches courses in the general area of thermal sciences and is currently in charge of departmental courses in instrumentation and measurements.



Bernie A. McGraw is director of the Tennessee Energy Conservation in Housing Facility. He received the B.A. degree from Maryville College (environmental science, 1977) and the B.S. degree from the University of Tennessee (mechanical engineering, 1979). As a licensed engineer in the State of Tennessee, his major areas of research have been in solar-assisted and ground-coupled heat pumps. He has conducted several industrial energy audits for the Energy Analysis and Diagnostic Center. He is also serves as a consulting engineer in designing commercial heating, ventilation, and air-conditioning systems.



Robert N. Baugh received his B.S. degree in mechanical engineering from the University of Tennessee in 1983. He is presently a mechanical engineer with the Energy, Environment, and Resources Center at the University and is actively involved in the testing and evaluation of earth-coupled heat pump systems. He is a member of the American Society of Heating, Refrigeration, and Air-Conditioning Engineers and the American Society of Mechanical Engineers.



William A. Griffith received his B.S. degree in mechanical engineering from the University of Tennessee. He is currently working toward the M.S. degree in the area of thermal and fluid sciences. He is a student member of the American Society of Mechanical Engineers and a member of the National Society of Professional Engineers. Since February 1984 he has been working on the University of Tennessee's ground-coupled heat pump project.

Ground-Coupled Heat Pump Research at the University of Tennessee

William S. Johnson and Bernie A. McGraw
Robert N. Baugh and William A. Griffith

University of Tennessee

Knoxville, Tennessee 37996

Introduction

The ground has been considered for the heat sink and heat source for heat pumps almost since their initial application in residential heating and cooling because the ground is normally at a more favorable temperature than the ambient air.

Metz[1], reports on ground coupled heat pump research at Brookhaven National Laboratory including the testing of four buried tanks, six earth coils and six sealed vertical wells. He reports a heating seasonal performance factor of 2.3 for a horizontal ground coil in Upton, New York. Bose[2] reports studies with six experimental earth coils involving different pipe materials, water/antifreeze solutions and several brands of water source heat pumps. Seasonal performance factors of 2.8 for heating and 2.7 for cooling have been reported for his site in Oklahoma. In a 1982 survey of ground source heat pump investigations, Ball[3] summarized data from various investigations and reports a range of seasonal performance factors for heating from 2.0 to 3.35 for 17 different installations and two values for cooling, 1.8 and 2.7. Based on these studies, it appears that ground coupled heat pumps should be competitive with, and perhaps superior to air-air systems.

Objective

This study was undertaken to evaluate the seasonal performance of a horizontal-coil ground-coupled heat pump system in both the heating and cooling mode to determine its potential for use in the Knoxville, TN. area. In addition, the effects of the most significant parameters on performance were to be determined along with the required data to develop an optimized system.

The ground-coupled heat pump system is installed in TECH House I which is part of a research and demonstration project known as the Tennessee Energy Conservation in Housing (TECH) Program. Initiated in 1975 as a joint venture by the University of Tennessee, the Tennessee Valley Authority, the Department of Energy (through the Oak Ridge National Laboratory), and a consortium of private industry, the facility consists of two additional solar houses (TECH House IV and TECH House V), an annual cycle energy system (ACES) house (TECH House II), and a control house (TECH House III) located on the University of Tennessee Institute of Agriculture's Experiment Station, in Knoxville, Tennessee.

Several heating and cooling systems have been tested in TECH House I since its construction. It was initially operated as a solar house with liquid circulating roof mounted collectors for space heating and has since been operated with a solar assisted heat pump using both solar panels and the ground for solar assist.

System Description

In 1982 a ground-coupled heat pump system utilizing an isolated horizontal coil was installed in TECH House I. Design of this system was performed by Fisher[4]. This system utilizes a Tetco hydronic, heating only heat pump, with space cooling being accomplished by re-directing the water-methanol brine with three way valves. The position of the three way valves (controlled manually) determines the mode of operation (heating or cooling). When the thermostat engages, the compressor, indoor fan, and circulation pumps are powered.

Both circulation pumps are Gorman-Rupp series 14520 (240 volt) pumps. The measured flow rate in each loop is approximately 10 GPM for both the heating and cooling modes of operation. The pipe is 1¼ inch polybutylene with an inside diameter of .380 inches and a wall thickness of 0.120

inches. After installation, pipe length was measured to be 675 feet. The pipe was buried approximately four feet deep.

Just prior to the 1984 cooling season, the ground coil was excavated and replaced with a 1¼ inch high density polyethelyne pipe. The material was changed in order to use butt fusion welded joints rather than barbed fittings. Also in lieu of using indigenous clay backfill around the pipe, fine river sand was used and the pipe was surrounded with about 3 inches of sand on all sides. The clay was then backfilled and carefully tamped. In addition to the 675 ft. coil at 4 ft. depth, an additional 675 ft. of coil was buried in the same trench at a depth of two feet and likewise surrounded by sand. Also, a ground soaker pipe and a brine to domestic water heat exchanger were installed at this time. During the 1984 cooling season, however, only the 675 ft. coil at a depth of 4 feet was used in order to determine the effect of sand backfill on system performance. The other features are for future use.

It was estimated that the installed cost of the initial ground coupled heat pump system was $5290. This estimate is for the ground coil, heat pump and associated piping and controls and does not include costs for ductwork. Table 1 shows a breakdown of the costs of the major items associated with the system installation.

TABLE 1

ESTIMATED GROUND COUPLED HEAT PUMP COST BREAKDOWN

	1982 Dollars
Earth Coil	$ 585.00
Earth Coil Installation	350.00
Heat Pump	2,500.00
Water-to-Air Coil	655.00
Air Handling Unit	200.00
Heat Pump & Water Coil Installation	1,000.00
TOTAL	$ 5,290.00

Instrumentation

Most data at the TECH Facility are automatically fed into a hybrid data acquisistion system (DAS). Every hour the integrated digital signals from electronic counters and the instantaneous analog signals are recorded on magnetic tape. These data include hourly heat flows, energy consumptions, various ground and conditioned space temperatures as well as hourly meteorogical conditions consisting of dry bulb and dew point temperature, wind speed and direction, barometric pressure, total global solar insolation, and rainfall.

Ceramic blocks having electrical resistivity which changes with moisture are used to measure the moisture content of the soil. These sensors were calibrated using actual soil samples from the site. Because of the slow response of these sensors, moisture readings are taken weekly by hand using an AC bridge to measure the probe resistance.

Procedure

Between September 1982 and October 1984, data for two heating seasons and two cooling seasons have been obtained. Data have been collected hourly with the DAS. The exception being soil moisture and far field soil temperature measurements which have been made weekly.

All instruments were calibrated prior to their installation and the calibration of each is re-checked monthly.

Performance Measurement

Heat pump systems are normally rated by a coefficient of performance (COP), defined as the energy added to (removed from) the conditioned space at the heating (cooling) coil divided by total electrical energy input. A better indicator, however is performance factor[5] (PF) defined as the total useful energy supplied to (removed from) the conditioned space divided by the total purchased power required to deliver that energy for the appropriate time period. Since the ground coupled heat pump system is packaged in such a manner that it could be installed either inside or outside the conditioned space, two separate performance factors are presented.

PF_i is the performance factor for the case in which the heat pump package and all other power consuming components (pumps and the blower) are located within the conditioned space. The ground coupled heat pump system is presently installed in this manner. For example, the heating PF_i is calculated as the sum of the ground heat transfer and the total system purchased

electric power divided by the total purchased power for the period under consideration. PF_o is calculated assuming the entire package (compressor, refrigerant piping, two refrigerant to brine heat exchangers and pumps) are located outside the conditioned space. Consequently, heat losses from these components are not included in heat delivered to the space. Heating PF_o is then calculated as the sum of the heat transferred through the condensing coil and blower power consumption divided by the total purchased power. Likewise, for cooling operation, PF_i is ground heat transfer less total purchased power divided by total purchased power and PF_o is evaporating coil heat transfer less fan power divided by total purchased power. Locating the heat pump package within the conditioned space will be beneficial to the user during the heating season, but will be a detriment during the cooling season.

Since the heat pump is a heating-only unit, cooling performance data were not available and so data from the experiment were used to obtain the coefficient of performance of the package unit as a function of brine inlet temperature. In order to reduce the effects of transient operation, data were used only if the heat pump had operated for a significant fraction of a given hour. Since the conditioned space was controlled by a thermostat it was assumed that air temperature entering the inside coil was constant and 70°F (21°C) in winter and 80°F (27°C) in summer. According to the Directory of Certified Applied Air-Conditioning Products[6], water-source heat pumps should be rated at a water source temperature of 70°F (21°C) for heating and 85°F (29°C) for cooling. Our data show a COP of 3.6 for heating and 2.6 for cooling at these temperatures. ARI cited values range between 2.4 and 3.9 for heating and 2.3 to 3.4 for cooling, indicating that the performance of the system used in this study is in the upper-range of commercially available units in the heating mode and in the lower range for cooling. Also, since the water-to-water heat pump was intended for heating only, cooling capactities were not supplied. Consequently, the cooling capacity at an inlet brine temperature of 85°F (29°C) was measured and found to be approximately 2 tons.

System Performance Factors

The overall seasonal performance factors (PF_i since the unit was within the conditioned space) for the two winter seasons were 2.6 for each season and were 1.11 and 1.35 for the 1983 and 1984 cooling seasons respectively, Table 2 shows the monthly performance values along with the degree days. It is seen that system performance generally decreases through the season, probably due to changes in soil conditions.

During February, 1983 the indoor water-to-air coil was replaced with a larger one better suited to the heat pump system and this apparently produced the increase in performance during March. The winter of 1983-84 incurred a significant increase in heating degree days over 1982-83 but the ground-coupled heat pump system maintained the same performance as during the 1982-83 winter. The effect of degree days on a ground-coupled system is through run time and it has been demonstrated that shorter run times allow the ground temperature to recover somewhat between runs and produce more favorable temperatures, resulting in higher performance.

System performance during the summer of 1983 was very poor and this poor performance led to the changes made in the ground coil prior to the summer of 1984. Significant soil drying beginning in June and continuing throughout the summer of 1983 apparently produced the decreasing performance as shown in Table 2. Also, some evidence of voids was found adjacent to the pipe which would become dry in the summer due to moisture migration away from the pipe. The sand backfill held a high value of moisture content longer than the clay and did not begin to dry out until early September when the cooling season was virtually over. However, the summer of 1984 had fewer degree days with resultant shorter average run times than in 1983 which indicates that perhaps not all of the increase in performance noted is due solely to the backfill improvement.

Soil Conditions

In order for a ground-coupled heat pump to effectively compete with an air-air system, the ground must normally be at a more favorable temperature than the ambient air. Figure 1 shows a plot of the difference between the far-field ground temperature and average daily ambient temperature for the 1983-84 heating season and the 1983 and 1984 cooling seasons. During the winter, the ground is consistently at a higher temperature than the ambient air with a mean difference of about 18°F (10°C). During summer the ground temperature was below ambient air, but the difference is smaller, indicating much less summer potential for system performance improvement over a standard air-air system. It should be noted, however, that because of the daily excursions of the ambient temperature above and below the mean, there are certain times of the day when the ground coupled system may have a significant advantage. For example, during winter

TABLE 2

COMPARISON OF PERFORMANCE FACTORS

Month	1982-83 Season			1983-84 Season		
	Degree Days °F*	PF_i	PF_o	Degree Days °F	PF_i	PF_o
November	457	2.94	2.03	544	2.79	2.29
December	592	2.64	2.27	900	2.77	2.25
January	845	2.39	1.80	969	2.53	2.07
February	661	2.44	1.91	682	2.53	2.02
March	371	2.81	2.30	585	2.38	2.00
Seasonal	2926	2.60	2.04	3680	2.59	2.15
June	235	1.37	1.67	289	1.34	1.72
July	425	1.14	1.36	255	1.43	1.70
August	462	1.14	1.21	326	1.35	1.69
September	217	1.06	1.22	77†	1.20	1.77
Seasonal	1339	1.11	1.31	947	1.35	1.71

* base 65°F

† September 1 - 16 only

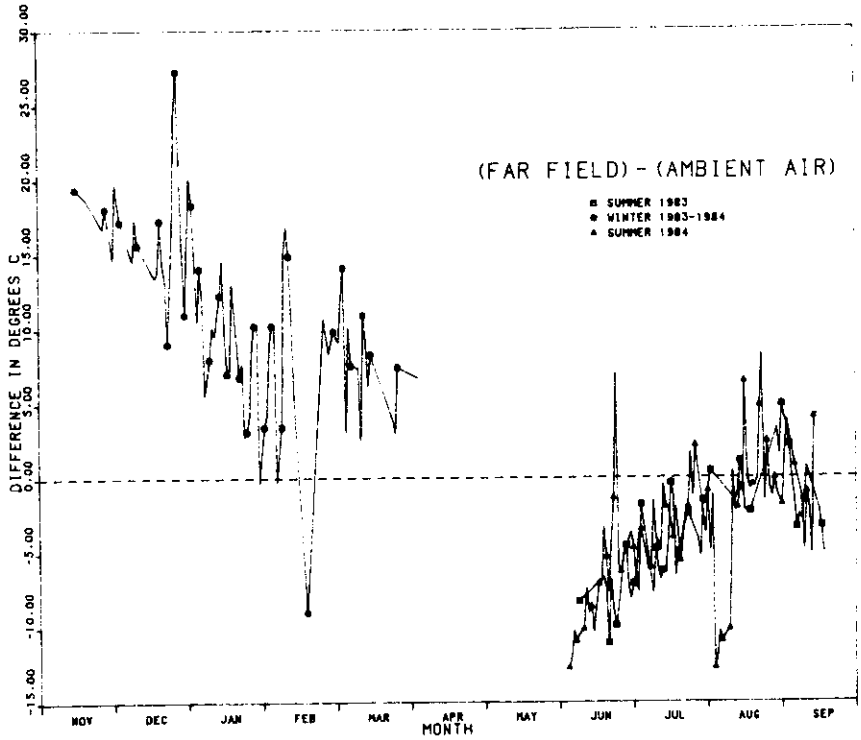


Figure 1 - Difference in the far-field soil temperature at 4 ft. depth and the average daily ambient air temperature.

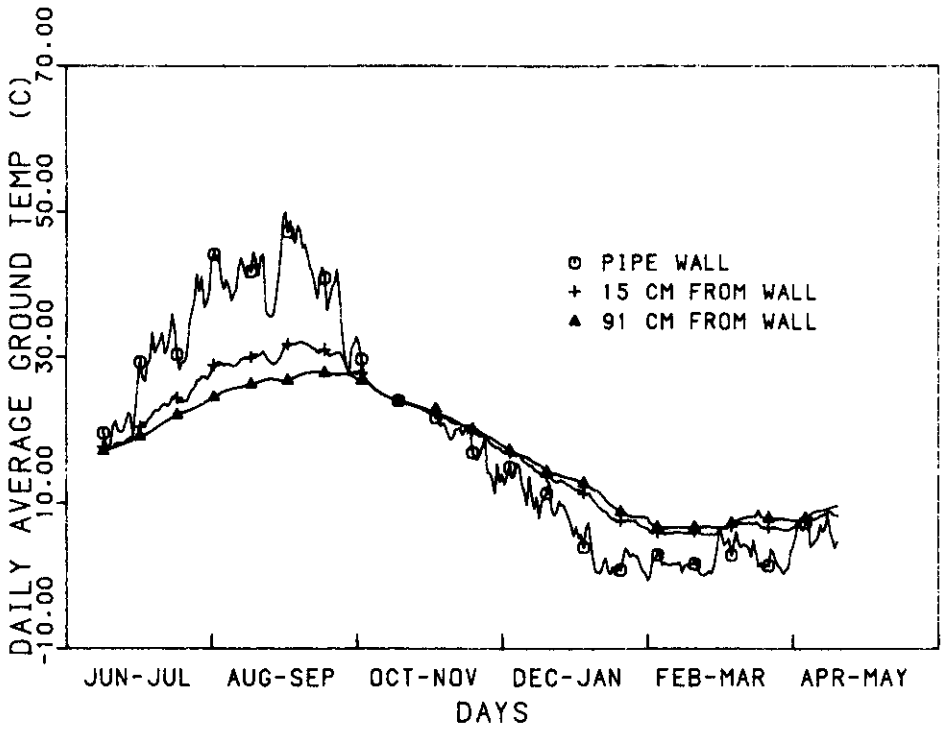


Figure 2 - Daily average ground temperatures near the ground coil (June, 1983 through May, 1984).

evenings, when the ambient temperature is well below the daily average, heating requirements are highest and system performance is most critical. It is at this time that the ground coupled system should show its greatest advantage.

The thermal response of the soil is a primary indicator of the ability of the ground to both store thermal energy and to transfer it away from (toward) the pipe. The average daily ground temperatures for different horizontal distances from the buried pipe are shown for the 1982-83 seasons on Figure 2. As expected, the temperature fluctuations of the soil are less prominent as distance from the pipe increases. In addition, the temperatures begin to approach the far-field values as the distance from the pipe reaches about 15 cm.

The difference between the brine outlet and far field temperatures as shown in Figure 3 shows the relative effectiveness of the ground as a heat exchanger between the cooling and heating seasons. The larger difference seen in the summer months shows the difficulty of dissipating the summer heat flux. This large temperature difference can be explained by a combination of higher ground heat flux during the summer and the lower soil thermal conductivity expected for the summer conditions.

Freezing temperatures were frequently observed in the soil near the pipe during January and February, 1983. Since these temperatures were measured at the midpoint of the ground coil, it is likely that more extensive freezing occurred near the first segment of pipe length where brine temperatures would be lowest. However, there were no temperature probes located in this region. Freezing of the soil and subsequent release of the latent heat has the effect of increasing the effective thermal conductivity of the soil during the winter season. Freezing effects are not likely to be dominant in the Knoxville area due to relatively mild winters and the additional lengths of ground coil required by summer design conditions.

Conclusions

Based on this study the following conclusions are presented:

1. Although at the outset of the study the system was considered optimized as designed by Batelle, the actual data indicate that the ground coil heat exchanger is undersized for summer operation. Operation during the winter season, however, was acceptable with an overall seasonal performance factor of 2.6 which is higher than a typical air-air heat pump.
2. An undersized ground coil in a ground-coupled heat pump system will produce a significant performance penalty.
3. Backfilling with clay must be carefully done to avoid having air pockets in the ground due to the tendency of clay to agglomerate.

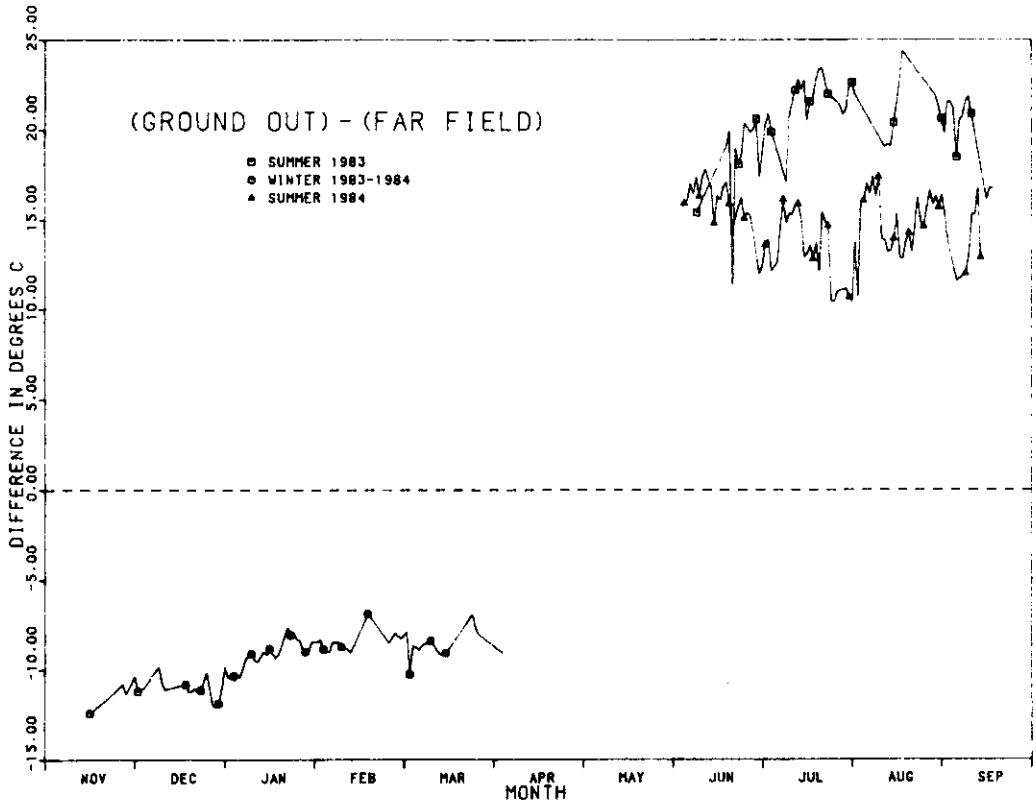


Figure 3 - Difference between brine temperature leaving the ground and the far-field temperature at 4 ft. depth.

4. Backfilling with sand around the ground coil produced improved performance of the system during the initial summer of operation.
5. During winter operation, no electric resistance backup heating was required.
6. A ground-coupled heat pump in this area of the country must be sized based on the cooling peak which will normally result in an over-design for the winter season. Such an overdesign may, however, be desirable because of reduced need for backup resistance heat.

Recommendations

Although numerous questions have been answered as a result of this study, many others have been discovered. Some of these are listed below.

1. What is the effect of keeping the ground artifically moist during the summer?
2. What is the effect of increasing the length of the ground coil?
3. Would the use of the heated brine to preheat domestic hot water in the summer be attractive?
4. How should the heat pump system be sized to take advantage of the recovery of the ground temperature during system off periods?

Items 1 thru 3 in this list are already included in the configuration of our test setup and future work will be directed at finding appropriate answers.

We are currently working with ORNL and some private industrial firms to design and install an improved, near optimum horizontal-coil ground coupled heat pump system in TECH House V at the TECH site. The system is scheduled to be tested during the summer of 1985 with the system described herein in TECH House I used as a standard for comparison.

References

1. Metz, P. D., "Design, Operation and Performance of a Ground Coupled Heat Pump System in a Cold Climate", Solar Technology Group, Brookhaven National Laboratory, 1981.

2. Bose, J. E., Oklahoma State University Division of Engineering Technology and Architecture, Proceedings from the 6th Heat Pump Technology Conference, Tulsa, Oklahoma, November 4-5, 1982.

3. Ball, D. A., Fisher, R. D., Talbert, S. G., "State-of-the-Art Survey of Existing Knowledge for the design of Ground-Source Heat Pumps, Batelle Columbus Laboratory, Published by the United States Departments of Energy, November, 1983.

4. Fisher, R. D., "Design of the Ground-Coupled Heat Pump System for the University of Tennessee Solar House",

Bateile Columbus Laboratories, Columbus, Ohio 1981.

5. Bedinger, A. F. G., Sullivan, W. G., "Economic Analysis of Solar Space Heating Based on Seasonal Performance Factor," ASME Solar Energy Division, 4th Annual Technical Conference, Albuquerque, NM, April 26-30, 1982.

6. Directory of Certified Applied Air-Conditioning Products, Air-Conditioning and Refrigeration Institute, 1983.

ORNL Ground Coil Analytical and Experimental Studies

Vince C. Mei

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

Vince C. Mei received the B.S., M.S., and Ph.D. degrees in mechanical engineering from the National Cheng-Kung University in Taiwan, the University of Toronto in Canada, and the Illinois Institute of Technology in Chicago, respectively. His major professional experience has been in the areas of heating, ventilation, air-conditioning, and refrigeration design and manufacturing. He has also had considerable experience in research and development in energy conservation (heat pump application).

ORNL GROUND COIL ANALYTICAL AND EXPERIMENTAL STUDIES*

V. C. Mei
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Objective

The goal of this research program is to understand the phenomenon of soil heat transfer coupled with fluid flow, soil moisture freezing, soil moisture migration, and ground surface temperature variation. From these studies, comprehensive ground coil mathematical models are being developed to better predict ground coil heat exchanger performance.

Background

The ORNL ground coil research program began in 1980. The fundamental concept is to be more rigorous in the theoretical analyses underlying development of mathematical models for different types of ground coils. The resulting models are then validated with laboratory or field experimental results. Once validated, the models can be used to design the ground coil.

The program has been loosely divided into three phases. Phase 1 was for ground coils involving heat transfer only, which was suitable for deep-well, tube-in-tube type ground heat exchanger analyses. Phase 2 is for ground coils involving heat transfer and soil moisture freezing, which are designed for horizontal ground coil winter operation. Phase 3 will be for ground coils involving heat transfer and soil moisture migration, which are designed for horizontal ground coil summer operation. The effect of ground surface temperature variation and thermal interference from the adjacent coil legs will be added to phases 2 and 3 at a later stage of the model development.

Our present analytical approach is based on calculating the amount of energy that the ground can provide or dissipate under specific operating conditions. Past analytical work[1-5] was based primarily on line source theory, in which the strength of the line source is specified. This latter approach thus predetermines the amount of energy to be absorbed or dissipated. Other factors, such as soil freezing and ground surface temperature variation, can easily be incorporated into our models. The models are also intended to be flexible and readily adaptable for different operating situations.

*Research sponsored by the Building Equipment Research Division, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

Technical Accomplishments

Completed the Analytical and Experimental Analyses for Deep-Well, Concentric-Tube Ground-Coupled Heat Exchangers

A mathematical model based on energy balance coupled with fluid flow[6,7] was solved numerically. The model was validated with laboratory experimental data obtained from a vertical, tube-in-tube heat exchanger 47 m (155 ft) deep. Figure 1 shows the configuration of the heat exchanger. Figure 2 shows the excellent agreement between the calculated and experimental results. The parametric study indicates that the size and material of the

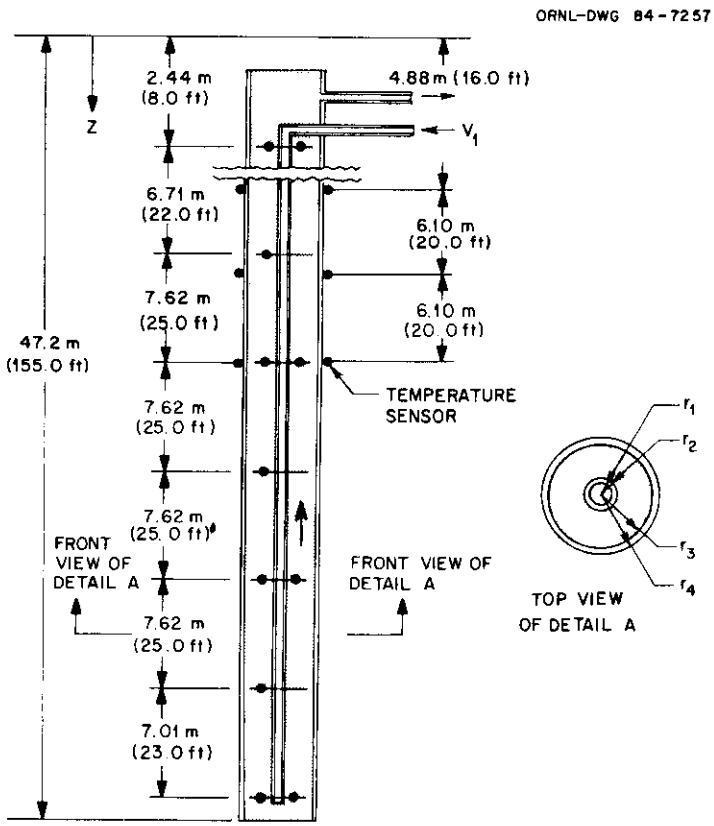


Fig. 1. Schematic of vertical, concentric-tube ground-coupled heat exchanger.

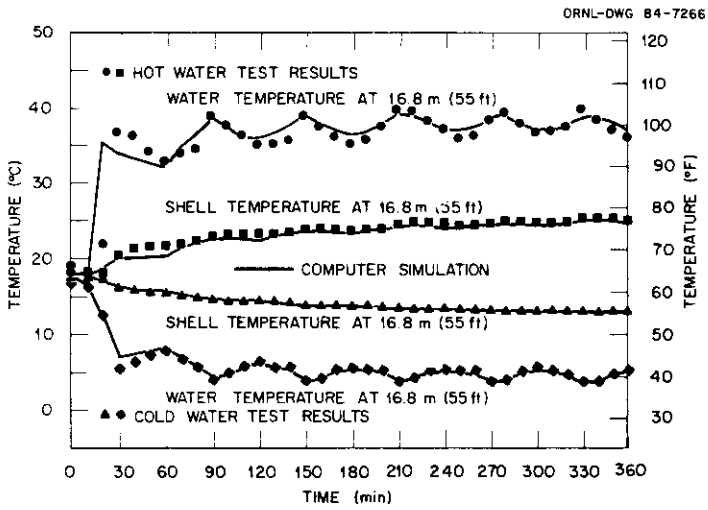


Fig. 2. Validation of hot and cold inlet water cyclic operation.

exchanger casing and the length of the exchanger are important factors in determining the performance of the ground heat exchanger (Figs. 3 and 4).

Completed the Horizontal Ground Coil Model Coupled with Soil Freezing

This model was designed for horizontal ground coil winter operation, particularly in the northern part of the United States

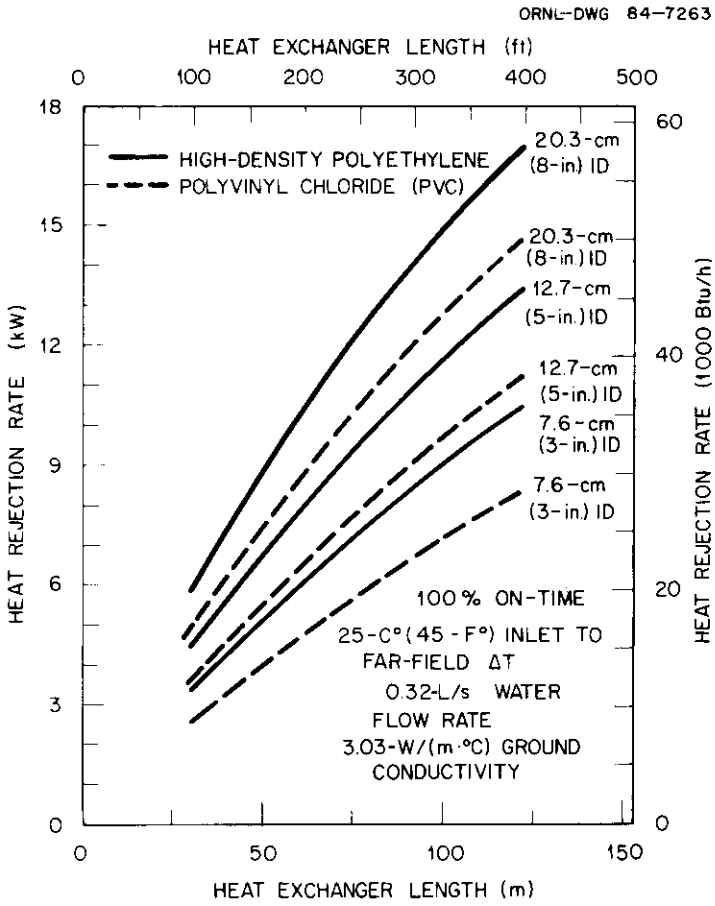


Fig. 3. Effect of casing size and material on heat rejection rate.

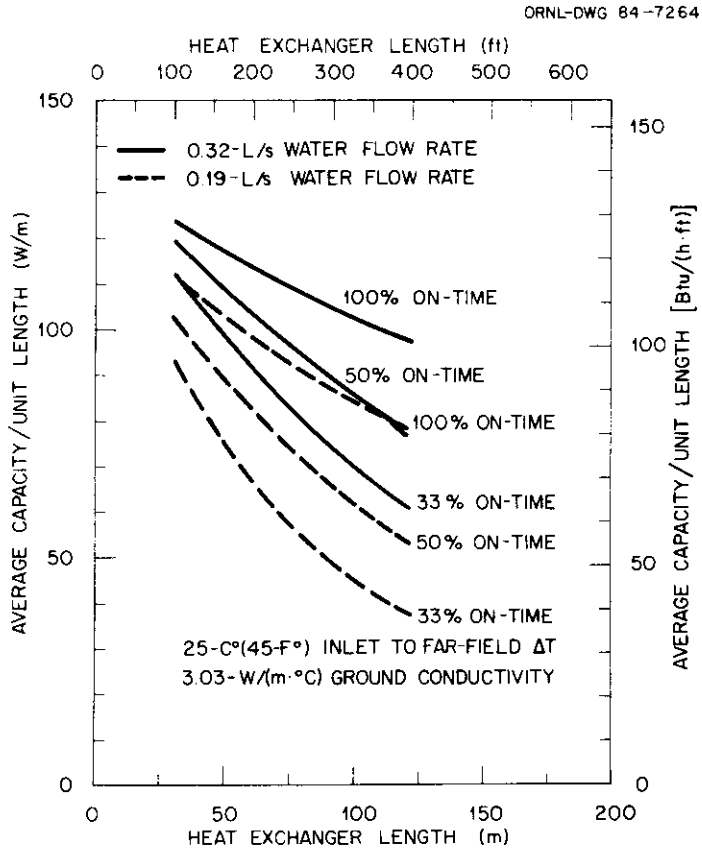


Fig. 4. Effect of well depth on heat exchanger performance.

where the soil around the coil would be frozen because of low fluid temperature. This model has been solved numerically and has been partially validated with field test data provided by Brookhaven National Laboratory (BNL).[8]

Soil freezing around the ground coil has several major effects on ground coil performance:

1. the thermal conductivity of the soil frozen region changes, depending on type of soil, moisture content, and temperature;
2. latent heat released by the moisture freezing is absorbed by the ground coil; and
3. a storage effect is created which enables the ground coil to absorb energy continuously even when the heat pump is not in operation.

Figure 5 shows the comparison between calculated and experimental results of total energy absorption by the ground coil. The calculated results are about 9% less than the experimental data. In other words, the computer code seems to give a conservative design. Figure 6 shows the importance of soil freezing around the coil. If the inlet ground coil fluid temperature is −12.2°C (10°F) instead of −3.2°C (26.3°F), as for the base run, the total energy absorbed will be 2.5 times higher. If the effect of soil freezing is not considered, there is a substantial error in predicting the total energy absorbed by the coil. As Fig. 6 illustrates, the lower the allowable coil inlet fluid temperature, the more important it is to account for the effect of soil freezing.

This model has been described in detail in ref. 9.

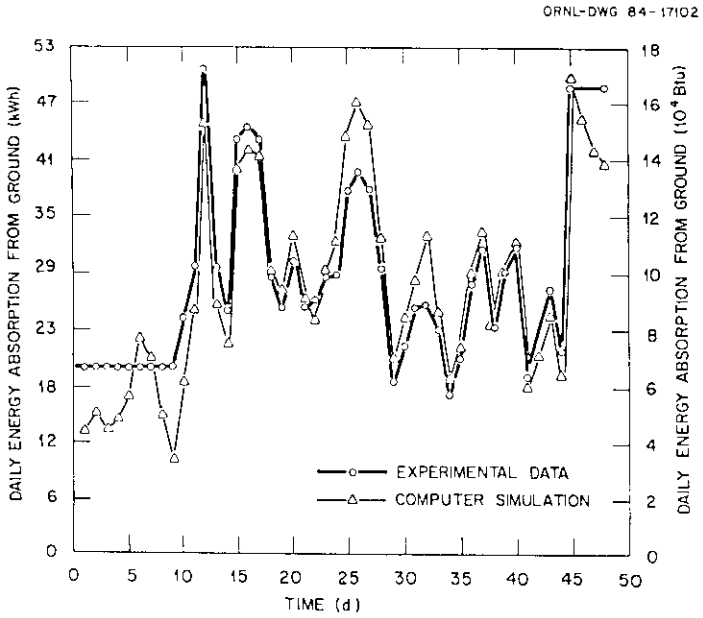


Fig. 5. Comparison of horizontal ground coil theoretical and experimental results on total energy absorption.

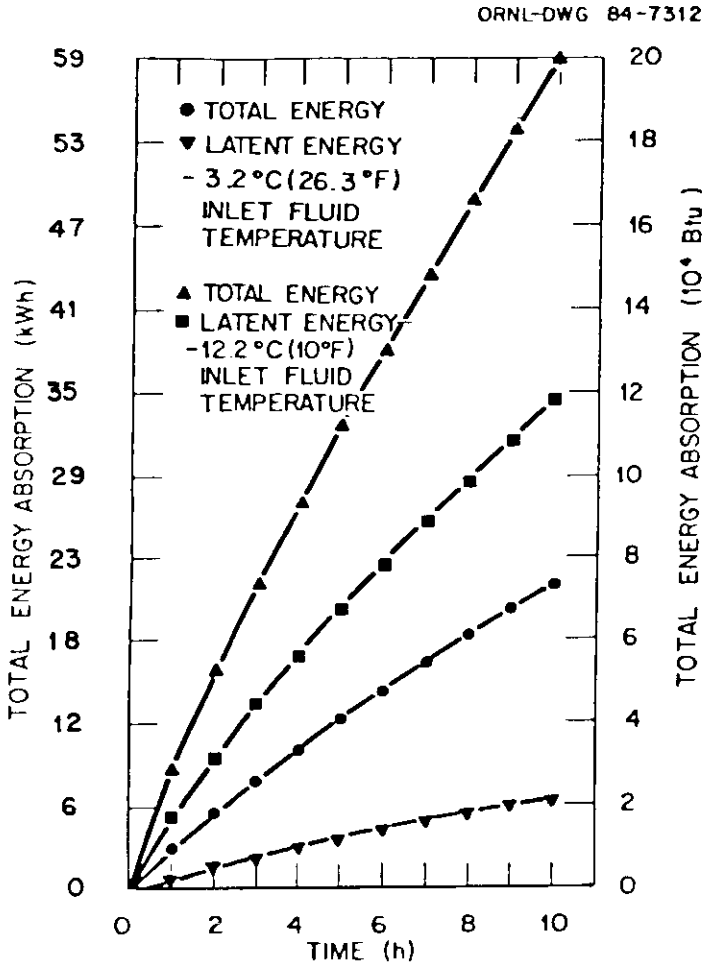


Fig. 6. Effect of soil freezing on ground coil energy absorption.

Completed the Horizontal Ground Coil Model Coupled with Ground Surface Temperature Variation

This model is two dimensional instead of one dimensional as the previous ones. The far-field boundary conditions become a function of time and depth, which is calculated by Kusuda's equation.[10] The advantage of this model is that the effect of coil burial depth can be taken into account. However, the soil freezing effect cannot be added easily without assuming that the soil frozen region is radially symmetrical.

Figure 7 shows a 44-d simulation of the BNL field experiment. The calculated daily total energy absorption is slightly lower than the measured ones. However, some factors that cannot be taken into account in the model, such as rain and snow, may have boosted the measured heat pump performance. Figure 8 shows the

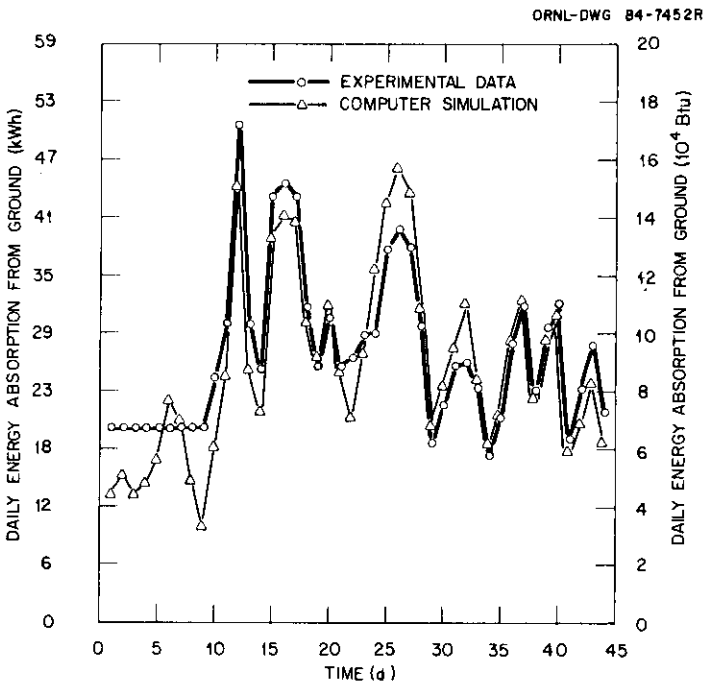


Fig. 7. Computer simulation of 44 d of BNL field test results on ground coil total energy absorption with effect of ground surface temperature variation.

calculated temperature distribution at the ground coil inlet after 41 d. Obviously, the temperature distribution is far from being radially symmetrical.

This model has been described in detail in ref. 11.

Completed the Test for Shallow-Well "U"-Tube Ground Coil Heat Pump System

Table 1 shows monthly ground coil cooling and heating performance. The ground coil system performs better than an air-source heat pump system in the heating season, but the latter is

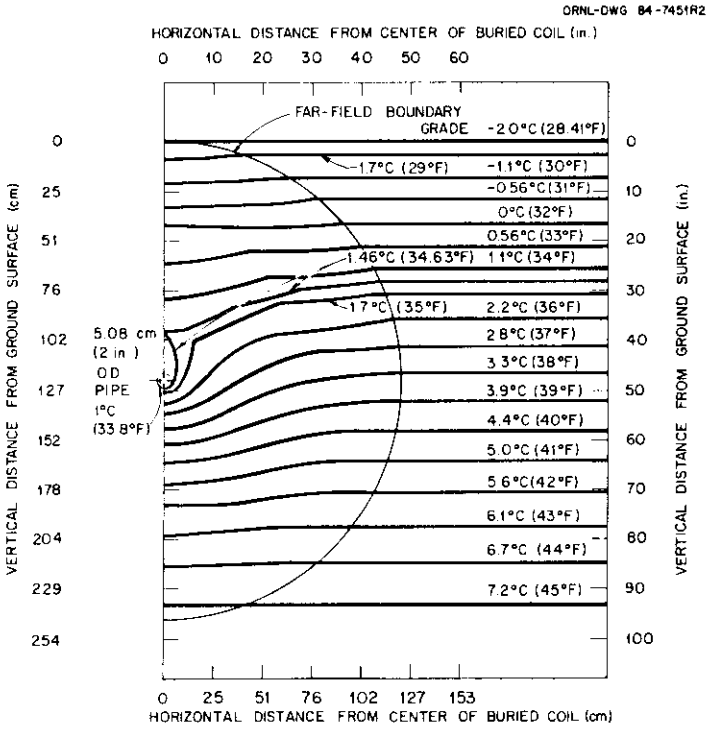


Fig. 8. Computer calculated ground temperature distribution after 42-d simulation.

Table 1. Monthly ground coil system performance for 1983 cooling and 1983–84 heating seasons[a]

	Cooling		
	July	August	September
Capacity output, kWh	2095	2196.3	1261.6
Power output, kWh	1174	1284.4	700.9
COP[b]	1.78	1.71	1.80

	Heating				
	November	December	January	February	March
Capacity output, kWh	1203.9	2381.8	3479.6	3216.5	2136.2
Power output, kWh	389.6	783.5	1242.7	1157.0	771.2
COP[b]	3.09	3.04	2.80	2.78	2.77

Seasonal performance factor		Annual performance factor
Cooling	Heating	
1.76	2.86	2.4

[a]Compressor is located indoors.
[b]Coefficient of peformance.

better for the cooling season. Nevertheless, the ground coil system achieved a respectable annual performance factor of 2.4.

Future Activities

Thus far, we have developed various advanced ground coil models for different types of ground coils and various factors affecting soil heat transfer. However, at this writing, we have not validated our models adequately except for the deep-well, concentric-tube ground heat exchangers. Cooperation between ORNL and Niagara-Mohawk Power Company will enable us to obtain much-needed horizontal coil field experimental results for a variety of soils which can be used to validate our models. However, the task to combine soil moisture freezing and surface temperature variation into a single model will have to be performed first.

For horizontal ground coil summer operation, the temperature differential may cause soil moisture migration. Efforts are under way to obtain computer codes for buried cable analyses,[12,13] which describe soil moisture migration at constant heat flux, so that the severity of moisture migration can be estimated first before investing major effort in this phase 3 model development. There are cooperative activities between ORNL and the Canadian National Research Council (CNRC) in soil heat and mass transfer research. ORNL will perform theoretical and numerical analyses, and CNRC will conduct the experimental studies. The decision to develop a horizontal ground coil model with soil moisture migration will be based on these preliminary theoretical and experimental studies.

For the vertical, shallow-well, "U"-tube type ground coil, efforts will be focused on developing a satisfactory mathematical model.

References

1. L. R. Ingersoll and H. J. Plass, "Theory of the Ground Pipe Heat Source for the Heat Pump," *ASHVE* **54,** 339–94 (July 1948).
2. C. H. Coogan, "Heat Transfer Rates, Experimental Determination for Heat Absorbing Coil Buried in the Earth," *Mech. Eng.* **71,** 495–98, 1055–56 (1949).
3. E. B. Penrod, "Sizing Earth Heat Pumps," *Refrig. Eng.* **62**(4), 57–61, 108 (April 1954).
4. D. M. Vestal and B. J. Fluker, "Earth as Heat Source and Sink for Heat Pumps," *ASHRAE Trans.* **63,** 41–69 (1957).
5. M. D. Kalman, "Earth Heat Exchangers for Ground Source Heat Pumps," M.S. thesis, Georgia Institute of Technology, Atlanta, 1980.
6. V. C. Mei and S. K. Fischer, "Vertical Concentric Tube Ground-Coupled Heat Exchangers," *ASHRAE Trans.* **89,** 391–406 (1983).
7. V. C. Mei and S. K. Fischer, *A Theoretical and Experimental Analysis of Vertical, Concentric-Tube Ground-Coupled Heat Exchangers,* ORNL/CON-153, Oak Ridge National Laboratory, October 1984.
8. P. D. Metz, Brookhaven National Laboratory, personal communication to V. C. Mei, Oak Ridge National Laboratory, Sept. 21, 1983.
9. V. C. Mei and C. J. Emerson, "New Approach for Analysis of Ground-Coil Design for Applied Heat Pump Systems," submitted for presentation at the ASHRAE 1985 Summer Meeting.

10. T. Kusuda et al., "Earth Temperature and Thermal Diffusivity at Selected Stations in the United States," *ASHRAE Trans.* **71** (Pt. 1), 61–75 (1965).

11. V. C. Mei, "Theoretical Heat Pump Ground Coil Analysis with Variable Ground Far-Field Boundary Condition," to be presented at the National Heat Transfer Conference, Denver, Colo., August 1985.

12. J. G. Hartley, "An Analysis of the Thermal Stability of the Soil Environment of Underground Electric Cables," Ph.D. dissertation, Georgia Institute of Technology, Atlanta, 1978.

13. R. J. Couvillion, "Heat and Mass Transfer in a Semi-Infinite Moist Soil with a Drying Front Present," Ph.D. dissertation, Georgia Institute of Technology, Atlanta, 1981.

INTERNATIONAL ACTIVITIES

U.S. Department of Energy Cooperation in Advanced Heat Pump Research Through the International Energy Agency

John D. Ryan

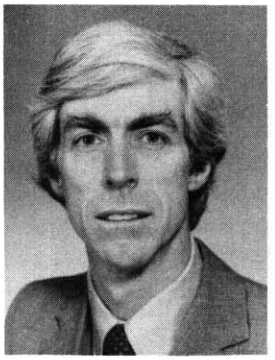
U.S. Department of Energy

Washington, D.C. 20585

John D. Ryan received B.S. and M.S. degrees in mechanical engineering from Virginia Polytechnic Institute and Stanford University, respectively. His research at Stanford was principally in engineering economic systems. He began his professional career at Pratt & Whitney Aircraft (now United Technologies) conducting analytical studies of new gas turbine systems for industrial and aircraft applications.

Under the Presidential Internships in Science and Engineering Program, Ryan joined the National Bureau of Standards in 1972 to work on energy and environmental programs. His main contributions were in the evaluation of cogeneration systems, for which he received the U.S. Department of Commerce Bronze Medal in 1975.

Ryan joined the U.S. Department of Energy (DOE) in 1979. Since 1981 he has been directing DOE's Energy Conversion Equipment Branch, with a program in research and development of building energy equipment. The scope of this program includes gas and oil furnaces and boilers, wood and coal-fired equipment, electric and thermal heat pumps, refrigeration systems, appliances, and lighting.



U.S. Department of Energy Cooperation in Advanced Heat Pump Research Through the International Energy Agency

John D. Ryan

U.S. Department of Energy
Washington, D.C. 20585

The International Energy Agency

The growing need for international co-operation in energy led to the establishment of the International Energy Agency (IEA) in 1974 as a forum for the 21 participating countries to coordinate their energy planning. The IEA provides a framework within which the co-operative efforts of its participating countries will work to improve the overall energy situation. The Agency carries out the International Energy Program and a Long-Term Co-operation Agreement and works:

- to mold a better world energy supply and demand structure, now and for the future;
- to prepare participating countries against risk of oil supply disruptions and to share remaining oil supplies in a period of severe supply disruption;
- to develop alternative energy sources and to increase the efficiency of energy use through co-operative research and development programs;
- to promote co-operative relations with oil-producing nations and other oil consuming countries.

Organization and Secretariat

The IEA is an autonomous agency within the Organization for Economic Co-operation and Development (OECD). Its principal decision making body is the Governing Board, composed of senior level representatives from each participating country. The Governing Board also meets at Ministerial Level from time to time. A Secretariat, headquarters in Paris, with a permanent staff of energy experts drawn from Member countries, supports the work of the Governing Board and subordinate bodies.

The IEA Secretariat collects and analyses energy data, reviews Member countries' domestic energy policies and programs, makes forecasts, undertakes and publishes special studies. (A list of recent publications can be found at the end of this paper.)

The Governing Board directs the activities and makes the major policy decisions of the IEA. In most cases, issues are decided by consensus; where a vote is required, a system of weighted voting would apply.

Standing Groups

The Governing Board is supported in its work by four Standing Groups and a High-Level Committee. These groups comprised of senior-level energy specialists from the participating countries, convene in Paris on a regular basis. Their responsibilities related to a particular area of the IEA program:

- Long-Term Co-operation - promotes conservation and efficient use of energy, increased production and use of alternative sources of energy and other measures designed to reduce long-term dependence on oil;

- o Oil Market Information - receives, collects, and analyses information on the international oil market and oil company activities.

- o Emergency Questions - refines, tests and maintains on a stand-by basis the IEA Dispute Settlement Center for supply emergencies;

- o Relations with Producer and Other Consumer Countries - encourages co-operative relations with other nations, both oil producers and non-Member oil consumers;

- o Committee on Energy Research and Development - encourages accelerated research use and develop alternative sources through co-operative multi-national projects.

Energy Research and Development

The Committee on Energy Research and Development (CRD) supervises IEA efforts to reduce Member countries' long-term dependence on imported oil through energy research, development and technology demonstration. It conducts annual reviews of Member countries RD&D programs and coordinates multi-national energy RD&D projects.

The Committee seeks to encourage the implementation of a IEA Group Strategy for Energy Research, Development and Demonstration for more effective national energy RD&D programs. A group strategy report suggests which new technologies could contribute most to IEA energy needs of the next 40 years, and also serves as a guide for individual IEA national energy RD&D programs.

Each year the Committee reviews IEA Countries' RD&D policies and programs. Domestic reports are reviewed in order to monitor the relationship between national energy RD&D programs and national needs and resources, as well as the potential contributions these programs can make to IEA's energy RD&D objectives, both individually and as a group of nations.

Emphasis is placed on early commercialization of the most promising oil alternative and energy conservation technologies. Their accelerated introduction could provide Member countries with a wider choice in reshaping their energy future.

More than 70 international collaborative energy research projects are now in progress under the sponsorship of the IEA. International teams participate in research, development and demonstration projects in areas of:

- o End-use efficiency

- o Fossil fuels

- o Renewable Energy

- o Fusion

For example, at Grimethorpe in the United Kingdom, a cleaner and more efficient method of burning all types of coals is being tested. In Almeria, Spain, two electric power producing solar plans are being compared to find the most favorable technology for the future. In New Mexico, an artifical geothermal system to extract heat from hot dry rocks is being examined.

Advanced Heat Pumps Research

In the End-Use Efficiency Area, one of the most prolific areas of cooperative research is Advanced Heat Pumps. The implementing agreement for this area was signed by thirteen IEA member countries in 1978. The inaugural project was a Common Study on Advanced Heat Pumps (Annex I). This project reviewed the state-of-the art in advanced and electric and thermally-activated heat pumps and examined the market potential in each of the thirteen countries. This comprehensive study, costing in excess of $800,000, set the stage for future cooperation by defining areas of mutual interest in all countries and by defining R&D needs.

To date, a total of seven additional projects (annexes) have been intiated and several more are under negotiation. These seven projects are discussed briefly below and summarized with pertinent data in Table 1. The projects can be either Task Sharing whereby work elements are carried out in each country, without an exchange of funds, or Jointly Funded, whereby work is carried out by a single organization funded by the participating countries. A combination of the two appears to be the most productive approach to date.

Annex II, Vertical Eath Heat Pump Systems, was directed analytically studying the potential of vertical-configuration earth source heat pumps. The principal investigator (Operating Agent) was Sweden who was able to draw upon substantial national experience in earth coupling and apply this to the situation in four other countries working with researchers thereform, including the U.S.

Annex III, Industrial Heat Pumps, was directed toward summarizing the state-of-the art and working experiences with industrial heat pumps. A series of case studies was produced, one or more from each country and a number of site visits were arranged to interesting plants. The U. S. did not participate in this now-completed Annex.

Annex IV, the Heat Pump Center, is an effort to improve lines of communication on heat pumps by establishing a standing organization to facilitate information exchange. The Center is located at the Fachinformationszentrum in Karlsruhe, West Germany and also draws on technical resources at a research institute in Graz, Austria. A feature of this Center is the goal of synthesizing information and focusing attention on major information sources and topics to assist users in finding only the most revelant information. The U.S. organization which is facilitating transfer to and from the U.S. is the Oak Ridge National Laboratory.

Annex V, Large Heat Pumps for District Heating, focused on an intensive case study of a few actual examples of large heat pump systems. The United States did not participate.

Annex VI, Working Fluid Mixtures and High Temperature Working Fluids for Compressor Driven Systems, consists of a literature search and state-of-the-art review of technology in six countries, including the United States. The Operating Agent is Sweden.

Annex VII, New Developments in Heat Pump Evaporators, is a new annex aimed at exploring novel techniques in heat source/heat exchange technology. The United States is not participating.

Annex VIII, Advanced In-Ground Heat Source and Storage Technology, is a follow-on project to Annex II. This consists of a cooperative exchange between the U.S. and Canada. Canada is producing new data on heat source/sink heat transport phenomena and the United States is producing analytical models, verified by the new data.

A number of other new annexes are under discussion or negotiation at present. The topic areas for these new annexes are:

- o modeling techniques for simulation and design of compression heat pumps
- o Stirling engine technology assessment
- o calculation of heat and mass transfer in absorption systems.
- o high temperature industrial heat pumps
- o non-thermophysical properties of refrigerants and working fluid candidates

Decisions regarding U.S. participation in these potential new annexes plus the identification of new areas of research appropriate for international collaboration should include strong input and guidance from U.S. manufacturers, utilities and research organizations. These groups are strongly encouraged to provide such input to DOE (John Ryan or Ronald Fiskum) and ORNL (Fredrick Creswick).

Publications

IEA Secretariat

The IEA Secretariat has prepared a number of publications which are of potential interest to heat pump researchers. These are:

- o Heat Pump Systes, A Technology Review
  This report is one of a series of technology reviews published by the IEA. The report includes a brief overall state-of-technology overview for each of fourteen countries. (ISGN 92-64-12378-4)

- o Annual Report on Energy Research, Development and Demonstration, Activities of the IEA 1982/83
  This report summarizes RD&D collaborative work undertaken through the IEA in end-use efficiency, fossil fuels, renewable energy and fusion energy.

- o Energy Research, Development and Demonstration in The IEA Countries, 1982 Review
  This annual review of energy policy and energy RD&D policy provides government and industrial RD&D budgets and program elements. Individual reports are provided on twenty-one countries. (ISBN 92-64-12518-3)

General publications of the IEA Secretariat can be obtained through OECD sales agents worldwide. In the United States the sales agent is:

OCED Publications and Information Center
Suite 1207
1750 Pennsylvania Avenue, N.W.
Washington, DC 20006
Telephone (202) 724-1857

Advanced Heat Pumps

Each Annex under the implementing agreement would normally have a final report documenting the project's results. These are usually available only within the countries which have participated in the project. Currently, there are two completed projects in which the United States participated. The resultant publications and their availability are noted below:

Annex I. Common Study

Common Study on Advanced Heat Pump Systems
Final Report, Volume I, Results and Conclusion
Available from: Verlag TUV Rheinland GmbH
Sicherheitstechnik, Energie &
Umweltschutz
Am Grauen Stein
5000 Koln 91
Federal Republic of Germany

Common Study On Advanced Heat Pump Systems
Final Report, Volume II, Technology Survey
Available from: Oak Ridge National Laboratory

Common Study On Advanced Heat Pump Systems
Final Report, Volume III, Market Study
Available from: Oak Ridge National Laboratory

Annex II. Vertical Earth Heat Pump Systems

International Energy Agency Advanced Heat Pumps,
Annex II
Vertical Earth Heat Pump Systems,Final Report
Available from: Oak Ridge National Laboratory

Annex VI. Fluid Mixtures and High Temperature
Fluids (Partly-Completed Project)

High Temperature Working Fluids and Non-
azeotropic Mixtures in Compressor-Driven Heat
Pumps - State of the Art Report
Available from: Oak Ridge National Laboratory

TABLE 1

IEA IMPLEMENTING AGREEMENT ON ADVANCED HEAT PUMPS

STATUS SUMMARY

ANNEX #	TITLE	STATUS	PARTICIPANTS	OPER AGENT	WORKING PERIOD	TYPE	TOTAL BUDGET	UNITED STATES CONTRIBUTION
I	Common Study On Advanced Heat Pumps	COMPL	AUS, BEL, CAN DEN, GER, ITA JAP, NET, SPA SWE, SWI, UK USA	GER USA	10/78 - 7/80	JF	DM 1,506,830 i.e. $824,990	DM 400,000 (27%) i.e. $219,000
II	Vertical Earth Heat Pump System	COMPL	AUS, CAN, DEN SWE, USA	SWE	1/81 - 7/84	COM	SK 900,000	SKR 320,000 (36%) i.e. $63,000
III	Industrial Heat Pumps	COMPL	AUS, BEL, CAN DEN, IRE, ITA JAP, NETH, SPA SWE	BEL	1/81 - 11/84	COM	?	Not Participating
IV	Heat Pump Center	ACTIVE	AUS, CAN, GER ITA, JAP, NETH SWE, USA	FIZ	12/82 - 12/85	COM	DM 1,777,500 i.e. $592,500	DM 420,0000 (24%) i.e. $140,000
V	Large Heat Pump for District Heating	ACTIVE	DEN, GER, ITA SWE	DEN	2/82 - 2/85	COM	DM 230,000	Not Participating
VI	Working Fluid Mixtures and High Temperature Working Fluids for Compressor Driven Systems	ACTIVE	AUS, DEN, FIN GER, SWE, US	SWE	3/82 - 9/84	COM	SK 552,500 i.e. $110,600	SK 107,000 (19%) i.e. $21,000
VII	New Developments in Heat Pump Evaporators	NEW	CAN, DEN, FIN GER, SWE	SWE	1/85 - 6/86	JF	$50,400	Not Participating
VIII	Advanced In-ground Heat Source & Storage Technology	NEW	CAN, US	CAN	1/85 - 1/87	TS		(50%)

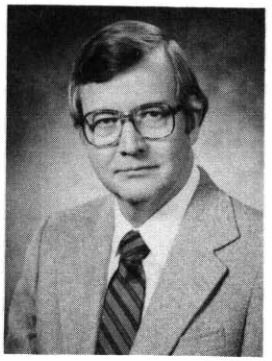
*TYPE: JF - Jointly Funded TS - Task Sharing COM - Combination of Joint Funding and Task Sharing

The International Energy Agency Heat Pump Center

Frederick A. Creswick

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831



Frederick A. Creswick is a group leader in the Building Equipment Research Program at ORNL. One of his responsibilities in this assignment is to serve as alternate U.S. delegate to the International Energy Agency Working Party on Advanced Heat Pumps. Prior to joining ORNL in 1977, he worked at Battelle-Columbus Laboratories for 18 years on automotive, aerospace, and heating, ventilation, and air-conditioning thermal systems. He also worked for 7 years at General Motors Research Laboratories on gas-turbine engine development.

THE INTERNATIONAL ENERGY AGENCY HEAT PUMP CENTER

F. A. Creswick
Oak Ridge National Laboratory
Oak Ridge, Tennessee

The previous speaker has described the International Energy Agency (IEA) Working Party on Advanced Heat Pumps and DOE participation in that group. One of the activities of the organization has been to establish and operate the IEA Heat Pump Center (HPC). The purpose of my presentation is to give a brief description of the HPC, its activities, and the potential benefits to U.S. organizations.

Objectives

It might be said that the perceived need for a heat pump center was basically a European perspective. All but three of the member countries of the IEA Working Party on Advanced Heat Pumps are European; the exceptions are the United States, Canada, and Japan. Both the energy situation and the situation regarding heat pumps are different in Europe than in North America and Japan. In Europe, residential heat pumps are marginally cost-effective, primarily because there is essentially no residential cooling market. Thus, the base-case competitive system is an oil-fired boiler rather than a boiler and a chiller. Accordingly, European heat pump sales have been very small in comparison with U.S. and Japanese sales.

Nevertheless, most European countries could improve their energy situation significantly through widespread use of heat pumps. From this viewpoint, the planners of the HPC, who were mostly government officials or their representatives, formulated the following set of objectives of the HPC:

1. to expedite commercialization of heat pumps by eliminating noneconomic barriers;
2. to promote regular contacts and information exchange among researchers, engineers, and government officials in the field of heat pumps;
3. to encourage international exchange of technologies; and
4. to improve the quality, reliability, and accessibility of information concerning heat pumps.

Some of the noneconomic barriers implied by the first objective are lack of installation and maintenance infrastructure; lack of uniform, reliable information on heat pump products; lack of uniform testing and approval procedures; and institutional and regulatory barriers.

Heat Pump Center Contractor and Organization

The concept of the HPC was approved in 1980 by the IEA and supported by Austria, the Federal Republic of Germany, Italy, Japan, the Netherlands, Sweden, and the United States of America. As of this writing, Canada and the United Kingdom have also expressed an interest in supporting the HPC.

Requests for proposals were issued in a competitive procurement, and the prime contract was awarded to Fachinformationszentrum (FIZ) Energie Physik Mathematik GmbH of Karlsruhe, Federal Republic of Germany. Subcontractors to FIZ were a consortium of organizations in Graz, Austria, which included the Graz Technical University. Under the present HPC organization, FIZ provides overall management as well as information services, and the Graz group provides various analytical functions. The HPC is also supported by a national team in each participating country. Each team has the function of providing certain types of information to the HPC and serving as a point of contact for the services of the HPC. In the United States, the national team effort is provided by Oak Ridge National Laboratory.

Activities of the Center

The HPC has now been in operation for about one and one-half years. Its main activities have been the following:

1. quarterly newsletter,
2. RD&D project summaries,
3. product information data base,
4. survey of heat pump standards,
5. bibliography on nonazeotropic mixed refrigerants, and
6. workshop on heat pumps for retrofit in the residential sector.

The RD&D project summaries and the bibliography on nonazeotropic refrigerant mixtures will be available in print in the

next few months. The workshop on retrofit will take place around the middle of 1985.

Benefits to U.S. Organizations

Understandably, U.S. objectives in participating in the HPC do not fully coincide with the stated objectives of the center. As researchers, we have been aware that publicly supported R&D in Europe and Japan is substantially greater than that being conducted in the United States. Accordingly, we feel an obligation to keep abreast of foreign developments to ensure that U.S. activities do not wastefully duplicate work already conducted elsewhere and to make available foreign technology that could enhance U.S. R&D efforts. We feel that these benefits alone are well worth the cost of U.S. participation.

Another benefit—one which has not been exploited as yet—is that the HPC is a potential source of information on the European heat pump market. While this market is not large at present, it has attracted some major Japanese manufacturers. Although we do not know the extent of U.S. manufacturers' interest in the European market, we do believe that the HPC can be a useful source of information for those who maintain surveillance of this market.

How to Obtain Services of the HPC

Individuals or organizations in the United States can contact the HPC directly or can make contact through ORNL. To contact the HPC directly, call, write, or telex

Mr. K. O. Holzapfel
IEA Heat Pump Center
Fachinformationszentrum-Karlsruhe
D-7514 Eggenstein-Leopoldshafen 2
Federal Republic of Germany
Telephone 49 (7247) 824-541
Telex 7 826 487 FIZE D

To make contact through ORNL, call or write

Fred Creswick or
Kay Zimmerman
ORNL
P.O. Box Y
Oak Ridge, TN 37831
Telephone (615) 574-2009 (Fred)
576-4921 (Kay)

We are eager to have the United States derive maximum benefit of our participation in the HPC, and we welcome your requests or questions.

HEAT—ACTUATED HEAT PUMPS—I

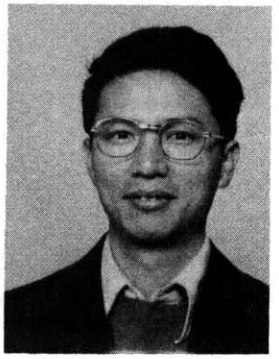
Stirling-Engine-Driven Heat Pump Program Overview

Fang C. Chen

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

Fang C. Chen is a group leader in the Energy Division of Oak Ridge National Laboratory and is responsible for Stirling-engine-driven and shipboard heat pump projects. He received his B.S. degree in mechanical engineering from National Taiwan University and his Ph.D. in applied physics from Harvard University. Before joining the Laboratory, he was with the NUS Corporation in Rockville, Maryland, where he conducted studies in thermal environmental engineering. He also worked with the TRW Energy Systems Group in McLean, Virginia, in energy conversion and conservation.



STIRLING-ENGINE-DRIVEN HEAT PUMP PROGRAM OVERVIEW*

F. C. Chen
Oak Ridge National Laboratory
Oak Ridge, Tennessee 37831

Abstract

This report presents a brief summary of the Stirling-engine-driven heat pump project being carried out at Oak Ridge National Laboratory (ORNL) for the Department of Energy (DOE). Aspects of the project discussed herein are: objective and approach, early development, current status and accomplishments, and future technology options.

Introduction

Approximately 55% of the 75 million homes in the United States are heated by natural gas. For summer space cooling, these gas-heated homes require electric air conditioners. Conventional combustion heating equipment currently achieves less than 70% service efficiency. More advanced equipment achieves near 90%; however, even when developed to its ultimate potential, it cannot exceed 100% efficiency nor provide space cooling.

Stirling-engine-driven heat pumps are a class of thermally activated heat pumps that (1) have a potential heating coefficient of performance (COP) exceeding 1.0 (i.e., delivering a heating effect greater than the input fuel heating value) and (2) can provide space cooling in the summer. The gas-fired, Stirling-engine heat pump is emerging as a promising option for improved fuel utilization and economy in future building space-conditioning.

Many different system concepts of the Stirling-engine-driven heat pumps have been conceived. Development of the technology has focused on Stirling-engine-driven, Rankine-cycle heat pumps. This report presents a brief overview of the Stirling-engine-driven heat pump project at ORNL for DOE.

Project Objective and Approach

The project objective is to promote the development of Stirling-engine-driven heat pumps that will lead to more efficient use of fossil fuels in the space conditioning of residential and commercial buildings. The technical approaches taken to pursue the project objective include systems analysis, engineering systems development, and research and technological development in supporting fields.

The systems analysis activities consist of technology assessment to evaluate state-of-the-art development, analytical studies to determine savings potential and economic viability for each candidate system, and market application studies to encourage the involvement of industry. System analysis results will define the cost and performance goals of Stirling-engine heat pump technology, determine the possibilities of competition with other thermally activated heat pump technologies, and predict the technology development required to realize the potential of the system.

Engineering system development that consists of research and development of Stirling-engine heat pump systems and/or subsystems is ongoing. We intend to establish experimentally the feasibility of the most promising concepts. Several hardware iterations based on fundamental research and component analysis of perspective concepts may be required before cost and performance goals can be achieved.

Supporting research consists of fundamental research on basic transport processes and materials and development of component technology and design methodology. These activities will contribute to the data base necessary to develop cost-effective Stirling-engine heat pump technologies.

The performance targets for the near term, excluding auxiliary parasitic losses, are as follows.

	Residential	Commercial
HCOP[a,b]	1.7	1.9
CCOP[a]	1.0	1.1

[a]Steady state heating and cooling COPs at 47°F and 95°F, respectively.

[b]80% waste-heat utilization.

The major design requirements of a Stirling-engine-driven heat pump system, according to a subcontractor study,[1] may include the following:

Engine efficiency	30%+
Life	15,000–50,000 h
Maintenance	Once per year
Emission	Comparable to conventional gas-fired furnace
Noise and vibration	Low
Cost	$300-$400 per kW (max)

*Research sponsored by the Building Equipment Research Division.

Early Development

Stirling-engine-driven heat pump technology originated in private industry. Two of the projects [the General Electric (GE) Stirling-engine heat pump project, funded by the American Gas Association (AGA), and the Mechanical Technology, Inc. (MTI) project, partially funded by Consolidated Natural Gas Company (CNG)] were later sponsored by government funds. The Stirling-engine concept adopted for these projects was the free-piston engine, which has the advantages of being hermetically sealed and being ideal for residential applications. These two projects focused on laboratory proof-of-concept testing to verify the technical feasibility and meet projected performance targets. The system characteristics and accomplishments are shown in Figs. 1 and 2 and Table 1. After two hardware iterations, the GE project has been terminated; the MTI project is currently in the engineering development phase.

Current Status and Accomplishments

Because of the technology advancement made by these two hardware experimental projects, many technical issues relating to technology development started to appear. Some of the issues are systems related, such as power transmission from a free-piston engine to a vapor compressor, compressor load effects on engine performance, engine-compressor controllability under variable loads, and relative advantages of kinematic and free-piston engines for use as heat pump drives. Others issues relate to supporting research areas: the performances of certain engine-specific heat exchangers are not quantified; the reliability and durability of gas bearings and seals are not really known; many of the transport processes involved are not well understood.

The program has evolved from development of hardware into a set of projects aimed at developing a better technology data base for accelerating hardware development through experimental (hardware) and analytical (software) studies. The ongoing projects to date are listed in Table 2 and include compressor hardware testing, technology assessment, component development, and fundamental research. This set of projects covers only a few of the technical issues identified. Some of the unfunded issues common to Stirling technology could be learned from the DOE Automotive Stirling and GRI programs. Others need to be studied in the future. The technical accomplishments and status of the ongoing projects are shown in Table 3.

Future Options

In addition to the free-piston Stirling/Rankine systems, options for Stirling-engine heat pump technology include kinematic Stirling/Rankine systems, duplex Stirling systems, and regenerative cycle systems, such as the Vuilleumier- and Ericsson-cycle concepts. Space-conditioning projects using these concepts are listed in Table 4.

The free-piston Stirling/Rankine system has several desirable attributes, including the hermetic sealing of the free-piston engine and the good thermal performance of the Rankine vapor-compression cycle at moderate ambient temperature. It is one of the most promising concepts for residential applications. To meet

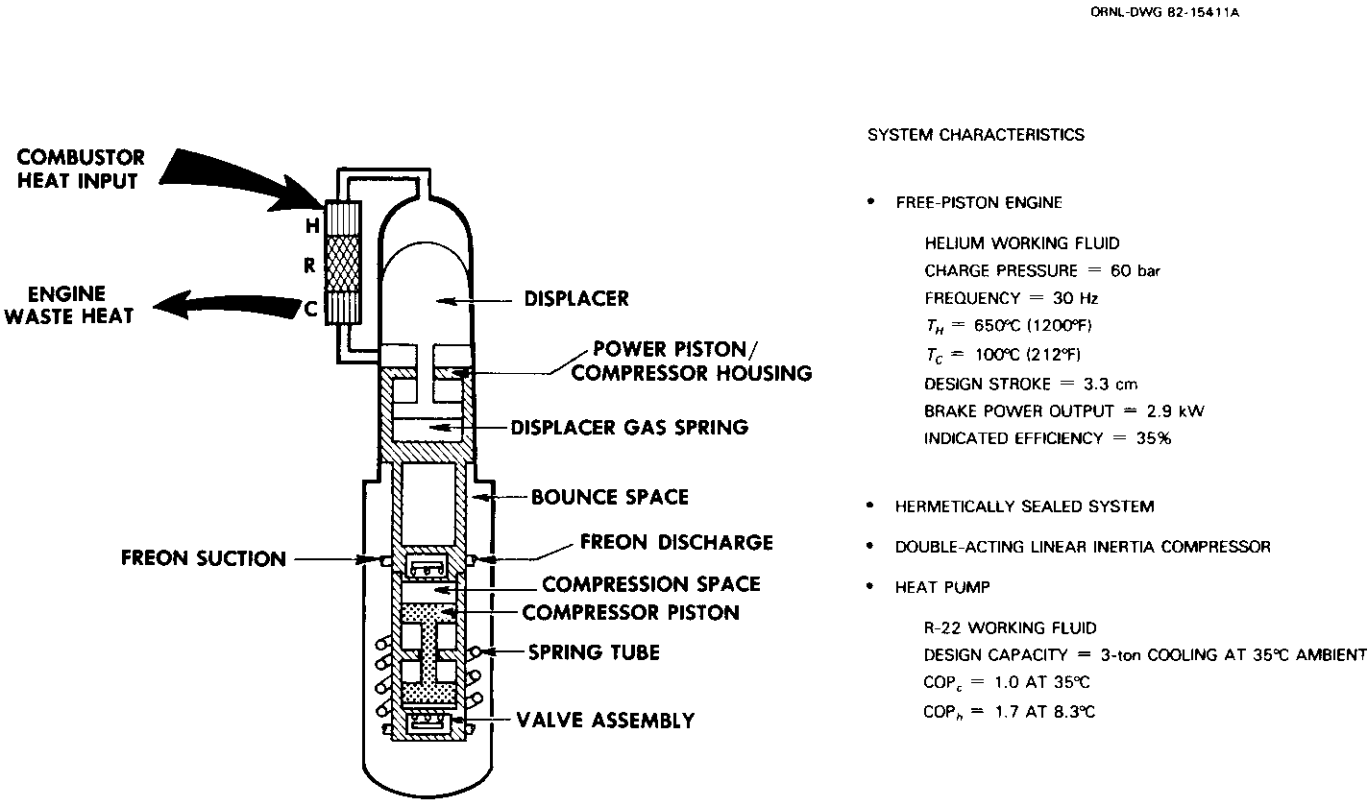


Fig. 1. GE engine/compressor assembly.

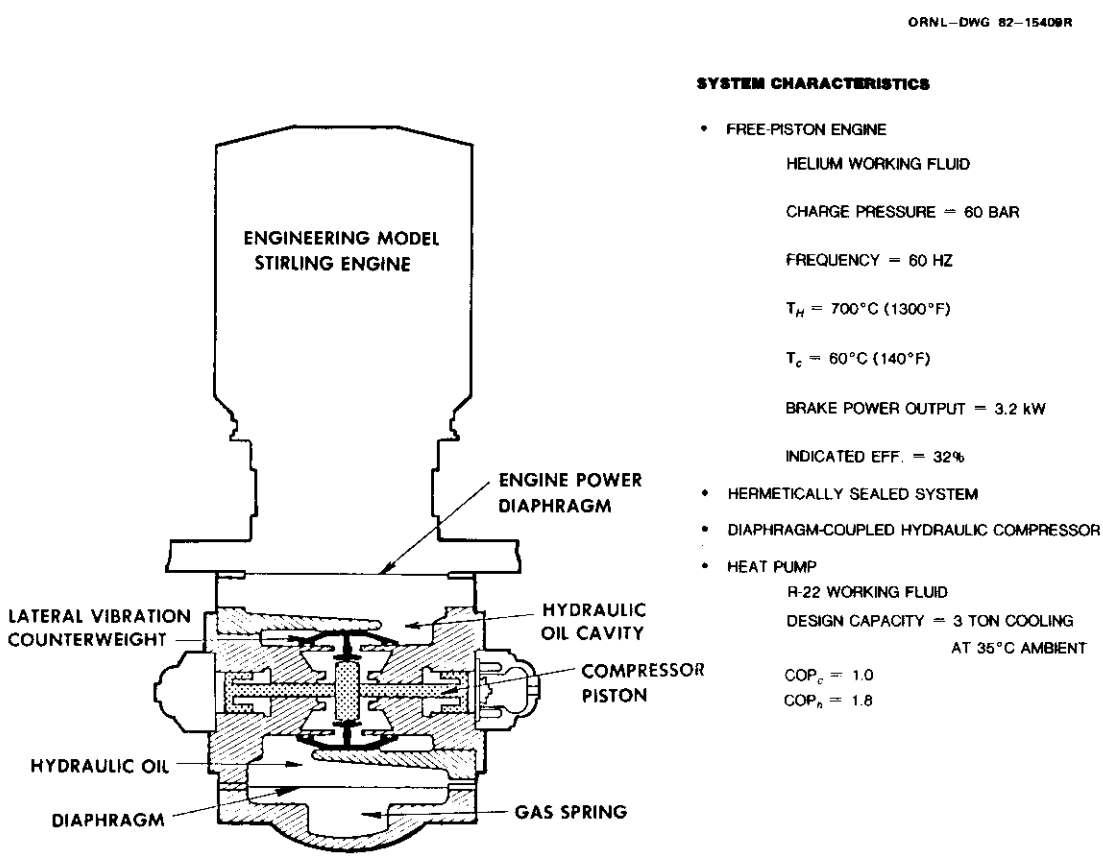


Fig. 2. MTI engine/compressor assembly.

Table 1. Early development effort and accomplishments

- Free-Piston Stirling/Rankine Diaphragm-coupled Compressor Heat Pump—MTI
- Designed, fabricated and tested a breadboard engine, hydraulic transmission, and compressor power module
- Free-Piston Stirling/Rankine Inertia Compressor Heat Pump—GE
- Designed, fabricated and tested two prototype engine-compressor assemblies; demonstrated the principle of operation

Table 2. Stirling engine–driven heat pumps—ongoing projects

- Stirling/Rankine Diaphragm-coupled Compressor Heat Pump—MTI
- RE-1000 Free-Piston Stirling Engine Test—NASA
- FPSE (MTI Engine) Endurance Test—NASA
- FPSE Linear Harmonic Analysis—ORNL
- Design of an Alternate Dynamometer—ORFMA
- Regenerator and Oscillating Flow Tests—ANL
- Second-order Code Modification—UT
- FPSE Technology Assessment—ADL

Table 3. Current status and accomplishments

- MTI—Conducting performance improvement tests and examining alternative concepts
- NASA—RE-1000 engine delivered 1 kW power at 27.4% brake efficiency; engine maximum power output is 1.45 kW.
- NASA—Completed 1000-h endurance test
- ORNL—Completed the mathematical validation of the linear harmonic analysis for Stirling engine
- ORFMA—Alternate dynamometer in progress
- ANL—Completed test plan
- UT—Completed nearly isothermalized code modification; at large heat transfer rate in expansion and compression spaces, the polytropic coefficient of the process approaches one.
- ADL—Technology assessment in progress

Table 4. Major technology options and activities

Options	R&D activity sponsors
Free-Piston Stirling/Rankine	DOE, GRI
Kinematic Stirling/Rankine	GRI, Japan-MITI
Duplex Stirling	Private industry—US, Japan
Regenerative cycle	Denmark, Finland

near-term performance targets, its major components have to achieve efficiency levels. Major components of a Stirling/Rankine system include combustor, engine, transmission, and compressor. The major components targeted and the corresponding state-of-the-art (SOA) efficiencies are listed in Table 4. As shown in Table 5, the SOA free-piston engine is able to achieve relatively good efficiency. Still better engine efficiency is being pursued; however, much work still is required in determining effective means of power transmission from an engine to a vapor compressor and in system integration. Once the performance targets are reached, system reliability and cost-effective alternatives to achieve economic viability should be analyzed to make the technology commercially attractive to private industry.

Table 5. Component efficiencies of a Free-Piston Stirling/Rankine heat pump system

Component	Efficiency (%)		Project
	Targeted	SOA[a]	
Combustor	80	80	MTI EM-1 engine
Engine[b]	40	35	MTI EM-1 engine
		32	RE-1000 engine
Transmission	80	60	GE inertial configuration
		55	MTI hydraulic configuration
Compressor[c]	70	70	GE inertial configuration

[a]State of the art.
[b]Indicated efficiency.
[c]Compression efficiency.

The kinematic Stirling/Rankine system is being pursued vigorously by Japanese government-sponsored research for both residential and small commercial applications.[2] In the past, kinematic Stirling engines have been known to have seal problems. The difficulty in providing a hermetically sealed kinematic engine has contributed impetus to the development of free-piston engines. Recent advancement in kinematic engine seal technology has alleviated the problems with reliability, durability, and maintainability of the kinematic engine, making it more suitable for air-conditioning applications. Because the Stirling/Rankine technology is relatively mature and near commercialization, government involvement may be limited to funding supporting research and technology.

The duplex Stirling system, which has the advantages of being hermetically sealed and mechanically simple, is another promising option. The thermal performance of a Stirling-cycle heat pump subsystem may not be as good at moderate ambient temperature as that of a Rankine vapor-compression system, but it is superior at low ambient temperature. A duplex Stirling system could have a place in the technology spectrum for space conditioning in cold climates and for heat-activated refrigeration applications. Although the duplex Stirling concept shares many technological aspects with the free-piston engine, its uniqueness has not been adequately analyzed. Additional government-supported R&D should serve to augment private industry's effort[3] in demonstrating the viability and potential of this option.

A regenerative cycle system may be considered to be a generalization of a Stirling-engine-driven heat pump system that includes Vuilleumier- and Ericsson-cycle concepts. These concepts also share many of the Stirling engine features, including external combustion and the use of regenerative heat exchangers. The potential of the Vuilleumier (VM) cycle has been analyzed by Nykyri et al.[4] for space heating and Anderson[5] for heat-activated refrigeration. Benson[6] has shown the possibility of a heat-activated Ericsson-cycle heat pump design. The VM-cycle concept is known to be used in cryogenic cooling. The potential and feasibility of these concepts for building space conditioning have yet to be investigated.

References

1. *Topical Report—Status of Free Piston Stirling Engine-Driven Heat Pumps/Identification of Development Issues*, Arthur D. Little, Inc., Cambridge, Mass., September 1984.
2. *Stirling Engine R&D Project in Japan—1983 Progress Report*, New Energy Development Organization, Tokyo, Japan, April 1984.
3. Gedeon, D., et. al., *Duplex Stirling Heating-Only Gas-Fired Heat Pump, Phase II. Breadboard Demonstration*, GRI Contract No. 5081-341-0464, Gas Research Institute, Chicago, March 1983.
4. Nykyri, M., et al., *Evaluation of the Vuilleumier Heat Pump for Heating Applications*, Technical Research Centre of Finland, NTIS, PB82-205212, August 1981.
5. Qvale, B., Director, Laboratory for Energetics, The Technical University of Denmark, DK2800 Lyngby, Denmark, personal communication with F. C. Chen, Oak Ridge National Laboratory, Oct. 30, 1984.
6. Benson, G. M., "Thermal Oscillators," pp. 1478–87 in *Proceedings of the 12th IECE Conference, Washington, D.C.*, vol. 2, Aug. 30, 1977.

Breadboard Development of a Hydraulically Coupled Free-Piston Stirling Engine Heat Pump Compressor

Thomas J. Marusak

Mechanical Technology, Inc.

Latham, New York 12110

Thomas J. Marusak is program manager for the Residential Free-Piston Stirling Engine Programs at MTI. Since joining MTI in 1978, he has held responsible positions in program management, business development, marketing management, and systems engineering. Prior to joining MTI, he worked for General Electric's Space Division where he served as task leader for the U.S. Department of Energy-sponsored Free-Piston Stirling Engine Heat Pump Program. The author holds the M.S. and B.S. degrees in mechanical engineering from Stanford University and Pennsylvania State University, respectively, and has completed advanced courses toward the M.B.A. degree at Rensselaer Polytechnic Institute.

BREADBOARD DEVELOPMENT OF A HYDRAULICALLY COUPLED FREE-PISTON STIRLING ENGINE HEAT PUMP COMPRESSOR

Thomas J. Marusak
Mechanical Technology Incorporated
968 Albany-Shaker Road
Latham, New York 12110

Abstract

For more than a decade the Free Piston Stirling Engine (FPSE) has been considered as an attractive candidate for a thermally activated heat pump because of its potential for high efficiency coupled with long life and high reliability. The distinguishing features of the FPSE, one moving part and hermetic separation of the power cycle and refrigeration cycle working fluids, makes it ideally suited for a heat pump application. However, two major design challenges have kept the FPSE in the realm of laboratory rather than product development. Specifically, these challenges involve; effective control of a tuned resonant system over a wide range of loads and hermetic coupling of the driver and driven mechanical members.

Program Background

In 1980, Mechanical Technology Incorporated (MTI) proposed a development program to DOE for the development of an FPSE heat pump concept which would uniquely address the above mentioned design problems. The concept consists of a power conversion system which is tuned by a linear motor displacer control and hermetically linked to a freon compressor via a diaphragm and hydraulic transmission coupling. The design is shown, schematically, in Figure 1. The major components consist of a natural gas combustor, a thermal amplifier type FPSE, a hydraulic transmission with counterweight dynamic balancing, and a double acting freon compressor. From 1981 through 1983, the Department of Energy (DOE) embarked on a short term development program with MTI to demonstrate technical feasibility of the concept. Although not all of the technical performance goals were achieved, the program did accomplish sufficient progress to warrant further development.

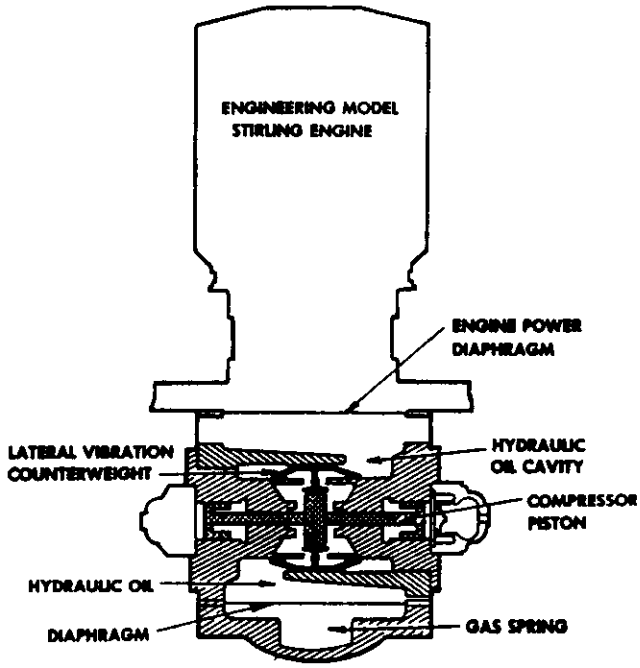


Fig. 1 Engineering Model Stirling Engine and Combustor

In December, 1982, the Gas Research Institute (GRI) initiated a second FPSE heat pump development program with MTI, concentrating on the advancement of engine and combustor efficiency and life rather than on system development. Furthermore, until recently, the scope of the GRI work did not involve any compressor development, leaving that portion of the effort to DOE. The specific objective of the GRI program is to establish a laboratory evaluation of the FPSE heat pump power conversion unit at Lennox Industries Research and Development Laboratory. Lennox is the participating commercialization partner on the development program.

During 1984, the budgets and task schedules of the GRI and DOE programs were coordinated to complement each other. Although significant progress has been achieved on the combustor and engine components, a major development effort continues on the hydraulic transmission. Successful development of the transmission will lead to the fabrication of several laboratory development units that will directly preceed full heat pump system development. The project continues to be viewed as a long term, high-risk - high payback opportunity; with possible product introduction in the early 1990's.

Technical Status

Component development has centered on three items: the natural gas combustor, the engine/displacer control, and the transmission/compressor section. Only a superficial effort has been given to auxiliaries and system controls. The technical objective has been to establish the minimum level of efficiency and reliability suitable for a laboratory prototype.

Combustor

The technical goals for the combustor are to achieve at least 75 percent efficiency at 12 kW firing rate (based on natural gas Higher Heating Value (HHV)) with a gas supply pressure of 3-inches H_2O. Furthermore, an endurance goal of 1000 hours was established for the ceramic liner of the combustion zone. Three generations of combustors have been designed and fabricated. The latest model, termed G-7, has achieved all of the above performance goals. A schematic of the G-7 combustor is shown in Figure 2. Its operation is based on sub-atmospheric combustion of preheated air with natural gas; producing a turbulent diffusion flame. The proper airflow is accomplished via a suction gas blower and air bypass valve. The exhaust blower inlet gas temperature is kept below 190°F to comply with manufacturer's requirements for long blower life. This is accomplished by both waste heat recovery and bypass air exhaust dilution. The measured performance of the G-7 combustor is displayed in Figure 3. At the 12 kW rating point, 82% efficiency was measured; far surpassing the 75% performance goal. The data shown indicates that the required fuel pressure was slightly above the target of 3-inch H_2O. Subsequent testing verified that the fuel nozzle jets were restricting the fuel flow for the reported data. The nozzle has since been modified to achieve the same operating performance at pressures below the 3-inch target.

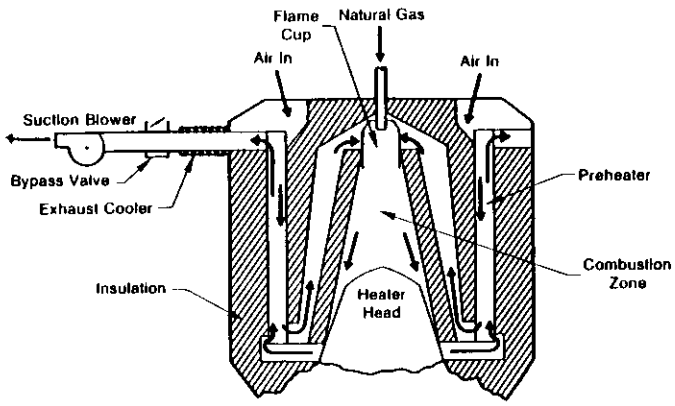


Fig. 2 G7 Combustor Schematic

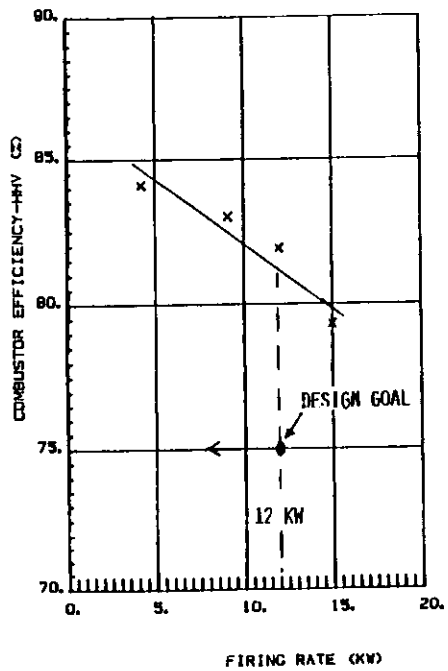


Fig. 3a

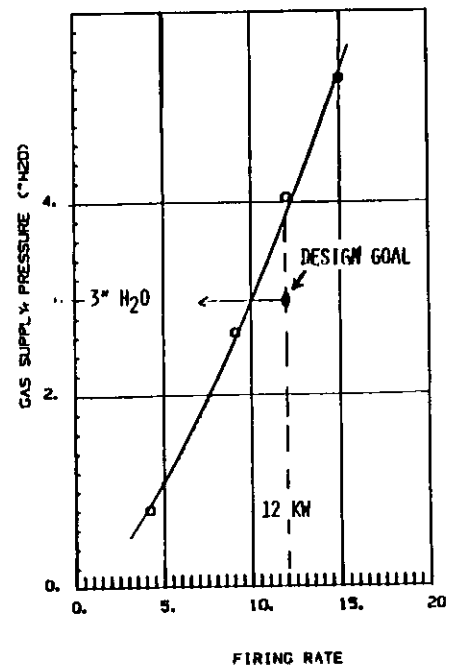


Fig. 3b.

Fig. 3a-b G7 Combustor Performance

Combustor Liner - The combustor liner consists of a Kaowool ceramic material that had suffered frequent mechanical failures earlier in the development effort. During 1983, the liner was redesigned to provide at least 500 hours of trouble free operation. Since that time several liners have been operated for extended periods. The latest version has completed over 1600 hours of trouble free operation and continues to be tested. Figure 4 shows that liner after the first 1000 hours of testing and compares it to an identical liner that has not yet been used. The clean, unused appearance of the 1000 hour liner is visual evidence of the good integrity of this design.

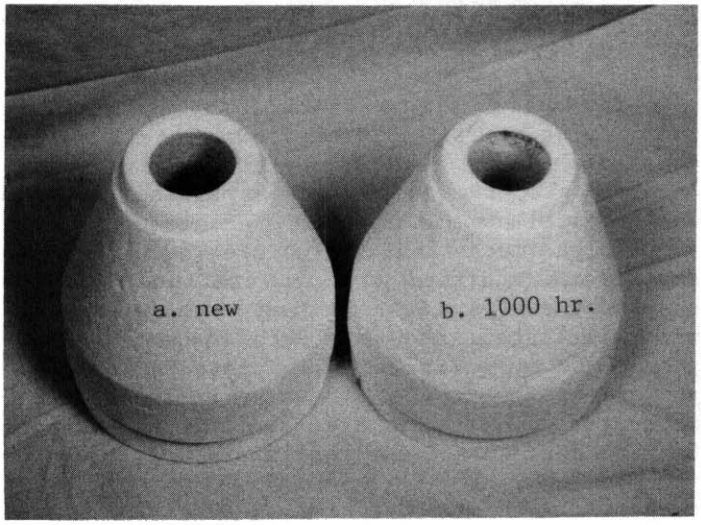


Fig. 4 Combustor Liner

Engine

Early in the GRI sponsored program, one of the major technical objectives was to increase the overall efficiency and reliability of the Stirling engine through development of the displacer drive unit and engine thermodynamic components. The approach used was to analyze power module design parameters through computer modelling in order to identify areas for achieving increased conversion efficiency and power output. Based on the analysis, component modifications were made to the Enginneering Model (EM) Performance Engine Hardware, and tests were conducted to measure performance improvements. The effort concentrated on two major areas: displacer drive development and engine thermodynamic development.

Displacer Drive - One of the unique features of the MTI free piston engines is that they use linear motors to control the motion of the displacer and consequently the engine power output. In a gas fueled heat pump operation, this motor is seen as an electrical parasitic which would detract from the overall attractiveness of the heat pump system in direct proportion to the amount of electricity consumed. To minimize this parasitic, the motor stator section was redesigned to strengthen the magnetic flux pattern at long strokes. The modified motor resulted in significantly lower power input requirements as compared to the original displacer drive for similar operating conditions. Figure 5 illustrates the improvement achieved. Note; significant amounts of electrical power are consumed only at the ambient temperature extremes (i.e., as the 95°F and 17°F day outdoor conditions are approached). At the more moderate outdoor conditions, which account for the bulk of the space conditioning requirements, relatively small amounts of electrical power are required.

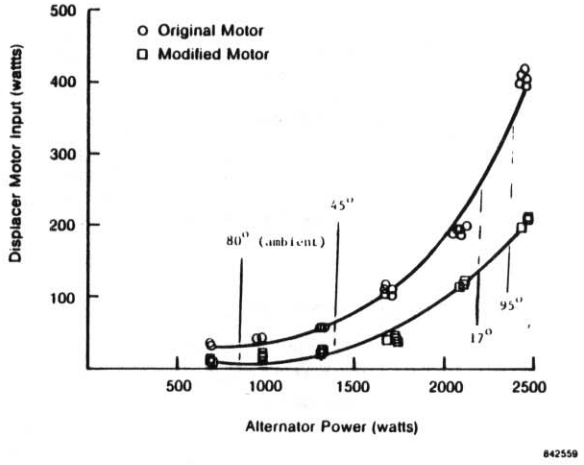


Fig. 5 Displacer Drive Improvements

Thermodynamic Development - A thermodynamic analysis showed that the engine's level of efficiency could be improved significantly through thermodynamic component development. Hardware modifications were made, and tests were conducted involving continuous correlation between test and analysis. Based on a computer analysis, several engine modifications were made to effect improved performance. Specifically, two major modifications were performed:

- all critical seals were machined to the original design requirements; and,
- the regenerator matrixes were optimized with the regard to porosity and wire geometry.

Figure 6 illustrates the effect of decreasing the leakage losses through reduction of the seal clearances to the original engine design specifications (it should be noted that the original engine builds included hardware that was not machined to the specified tolerances).

A second area predicted to provide a major performance improvement was the regenerator configuration. The original EM configuration incorporated woven-wire regenerators. Analyses predicted that the significant gains in efficiency could be realized by changing to screen regen-

erators. A change of material was made, and test data of the EM with screen regenerators was taken over a range of power conditions. As shown in Figure 6, the improvement in efficiency due to screens accounts for up to four percentage points. Further analysis reveals that with additional improvements in the next generator engine, a target efficiency level of over 40% should be attainable.

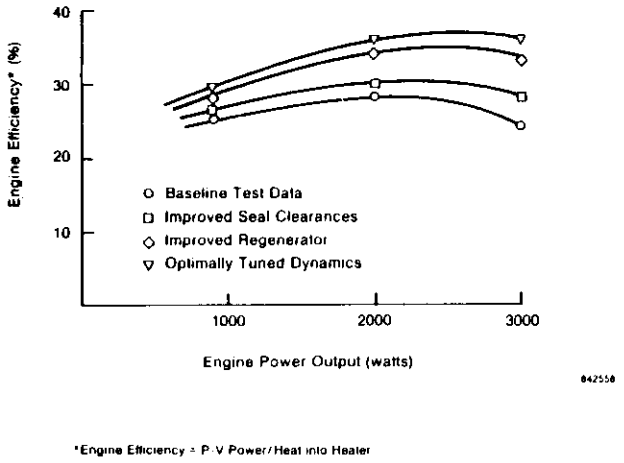


Fig. 6 Regenerator Efficiency Improvements

The combined combustor and engine efficiency, relative to alternative internal combustion engine in the FPSE power class (4 hp), is shown in Figure 7. As shown, the present level of FPSE efficiency is competitive with the other engines to which it is compared. Also indicated in Figure 7 is the development potential for the engine which exceeds the current performance by nearly 25%.

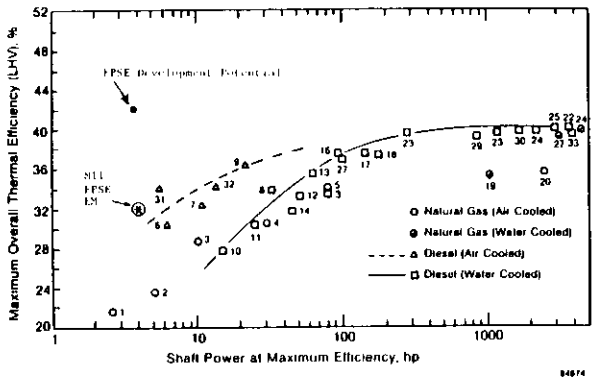


Fig. 7 FPSE efficiency relative to diesel and gas engine overall thermal efficiency

It should be pointed out that the bulk of the operating experience achieved with MTI's FPSE involves linear alternator loads rather than freon compressors. With the linear alternator load, the helium working fluid and the gas bearings operate both the displacer and load power piston. In the heat pump configuration only the displacer operates in a helium environment with gas bearings, while the power piston operates in an oil and freon environment. This results in some simplifications, as well as some additional complexities to the system. In any case, a new internally generated gas bearing system had to be developed to power the displacer bearing for the heat pump application. This was accomplished by pumping helium working fluid from the displacer gas spring with a check valve arrangement that is designed to "clip" the pressure wave generated in the spring cavity; thereby supplying the bearings with the high pressure helium.

The design effort was also directed toward simplifying the displacer mechanical arrangements to enhance reliability and ease assembly. As a result, the number of gas spring chambers was reduced from three to two, leading to a simplified mechanical structure. An existing engine was modified to simulate the heat pump displacer design. Self pumping bearing operation was established, and test data showed that very good bearing efficiency was achieved. Table 1 summarizes the results of these tests and indicates that based on these results, the expected bearing performance for the laboratory prototype (termed Mark 1) will be better than the design target.

Table 1
Gas Bearing Design Validation Test Results

Parameter	Prediction	Measurement	Design Target
Bearing Flow (lb/hr)	3.7	5.4	11.1
Power Loss (watts)	36	53	150
Offset (in.)	0.015	0.005	0.015

To evaluate the reliability potential of the MTI FPSE, a 1000 hour endurance test was performed with a linear alternator load. Only two major unplanned shutdowns occurred during the course of this program. The first was caused by heater head instrumentation problems, and the other from a magnet segment which broke loose due to improper assembly.

Throughout the task effort, the ratio of the actual test hours accumulated relative to the hours available increased dramatically. The last 400 hours of duty cycle testing were accomplished within 432 available hours. Currently an additional 9000 hours are being attempted on the same

engine. To date, 1600 hours have been achieved without major incident.

Compressor Development

The development of the diaphragm coupled compressor is the focal point of the DOE sponsored program. Earlier attempts to couple a FPSE with a freon compressor involved a spring tube coupling to either resonant or inertia refrigerant compressors. These approaches were independently pursued by both MTI (privately funded) and General Electric (DOE and GRI funded). Because of slow progress in solving the cost, performance, reliability, and control problems associated with the spring tube approach, both programs came to unsuccessful finishes. The diaphragm coupled compressor concept was initiated in an effort to overcome much of the difficulties that plaqued the earlier projects.

By the end of 1983, nearly all of the major technical goals of the DOE compressor development program were achieved, i.e.:

- stable and controlled operation was achieved at the four ARI rating points;
- modulation capability was demonstrated; and,
- good breadboard level reliability was established.

The one remaining unaccomplished goal was to achieve the full design capacity of three tons at the 95°F day conditions. Table 2 lists the performance that was accomplished earlier in the program. 1984 was devoted to identifying the cause of the capacity short-fall, and to improving the machine's mechanical capabilities.

Table 2
FPSE Heat Pump Compressor Development Progress

Parameter	1/84*	6/84*	11/84*	Goal
Capacity @ 95°F (BtuH)	16,200	28,200	18,240	36,000
Transmission Loss (watts)	1,635	736	1,088	406
Lower End Efficiency	45%	63%	47%	69%

*Tested without counterweight balance

An analyses was performed which showed that the capacity problem was the result of excessive transmission losses and inadequate power matching between the engine and load characteristics. To facilitate the development of the transmission, the counterweight balancer was removed and the direct drive compressor (schematically illustrated in Figure 8) was utilized to develop the transmission. Two modifications to the compressor resulted in sizeable improvements to its performance. The first was the substitution of a low viscosity (10 centistrokes) transmission fluid for a medium viscosity one. This resulted in a hydraulic loss reduction of as much as 50%.

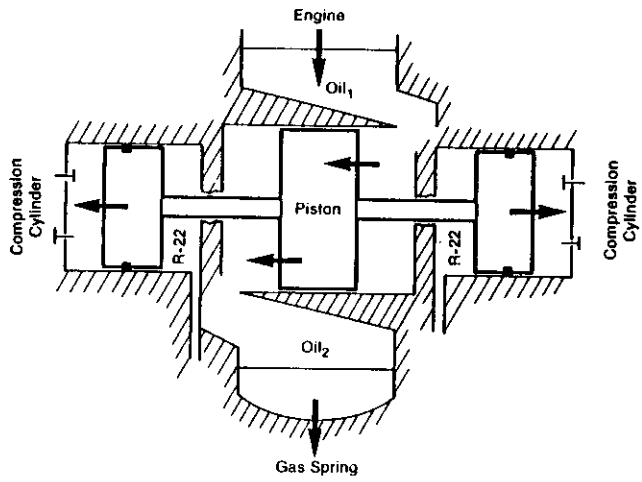


Fig. 8 Direct-Drive Compressor Schematic

The second item resulting in major improvement was the modification of the compressor ring seal to minimize freon leakage during compression. Together with the gains achieved by lowering the fluid viscosity, the overall compressor efficiency was improved by 40%.

At that point, it was decided to reinstitute the counterweight balancer into the hardware in order to reduce engine vibration and begin development of the full transmission and compressor system. Unfortunately, the introduction of the balancer introduced additional large transmission losses that further reduced the system's ability to achieve its capacity goals. Extensive testing of the counterweight provided evidence that the counterweight outer diameter (O.D.) clearance seals were exceedingly inefficient as a result of hydraulic fluid blowby. Subsequently, the seal clearances were reduced to improve the blowby loss mechanism. The reduced clearance resulted in a binding of the counterweight clearance seals against the cylinder inner diameter (I.D.). To alleviate this problem, a compliant seal was designed and fabricated. The seal was tested for both sealing effectiveness and frictional characteristics prior to operation in the breadboard system. Both parameters were measured to be acceptable. Subsequent testing of the counterweight and compliant seal in the breadboard unit revealed essentially no change in the overall performance.

At this point, several other possibilities are being investigated for achieving improved performance. Namely, flow straighteners and lower viscosity fluids will be utilized to further reduce hydraulic flow losses. Due to the complex geometry of the compressor housing, it is

difficult to perform a meaningful fluid dynamics analysis to predict the effect of flow losses. It is not known at this time whether a distinguishable improvement can be achieved in this fashion.

Another modification will be to identify and improve any freon shuttling losses that may result in the aft piston chambers of the double acting compressor. Finally, the counterweight area will be increased so as to improve the swept volume ratio between the engine and compressor in order to overcome any residual compressor losses in excess of the original development target.

Conclusion

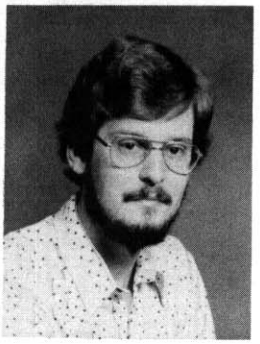
As with most new concepts, the diaphragm coupled compressor has unique problems of its own to be solved. Substantial progress has been made, but there are still problems that remain to be solved. At this point it is not clear whether the diaphragm/hydraulic coupling can be economically engineered to achieve all of its design goals. Nor is it clear that the concept cannot be made to overcome its current difficulity - low transmission efficiency and corresponding low capacity. Another six months of development are required to fully address this issue.

Initial Results of Sensitivity Tests Performed on the RE-1000 Free-Piston Stirling Engine

Jeffrey G. Schreiber

NASA Lewis Research Center

Cleveland, Ohio 44135



Jeffrey G. Schreiber received B.S. and M.S. degrees in mechanical engineering from Case Western Reserve University (1977) and the University of Toledo (1982), respectively. He has done considerable work with the free-piston Stirling engines and is currently involved with RE-1000 test cell activities.

Initial Results of Sensitivity Tests Performed on the RE-1000 Free-Piston Stirling Engine

Jeff G. Schreiber

NASA Lewis Research Center
Cleveland, Ohio 44135

SUMMARY

A 1 kW (1.33hp) single cylinder free-piston Stirling engine has been tested in the test facilities at the Lewis laboratory. Tests have been performed over the past several years on an engine designed to investigate the dynamics of a free-piston Stirling engine for the purpose of computer code validation. A description of the engine and its instrumentation is given in a prior NASA report TM-82999. Some initial test results are given in NASA Report TM-83407. Both reports are by J. G. Schreiber.

Tests to investigate the sensitivity of the engine performance to variations in working space pressure, heater and cooler temperatures, regenerator porosity, power piston mass and displacer dynamics have been initiated at Lewis. Maps of engine performance have been recorded with the use of an 81.2% porosity regenerator; both a high efficiency displacer and a high power displacer were tested; efficiencies up to 33% were recorded and power output of approximately 1500 watts was recorded.

This report presents preliminary results from the Lewis sensitivity tests being performed on the RE-1000 free-piston Stirling engine. Descriptions of future tests are also given.

INTRODUCTION

A 1kW (1.33 hp), single-cylinder, free-piston Stirling engine has been under test at the Lewis Research Center for several years. The RE-1000 test program currently being conducted was designed to give detailed information about the operational performance of a free-piston Stirling engine. Future tests will include the evaluation of alternative load devices on the engine.

This work is funded under a joint cooperative interagency agreement, number DE-A105-820R21005, between the National Aeronautics and Space Administration Lewis Research Center (NASA LeRC) and the Department of Energy, Oak Ridge National Laboratory (DOE/ORNL).

The effort over the past year included tests to determine the size of the power piston centering port best suited for the engine, performance tests with the high porosity (81.2%) regenerator, and detailed steady state flow tests of the heat exchangers.

This report gives a description of the sensitivity tests and covers the initial results of the sensitivity tests being conducted on the RE-1000. It also gives a description of future tests planned.

ENGINE DESCRIPTION

The test cell for the RE-1000 engine is shown in Figure 1. A cutaway of the engine is shown in Figure 2 with key parts labeled. The RE-1000 was built at Sunpower Inc., Athens, Ohio. It was built for the Lewis Research Center and was intended to be a rugged test bed for gathering data to be

used in the validation of free-piston Stirling engine computer simulations. Many instrumentation penetrations were incorporated in the engine to aid in the data gathering process.

The RE-1000 was designed with a displacer sprung to ground, an annular regenerator and cooler, and an electric resistance heater head. The load is provided by a dashpot built inside of the pressure vessel. The power piston is shown in Figure 3 and the displacers and displacer rods are shown in Figures 4 and 5.

The engine uses hardened coatings for wear resistance on sliding surfaces. Chromium oxide is used on the displacer rod and the power piston cylinder. The bore of the displacer is hardened stainless steel. The power piston is coated with a wear resistant flourocarbon coating.

Some of the steady state engine parameters measured include the mean pressure of both the working space and the bounce space, and many gas, metal and coolant temperatures. Cooler heat rejection is measured by monitoring the coolant flow rate and using a pair of thermocouples wired in a differential temperature configuration for best accuracy. A more complete description of the engine and the data system can be found in References 1 and 2.

SENSITIVITY TEST MATRIX

In order to investigate the sensitivity of the free-piston Stirling engine to variations in design parameters, a sensitivity test plan was devised. Thetests defined will generate performance maps of the engine for many different configurations. Two different displacers will be utilized. The two displacers are described in Reference 1. One of the displacer is built to produce relatively high engine efficiency while the other displacer is built to produce relatively high power output.

Two different regenerators will be used in the sensitivity tests. Both of the regenerators to be used are formed from similar knitted material as described in Reference 1. The high power regenerator weighs 109.0 grams (0.24-lb) and has a porosity of 81.2%. This regenerator was intended for use with the high power displacer. This displacer and regenerator shall be referred to as displacer number two and regenerator number two respectively.

The high efficiency regenerator weighs 139.0 grams (0.31-lb) and therefore has a porosity of 76.0% This regenerator was intended for use with the high efficiency displacer. This regenerator and displacer shall be referred to as regenerator number one and displacer number one respectively.

Tests will also be performed to characterize the engine's reponse to variations in the power piston mass. The large diameter section of the power piston shown in Figure 3 is made of steel to obtain the proper power piston mass as per the original design. The steel mass will be replaced by an aluminum mass of the same dimensions to reduce the overall power piston mass. This lower mass power piston will permit the heat exchangers and displacer to be excited at a higher resonant or driving frequency.

The order in which the tests on the RE-1000 are scheduled to be run is as follows:

Displ.1 Regen.2 (Orig.power piston)
Displ.2 Regen.2 (Orig.power piston)
Displ.1 Regen.1 (Orig.power piston)
Displ.2 Regen.1 (Orig.power piston)
Displ.1 Regen.1 (Light power piston)

After the completion of these tests, the engine will be operated on a different working fluid if time permits. A matrix showing the configurations to be tested is given in Figure 6.

For each of the engine configurations a map of engine performance will be obtained. Engine heater temperature will be varied over four different levels. Engine coolant temperature will be varied over three different levels. Engine operating pressure will be varied over three different levels. This would produce a matrix of 36 combinations of temperature and pressure for each engine configuration. However, not all of the combinations will be investigated. Figure 7 shows the areas of the matrix that will be investigated.

At each combination of settings to be investigate the engine performance will be recorded as a function of power piston stroke. The stroke will be varied in increments of 0.20cm (0.079 in) starting with 1.80cm (0.709in) and increasing to some upper limit. The upper limit is somewhat dependent on the pressure level and temperatures of the heat exchangers. In some cases the engine has been operated at strokes as high as 3.60 cm (1.417 in).

TEST PROGRAM STATUS

The first two configurations listed have been tested, that is to say that the engine tests with regenerator number two have been completed. Figures 8 and 9 show the brake power and brake efficiency for the

combination of displacer number one and regenerator number two. Figures number 10 and 11 show the brake power and brake efficiency for the combination of displacer number two and regenerator number two.

Table 1 gives six of the data points recorded during the sensitivity tests. Readings 600, 601 and 602 were taken with the high power regenerator (#2) and the high efficiency displacer (#1). All of the parameters used in the data have previously been defined in Reference 1.

For the combination of the high power regenerator (#2) and the high power displacer (#2), data reading 941, 943 and 945 are given. The displacer rod used with the high power displacer has a port drilled in to allow the measurement of the pressure variations in the displacer gas spring. The gas spring dynamic pressure measurement is labed PDYNDB. This measurement cannot be made with the high efficiency displacer since the diameter of the displacer rod is too small to permit both the centering port passages and an adequate size gas spring pressure tap. The amplitude of the PDYNDB signal is listed in Table 2 for the three readings 941, 943 and 945. Also listed in Table 2 are measurements related to the working space pressure amplitude and to the center position of the power piston and the displacer. Neither the power piston nor the displacer oscillate symmetrically about the centering ports. Due to nonlinearities in the operation of the engine and fabrication tolerences the piston stroke and the displacer stroke center positions will vary from data point to data point. A positive value for the centering position indicates the center of the stroke has moved toward the expansion space or farther into the cylinder. A negative value for the centering position indicates the center of the stroke has moved away from the expansion space or farther out of the cylinder. The working space pressure amplitude is measured in the compression space and is labeled PDYNC.

The farthest the piston and displacer can physically move towards the expansion space or into the cylinder is called their in-limits. The farthest the piston and displacer can physically move away from the expansion space is called their out-limits. Table 3 gives the largest physical stroke for the piston and the two displacers. It also gives the distance of travel from each component's in-limit to the centering port position. It is important to know the true stroke center position during tests if the data is to be used for computer code validation since the volume distribution in the engine is a function of the center position and the dynamics are influenced by the volume distribution.

Figure 12 shows the steady state pressure drop through the heat exchangers as a function of mass flow rate. The tests were performed with nitrogen. The temperature of the nitrogen was approximately 0°C (32°F) during the tests. The cooler and regenerator were tested with the nitrogen inlet pressure at 1.35 MPa (200 psi). The mass flow rates at which the steady state tests were performed yield Reynolds numbers similar to those occurring in the heat exchangers during actual engine operation.

FUTURE TESTS

Work will continue to complete the sensitivity tests as described. It is anticiplated that all of the data recorded in these sensitivity tests will be publisheo in detail. The total number of data points to be recorded in the sensitivity tests are anticipated to be approximately 900.

At the completion of the sensitivity tests the engine will be rebuilt with a hydraulic output load device. The hydraulic output system is being designed and fabricated to be a general purpose test bed load. Several load para-meters, such as the spring component of the load and the dynamic balance of the load, will be adjustable. The load device has been designed and is being fabricated by Foster-Miller Inc. of Waltham, Massachusetts. Sunpower Inc. of Athens, Ohio is a subcontractor in this effort.

CONCLUDING REMARKS

The RE-1000 has been a rugged and reliable test bed for general free-piston Stirling engine tests. With the engine built in a high power output configuration, the engine has produced 1500 watts. Efficiencies of 33% had been measured prior to the initiation of the sensitivity tests while the engine was in a high efficiency configuration. It is anticipated that even higher efficiencies are possible. The efficiency is defined as:

$$\text{EFFICIENCY} = \frac{\text{Brake Power Output}}{\text{Heat Input To The Head}}$$

The engine has yet to be tested with both the high efficiency displacer and the high efficiency regenerator as part of the sensitivity tests.

The maps that are to be generated by the sensitivity tests should provide a substantial data base to aid in the validation of computer codes. The matrix over which the engine will be tested will

provide information regarding the sensitivity of free-piston Stirling engines to variations in heater temperature, cooler temperature, regenerator porosity, mean operating pressure, displacer dynamics and power piston mass.

Appendix

This report was first published as part of the proceedings for the 19th International Energy Conversion Engineering Conference (IECEC) held in San Francisco, California in August of 1984. Since that time a great amount of work has been done in the RE-1000 test program. This appendix has been added to the report for this printing to provide an update on the activities and accomplishments of the RE-1000 test program.

The update will be divided into three sections. The first section deals with the sensitivity tests as described in the main body of the report. The second section discusses the testing and results of a resonant balance system designed and built by Sunpower of Athens, Ohio, for the RE-1000. The third section tells of progress made on the hydraulic output device being built for the RE-1000 by Foster-Miller Inc., of Waltham, Mass.

The sensitivity tests have almost been completed. Only a few test points need to be run to complete the test matrix described in the report. All of the test conditions with the standard power piston mass have been completed. This includes tests run with Argon and Nitrogen as the working fluid. Since the reason for performing these tests was to generate data for use in the validation of computer codes, the data taken with Nitrogen and Argon should be beneficial in developing good simulations of the flow loss mechanism and heat transfer characteristics in Stirling engines. The data includes 625 data points taken with Helium as the working fluid, 64 data points taken with Nitrogen as the working fluid and 12 data points taken with Argon as the working fluid. Tests with the reduced power piston mass have almost been completed. When the mass was reduced, the operating frequency of the engine went from the 30hz level to approximately 37.5hz. When the tests with the reduced power piston mass have been completed a total of nearly 780 data points will have been accumulated in the sensitivity test program. The data will be presented at the 1985 IECEC Conference and published in detail soon afterward.

A resonant balance system was tested on the RE-1000. The balance system consisted of a mass located above the engine housing and attached to the engine housing by a spring system. The springs and the mass were sized such that the resonant frequency of the spring-mass balance system was 30hz. This frequency was intended to match the operating frequency of the engine. As the operating frequency of the engine approached the natural frequency of the balance system, the balance mass would become excited. The forces caused by the motion induced in the balance mass would counteract the vibration forces generated in the engine to yeld substantially reduced vibration levels of the engine housing. At design conditions of 2.5cm power piston stroke and 30 hz operating frequency, the balancing system reduced the engine vibrations level by a factor of thirty. The results of these tests will be published in the final report from the SP-1000 effort by Sunpower in 1985.

The hydraulic output system has been fabricated and is ready to be installed in the RE-1000 test facility. The hydraulic output system is intended to be a rugged, modular, research device to use in the investigation of interaction between free-piston Stirling engines and the load being applied to the engine. The device uses annular diaphrams to seal the gas and hydraulic chambers. The hydraulic power output is provided by a double acting pump which can be rebuilt into different configurations - a pair of back to back single acting pumps for example.

To date the RE-1000 has accumulated over 265 hours of operation. No engine failures have occurred to date. The engine will be rebuilt soon in the form of a hydraulic output engine at which time testing will continue. The data taken with the hydraulic load will be used to validate computer simulations with emphasis on engine-load interactions.

REFERENCES

(1) Schreiber, Jeff: Testing and Performance Characteristics of a 1kW Free-Piston Stirling Engine. NASA TM-82999, 1983.

(2) Schreiber, Jeff: Test Results and Description of a 1kW Free-Piston Stirling Engine with a Dashpot Load. NASA TM-83407, 1983.

TABLE I. - RE-1000 DATA

NASA LEWIS PRELIMINARY DATA 01/03/84 RE 1000 FREE PISTON STIRLING ENGINE TEST D001, REC 11/17/83 10:12:06.78 RDG 601

FLUID HELIUM BAROM 14.268 PSI

HEAT TO DASHPOT COOLING
FLODP 4.12 L/MIN
TWINDP 17.1 DEG.C
TDLDP 2.55 DEG.C
TWODPR 20.2 DEG.C

POWER IN
AMPS1 742. AMPS
AMPS2 627. AMPS
VOLTG 2.98 VOLTS

ENGINE CHARGE PRESSURE
PRESUP 0. KPA
MEANBP 0. KPA
MEANCP 0. KPA

GAS TEMPERATURES
TGEXP 557.6 DEG.C
TGREGH 543.3 DEG.C
TGREGC 101.9 DEG.C
TGCOMP 57.1 DEG.C
TGBOUN 34.1 DEG.C

SURFACE TEMPERATURES
T01HTR 598.3 DEG.C
T02HTR 615.7 DEG.C
T03HTR 616.6 DEG.C
T04HTR 599.0 DEG.C
T05HTR 609.3 DEG.C
T06HTR 590.2 DEG.C
T07HTR 580.9 DEG.C
T08HTR 603.6 DEG.C
T09HTR 622.6 DEG.C
T10HTR 545.0 DEG.C
T11HTR 601.4 DEG.C
T12HTR 607.7 DEG.C

HEAT TO COOLER
FLOCLR 4.13 L/MIN
TWINCL 25.1 DEG.C
TDLCL 9.03 DEG.C
TWOCLR 34.11 DEG.C

CALCULATED PARAMETERS
1 PWRIN 4080. WATTS
2 QCOOLR 2589. WATTS
3 QDSHPT 730. WATTS
4 EXTEFF 26.4 %
5 TAVHTR 599.2 DEG.C
6 INTEFF 29.4 %
8 AMPS 1369. AMPS
9 QDISPG 3. WATTS
10 QDISP 14. WATTS
11 QREG1 102. WATTS
12 QREG2 120. WATTS
13 QREG3 34. WATTS

VIBRATION
VX1HOR 0.2 CM/S
VY1VER 2.3 CM/S

PHASE ANGLES
PADISP 57.1 DEG.
PAPRES -22.5 DEG.

ENGINE SPEED
FREQ 29.9 HZ

REMOTE CALCULATIONS
PWROUT 1077. WATTS
INDPWR 1125. WATTS
PISTST 2.38 CM
DISPST 2.65 CM

T13REG 561.0 DEG.C
T14REG 525.9 DEG.C
T15REG 397.9 DEG.C
T16REG 120.7 DEG.C
T17REG 357.2 DEG.C
T18REG 378.2 DEG.C
T19REG 252.1 DEG.C

T03HED 2[illegible].9 DEG.C

NASA LEWIS PRELIMINARY DATA 01/03/84 RE 1000 FREE PISTON STIRLING ENGINE TEST D001, REC 11/17/83 10:17:21.80 RDG 602

FLUID HELIUM BAROM 14.268 PSI

HEAT TO DASHPOT COOLING
FLODP 4.10 L/MIN
TWINDP 17.3 DEG.C
TDLDP 2.94 DEG.C
TWODPR 20.9 DEG.C

POWER IN
AMPS1 776. AMPS
AMPS2 651. AMPS
VOLTG 3.11 VOLTS

ENGINE CHARGE PRESSURE
PRESUP 0. KPA
MEANBP 0. KPA
MEANCP 0. KPA

GAS TEMPERATURES
TGEXP 554.8 DEG.C
TGREGH 547.0 DEG.C
TGREGC 101.9 DEG.C
TGCOMP 61.6 DEG.C
TGBOUN 36.9 DEG.C

SURFACE TEMPERATURES
T01HTR 609.3 DEG.C
T02HTR 609.9 DEG.C
T03HTR 619.7 DEG.C
T04HTR 594.8 DEG.C
T05HTR 604.4 DEG.C
T06HTR 602.3 DEG.C
T07HTR 573.2 DEG.C
T08HTR 605.9 DEG.C
T09HTR 619.2 DEG.C
T10HTR 546.2 DEG.C
T11HTR 603.4 DEG.C
T12HTR 603.9 DEG.C

HEAT TO COOLER
FLOCLR 4.17 L/MIN
TWINCL 25.2 DEG.C
TDLCL 9.63 DEG.C
TWOCLR 34.80 DEG.C

CALCULATED PARAMETERS
1 PWRIN 4433. WATTS
2 QCOOLR 2789. WATTS
3 QDSHPT 840. WATTS
4 EXTEFF 27.4 %
5 TAVHTR 599.3 DEG.C
6 INTEFF 30.4 %
8 AMPS 1427. AMPS
9 QDISPG 3. WATTS
10 QDISP 14. WATTS
11 QREG1 106. WATTS
12 QREG2 121. WATTS
13 QREG3 34. WATTS

VIBRATION
VX1HOR 0.2 CM/S
VY1VER 2.5 CM/S

PHASE ANGLES
PADISP 57.1 DEG.
PAPRES -21.6 DEG.

ENGINE SPEED
FREQ 29.9 HZ

REMOTE CALCULATIONS
PWROUT 1216. WATTS
INDPWR 1262. WATTS
PISTST 2.58 CM
DISPST 2.72 CM

T13REG 553.7 DEG.C
T14REG 514.8 DEG.C
T15REG 376.7 DEG.C
T16REG 113.7 DEG.C
T17REG 355.8 DEG.C
T18REG 366.0 DEG.C
T19REG 235.5 DEG.C

T03HED 273.5 DEG.C

TABLE I. - RE-1000 DATA (continued)

NASA LEWIS PRELIMINARY DATA 01/03/84 RE 1000 FREE PISTON STIRLING ENGINE TEST D001, REC 11/17/83 10:42:18.80 RDG 603

FLUID HELIUM BAROM 14.278 PSI

HEAT TO DASHPOT COOLING
FLODP 4.19 L/MIN
TWINDP 18.3 DEG.C
TDLDP 1.81 DEG.C
TWODPR 20.8 DEG.C

POWER IN
AMPS1 608. AMPS
AMPS2 525. AMPS
VOLTG 2.45 VOLTS

ENGINE CHARGE PRESSURE
PRESUP 0. KPA
MEANBP 0. KPA
MEANCP 0. KPA

GAS TEMPERATURES
TGEXP 519.9 DEG.C
TGREGH 508.3 DEG.C
TGREGC 93.2 DEG.C
TGCOMP 47.8 DEG.C
TGBOUN 34.9 DEG.C

SURFACE TEMPERATURES
T01HTR 547.2 DEG.C
T02HTR 567.6 DEG.C
T03HTR 558.6 DEG.C
T04HTR 554.3 DEG.C
T05HTR 558.4 DEG.C
T06HTR 542.8 DEG.C
T07HTR 537.5 DEG.C
T08HTR 560.3 DEG.C
T09HTR 572.3 DEG.C
T10HTR 502.1 DEG.C
T11HTR 556.9 DEG.C
T12HTR 559.0 DEG.C

HEAT TO COOLER
FLOCLR 4.19 L/MIN
TWINCL 25.2 DEG.C
TDLCL 6.33 DEG.C
TWOCLR 31.52 DEG.C

CALCULATED PARAMETERS
1 PWRIN 2775. WATTS
2 QCOOLR 1840. WATTS
3 QDSHPT 527. WATTS
4 EXTEFF 23.9 %
5 TAVHTR 551.4 DEG.C
6 INTEFF 26.5 %
8 AMPS 1133. AMPS
9 QDISPG 1. WATTS
10 QDISP 13. WATTS
11 QREG1 87. WATTS
12 QREG2 118. WATTS
13 QREG3 32. WATTS

VIBRATION
VX1HOR 0.1 CM/S
VY1VER 1.7 CM/S

PHASE ANGLES
PADISP 56.2 DEG.
PAPRES -24.5 DEG.

ENGINE SPEED
FREQ 29.8 HZ

REMOTE CALCULATIONS
PWROUT 662. WATTS
INDPWR 686. WATTS
PISTST 1.80 CM
DISPST 2.17 CM

T13REG 518.4 DEG.C
T14REG 486.5 DEG.C
T15REG 371.1 DEG.C
T16REG 116.8 DEG.C
T17REG 332.4 DEG.C
T18REG 340.8 DEG.C
T19REG 233.9 DEG.C

T03HED 314.2 DEG.C

NASA LEWIS PRELIMINARY DATA 01/24/84 RE 1000 FREE PISTON STIRLING ENGINE TEST D001, REC 01/24/84 16:32:22.08 RDG 945

FLUID HELIUM BAROM 14.258 PSI

HEAT TO DASHPOT COOLING
FLODP 4.32 L/MIN
TWINDP 17.0 DEG.C
TDLDP 3.24 DEG.C
TWODPR 20.8 DEG.C

POWER IN
AMPS1 803. AMPS
AMPS2 721. AMPS
VOLTG 3.40 VOLTS

ENGINE CHARGE PRESSURE
PRESUP 7141. KPA
MEANBP 7036. KPA
MEANCP 7105. KPA

GAS TEMPERATURES
TGEXP 921.7 DEG.C
TGREGH 531.0 DEG.C
TGREGC 110.4 DEG.C
TGCOMP 67.4 DEG.C
TGBOUN 42.1 DEG.C

SURFACE TEMPERATURES
T01HTR 604.3 DEG.C
T02HTR 604.0 DEG.C
T03HTR 613.6 DEG.C
T04HTR 590.8 DEG.C
T05HTR 596.2 DEG.C
T06HTR 599.8 DEG.C
T07HTR 560.6 DEG.C
T08HTR 603.5 DEG.C
T09HTR 614.0 DEG.C
T10HTR 603.4 DEG.C
T11HTR 597.8 DEG.C
T12HTR 599.2 DEG.C

HEAT TO COOLER
FLOCLR 4.25 L/MIN
TWINCL 25.1 DEG.C
TDLCL 11.19 DEG.C
TWOCLR 36.56 DEG.C

CALCULATED PARAMETERS
1 PWRIN 5185. WATTS
2 QCOOLR 3303. WATTS
3 QDSHPT 975. WATTS
4 EXTEFF 22.9 %
5 TAVHTR 598.9 DEG.C
6 INTEFF 26.4 %
8 AMPS 1524. AMPS
9 QDISPG 5. WATTS
10 QDISP 24. WATTS
11 QREG1 105. WATTS
12 QREG2 107. WATTS
13 QREG3 33. WATTS

VIBRATION
VX1HOR 0.2 CM/S
VY1VER 3.2 CM/S

PHASE ANGLES
PADISP 82.9 DEG.
PAPRES -12.8 DEG.

ENGINE SPEED
FREQ 31.7 HZ

REMOTE CALCULATIONS
PWROUT 1187. WATTS
INDPWR 1260. WATTS
PISTST 2.99 CM
DISPST 2.42 CM

T13REG 424.1 DEG.C
T14REG 499.3 DEG.C
T15REG 378.6 DEG.C
T16REG 90.3 DEG.C
T17REG 365.1 DEG.C
T18REG 367.9 DEG.C
T19REG 238.3 DEG.C

T03HED 505.8 DEG.C

TABLE I. - RE-1000 DATA (concluded)

NASA LEWIS PRELIMINARY DATA 01/24/84 RE 1000 FREE PISTON STIRLING ENGINE TEST D001, REC 01/24/84 16:24:31.06 RDG 943

FLUID HELIUM BAROM 14.258 PSI

HEAT TO DASHPOT COOLING
FLODP 4.34 L/MIN

TWINDP 16.9 DEG.C
TDLDP 2.75 DEG.C
TWODPR 20.3 DEG.C

POWER IN
AMPS1 730. AMPS
AMPS2 660. AMPS
VOLTG 3.10 VOLTS

ENGINE CHARGE PRESSURE
PRESUP 7459. KPA

MEANBP 7007. KPA
MEANCP 7059. KPA

GAS TEMPERATURES
TGEXP 900.7 DEG.C

TGREGH 532.2 DEG.C
TGREGC 103.1 DEG.C
TGCOMP 58.6 DEG.C
TGBOUN 40.2 DEG.C

SURFACE TEMPERATURES
T01HTR 602.5 DEG.C
T02HTR 605.8 DEG.C
T03HTR 612.3 DEG.C
T04HTR 592.2 DEG.C
T05HTR 597.3 DEG.C
T06HTR 600.1 DEG.C
T07HTR 564.5 DEG.C
T08HTR 602.7 DEG.C
T09HTR 611.6 DEG.C
T10HTR 601.8 DEG.C
T11HTR 595.6 DEG.C
T12HTR 601.5 DEG.C

T13REG 420.0 DEG.C
T14REG 502.8 DEG.C
T15REG 373.7 DEG.C
T16REG 99.3 DEG.C
T17REG 359.9 DEG.C
T18REG 370.5 DEG.C
T19REG 237.4 DEG.C

T03HED 509.0 DEG.C

HEAT TO COOLER
FLOCLR 4.26 L/MIN
TWINCL 25.0 DEG.C
TDLCL 9.20 DEG.C
TWOCLR 34.45 DEG.C

CALCULATED PARAMETERS
1 PWRIN 4314. WATTS
2 QCOOLR 2723. WATTS
3 QDSHPT 831. WATTS
4 EXTEFF 23.2 %
5 TAVHTR 599.0 DEG.C
6 INTEFF 26.9 %
8 AMPS 1390. AMPS
9 QDISPG 5. WATTS
10 QDISP 24. WATTS
11 QREG1 108. WATTS
12 QREG2 107. WATTS
13 QREG3 33. WATTS

VIBRATION
VX1HOR 0.1 CM/S
VY1VER 2.8 CM/S

PHASE ANGLES
PADISP 83.7 DEG.
PAPRES -14.3 DEG.

ENGINE SPEED
FREQ 31.9 HZ

REMOTE CALCULATIONS
PWRDUT 1001. WATTS
INDPWR 1078. WATTS
PISTST 2.59 CM
DISPST 2.20 CM

NASA LEWIS PRELIMINARY DATA 01/24/84 RE 1000 FREE PISTON STIRLING ENGINE TEST D001, REC 01/24/84 16:10:18.91 RDG 941

FLUID HELIUM BAROM 14.258 PSI

HEAT TO DASHPOT COOLING
FLODP 4.35 L/MIN

TWINDP 16.9 DEG.C
TDLDP 2.28 DEG.C
TWODPR 19.8 DEG.C

POWER IN
AMPS1 660. AMPS
AMPS2 601. AMPS
VOLTG 2.81 VOLTS

ENGINE CHARGE PRESSURE
PRESUP 7124. KPA

MEANBP 6999. KPA
MEANCP 7045. KPA

GAS TEMPERATURES
TGEXP 986.8 DEG.C

TGREGH 536.7 DEG.C
TGREGC 97.4 DEG.C
TGCOMP 51.3 DEG.C
TGBOUN 39.0 DEG.C

SURFACE TEMPERATURES
T01HTR 601.7 DEG.C
T02HTR 606.8 DEG.C
T03HTR 610.6 DEG.C
T04HTR 594.1 DEG.C
T05HTR 597.2 DEG.C
T06HTR 601.5 DEG.C
T07HTR 567.6 DEG.C
T08HTR 604.3 DEG.C
T09HTR 610.3 DEG.C
T10HTR 601.3 DEG.C
T11HTR 596.1 DEG.C
T12HTR 602.9 DEG.C

T13REG 417.9 DEG.C
T14REG 505.4 DEG.C
T15REG 374.0 DEG.C
T16REG 110.2 DEG.C
T17REG 360.6 DEG.C
T18REG 369.3 DEG.C
T19REG 233.3 DEG.C

T03HED 509.8 DEG.C

HEAT TO COOLER
FLOCLR 4.22 L/MIN
TWINCL 24.8 DEG.C
TDLCL 7.51 DEG.C
TWOCLR 32.55 DEG.C

CALCULATED PARAMETERS
1 PWRIN 3547. WATTS
2 QCOOLR 2199. WATTS
3 QDSHPT 690. WATTS
4 EXTEFF 23.2 %
5 TAVHTR 599.5 DEG.C
6 INTEFF 27.2 %
8 AMPS 1261. AMPS
9 QDISPG 5. WATTS
10 QDISP 27. WATTS
11 QREG1 110. WATTS
12 QREG2 110. WATTS
13 QREG3 34. WATTS

VIBRATION
VX1HOR 0.1 CM/S
VY1VER 2.4 CM/S

PHASE ANGLES
PADISP 83.9 DEG.
PAPRES -15.8 DEG.

ENGINE SPEED
FREQ 32.2 HZ

REMOTE CALCULATIONS
PWRDUT 822. WATTS
INDPWR 892. WATTS
PISTST 2.19 CM
DISPST 1.97 CM

TABLE II. - RE-1000 DATA

Reading number	PDYNC amplitude, MPa	PDYNB amplitude, MPa	Piston center, cm	Displacer center, cm
600	0.955	-----	0.58	-0.11
601	1.034	-----	-.59	-.16
602	1.127	-----	-.58	-.21
941	1.142	1.304	-.38	.02
943	1.344	1.452	-.43	-.04
945	1.540	1.599	-.37	-.08

TABLE III. - PISTON STROKES AND CENTERING PORT LOCATIONS

Maximum power piston stroke, cm	4.35
Power piston centerline port[a], cm	2.05
Maximum displacer 1 stroke, cm	4.01
Displacer 1 centering port[a], cm	1.90
Maximum displacer 2 stroke, cm	5.18
Displacer 2 centerline port[a], cm	2.64

[a]Centering port location is the distance from the in-limit to the position where the centering port is open.

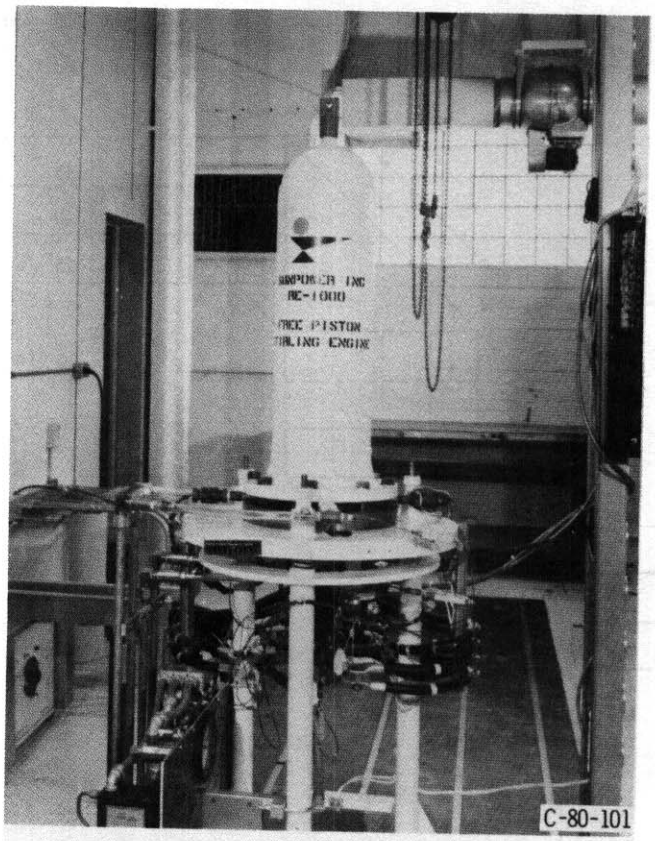


Figure 1. - RE-1000 free-piston Stirling Engine in test cell.

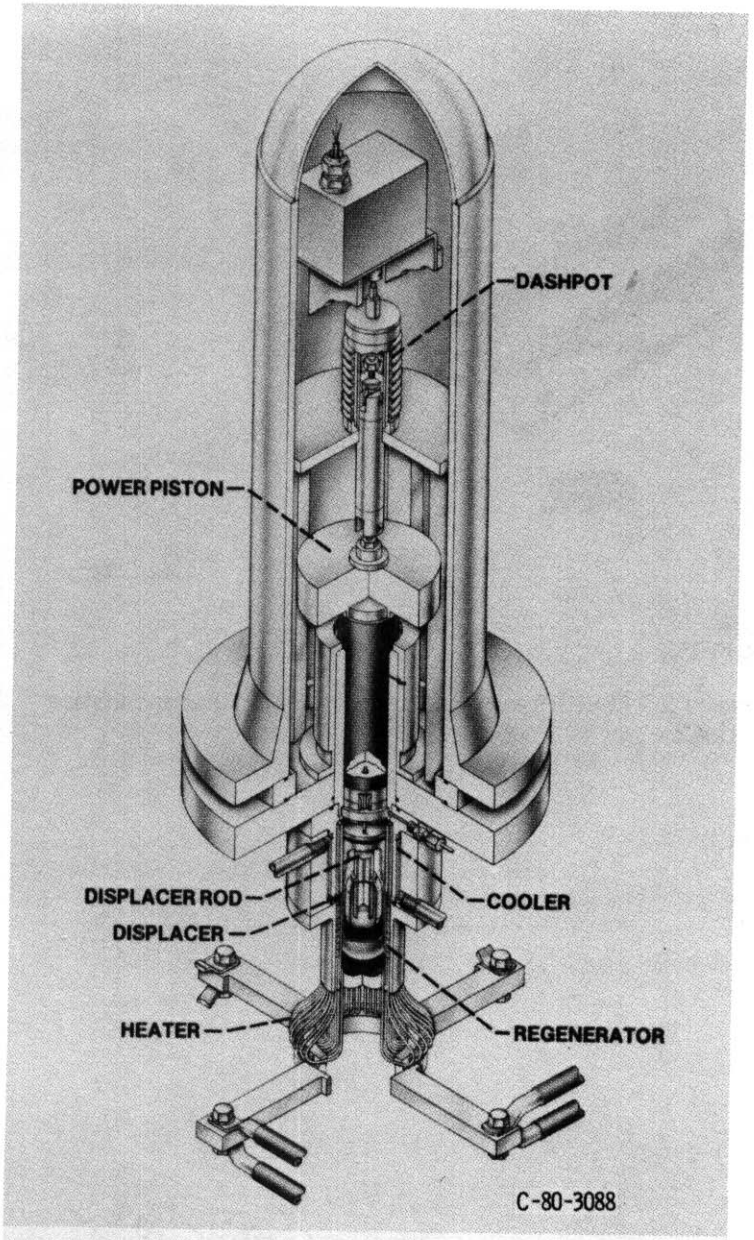


Figure 2. - Cutaway view of RE-1000 free-piston, free-displacer Stirling Engine.

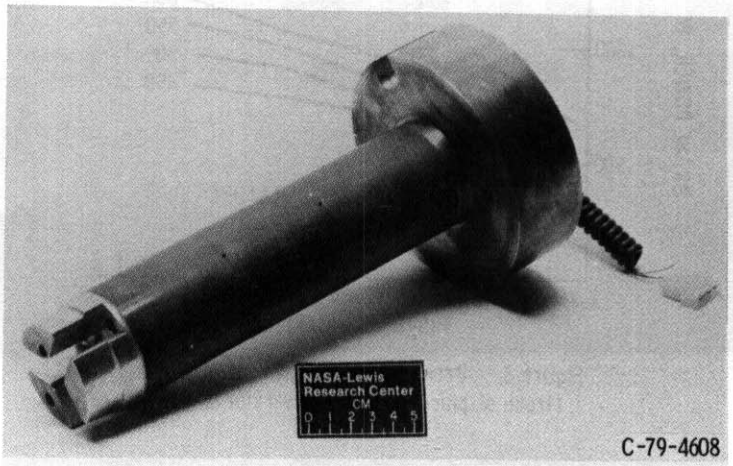


Figure 3. - RE-1000 free-piston Stirling Engine power piston.

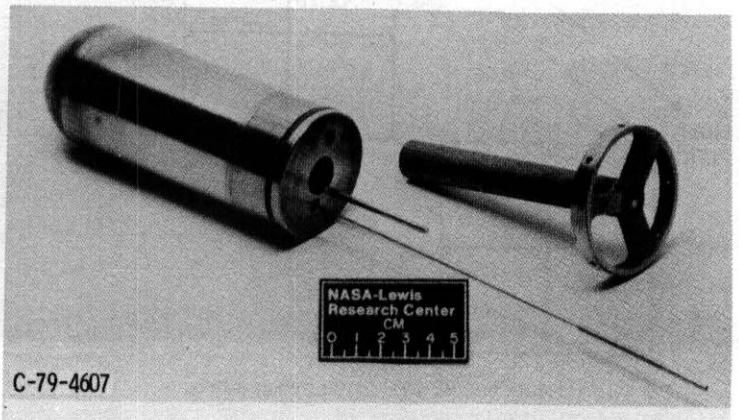


Figure 4. - RE-1000 free-piston Stirling Engine displacer and displacer rod 1.

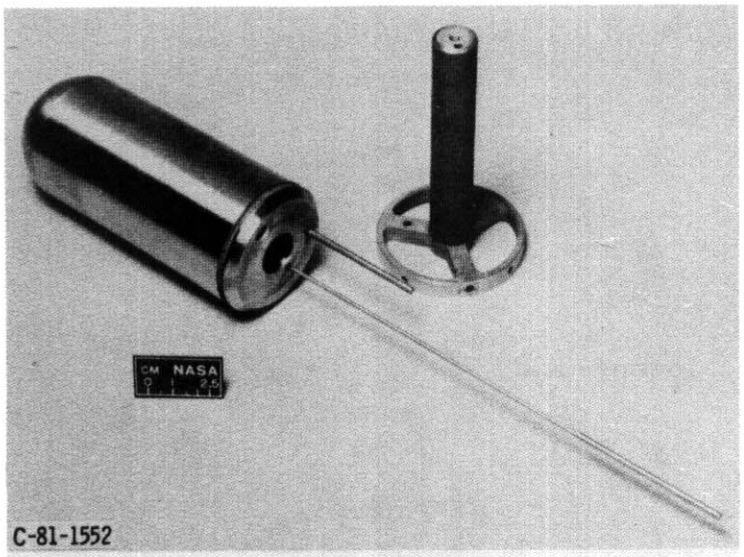


Figure 5. - RE-1000 free-piston Stirling Engine displacer and displacer rod 2.

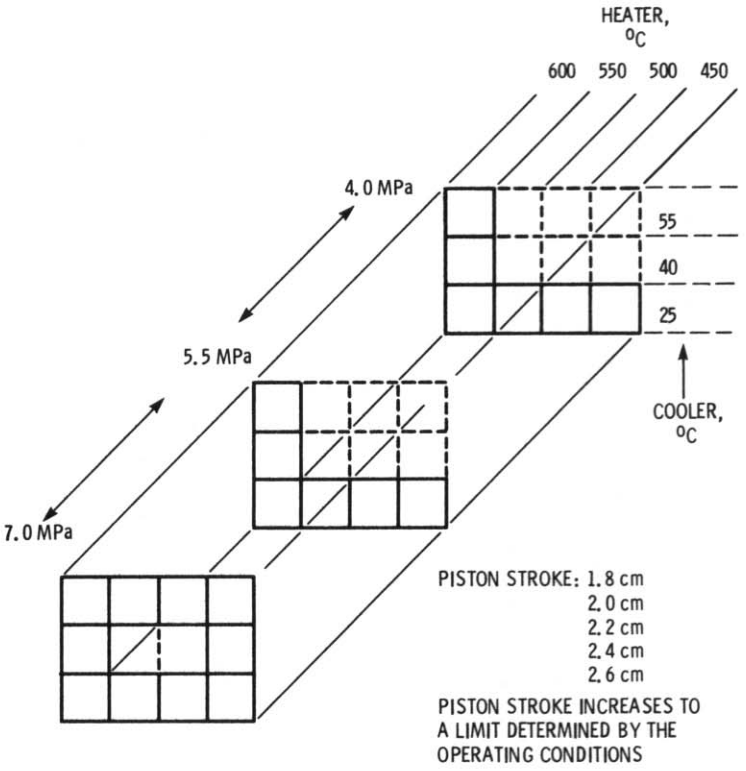


Figure 7. - RE-1000 test matrix.

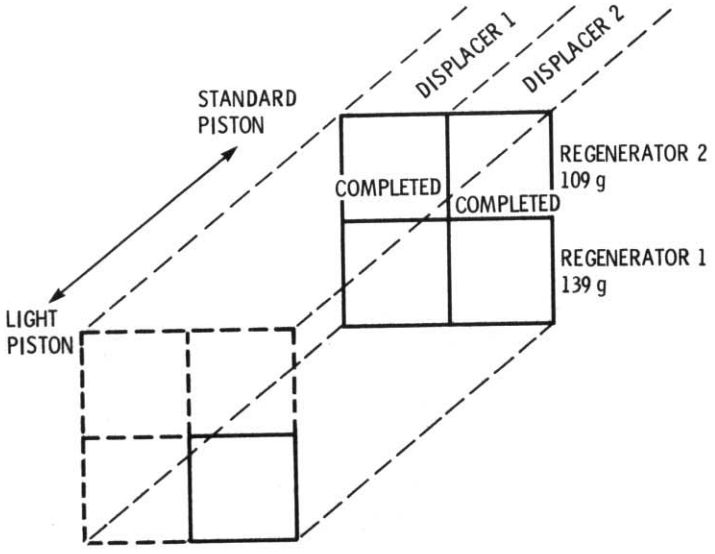


Figure 6. - RE-1000 configuration matrix.

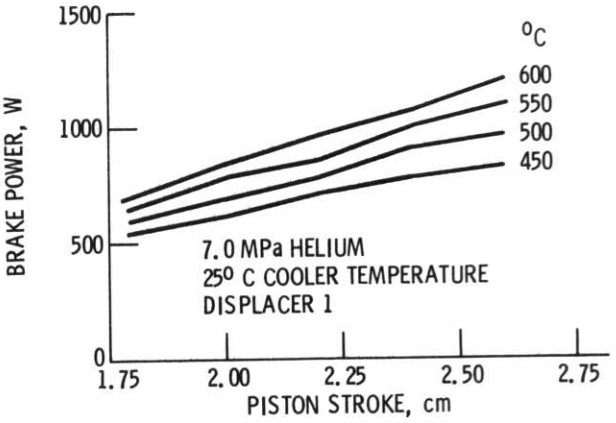


Figure 8. - Brake power output versus piston stroke displacer 1 and regenerator 2.

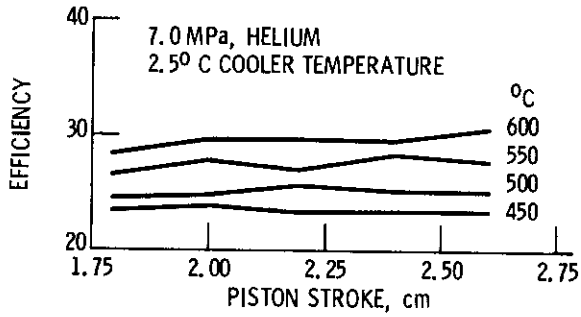


Figure 9. - Efficiency versus piston stroke displacer 1 and regenerator 2.

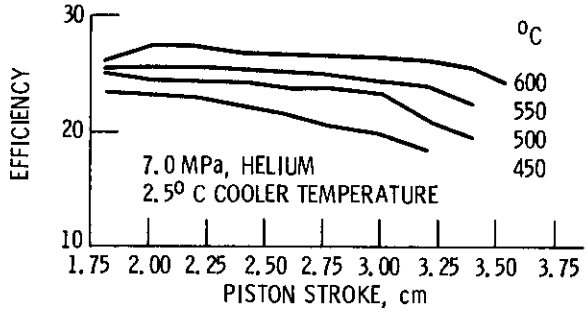


Figure 11. - Efficiency versus piston stroke displacer 2 and regenerator 2.

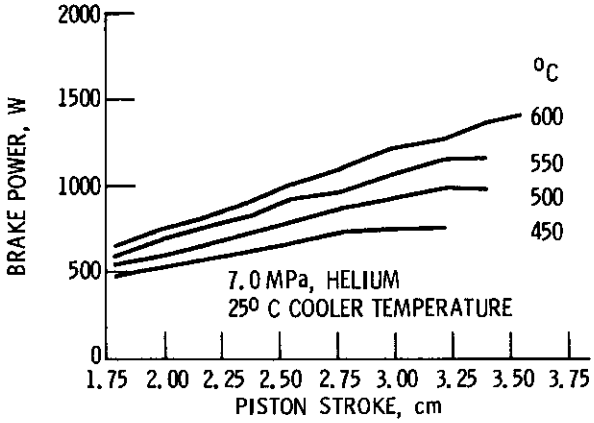


Figure 10. - Brake power output versus piston stroke displacer 2 and regenerator 2.

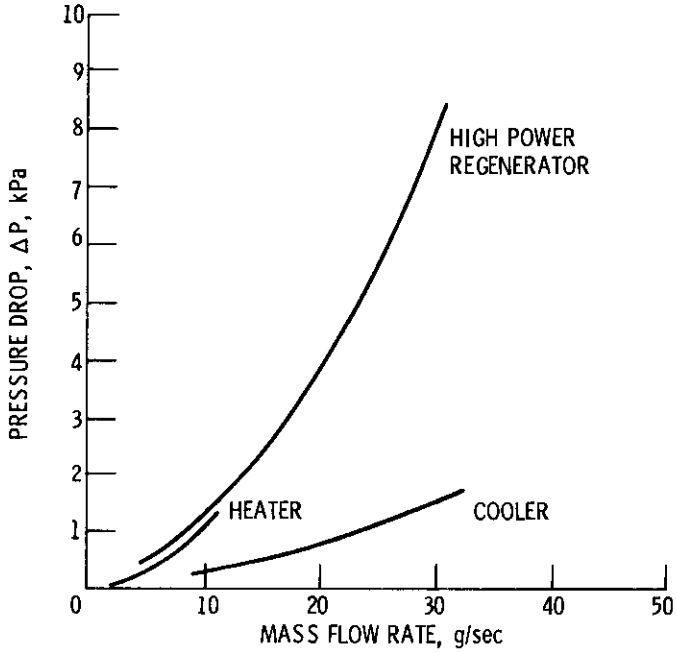


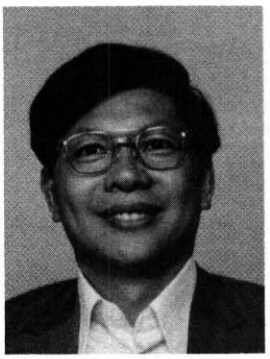
Figure 12. - Pressure drop as a function of mass flow rate for the RE-1000 with nitrogen at 2.07 MPa inlet pressure for the cooler and regenerator, and 1.38 MPa for the heater; 109-g regenerator.

Simplified Analysis of Stirling Engines and Heat Pumps

Norbert C. J. Chen, Frederick P. Griffin, and Colin D. West

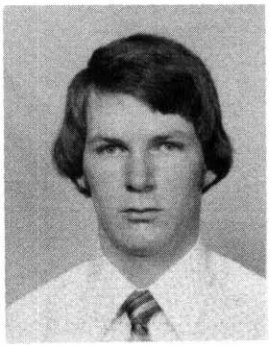
Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831



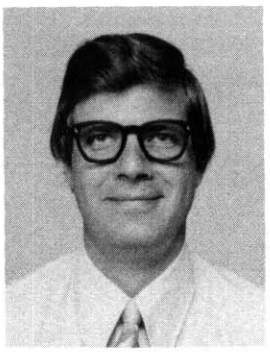
Norbert C. J. Chen received a B.S. in mechanical engineering in 1963 (National Taiwan University), an M.S. in aerospace engineering in 1968 (University of Cincinnati), and a Ph.D. in astronautics, aeronautics, and engineering sciences in 1972 (Purdue University). His master's and doctoral topics were in the areas of magnetohydrodynamics, with applications to solar physics. He worked as a postdoctoral fellow at the National Center for Atmospheric Research (Boulder, Colorado) and a research engineer at General Dynamics (San Diego, California).

Dr. Chen has been a staff member with the Experimental Engineering Section of the Engineering Technology Division at Oak Ridge National Laboratory for 10 years. He has had a leading role in free-piston Stirling engine analytical development, and has been equally active in aerosol transport research.



Frederick P. Griffin received both the B.S. (mechanical engineering, 1978) and M.S. (thermal sciences, 1979) degrees from Rensselaer Polytechnic Institute. His master's thesis was in the area of thermal energy storage analysis. Current research work is primarily concerned with the Stirling-engine-driven heat pump. Some of his past activities have included analysis of fluidized-bed combustion and research on advanced power cycles for electric power plants.

Colin D. West is a senior staff member in the Engineering Technology Division at Oak Ridge National Laboratory where he is currently responsible for irradiation experiments and work relating to liquid-metal-cooled reactor safety. He has written two books on Stirling engines and coauthored another book on energy strategy for the United States. He has been organizer of the Stirling machine sessions at the Intersociety Energy Conversion Engineering Conference (IECEC) for the past 2 years, working to increase the number of U.S. and foreign papers in these sessions and making Stirling engine research one of the largest of the IECEC topic areas. The conference this year involved 8 sessions and 50 papers.



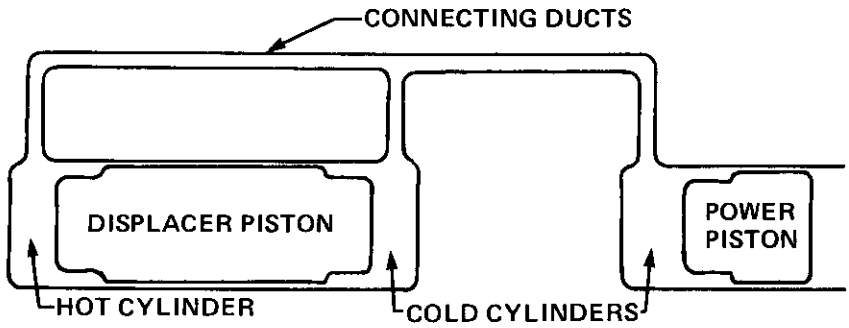
SIMPLIFIED ANALYSIS OF STIRLING ENGINES AND HEAT PUMPS

N. C. J. CHEN
F. P. GRIFFIN
C. D. WEST

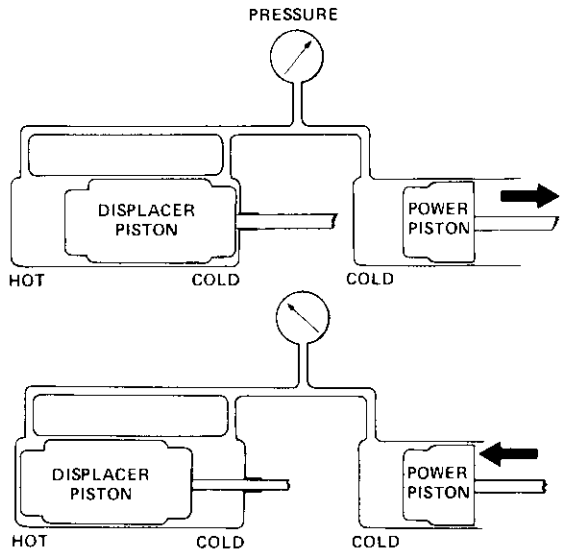
ENGINEERING TECHNOLOGY DIVISION
OAK RIDGE NATIONAL LABORATORY

DECEMBER 12, 1984

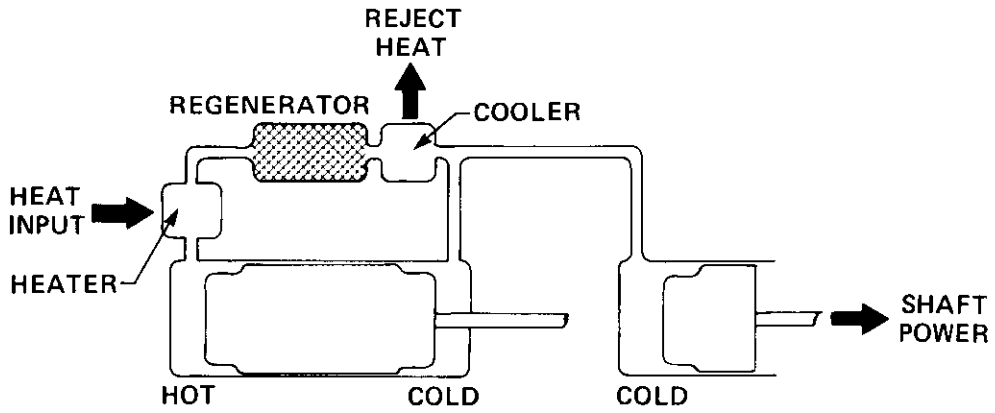
A SIMPLIFIED STIRLING MACHINE



THE POWER PISTON OF A STIRLING ENGINE WORKS ON THE PRESSURE CHANGES CREATED BY THE DISPLACER



STIRLING ENGINE WITH REGENERATOR AND HEAT EXCHANGERS



ANALYTICAL OBJECTIVES

- TO UNDERSTAND WHY FREE–PISTON STIRLING ENGINES FOR HEAT PUMPS DO NOT WORK AS WELL AS EXPECTED
    - IF THE MACHINES ALREADY BUILT HAD WORKED AS WELL AS PREDICTED, THE TECHNICAL OBJECTIVES OF THE PROGRAM WOULD HAVE BEEN MET
- TO UNDERSTAND THE NATURE OF THE LOSS MECHANISMS IN THE FREE–PISTON STIRLING ENGINE HEAT PUMP SYSTEMS SO THAT IMPROVEMENTS CAN BE MADE AND THE FULL POTENTIAL OF THESE MACHINES REALIZED

LINEAR HARMONIC ANALYSIS (LHA) (1)

- THE ANALYTICAL APPROACH ADOPTED CLOSELY REFLECTS THE PHYSICS OF REAL STIRLING ENGINES, WHICH TEND TO HAVE PISTON MOVEMENTS AND GAS PRESSURE VARIATIONS THAT ARE APPROXIMATELY SINUSOIDAL
- THE MAJOR VARIABLES (PRESSURE, GAS TEMPERATURE, PISTON POSITION, ETC.) ARE REPRESENTED AS A MEAN VALUE PLUS SINE AND COSINE TERMS, THAT IS, A LINEAR HARMONIC REPRESENTATION
- NONLINEAR COMBINATIONS OF THE BASIC VARIABLES ARE ALSO REPRESENTED AS A MEAN VALUE PLUS SINE AND COSINE TERMS BY A FOURIER SERIES TRUNCATED AFTER THE FIRST THREE TERMS

LINEAR HARMONIC ANALYSIS (2)

- WHEN THE MEAN VALUES, SINES, AND COSINES ARE SUBSTITUTED INTO THE DIFFERENTIAL EQUATIONS OF MASS AND ENERGY CONSERVATION, THE DIFFERENTIAL EQUATIONS BECOME SIMULTANEOUS ALGEBRAIC ONES THAT ARE EASILY SOLVED

- BECAUSE THE EQUATIONS ARE SOLVED SIMULTANEOUSLY, ALL LOSS MECHANISMS IN THE MODEL ARE TAKEN INTO ACCOUNT AT ONCE; THEIR INTERACTIONS AS WELL AS THEIR INDIVIDUAL EFFECTS ARE INCLUDED

    - IN THE STIRLING MACHINE FIELD, THIS IS CALLED A THIRD–ORDER MODEL

CURRENT STATUS OF LHA (1)

- THE ANALYSIS HAS BEEN FORMULATED FOR BOTH KINEMATIC AND FREE–PISTON ENGINES

- THE FORMULATION INCLUDES THE FOUR KNOWN MAJOR LOSS MECHANISMS IN THE CYCLE

    - ADIABATIC CYLINDER EFFECTS, TRANSIENT HEAT TRANSFER IN THE CYLINDERS, PRESSURE DROP IN THE HEAT EXCHANGER ELEMENTS, AND GAS LEAKAGE FROM THE COMPRESSION SPACE

- A SECOND–LAW ANALYSIS OF THE THERMODYNAMICS ALLOWS THE LOSS DUE TO EACH MECHANISM TO BE CALCULATED UNAMBIGUOUSLY, ALTHOUGH THE LOSSES ARE ACTING AND INTERACTING SIMULTANEOUSLY

- AN APPROXIMATE CALCULATION OF THE EFFECTS OF IMPERFECT HEAT EXCHANGERS AND REGENERATORS HAS ALSO BEEN INCORPORATED

CURRENT STATUS OF LHA (2)

- THE MATHEMATICAL ACCURACY OF THE METHOD HAS BEEN VERIFIED BY COMPARISON WITH NUMERICAL CALCULATIONS THAT DO NOT MAKE THE LINEARIZED APPROXIMATION
- THE THERMODYNAMIC ANALYSIS HAS BEEN PUBLISHED FOR PEER REVIEW AND USE
- THE LOSS ALLOCATION ANALYSIS HAS BEEN PUBLISHED FOR PEER REVIEW AND USE
- AN EXTENSION TO INCLUDE FREE–PISTON DYNAMICS AND IMPERFECT HEAT EXCHANGER EFFECTS HAS BEEN MADE, AND A REPORT WILL BE PUBLISHED SHORTLY
- A FURTHER EXTENSION HAS BEEN MADE TO ALLOW ANALYSIS OF MACHINES WITH THREE DIFFERENT TEMPERATURE LEVELS (e.g., VILLEUMIER HEAT PUMP AND GAMMA COGENERATOR)

COMPARISON WITH A DIFFERENTIAL EQUATION–SOLVING ROUTINE THAT IS NOT LINEARIZED (CSMP) SHOWS THAT LOSS OF ACCURACY DUE TO LINEARIZATION IS VERY SMALL

	CALCULATED INDICATED EFFICIENCY (%)		
LOSS MECHANISMS	CSMP[a]	LINEAR	DIFFERENCE
NONE	66.7	66.7	0
ADIABATIC CYLINDERS	63.0	63.5	+0.5
FINITE–CYLINDER HEAT TRANSFER	57.6	57.4	–0.2
SEAL LEAK	54.1	54.2	+0.1
PRESSURE DROP	44.3	44.2	–0.1
ALL LOSSES	12.5	12.1	–0.4

[a]AN IBM ROUTINE FOR SOLVING DIFFERENTIAL EQUATIONS

INITIAL ACCOMPLISHMENTS OF LHA (1)

- THE ANALYSIS IS MUCH FASTER THAN NUMERICAL METHODS

CPU TIME	LHA (s)	CSMP (s)
COMPILE	1.84	3.01
RUN	0.02	11.11
TOTAL FOR 1 CASE	1.86	14.12
TOTAL FOR 10 CASES	2.04	114.11

INITIAL ACCOMPLISHMENTS OF LHA (2)

- IMPORTANT INTERACTIONS BETWEEN SOME OF THE VARIOUS LOSS MECHANISMS HAVE BEEN REVEALED
- THE FOLLOWING TABLE SHOWS THAT THE EFFECT OF PRESSURE DROP IS TO INCREASE THE OTHER LOSSES TOO

	EFFICIENCY LOSS (%)		
PRESSURE DROP COEFFICIENT	ADIABATIC AND TRANSIENT HEAT TRANSFER EFFECTS	SEAL LEAKAGE	PRESSURE DROP
ZERO	8.9	14.1	
HIGH	13.5	22.2	18.2

INITIAL ACCOMPLISHMENTS OF LHA (3)

- THE IMPORTANCE OF CYLINDER HEAT TRANSFER EFFECTS, ESPECIALLY IN HEAT PUMPS, HAS BEEN CONFIRMED AND ILLUSTRATED
- USING THE ANALYSIS, IT IS POSSIBLE TO ALLOCATE THE CYLINDER HEAT TRANSFER EFFECTS TO THREE DIFFERENT MECHANISMS
  - TRANSIENT HEAT TRANSFER
  - MIXING LOSS IN THE CYLINDERS
  - FINITE TEMPERATURE DIFFERENCE LOSSES IN THE HEATER AND COOLER

THE EFFICIENCY REDUCTION DUE TO PERFECTLY ADIABATIC CYLINDER EFFECTS IS LARGER FOR SMALL–TEMPERATURE DIFFERENCE MACHINES AND ESPECIALLY FOR HEAT PUMPS

EXPANSION–SPACE HEAT EXCHANGER TEMPERATURE[a] (°C)	ENGINE EFFICIENCY (%)			HEAT PUMP COP	
	ISOTHERMAL	ADIABATIC	REDUCTION	ISOTHERMAL	ADIABATIC
637	65.9	60.9	−5.0		
200	34.5	27.1	−7.4		
37	0	−10.5	−10.5		
+10				11.5	5.8
0				8.4	4.9
−10				6.6	4.3

[a]COMPRESSION–SPACE TEMPERATURE IS 37°C

PROPOSED FUTURE WORK WITH LHA (1)

- COMPLETE PUBLICATION OF RESEARCH FOR PEER REVIEW AND USE
- VALIDATE BY COMPARISON WITH EXPERIMENTAL RESULTS OBTAINED BY NASA UNDER THE BER PROGRAM
    - IT IS EXPECTED THAT DISCREPANCIES BETWEEN MEASUREMENT AND PREDICTION WILL, WHEN EXAMINED BY LHA, LEAD TO A GREATER UNDERSTANDING OF THE ENGINE PERFORMANCE SHORTFALLS
- COMPLETE EXTENSION TO A WIDER RANGE OF STIRLING ENGINE/HEAT PUMP CONFIGURATIONS
    - ESPECIALLY THE GE INERTIAL COMPRESSOR SYSTEM
- MAKE COMPARISONS WITH EXPERIMENTAL MEASUREMENTS IN THE GE SYSTEM OBTAINED IN THE BER PROGRAM

PROPOSED FUTURE WORK WITH LHA (2)

- ISOLATE THE SOURCES OF LOSS WHEN SEVERAL LOSSES ARE SIMULTANEOUSLY PRESENT, AND RECOMMEND IMPROVEMENTS
- EVALUATE THE EXPECTED ADVANTAGES OF BOTH CURRENT AND NEW PROPOSALS FOR STIRLING/HEAT PUMP SYSTEMS
    - THE EVALUATION SHOULD BE NOT ONLY OF PRESENT PERFORMANCE BUT ALSO THE PERFORMANCE EXPECTED FROM IMPROVED AND DEVELOPED SYSTEMS

CONCLUSIONS

- THE LHA IS A POWERFUL AND ACCURATE TOOL THAT IS WELL SUITED TO THE ANALYSIS OF STIRLING MACHINES
- IT APPLIES TO BOTH KINEMATIC AND FREE–PISTON ENGINES
- THE METHOD IS FAST, INEXPENSIVE, AND EASY TO USE, WHILE MAINTAINING A CLOSE PHYSICAL CONNECTION WITH THE EFFECTS TAKING PLACE IN REAL MACHINES
- PRELIMINARY WORK HAS SHOWN THE METHOD TO BE CAPABLE OF MATHEMATICALLY ACCURATE ANALYSIS OF STIRLING SYSTEMS WITH SEVERAL SIMULTANEOUS LOSS EFFECTS
- PROPOSALS FOR FUTURE WORK TAKE ADVANTAGE OF THE STRENGTHS OF THE ANALYTICAL METHOD TO SEEK AN UNDERSTANDING OF THE DISAPPOINTING PERFORMANCE OF FREE–PISTON/HEAT PUMP HARDWARE AND TO PROPOSE IMPROVEMENTS

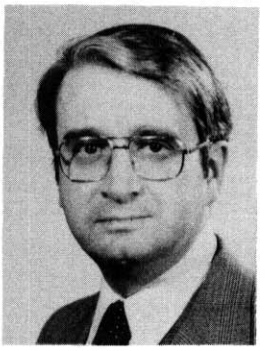
Assessment of Free-Piston Stirling Engines as Heat Pump Drives

W. Peter Teagan

Arthur D. Little, Inc.

Cambridge, Massachusetts 02140

W. Peter Teagan is manager of the Technology Assessment Unit for conducting technical, economic, and market evaluations on a wide range of advanced thermal and mechanical technologies such as heat pumps, Stirling engines, waste heat recovery equipment, and solar heating systems. Prior to joining Arthur D. Little, he was with Thermo Electron Corporation where he worked in the development of advanced heat engine systems for both terrestrial and space applications. He received his B.Sc. degree from Brown University and his Ph.D. in mechanical engineering from the Massachusetts Institute of Technology.



ASSESSMENT OF FREE-PISTON STIRLING ENGINES AS HEAT PUMP DRIVES

W. Peter Teagan
Arthur D. Little, Inc.
Cambridge, MA 02140

1.0 BACKGROUND

ORNL and the DOE have been supporting the development of FPSE/Heat Pump systems for over six years. These programs have contributed to advancing the state of knowledge on this technology on which to base future R&D and commercial development programs.

This paper describes the initial results of a program to assess the status and technology issues of FPSE/Heat Pump technology, formulate approaches for resolving technical problems which have arisen, and identify program planning options which will effectively use financial resources to undertake needed R&D leading to the development of systems with commercial potential. (Exhibit 1).

The approach pursued in undertaking Task 2 (per Exhibit 1) was to review the literature base which exists on FPSE systems and to meet with those organizations and individuals with significant activities in this field. Exhibit 2 lists the names of organizations visited or contacted during this effort.

The information relative to technical status and problem areas generated during this task will be used to:

- Assess the development problems and formulate approaches which would lead their solution (Task 3).
- Define specific R&D Program options (tasks, schedules, costs) which would implement the above program initiatives (Task 4).

In undertaking these tasks, many of the same organizations contacted in Task 2 will be recontacted to solicit their views on key technical issues and approaches to their resolution.

2.0 APPLICATION REQUIREMENTS

The review of the status and prospects for FPSE/Heat Pumps must take into account technical performance and cost constraints imposed by the economics of the application. The primary constraint is that the FPSE/Heat Pump System be economically competitive with other options which can be realistically considered for the same application. These include:

- o Gas furnace/electric air conditioning system.
- o Electric heat pumps.
- o Other engine driven heat pump options (I.C. engines, kinematic Stirling, etc.).
- o Absorption units.

This broadly stated constraint results in the approximate technical and cost specifications of Exhibit 3. Two key requirements, long life and high efficiency, of this application are discussed briefly below.

(a) Life and Reliability

The "system" must have a useful operating life of 10-15 years which implies, at least, 25,000 hours of operation and maybe over 50,000 hours. This statement is, however, not without its ambiguities since it does not exclude

replacing parts which are subject to wear or deterioration. For example, burner assembly parts which are readily accessible for maintenance could, in principle, be periodically replaced.

The life requirements for this application could be the most difficult goal for any engine driven system to achieve and impact significantly on the design flexibility for FPSE options.

(b) Engine Efficiency

A primary rationale for any gas driven heat pump unit is that annual energy costs for such a system will be significantly lower than for electric or gas driven alternatives. For a kinematic engine this usually implies that brake gross thermal efficiency (η_{bgt}) be in excess of 30 percent where efficiency is defined as:

$$\eta_{bgt} = \frac{\text{engine brake power output}}{\text{energy content of fuel at HHV + engine aid power at source efficiency}}$$

At this engine efficiency threshold the overall heating COP of an engine driven heat pump is in the range of 1.3 to 1.5 which results in significant cost savings as compared to a high efficiency furnace system. Significantly lower performance than this level could result in insufficient energy savings to make the system economically attractive.

3.0 RATIONALE FOR FPSE/HEAT PUMP

There are a number of engine drives which are being or have been considered for heat pump drives including kinematic Stirling, I.C. engines, and Brayton cycle engines. The FPSE is perceived as having potential advantages over the other, indicated in Exhibit 4.

(a) Life and Reliability

All Stirling engines have the advantage of not requiring valves or subjecting the working space or lubricants to hot combustion gases. These characteristics would tend to improve the life and reliability characteristics of this engine class. In addition, the FPSE eliminates the need for hermetic piston rods and a highly loaded crankcase mechanical drive with associated bearings and seals. In fact, in its simplest configuration the FPSE has only two moving parts, a displacer and a power piston. The superficial mechanical simplicity of the FPSE is used as the primary basis of the argument for very long life and good reliability. This argument can be further reinforced for those configurations using gas bearing and clearance seals in the engine and metal bellows and diaphragm seals on the outlet "shaft" thereby (ideally) resulting in no rubbing surfaces to contribute to wear. Such non rubbing seals have, in fact, demonstrated good life potential when used in specialized refrigeration equipment being developed for space use and in small heart pumps.

(b) High Efficiency

Ideally Stirling engines have very high efficiency potential when working at temperature levels attainable with metallic alloys (≡500-800°C). Kinematic Stirling engines have confirmed this efficiency potential with engine efficiencies as high as 35 percent measured on the automotive program. FPSE eliminate the losses associated with the tight shaft seals and other mechanical losses in the kinematic transmission. As a result, many FPSE proponents project that FPSE could have, at least, as high an efficiency as kinematic engines and possibly even higher.

(c) Low Cost Potential

In some configurations a FPSE/heat pump system can be fully mechanically integrated with no need for separate engine and compressor drive trains with associated connecting rods, crank rods, crankcases, bearings, and shaft seals.

The relatively small number of moving parts and configurational simplicity should have favorable cost ramifications.

(d) Low Noise and Vibration Levels

One potential advantage of all Stirling engines is low noise levels due to a lack of valves, and relatively smooth operation (as compared to I.C. engines). This advantage may be heightened for FPSE by eliminating the mechanical components associated with a kinematic drive shaft. Experience with selected FPSE tends to verify the low mechanical noise potential.

Most kinematic Stirling engines can be mechanically balanced resulting in low vibration levels. FPSE are, on the other hand, inherently unbalanced unless some form of opposed cylinder arrangement is utilized.* In principle, this unbalance can be isolated by using an appropriately designed spring mounting system. Isolation will be facilitated by the fact that the engine operates at a fixed frequency. The implications of this on overall system design have, however, only been superficially explored.

* Which is, in fact, used in the Mechanical Technologies Inc. design.

4.0 SYSTEM DESCRIPTIONS

There are at least 10 design options which have been considered for FPSE/Heat Pump systems. The following briefly describes two of the configurations which have received the most attention in government sponsored programs. The diagrams are greatly simplified to identify more clearly, the bearings, seals, and gas springs which are essential to system operation.

(a) Inertial Compressor (General Electric Co.)

Exhibit 5 is a simplified schematic of the G.E. concept utilizing an inertial compressor. Key features of this system include:

- The double acting freon compressor is contained within the power piston and moves relative to the power piston during system operation.
- Sliding bearings and seals are utilized on the power piston and displacer. All seals and bearings operate unlubricated in a dry helium environment.
- The connection of the compressor housing (power piston), which is oscillating with am amplitude of up to 1.7 inches, to the stationary heat pump housing is via flexible serpentine tubes which are flexed at engine frequency (30 Hz).
- The compressor subassembly is also a free piston device with an internal natural frequency determined by a regulated freon pressure (which defines the gas spring stiffness) and the mass of the compressor piston (6 lbs).

The system developed to date is designed to operate at 30 Hz with a helium working gas average pressure of 870 psi. In its present configuration, the motion is controlled only by internal dynamics of the 3 independently moving pistons. For reasons that are still not fully understood, the system natural frequency was lower (25 to 27 Hz) than the design value of 30 Hz with detrimental capacity and efficiency ramifications. An attempt was made to rectify this problem by adding a "second gas spring" for the power piston, thereby, increasing overall gas spring stiffness. The addition of this spring did not have the anticipated results and appeared to, if anything, reduce performance by virtue of increasing hysteresis losses. The disappointment with the results of adding the second gas spring, suggests that this complex dynamic system combined with the fluid flow issues associated with interactions of the engine and compressor are poorly understood making analytical projections of performance particularly difficult.

(b) Diaphram Coupled System (Mechanical Technologies Inc.)

Exhibit 6 is a simplified schematic of the system under development at MTI using a diaphragm actuated hydraulic system to transfer energy out of the Stirling engine.

In this system the "power piston" is incorporated into the mass of the hydraulic transmission. The restoring force is provided by the gas spring shown at the bottom of the hydraulic transmission assembly. During operation an increase in pressure in the engine due to the displacer shuttling gases to the hot space results in downward movement of the diaphragm. Via a hydraulic mechanism, (Exhibit 7) this movement is transformed into the movement of the refrigerant compressor assembly in a horizontal direction. A counterweight is provided in the hydraulic transmission which moves in the opposite direction of the compressor piston thereby providing dynamic balance for the power transfer assembly. This arrangement requires numerous seals both within the hydraulic system and between the high pressure (800 psi) hydraulic fluid and the low pressure freon contained in the compressor cylinder. The resultant friction and viscous fluid losses can be significant with present goals being about a 70 percent transmission efficiency at maximum load.

As indicated above, in this arrangement the "power piston" is incorporated into the diaphragm coupled hydraulically actuated compressor assembly. MTI has elected to divide the engine/heat pump into two assemblies; the "engine", the boundary of which is at the diaphragm and a compressor assembly which includes the diaphragms, hydraulic transmission, and lower gas spring. This paper continues with these definitions since they correspond to physically identifiable system subassemblies having measurable input/output characteristics. With these definitions the "engine" has no power piston contained in the helium envelop - only a displacer piston with associated seals, bearings, and gas springs. In the MTI system clearance seals and hydrostatic bearings are utilized on the displacer. As a result, there are no rubbing surfaces during operation which should virtually eliminate wear - except during startup and shutdown. The hydrostatic bearings, in turn, require extracting high pressure helium from the engine (during appropriate pressure pulses in the engine) to ensure a supply of helium at pressure levels above the mean. The means for doing this could include a pneumatic arrangement with check valves and buffer volumes. The mechanical and system efficiency implications of such arrangements have not been well defined at this time.

Capacity control is achieved by electromagnetically driving the displacer assembly at its natural frequency. The stoke of the displacer can thereby be controlled which, in turn, controls the output of the engine.

5.0 EXPERIENCE TO DATE/TECHNICAL ISSUES

Exhibits 8,9 and 10 summarize observations made relative to the potential advantages of FPSE/Heat Pumps as discussed in section 3. To date, FPSE/Heat Pump Systems have not demonstrated that stringent life, reliability, and efficiency requirements imposed by the economics of this application. However, significant progress has been made in FPSE technology both as a result of the DOE sponsored heat pump programs and other programs directed toward space and small generator applications with similar life requirements. This experience includes:

- Endurance testing of free piston equipment using combinations of gas bearing, clearance seals, and barrier seals (bellows, diaphrams) with over 1,000 hours on a 3kW FPSE alternator system (using the aforementioned M.I.T. technology) and up to 80,000 hours on small (100 watt) generators. This experience is supplemented by that from KSE where up to 10,000 hours of operation has been achieved using contract seals which could also be applied to FPSE.

- Steady improvements in the analytical design tools used to design FPSE which more accurately quantify basic loss mechanisms and allow for improved designs.

- Improvements in the design of major subassemblies, such as those in the hot end, to be more consistent with high efficiency and long life.

This experience indicates that the potential for the FPSE/Heat Pump might still be realized if the experience of the last six (6) years is effectively factored into the design of a well focussed R&D program.

The following discusses briefly some of the technical issues which were identified relative to the requirements of life, efficiency, and cost which must be addressed by future program efforts.

(a) Life and Reliability

Most of the effort on FPSE driven heat pumps has been directed toward achieving acceptable performance levels rather than demonstrating life and reliability. As a practical matter, therefore, there is only limited experience with operating equipment for sufficient periods of time to be meaningful.

Several observations can be made relative to life and reliability based on data from those programs which have received the major funding over the last 5 years.

(i) G.E. (Linear Compressor Program)

No endurance testing was undertaken with completely integrated FPSE/heat pump systems as part of this program. However, the system used rubbing bearing and seals. Limited endurance testing of these subassemblies indicated that they could be major life limiting factors given the operating conditions in the engines as currently designed. Additional life and reliability issues are associated with system control, the hot end subassembly, and the flexible tube arrangement used to connect the power piston/compressor assembly to the housing.

(ii) MTI - Hydraulic Transmission

Relatively few operating hours (several hundred) have been accumulated on the engine/-heat pump system. Recently, similar engines driving a linear alternator have undergone life testing in order to demonstrate their applicability to space applications. Over 1,000 hours of testing has been accumulated to date and life testing goals for this equipment have been raised to 10,000 hours.

It should be noted, however, that the heat pump system differs from the linear alternator systems since it uses a relatively complex hydraulic transmission coupled to the Stirling engine via flexing diaphragms. The hydraulic transmission itself requires four rubbing bearing/seals and two shaft seals between the high pressure hydraulic oil and the freon working spaces of the compressor. The hydraulic subassembly could be a more life limiting factor than the power side of the engine itself. Little operating experience has been gained on a complete transmission/ compressor subassembly.

There are a number of technical issues which could limit the life and reliability of FPSE driven heat pumps. These include:

- Bearing and seals which must operate unlubricated in a dry helium environment.

- Hot end components which must operate over thousands of on-off cycles annually (at least 10,000*) at temperatures in excess of 1400°F.

- Controls which must match FPSE output with highly variable load (and, in some cases, variable dynamic) conditions.

- Coupling arrangements between the FPSE and heat pump compressors which often, but not

always, involve some combination of numerous moving parts, diaphragms and bellows, or new design compressor assemblies (detracting from the superficial simplicity of the engine itself).

Of the above, the issues which could most influence the ultimate life and reliability and design of the FPSE system are the bearings and seals.

Contact bearings and seals are subject to wear, a problem exacerbated by their operation unlubricated in a dry helium environment. Low loss gas bearings and clearance seals require very small gaps (~.001") given current engine design approaches which, require very close tolerance with machining individual parts and extreme care in cleaning and assembly. In addition, the cyclic operation of a heat pump may result in frequent metallic contact during start and stop operation.

The issues of life and reliability of both types of seals and bearings has been under investigation for over ten years as the result of spaceborne cryocooler programs which attests to the importance of this issue.

(b) Efficiency

The FPSE/Heat Pump systems have not achieved the efficiency goals required by the heat pump application. As indicated in Exhibit 11, the "gross thermal efficiency" (efficiency at which fuel in converted into refrigerant compressor power) has generally been under 20%. By contrast a 30% kinematic engine driving on 85% efficient compressor would have on efficiency at 26%.

Experience to date is almost 30 percent below required levels. The reason for this are still not clear but include some combination of:

- Loss of positive phase angle and amplitude control between displacer and power piston possibly resulting in further (as compared to kinematic) engine deviations from ideal Stirling cycle performance.
- Gas springs required to result in naturally oscillating piston position have hysteresis (thermal and leakage) losses which may offset loss reductions associated with eliminating mechanical drives.
- Operation of electromagnetic displacer drive mechanisms which, in their present configurations, require external power thereby increasing parasitic power draws.
- Gas bearings (hydrostatic) require extraction of high pressure gas generated by the cycle and, therefore, represent a net loss of useful output power.
- Leakage losses of gas by clearance seals detracts directly from indicated power output.
- FPSE to Heat Pump coupling mechanisms which have relatively high losses.

Many, if not most, participants in the Stirling engine community feel that a well developed FPSE should be able to achieve efficiency levels comparable to those demonstrated in KSE. However, additional experimental and analytical work will be needed to assess the role of the various loss mechanisms and verify this efficiency potential. Of particular importance will be to identify FPSE to Heat Pump cycle coupling mechanisms which do not impose high losses on the system.

(c) Low Cost

Preliminary cost analysis prepared by the contractors indicated that advanced version of the FPSE/Heat Pump concepts under development might approach the constraints imposed by this application. These constraints imply a maximum manufacturing, cost of about $1,000 for an engine/compressor assembly in residential sizes.

It is not clear, however, that these cost estimates are based on designs that can also meet system performance requirements. Consequently, there are still open questions as to the ability of a FPSE/compressor system to simultaneously meet the cost and determine constraints of this application. These include the costs associated with:

- The design and material implications for the hot end subassemblies (burner, controls, air preheater, heater head,) to concurrently meet efficiency, life, and emission requirements.
- The coupling arrangements between the FPSE and the heat pump cycles.
- The high precision machining and care in cleaning and assembly implied by gas bearings and seals.

The impacts of the above and other issues need to be considered more fully in selecting design approaches for FPSE/Heat Pump systems.

Exhibit 1

PROGRAM TASKS

ASSESSMENT OF FREE-PISTON STIRLING ENGINE DRIVEN HEAT PUMP TECHNOLOGY

Task 1.0 MILESTONES AND COST SCHEDULE

Task 2.0 IDENTIFY AND CHARACTERIZE PRINCIPAL DEVELOPMENT PROBLEMS

Task 3.0 ASSESS DEVELOPMENT PROBLEMS AND FORMULATE APPROACHES TO THEIR SOLUTION

Task 4.0 DEFINE R&D PROGRAM OPTIONS

Task 5.0 FINAL REPORT PREPARATION

Exhibit 2

ORGANIZATIONS CONTACTED

Mechanical Technology, Inc.
968 Albany-Shaker Road
Latham, NY 12110

General Electric Company
Valley Forge
P.O. Box 8661
Philadelphia, PA 19101

Gas Research Institute
8600 West Bryn Mawr Avenue
Suite 1100
Chicago, IL 60631

Argonne National Laboratories
9700 South Cass Avenue
Argonne, IL 60439

Martini Engineering
2303 Harris
Richland, WA 907352

SUNPOWER, Inc.
6 Byard Street
Athens, OH 45701

Energy Research and Generation, Inc. (ERG)
Lowell and 57th Streets
Oakland, CA 94608

NASA-Lewis Research Center
1000 Brookpark Road
Cleveland, OH 44135

Stirling Power Systems
P. O. Box 1187
Ann Arbor, MI 48106

Joint Center for Graduate Study
University of Washington
100 Sprout Road
Richland, WA 99352

Oak Ridge National Laboratory
Oak Ridge, TN 37831

Exhibit 3

MAJOR PERFORMANCE REQUIREMENTS OF ENGINES FOR USE AS HEAT PUMP DRIVES

Brake Efficiency*	30%+
Life	25,000 - 50,000 hours
Maintenance	Once per year - minimal parts replacement required
Emissions	Comparable to conventional HVAC equipment
Low Noise and Vibration	
Manufacturing Cost	$300-$400 per kilowatt (max)

* Brake gross thermal efficiency based on fuel input and the equivalent of shaft power for a kinematic engine.

Exhibit 4

POTENTIAL ADVANTAGES OF FPSE/HEAT PUMP

- HIGH RELIABILITY/LONG LIFE
  - Few Moving Parts
  - Direct Coupling to HP CYCLE
  - No Shaft Seal
  - No Highly Loaded Bearings
- LOW COST
  - Simple Designs with Few Moving Parts
  - Direct Coupling to HP
- HIGH EFFICIENCY
  - Reduced Friction Losses
  - Reduced Engine → HP Transmission Losses

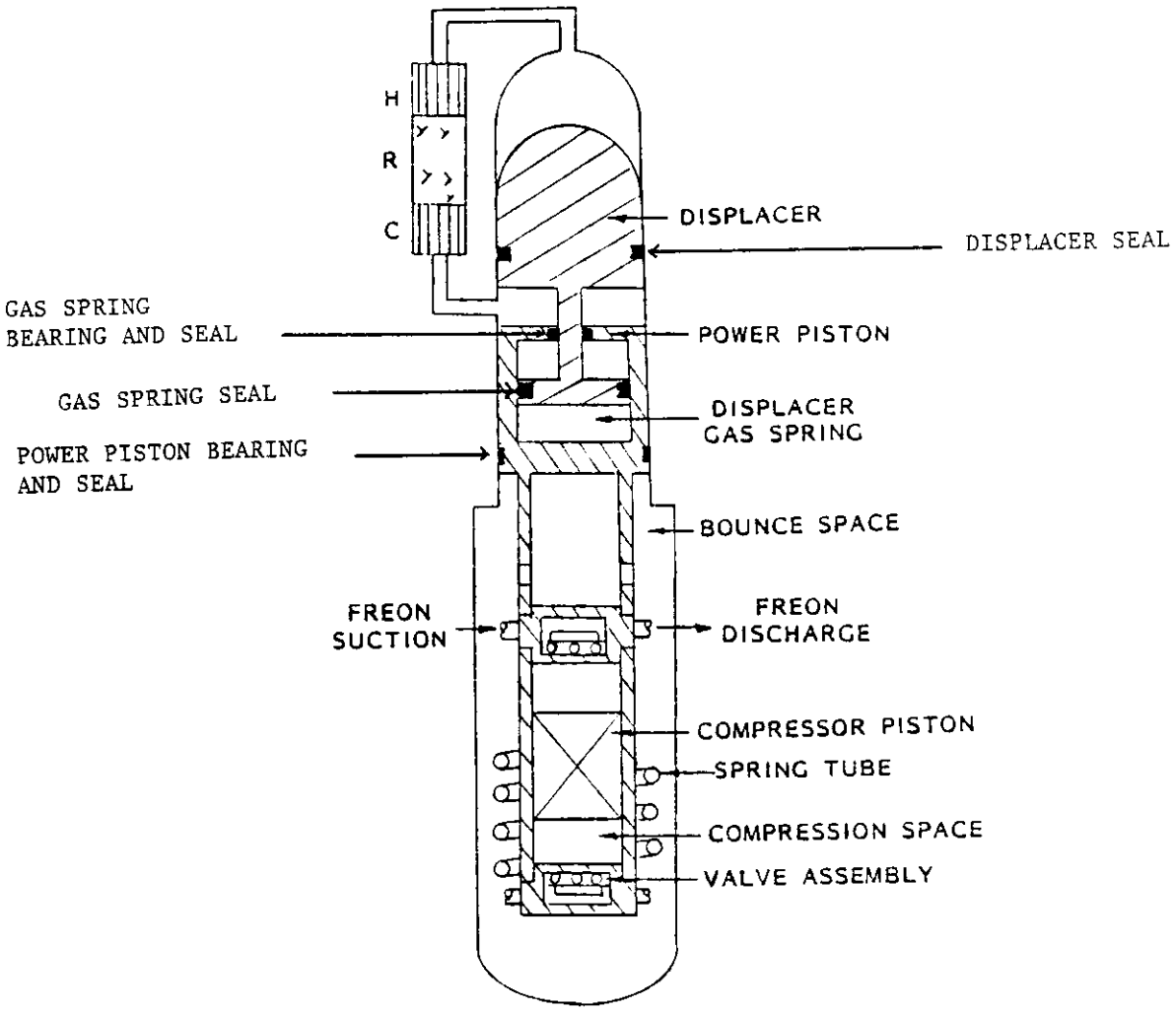


Exhibit 5

FREE PISTON STIRLING ENGINE WITH FREE PISTON LINEAR INERTIA COMPRESSOR

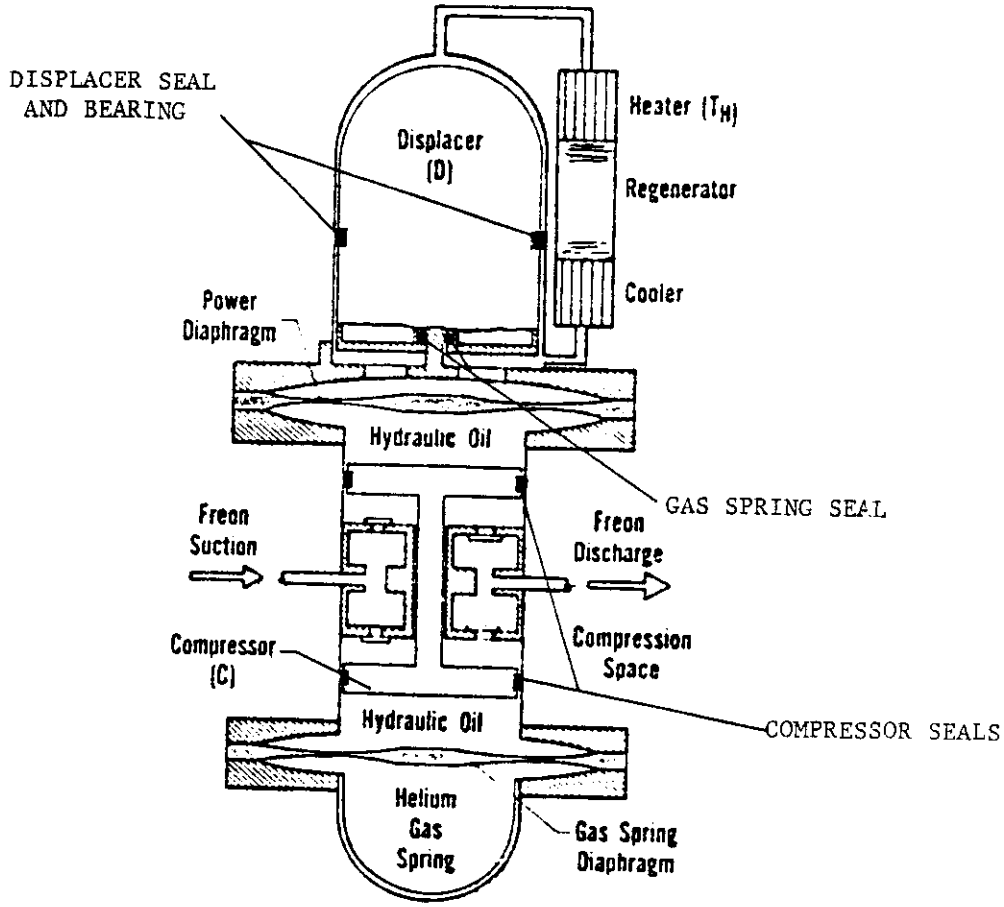


Exhibit 6

SCHEMATIC OF MTI HAHP (EARLY VERSION)

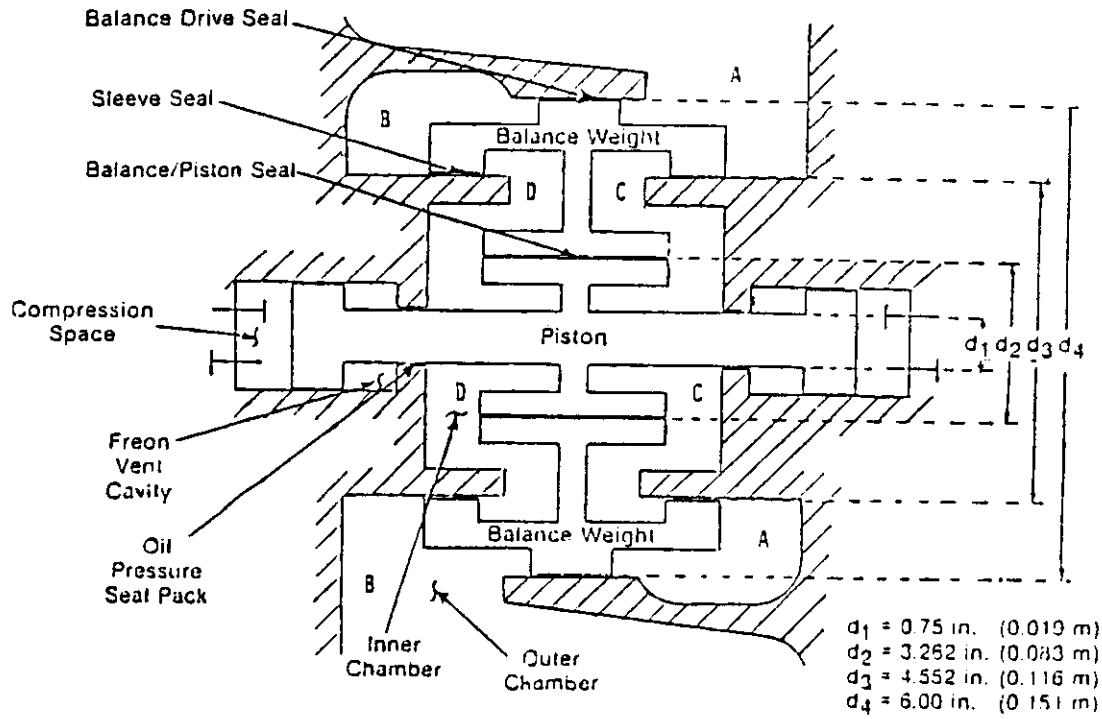


Exhibit 7

COMPRESSOR SCHEMATIC - MTI HAHP

Exhibit 8

OBSERVATIONS - LONG LIFE/HIGH RELIABILITY

- NOT DEMONSTRATED TO DATE AT REQUIRED POWER LEVELS
- STILL HAVE BEARINGS AND SEALS WHICH OPERATE IN DRY HELIUM - A MAJOR FACTOR WHEN USING CONTACTING SURFACES
- GAS BEARINGS AND CLEARANCE SEALS INTRODUCE A NEW SET OF RELIABILITY ISSUES
- ENGINE → HP CYCLE COUPLING MECHANISMS OFTEN COMPLEX
- LIFE OF HOT END COMPONENTS OPERATING IN CYCLIC MODE IS VERY UNCERTAIN

Exhibit 9

OBSERVATIONS - HIGH EFFICIENCY

- NET EFFICIENCY OF MOIST FPSE SYSTEMS TESTED TO DATE HAS BEEN DISAPPOINTING (COMPARED TO INITIAL PROJECTIONS)
  - APPARENTLY DUE, IN PART, TO HIGHER THAN EXPECTED BEARINGS, GAS SPRING, AND SEAL LOSSES
- THE LOSS OF PHASE ANGLE CONTROL FOR AN "UNCONSTRAINED" FPSE MAY COMPLICATE ACHIEVING HIGH EFFICIENCY OPERATION OVER A WIDE RANGE OF OPERATING CONDITIONS
- IT IS NOT CLEAR IF THE ANALYTICAL MODELS ARE SUFFICIENTLY WELL DEVELOPED TO ESTIMATE THE EFFICIENCY POTENTIAL OF FPSE/HEAT PUMPS ADEQUATELY FOR EVALUATION AND DESIGN PURPOSES

Exhibit 10

OBSERVATIONS - LOW COST

- MANY OF THE MORE COSTLY STIRLING ENGINE SUBASSEMBLIES ASSOCIATED WITH A KINEMATIC STIRLING ENGINE ARE RETAINED IN THE FPSE CONFIGURATION (HEATER HEADS, AIR PRE-HEATERS, CONTROLS, REGENERATORS, ETC.)
- SOME SYSTEMS EMPLOY COMPLEX TRANSMISSION/-COMPRESSOR SYSTEMS
- HIGH PRECISION REQUIRED FOR LOW LOSS GAS BEARINGS AND CLEARANCE SEALS

Exhibit 11

FREE PISTON STIRLING ENGINE PERFORMANCE COMPARISONS

	Engine Only	Engine and Heat Source	Load or Transmission	Engine Heat Source and Load	Heat Pump Thermal COP[5]	
	Brake[1] Thermal Efficiency	Brake Gross[2] Thermal Efficiency	Load [3] Efficiency	Gross Thermal[4] Efficiency	95°F Cooling	47°F Heating
MTI (6)						
• Advanced Heat Pump Goal	.40	.33	(C) (H)[7] .70, .60	(C) (H) .22, .20	1.1 (.74)	2.0 (1.3)
• Projection for Current[8] Heat Pump	.32	.26	(C) (H) .70, .60	(C) (H) .18 .16	.85 (.62)	1.8 (1.1)
• Linear Alternator (Test Load)	.32	.26	.72	.19	---	---
G.E.	(C) (H)	(C) (H)		(C) (H)		
• Proto #2 Projection	.30 .25	.23, .18	~1.0	.23, .18	.79 (.60)	1.6 (1.1)
• Proto #1 Accomp.	.19—.25	.14—.19	~1.0	.14—.19	.5—.7(.4—.6)	1.2—1.6(.9—1.1)
Sun Power						
• Linear Alternator		.18—.19	.87	.16—.17	---	---

Notes:

1. Brake Thermal Efficiency (ηbt) $\equiv \frac{\text{Engine Brake Power}}{\text{Power Input to Engine Head + Engine Aid Power at Source Efficiency}}$
2. Gross Brake Thermal Efficiency (ηbgt) $\equiv \frac{\text{Engine Brake Power}}{\text{Fuel Flow *HHV + Engine Aid Power at Source Efficiency}}$
3. Load Efficiency ($\eta \ell$) $\equiv$ Power to Refrigerant/Engine Brake Power
4. Gross Thermal Efficiency (ηgt) $\equiv \frac{\text{Gross Load Power (i.e. Power to Refrigerant)}}{\text{Fuel Flow *HHV + Engine Aid Power at Source Efficiency}} = \eta GBT * \eta_\ell$
5. Thermal COP = Heating (or Cooling) Delivered/Fuel Flow *HHV
   Numbers in Parenthesis Include Parasitics (added to Fuel Flow *HHV at Source Efficiency of .3).
6. In the case of the MTI heat pump the boundary between engine and load is the diaphragm between engine and transmission. Therefore transmission viscous and gas spring loss are accounted for in the transmission or load efficiency rather than the engine efficiency.
7. (C) and (H) refer to Cooling and Heating respectively.
8. Projections for Current MTI Heat Pump Design Based on Component Performance from Test and/or Analysis.

HEAT-ACTUATED HEAT PUMPS—II

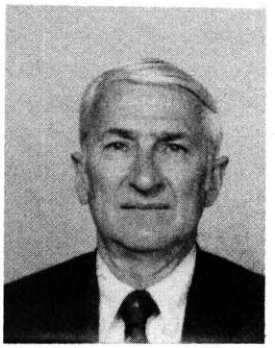
Engine-Driven and Absorption Heat Pump Programs

George T. Privon

Oak Ridge National Laboratory

Oak Ridge, Tennessee 37831

George T. Privon received B.S. and M.S. degrees from the University of Michigan and the University of Kentucky, respectively. He has had various professional responsibilities in the engineering areas of research, design, development, and product manufacturing. His primary area of expertise has pertained to reciprocating compressors and their related products. Since 1951 he has worked successively for Carrier Corporation, the Cardinal Compressor Company, Sundstrand Compressors, and Bristol Compressors. He joined the Laboratory staff in 1981.



ENGINE-DRIVEN AND ABSORPTION HEAT PUMP PROGRAMS*

G. T. Privon
Oak Ridge National Laboratory
Oak Ridge, Tennessee

Objective

The objectives of these programs are to accelerate the development of energy-efficient, thermally activated heat pumps for heating and cooling.

Background

The programs are funded out of the Energy Conversion Equipment Branch, Buildings Equipment Division of the Department of Energy. For each an effort has been made to have an integral program consisting of specific subcontractor projects and supporting areas of research, evaluation, and assessment.

The internal combustion engine-driven heat pump program started in March 1980 with a subcontract to Honeywell, Inc., for technology development, fabrication, and testing of a commercial-size breadboard system utilizing a free-piston linear engine. Included in the activities were the design and test of a hermetic seal. The overall purpose of the project was to demonstrate, by means of tests, proof of concept. The results from the Honeywell program were favorable in that components were developed and performed satisfactorily. Because of constraints of funding and time, steady-state performance results were taken only in the heating mode, although operation over a wide range of heating and cooling conditions was demonstrated. Because of constraints imposed by the hermetic seal, the carburetor engine operated in the 12 to 13% brake horsepower (BHP) efficiency range resulting in a heating coefficient of performance (COP) at 17°F of 1.1 to 1.2 (i.e., peak BHP efficiency of the engine was measured as 18%). At about the same time, June 1980, a subcontract was awarded to A. D. Little for an assessment of internal combustion engines as drivers for heat pumps. The assessment provided a record of prior experiences, addressed the capabilities and difficulties, and outlined some of the development options.

The ORNL/DOE absorption heat pump program was initiated in October 1979 by entering into an agreement with the Gas Research Institute to cofund an ongoing program at Allied Chemical. This was followed by a subcontract to Arkla Industries for a design and test of a heating-only hydronic heat pump system based on their existing absorption air conditioners. Both of these were relatively conventional systems and had a target heating COP at 47°F ambient of 1.25, which they subsequently attained. The advanced absorption-cycle development was conceived in late 1981, with a goal of developing, through proof of concept, high-efficiency absorption-cycle heat pumps. The minimum performance targets for the advance cycle systems with known fluids were set at 1.6 in heating at 47°F ambient and at 0.7 in cooling at 95°F ambient. The possibility of new fluids holds promise for yet higher efficiencies and will be covered by R. C. DeVault in the next session.

Figures 1 and 2 show the steady state COPs of alternative I.C. engine and absorption heat pump systems at the respective conditions of 47°F ambient heating and 95°F ambient cooling. In each program, the plan is to progress to the highest level feasible.

Summary

At this time, one subcontractor (Tectonics Research) is actively continuing the efforts started in the Honeywell, Inc., project. Tectonics has progressed to the point where a breadboard system with a 32.5%-BHP-efficient linear engine is in the final phase of steady state performance testing.

Three subcontractors, Carrier, Phillips, and Trane, are active in the absorption heat pump advanced cycle program. Each has completed analysis that confirms the level of potential indicated in Figs. 1 and 2 and has selected preferred cycles and fluids.

All four projects involve a degree of technical risk. Reliability for the linear engine heat pump and long life of the shaft seal must be established. The advanced absorption systems each require high performance from an internal counterflow heat exchanger and the development of reliable, high-performance fluid pumps.

Program Scope

The status of the individual subcontracted projects that comprise these two programs will be reported in other conference papers. Here an overview is presented to show how the projects fit together and what they involve.

*Research sponsored by the Office of Building Energy Research and Development, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

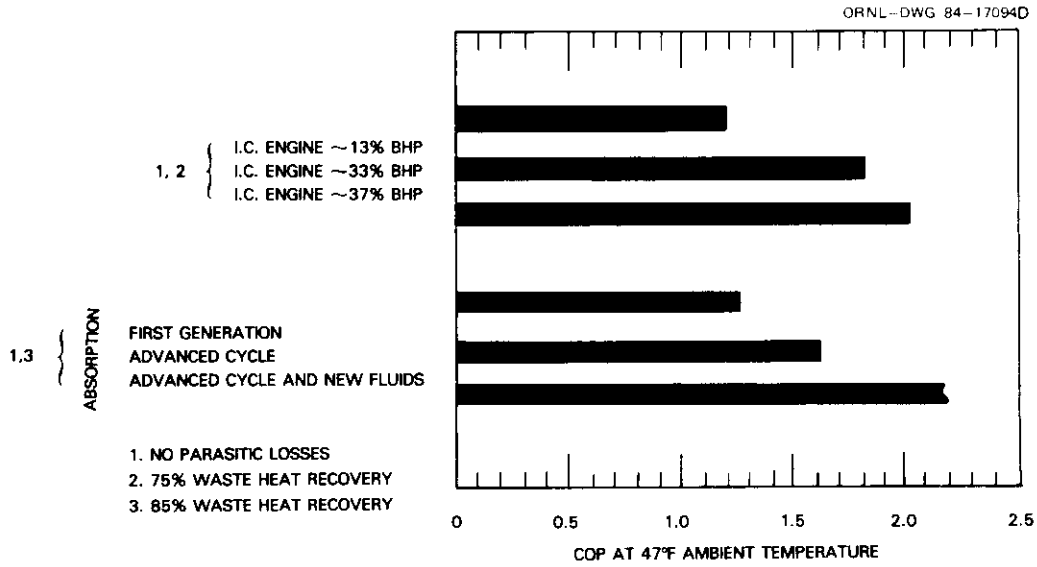


Fig. 1. Target heating performance of alternative heat pumps.

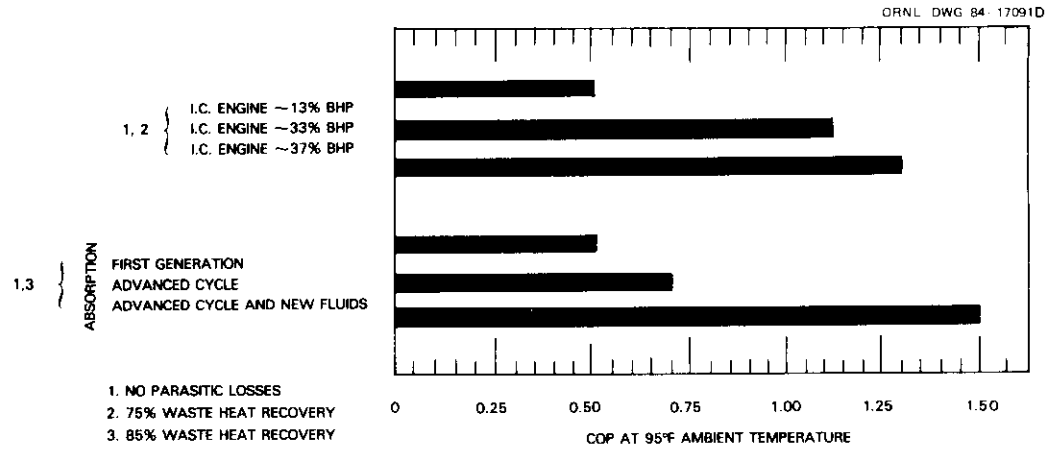


Fig. 2. Target cooling performance of alternative heat pumps.

I.C. Engine-Driven Heat Pump. Figure 3 lists the light commercial development subcontract in progress and shows activities that are intended to evaluate information, search for new opportunities, and perform supporting functions. Figure 4 shows in more detail the planned activities of the linear-engine development program. This project started with a subcontract to Honeywell, Inc., and is now the responsibility of Tectonics Research. The effort at this time deals with improvement of performance of the Braun linear free-piston engine operating in a heat pump system. The propane-fuel-injected engine in the breadboard system has achieved a peak indicated horsepower (IHP) efficiency of 38.5%, measured by an independent laboratory, and is in a high stage of development. The principal technical hurdles remaining are the confirmation of the reliability of the engine fuel injection system and the reliability of the refrigerant side shaft seal under extreme conditions of heat pump application. Both of these devices have been designed and tested at Tectonics Research. A field evaluation test of commercial-sized heat pump units, employing the equipment now under development, is tentatively planned for late 1986 startup.

Absorption Heat Pumps. Figure 5 shows, in general terms, the activities planned. The three subcontracters shown are Carrier and Trane, who are developing advanced cycle systems for commercial-sized equipment, and Phillips, who is developing advanced cycle systems for residential use. As noted previously, the subcontracted efforts are directed to development of high-performance systems through proof of concept testing of breadboard units. Out of each project a preferred unit will evolve for field evaluation. The supporting efforts shown consist of research in the basics of fluid properties, transport properties, chemical stability, and materials compatibility. In the future, it is expected that development and evaluation of new fluids and additives will present new opportunities for high-efficiency absorption systems.

In-house research. This is planned as a supporting function to provide the technology base with which to assess the activities of subcontractors, identify new research opportunities, and evaluate potential gains as well as technical and economic risks.

Future activities. It is our intent to implement the programs described herein, keeping some degree of flexibility to include new opportunities, as permitted by future funding and program philosophy restraints.

ORNL-DWG 84-17096D

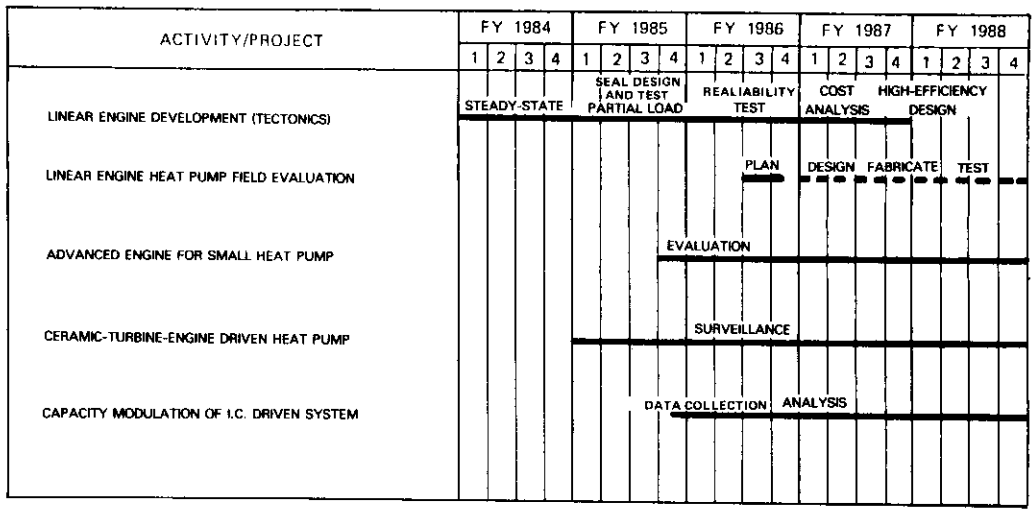


Fig. 3. Internal combustion engine–driven heat pump research.

ORNL-DWG 83-7560

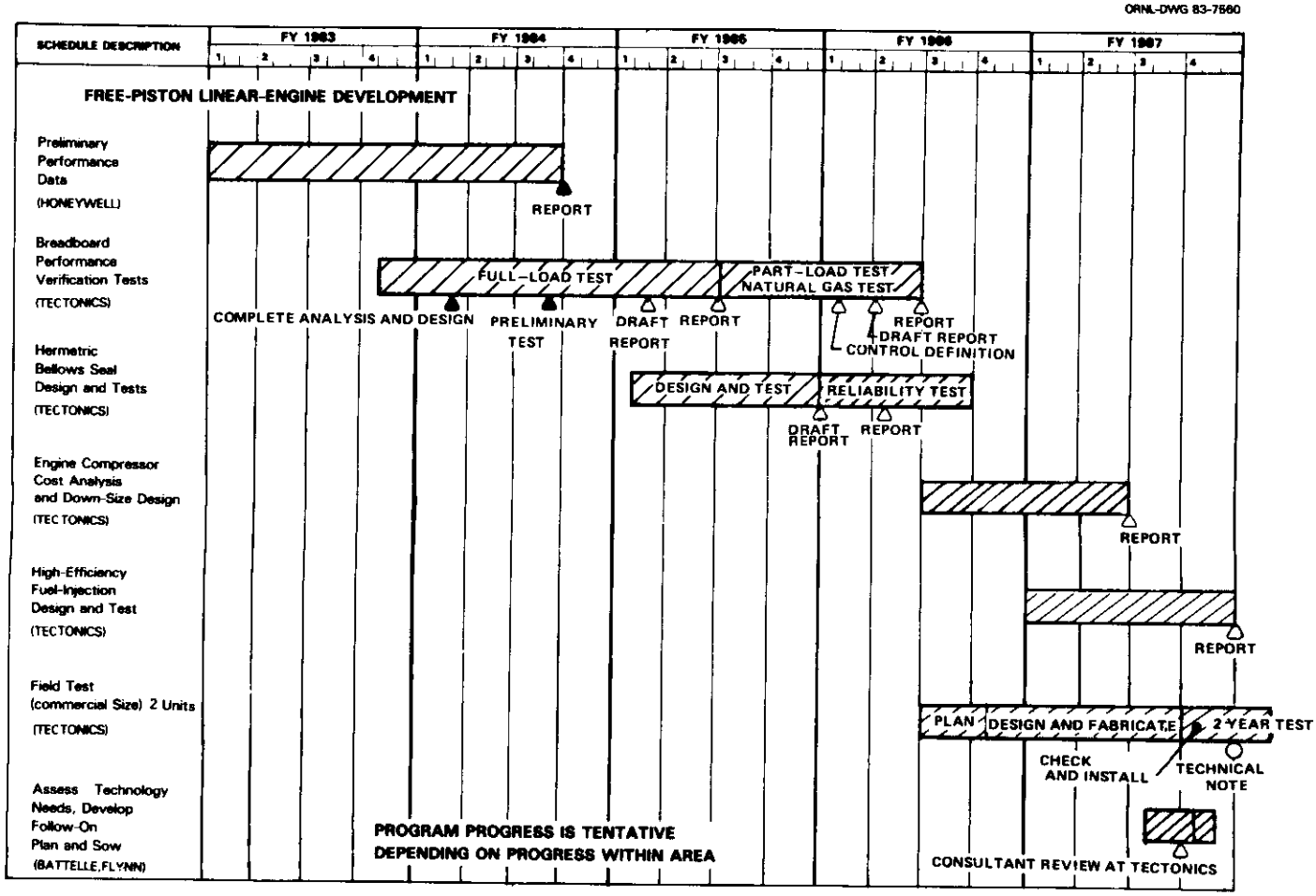


Fig. 4. Linear engine–driven heat pump research.

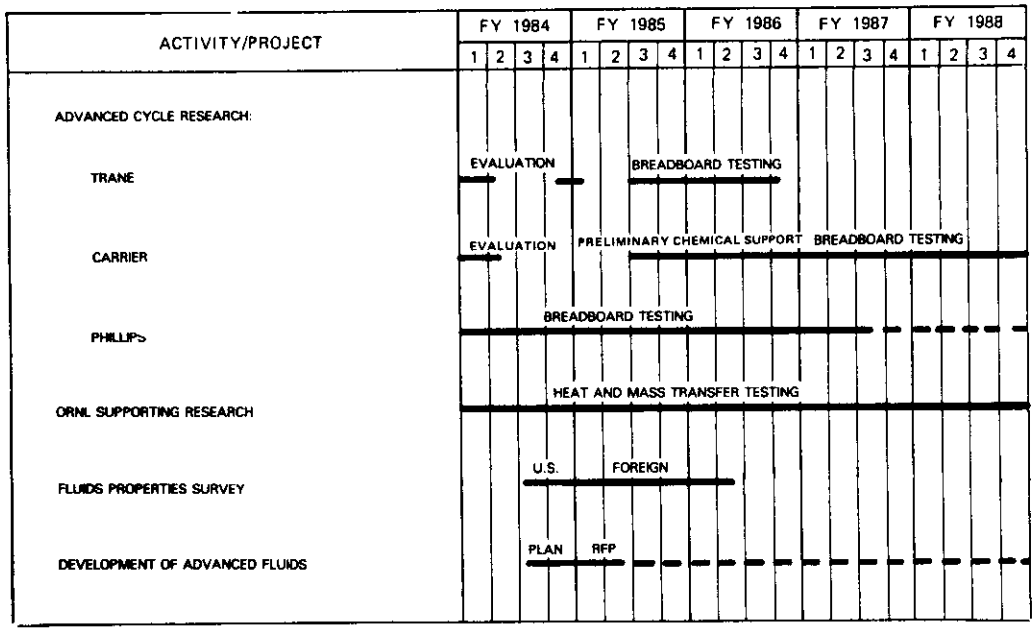


Fig. 5. Absorption heat pump research.

Status of the Braun Free-Piston Engine-Driven Heat Pump

A. Tom Braun

Tectonics Research, Inc.

Minneapolis, Minnesota 55431

A. Tom Braun, founder and president of Tectonics Research, Inc., graduated from Rhine Engineering College, West Germany, with a degree in mechanical engineering. He began his career in the Steam and Gas Turbine Division of the Siemens Works. It was during this period that he developed the interest for what was later to be a life-time dedication to the free-piston engine. After 3 years with Siemens, he moved to Canada (in 1952) where he worked in the aircraft and transportation industry on the development of a revolutionary snow blower.

After one year of postgraduate studies at Carleton University, Ottawa, Canada, he taught for 4 years at Queens University in Kingston, Ontario. While at Queens, Braun expanded his R&D work in free-piston engines, which later resulted in the demonstration of a new type of gasifier. Subsequently, with substantial funding from the Canadian Department of Defense and the Royal Canadian Navy, he designed an automotive-sized free-piston gasifier that reached the production-prototype stage during the early 1960s. Braun also conducted a comprehensive study for the Canadian Defense Research Board of the *10 Most Likely to Succeed New Engine Principles Over the Next 10 Years.*

In 1965 Braun moved to the United States at the invitation of a group of midwestern businessmen interested in his free-piston engine technology. In 1966 he formed Tectonics Research, Inc., for the performance of R&D work required in the development of production prototypes for his licensees. By the mid-1970s, diverse R&D efforts had led to the production and marketing of a line of fully balanced compressors by U.S. and European licensees (with subsequent sales in over 25 countries worldwide) and to the initiation of a test series for a highly developed free-piston engine-compressor production prototype. It was during this time that the first heat pump version based on these engines was built and tested. This effort, substantially expanded in 1980 through funding by the U.S. Department of Energy and Oak Ridge National Laboratory, resulted in the development of the present Braun free-piston engine-driven heat pump. The efficiencies and coefficients of performance obtained with these machines far exceed any of those reported in the United States or the world. Potential savings with these machines could be in billions of energy dollars annually.

Tom Braun holds close to 50 U.S. patents that protect the unique features of the Braun technology. He has authored and coauthored many publications relating to his work, which has been recognized by leading authorities in the engine, compressor, heating, and air-conditioning industries the world over.

STATUS OF THE BRAUN FREE-PISTON ENGINE-DRIVEN HEAT PUMP

A.T. Braun
Tectonics Research Inc.
Minneapolis, Minnesota

INTRODUCTION

The Braun Heat Pump has been under development for the past 3 1/2 years under three related DOE/ORNL contracts. The most recent test results have produced heating COP values in the 1.9 range. These results are well in excess of the minimum program goal of 1.4 and are very impressive when compared to heat actuated heat pumps under development elsewhere (Figure 1). The results are particularly attractive when the simplicity and low manufacturing cost of the BR-105R unit is taken into consideration.

Figure 2 shows the schedule of the work completed and the work yet to be performed in the development of the Braun engine driven heat pump.

Because of the critical nature of the seal between the engine and the refrigerant compressor used in the Heat Pump, the work was started in July 1980, with a contract (Seal I) for the adaptation of a proprietary seal to this particular application.

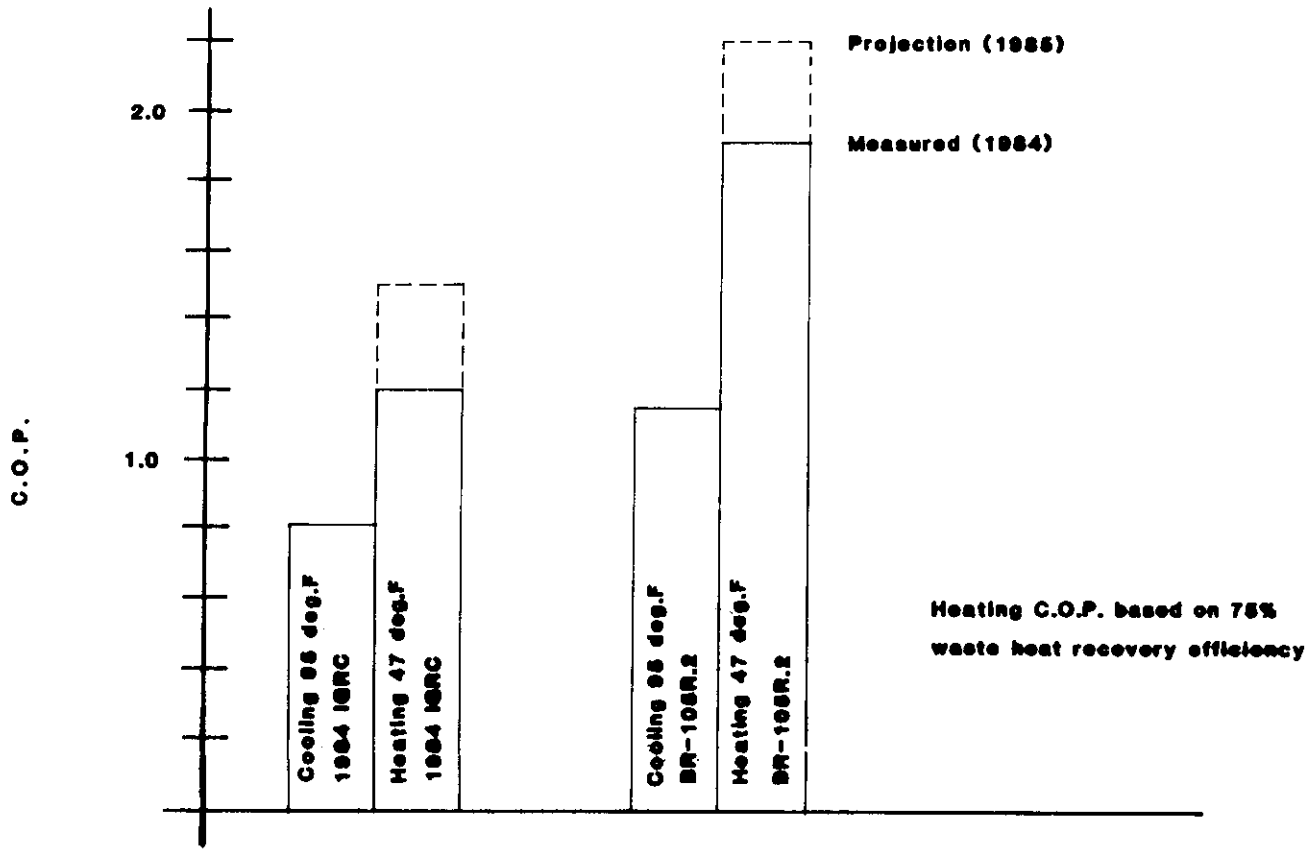


FIGURE 1 - COMPARITIVE C.O.P.'S FOR EXPERIMENTAL ENGINE DRIVEN GAS FIRED AIR-TO-AIR HEAT PUMPS

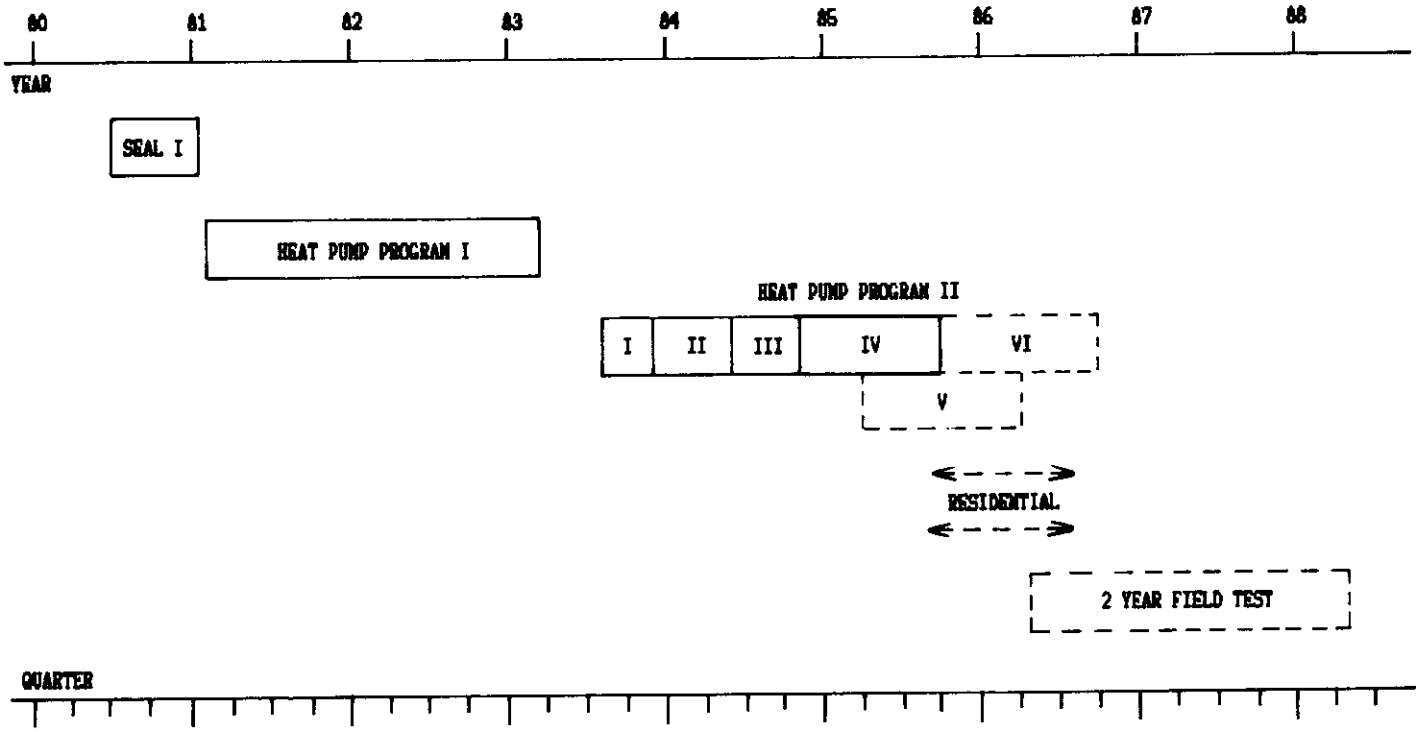


FIGURE 2 - BRAUN ENGINE DRIVEN HEAT PUMP, PROGRAM SCHEDULE

After obtaining proof-of-concept status under this Seal I contract, a second main R & D program for the development of a Braun Linear Engine-Driven Heat Actuated Heat Pump began in January 1981. This contract was completed in March 1983 on attainment of the proposed goals.

A third contract was started in August 1983 for the performance verification of an updated Braun Linear Engine-Driven Heat Pump. This contract is scheduled to continue through a number of phases until at least 1986.

A parallel effort has recently been started with initial planning work for a field test program under which several of the Heat Pumps will be placed in two or three typical commercial applications and tested for at least a two year period.

Hermetic Seal Program I,
Contract No. 86X61613C

The title of this program was "Hermetic Bellows Seal, Design and Application Phase I - Test and Demonstration of a Hermetic Compressor Seal." It began in July 1980 and continued through January 1981. The objective was to perform analysis, design, and tests of an early example of a hermetic seal for application in heat actuated Heat Pumps.

Program Description

In the course of this program, analytical techniques for the precise evaluation of the dynamic performance of complex linear metal bellows seals were verified. A specific seal solution was identified, analyzed and demonstrated to meet the general requirements of the hermetic refrigeration seal.

To remain within authorized funding levels, available off-the-shelf components were used.

The actual seals, defined by engineering specifications and available standard shelf stock, were procured from two seal manufacturers and were tested under various conditions and configurations. The test rigs adapted for the seal evaluation enabled investigation of the effects of both sinusoidal motion and actual engine dynamics of the heat pump drive on the seal. (figure 3) The sinusoidal drive permitted precise variation of speed and displacement. Correlation between these test results and the theoretical predictions validated the accuracy and suitability of the analytical methods.

The key to making a bellows seal practical is the ability to control the seal dynamics and thereby the maximum stresses in the bellows over a wide range of strokes and cyclic

speeds. The particular solution adopted resulted in a drastic reduction of seal stress levels over wide ranges of cyclic speeds and strokes. The effect is shown in figures 4 and 5 which compare respective maximum stress levels and stroke and speed range of the Braun seal to those of conventional bellows seals.

Using the most suitable of available seal components, a final design well within "infinite life" design criteria was worked out. This design uses a standard BR-105 linear engine model which had been selected as the initial drive unit in the projected heat pump program. The air compressor version of the BR-105 has been laboratory and field tested for well over 50,000 hours.

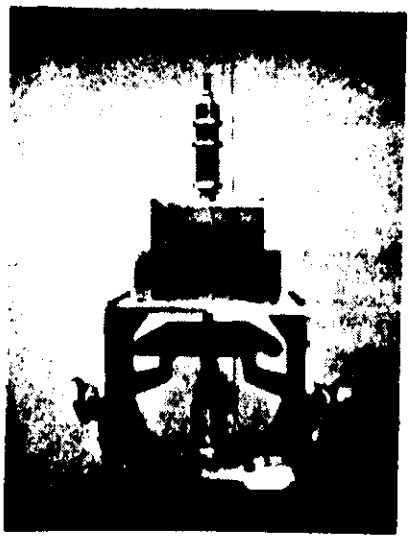


FIGURE 3 - SEAL TEST RIG

Heat Pump Program I,
Contract No. 86X-61619C

A program entitled - "Development of a Braun Linear Engine-Driven Heat Actuated Heat Pump", began in February, 1981 and continued through March, 1983. The objectives of the Program were to:

1) Obtain performance goals of 1.2 heating and 0.6 cooling COP's.

2) Develop and test basic components of a gas engine driven Heat Pump system for application in the light commercial sector.

3) Implement a breadboard for the system.

4) Measure steady state performance of the breadboard in the heating and cooling range.

Results from this program were as follows:

- o Hermetic Seal was successfully demonstrated in the BR-105R.1 Heat Pump.

- o Heat Pump performance with the carbureted engine at 20-25 deg.F ambient was measured at 1.25 heating COP (equivalent to approx. 1.35 at 47 deg. ambient).

- o An independent laboratory test confirmed previous engine brake efficiency measurements of 34% on the fuel injected version of the standard BR-105 engine.

Program Description

The BR-105R.1 breadboard Heat Pump was developed using parts of a standard Braun BR-105 free piston engine/air compressor (figure 6). Of two existing versions of the BR-105 engine, the first is carbureted and operates with an 18% brake efficiency, the second is fuel injected and operates with a 34% brake efficiency. Because of funding limitations and less costly adaptation, the carbureted engine was used on the BR-105.1 Heat Pump.

The effect of the engine brake efficiency on the Heat Pump COP can easily be demonstrated from fundamentals[1].

Figure 7 is a plot of this relationship indicating that if the fuel injected BR-105 is used in place of the carbureted engine to drive the heat pump, the COP should increase from 1.25 to about 1.85.

Since overall performance predictions of this kind are only as sound as the individual component efficiencies involved, the engine and compressor efficiencies were independently established.

Because of the substantial improvement of the overall COP expected from the fuel injected BR-105 engine and to obtain a credible base for future work, this engine was tested by a prominent U.S. laboratory. The engine brake efficiency obtained during these tests confirmed our own and earlier measurements by another independent laboratory of 34%.

The BR-105 refrigerant compressor underwent a comparative series of tests against a similar sized unit of a major US manufacturer. The results are shown in figure 8. The Braun compressor is shown in figure 8a.

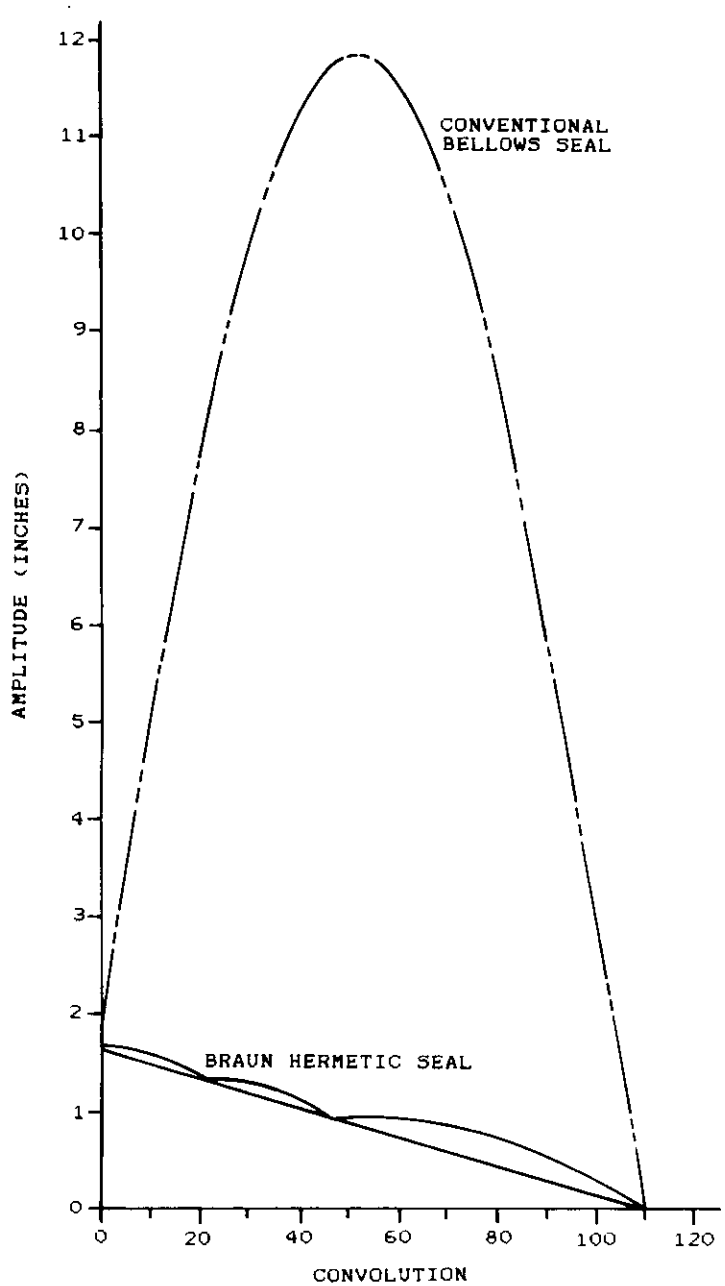


FIGURE 4 - BELLOWS SEAL COMPARISON

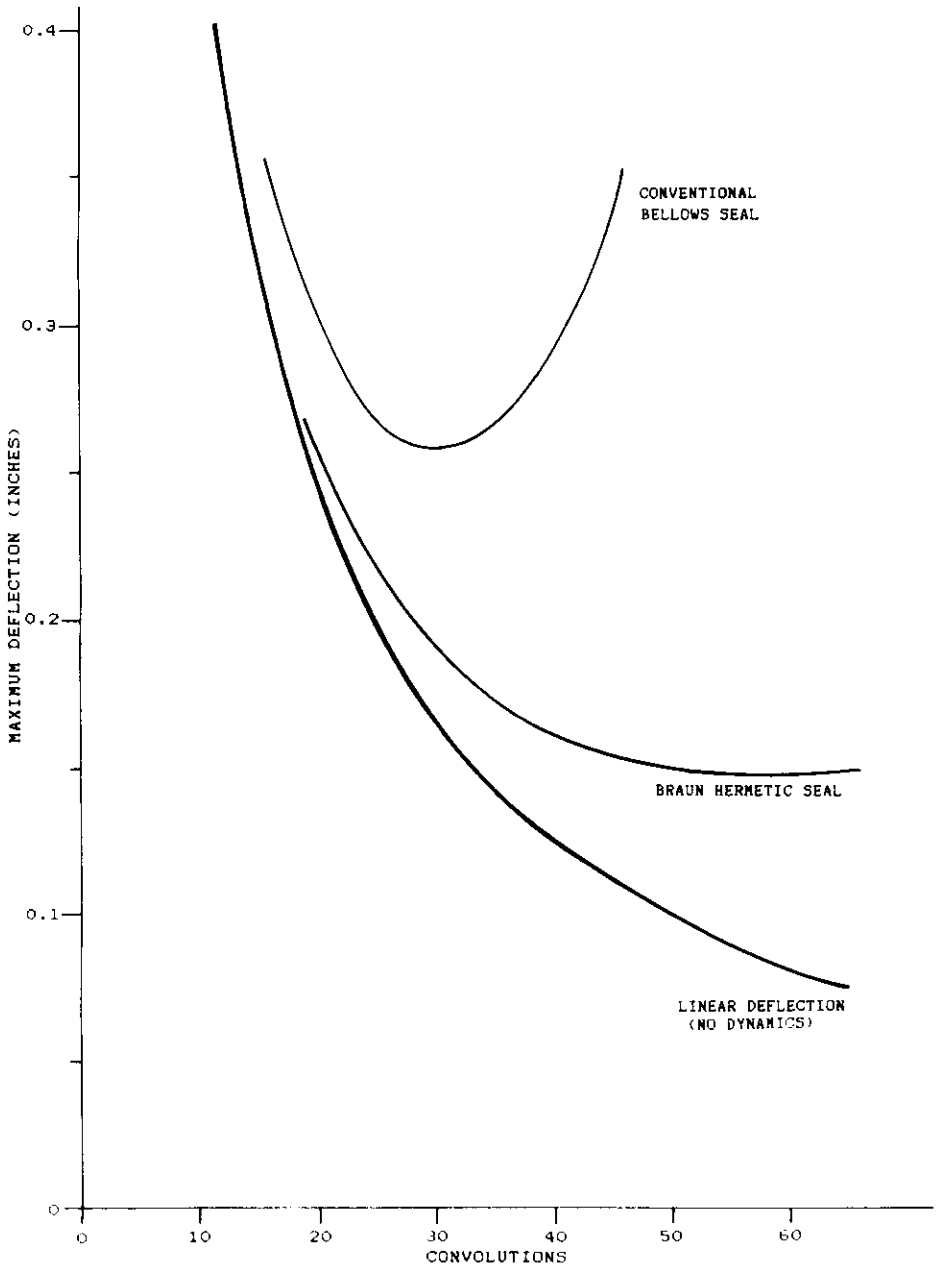


FIGURE 5 - SEAL DYNAMIC PERFORMANCE COMPARISON

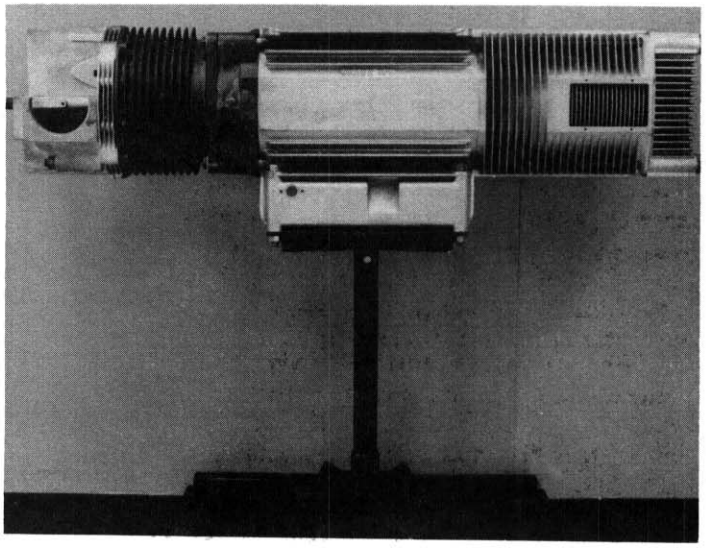


Figure 6 - BR-105 Engine/Air Compressor

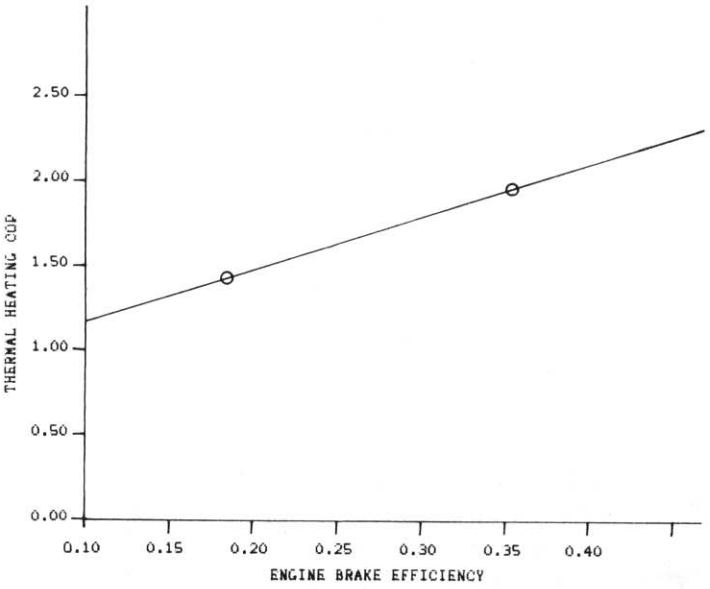


FIGURE 7 - THERMAL HEATING COP VS ENGINE BRAKE EFFICIENCY

A liquid-to-liquid system was selected to provide a refrigerant compressor test bed for the Braun heat pump demonstration. Figure 9 is a simplified schematic, and figure 10 a photograph, of this system in the Tectonics test lab. It is basically a quasi-adiabatic test condenser and evaporator and heat transfer tank. The heat source and heat sink are provided by the condenser and evaporator respectively. The thermal energy transfer is provided by the heat transfer fluid and occurs in the mixing header and the heat transfer tank. The excess heat (work of compression) is purged to the atmosphere.

Testing of the BR-105R.1 Heat Pump yielded a COP of 1.25 at a 20 -25 deg.F ambient temperature. Figure 11 shows the variation of the Heat Pump COP with ambient temperature. The reason for presenting the COP for the 20 - 25 deg.F ambient temperature was that above this

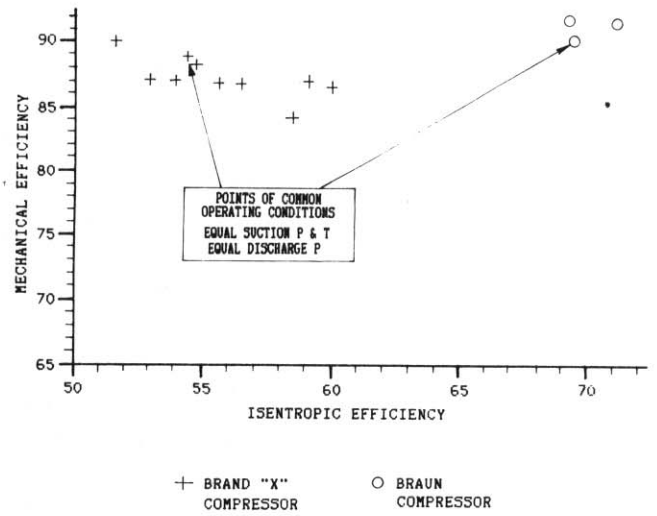
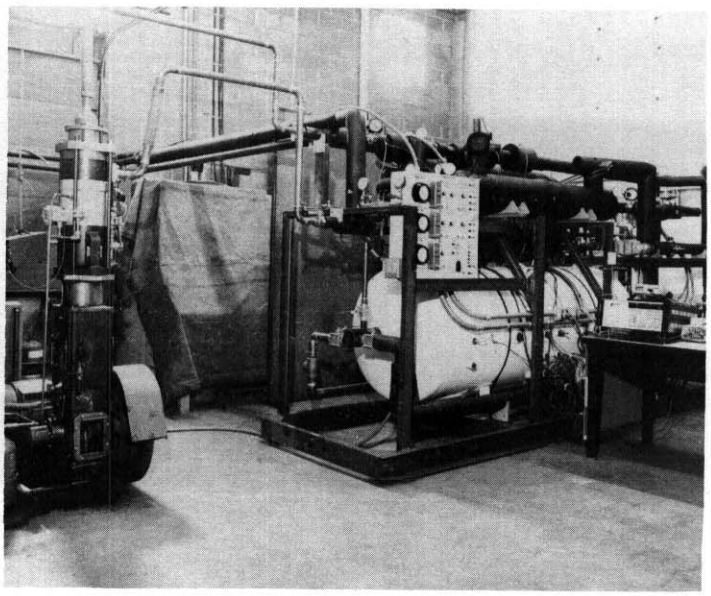


FIGURE 8 - COMPARISON OF MEASURED BRAND "X" AND BRAUN COMPRESSOR EFFICIENCIES

point conditions became unstable as a consequence of the low refrigerant flow rate of the BR-105R.1 with its low efficiency carbureted engine section. The test rig and the flow meters had been sized to handle the substantially higher flow rates of the future BR-105R.2 which was to have the high efficiency fuel injected engine drive.

In the absence of reliable test data due to the low flow rates of the BR-105R.1 at higher ambient temperatures, a reasonable extrapolation to the standard 47 deg.F ambient temperature is shown in Fig. 11. The COP of approximately 1.35 at that temperature was a remarkable achievement with the carbureted BR-105R.1.



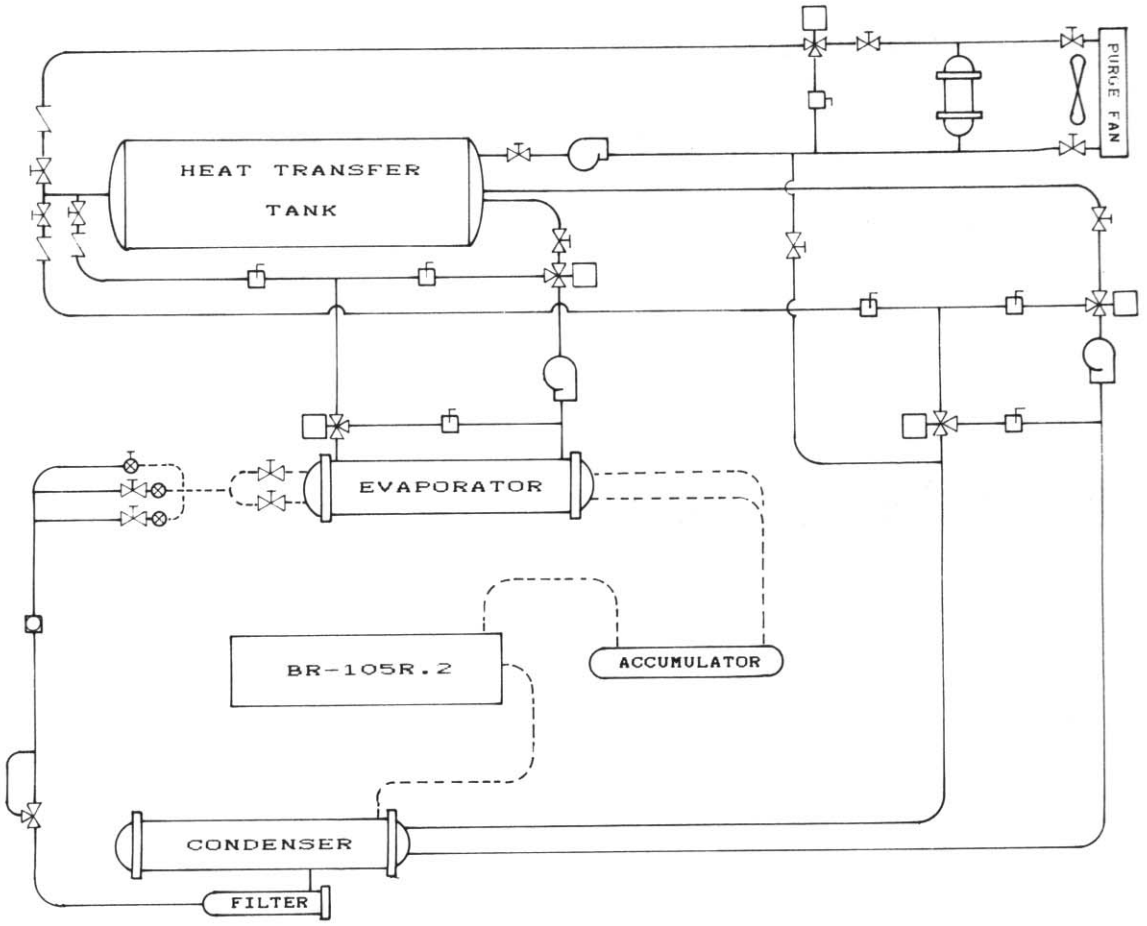


FIGURE 9 - SCHEMATIC OF TEST RIG

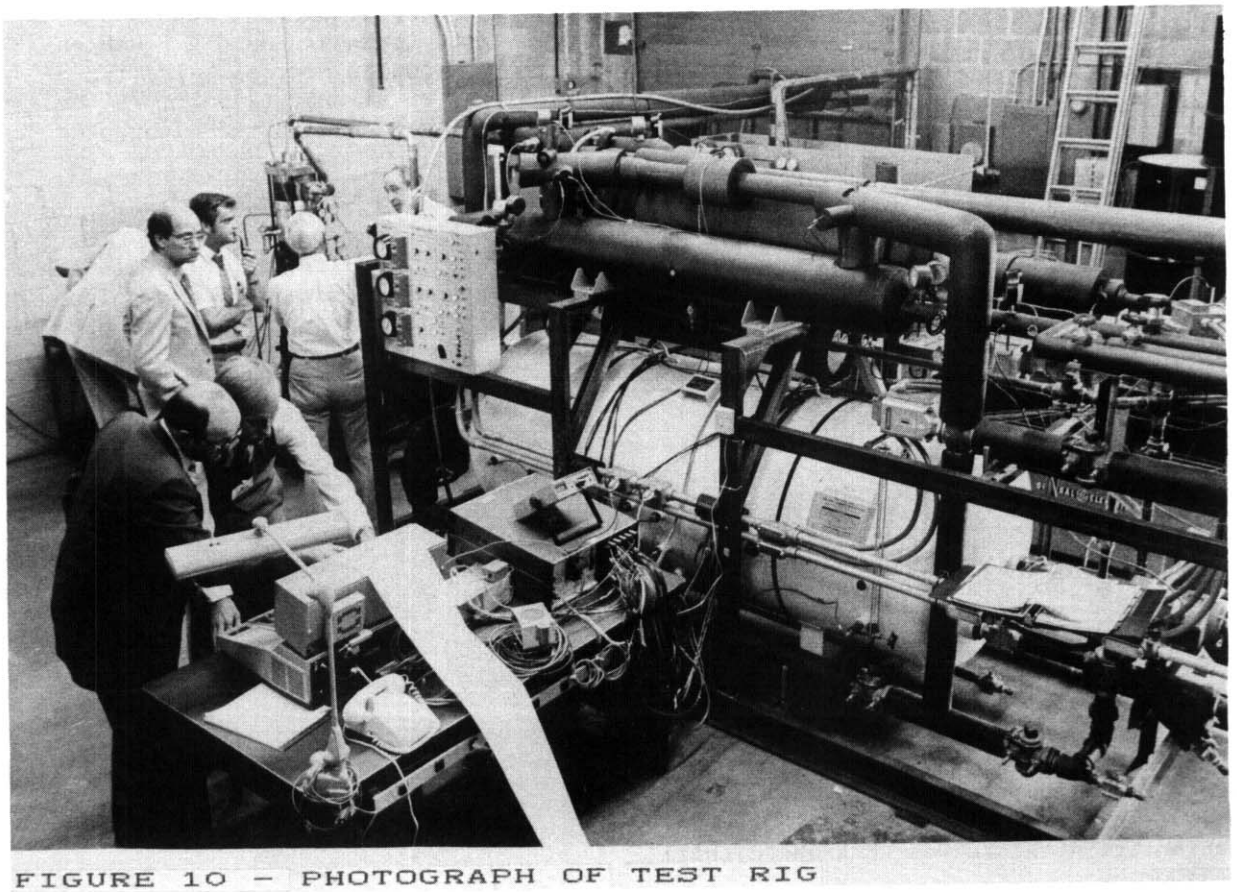


FIGURE 10 - PHOTOGRAPH OF TEST RIG

In spite of their lower accuracy, a substantial number of points were run at much higher ambient temperatures as shown in figure 12. This was to verify the analytical predictions of the engine/compressor unit. On completion of formal testing and after some minor modifications to the engine/compressor the operating range was even further extended. The corresponding data points are also displayed in figure 12.

From these encouraging results, the next step was to design, build and test an up-dated BR-105R.2 using the fuel injected version of the BR-105 engine with it's significantly higher brake efficiency of 34%. This unit would not only yield a much higher COP as indicated in figure 7, but also have a substantially higher refrigerant flow rate and operate effectively throughout the entire heating and cooling range.

Heat Pump, Program II, Contract No. 86X47972V

A program entitled- "Braun Linear Engine-Driven Heat Pump Performance Verification" began in August 1983 and is still in progress. At the time of this report, the first two phases of the contract have been completed. Progress is well into the third phase which consists of obtaining a first range of data points. Future phases of this contract include a Seal II program, performance mappings and the subsequent installation and testing of several field test units.

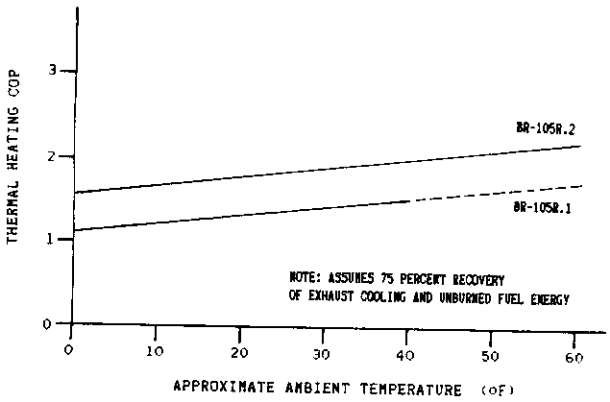


FIGURE 11 - BR105R.1 AND BR-105R.2 HEAT PUMP COP VS AMBIENT TEMPERATURE

Major objectives of program II:

1) Demonstrate operation from -20 to 115 deg. F ambient temp.

2) Achieve steady state performance figures of at least 0.7 cooling COP of 95 deg. F and at least 1.4 heating COP at 47 deg. F ambient.

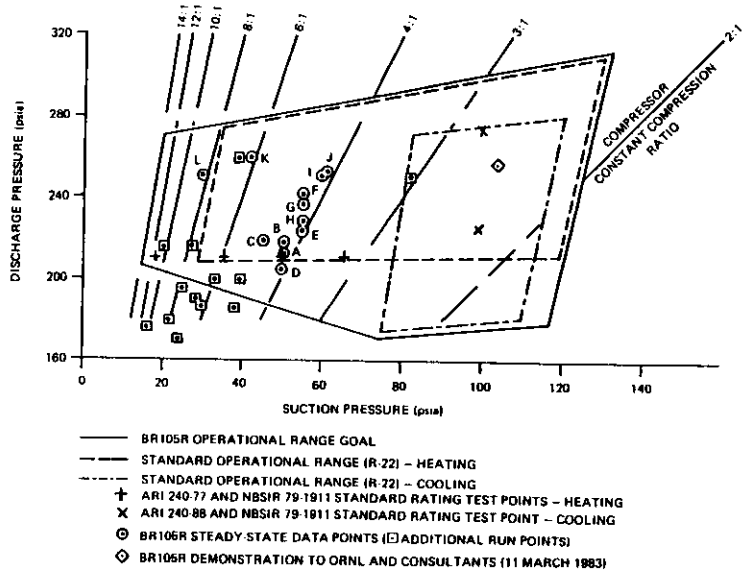


FIGURE 12 - BR-105R.1 OPERATING CONDITIONS

3) Develop, fabricate and test a commercially available hermetic seal assembly specifically sized for the BR-105R.2 unit.

4) Qualify the engine for efficient operation on natural gas.

5) Obtain part load performance data (mapping)

6) Subsequent field test program.

As of October 10, 1984 the majority of the data taken has been at the base reference point of 47 deg.F ambient temperature. Performance at this condition so far has produced heating COP's in the 1.9 -2.0 range with the engine/compressor unit running well within safe operating limits. Continued testing is in progress now to establish the ARI operating points of 17, 47, 80 and 95 deg. F as well as extremes of -20 deg.F for heating and 115 deg.F for cooling. Final results of these tests should be released shortly and results are expected to be again in excess of the program goals for the fuel injected BR-105R.2 unit.

Program Description

Heat Pump Program II is a continuation of the first heat pump program. The major difference is the high efficiency fuel injected engine will be used to drive the refrigerant compressor instead of the 18% efficient carbureted engine.

The program is divided into the following six phases:

I Analysis, Performance Prediction and Design.

II Selection, Procurement, Fabrication & Preliminary Testing

III Performance Testing

IV Hermetic Bellows Seal Design and Fabrication

V Natural Gas Qualification and Part Load Testing

VI Hermetic Bellows Seal Endurance Testing

Future plans include a two year field test program.

Phase I, entailed an analysis that was carried out to design and establish the best match between the engine and the compressor. A key consideration in the design of the compressor was to provide efficient operation throughout the entire heating and cooling range of -20 to 115 deg. F ambient temperature.

Phase II included re-designing of the test equipment and data acquisition system. Under the first heat pump contract, the data acquisition system employed was limited in its capability and slow at analyzing and presenting test results. The newly designed system is a computerized data acquisition and development system designed specifically for rapid data reduction and analysis of performance while testing is in progress (figure 13).

Phase III is the performance testing phase and should be concluded at the time of this paper. In this phase base point tests were carried out to establish a reference data base for comparison at later dates. The remaining tests are for demonstration of range and performance rating. The range tests are to demonstrate operation from -20 to 115 deg.F ambient. The performance rating is for establishing performance at four ARI conditions of 17, 47, 80 and 95 deg. F. These and a sufficient number of other points, will supply data for construction of performance maps.

Phases IV, V and VI are yet to be carried out. Phase IV calls for the design of a specifically sized proprietary hermetic seal utilizing techniques from the Seal I program. Testing will include: (1) bench testing the seal assemblies under conditions that the seal is expected to experience under various normal operating modes and certain extreme conditions, (2) testing as part of the engine/compressor assembly in the heat pump test loop, and (3) accelerated endurance testing. Endurance testing under Phase VI is to begin at the conclusion of phase IV and will overlap with parts of phase V. Phase V will include qualification of the heat pump for operation on natural gas which will involve the design, fabrication and testing of an appropriate natural gas fuel injection system. Testing will be carried out in a manner similar to phase III to determine full load characteristics and performance, followed by complete part load testing as well as mapping. The phase will be concluded with a repeat of the independent laboratory's testing of the natural gas injected engine for comparison to the previous independent tests performed on the propane injected engine.

Summary

The program for the development of the Braun Free-Piston Engine-Driven Heat Pump, initiated in July 1980, has produced through all it's different phases sound results. In all stages program goals have been met or substantially exceeded, as the units two basic blocks consist of a highly developed engine and compressor. No exotic materials, material systems, or methods are required. The only new components used, have been successfully commercialized by licensees worldwide in over twenty-five countries since 1974 (Braun Balancer) or used by suppliers in parallel applications for many decades (Braun Seal).

The orderly program and attainment of COP's well above any heretofore reported is partially attributable to a well understood and tested technology. As in the cases of licensees for related products, close cooperation with an appropriate manufacturer(s) should result in the early availability of a sound and exceptionally simple product. Introduction of the Braun Heat Pump into the US market could result in annual savings of billions of energy dollars.

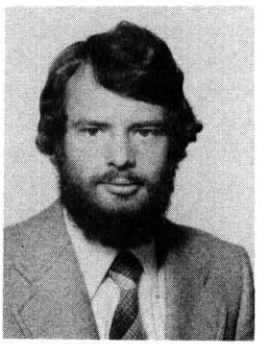
Laboratory Experiments on Absorption Heat Pumps

Reinhard Radermacher

University of Maryland

College Park, Maryland 20742

Reinhard Radermacher received the B.S. degree in physics from the Technical University in Munich in 1981. Following graduation he joined the National Bureau of Standards in Washington, D.C. Two years later he joined the Mechanical Engineering Department at the University of Maryland. His major research work has been concentrated on the experimental and theoretical analysis of advanced energy conversion cycles and on the measurement of thermodynamic and transport properties of fluid mixtures to be used in such cycles.



LABORATORY EXPERIMENTS ON ABSORPTION HEAT PUMPS

Reinhard Radermacher
University of Maryland
Mechanical Engineering Department
College Park, MD 20742

Abstract

A series of laboratory performance tests were conducted on an absorption cycle water chiller and heat pump. The part-load performance was compared to the full-load capacity and coefficient of performance. The causes of performance degradation with shorter operating times are speculated upon with partial substantiation resulting from tests on the chiller after it had been modified to prevent off cycle fluid migration.

Introduction

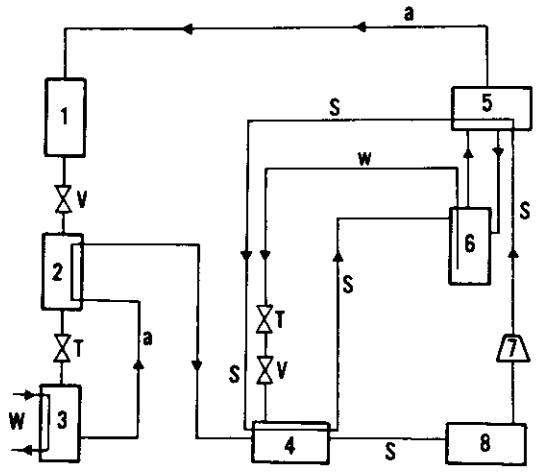
Absorption cycle equipment is, like other heating and cooling equipment, generally sized to meet the building load under design (i.e., maximum load) conditions. If both heating and cooling functions are performed by the same unit the sizing is usually based on the cooling load in order to maintain humidity control. The full-load steady-state performance values are essential for sizing criteria; however, the overwhelming operational time is under part load conditions, thus the best estimates of seasonal performance (or operating costs) require this information as well. Thermostat controls in use today require the equipment to meet part-loads by full-on full-off cycling. Comfort requirements usually require the on-period to be shorter than needed for the equipment to reach steady-state a significant part of the time; in fact, often steady-state operation is not reached at all in a given cycle. Thus, part-load performance should be less than full-load performance. This investigation quantitatively evaluated this phenomenon in a laboratory and attempted to explain their performance degrading causes. Because no reversible machine was available, separate heating and cooling machines of similar design were evaluated.

Description of the Equipment and Facilities

An air-cooled ammonia-water absorption chiller was used in the experimental investigation as the cooling equipment. This chiller was rated by the manufacturer as capable of cooling 0.45 l/s (7.2 gpm) of water from 12.8°C (55°F) to 7.3°C (45°F) with the condenser and absorber in a 35°C (95°F) dry-bulb environment, this corresponds to a nominal capacity of 10,550 W (36,000 Btu/hr). The unit was fired by natural gas and required an energy input of 23,150 W (79,000 Btu/hr) to the combustor. It was designed for residential service, with all the absorption system components in one modular outdoor package. Also included was a pump for circulating chilled water through an indoor coil. The electrical energy required to run the blower, the chilled water and the solution pumps as well as the controls amounted to 1050 W.

An air-to-water absorption heat pump with the working fluid pair ammonia/water was used as the heating machine. The heating capacity was 15,500 W under design conditions which were 8.3°C (47°F) outdoor air temperature, a water flowrate of 0.375 l/s (6 gpm) with a temperature rise from 40.5°C (105°F) to 51°C (125°F). The heat input to the combustor was 13,500 W (46,000 Btu/hr). The fuel required was natural gas.

Figures 1 and 2 show the process schemes of the absorption chiller and of the



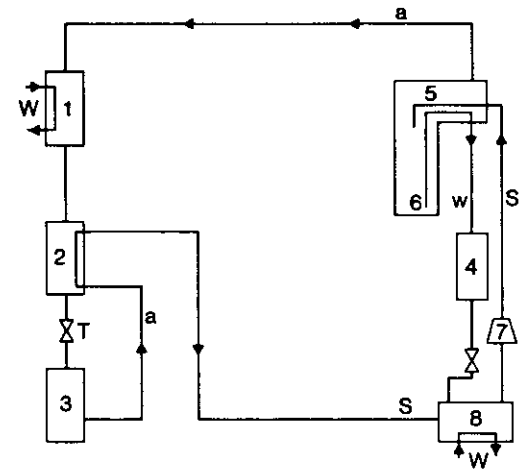
1 CONDENSER
2 REFRIGERANT HEAT EXCHANGER
3 EVAPORATOR
4 SOLUTION COOLED ABSORBER

5 RECTIFIER
6 GENERATOR
7 SOLUTION PUMP
8 AIR COOLED ABSORBER

S STRONG SOLUTION
w WEAK SOLUTION
a AMMONIA VAPOR
W CHILLED WATER

V VALVES BUILT IN TO SEPARATE HIGH AND LOW PRESSURE REGIONS
T RESTRICTORS

Fig. 1 Flow sheet of the absorption water chiller



1 - Condenser
2 - Refrigerant heat exchanger
3 - Evaporator
4 - Flue-gas heat exchanger

5 - Rectifier
6 - Generator
7 - Solution pump
8 - Absorber

S - Strong solution
w - Weak solution
a - Ammonia vapor
W - Chilled water

V - Valves built in to separate high and low pressure regions
T - Restrictors

Fig. 2 Flow sheet of the absorption heat pump

absorption heat pump. For both absorption machines the principal of operation is the same and all basic components have the same function. Therefore the discussion of the behavior of both units under part-load operation can be presented in a very similar manner with similar cause and effect relationships.

Both absorption machines were installed in an environmental chamber at the National Bureau of Standards in which operating conditions (i.e., the indoor coil's water temperature and the outdoor unit's dry-bulb and wet-bulb air temperatures) could be controlled. The air was circulated within the chamber to assure a uniform temperature distribution. During cyclic tests the air temperature was maintained within $\pm 1°C$ ($\pm 1.8°F$) of the set point, while during steady-state tests the accuracy of the air temperature was less than one third of this value. The test facility was designed to allow only one pass of the water through the unit's heat exchanger with the inlet being maintained at the required temperature within $\pm 0.5°C$ (0.9°F). Both units were instrumented such that it was possible to determine temperatures of inlets and outlets of all components, the high- and low-side pressures, capacity, the heat input to the combustor and the electrical energy consumed. The details of the instrumentation and the testing procedures which were the same as recommended in ASHRAE Standard 37-78, as far as applicable, are reported in references 1 and 2.

Results and Discussions

Figure 3 shows the steady-state capacity of the absorption water chiller. The capacity is decreasing with increasing outdoor temperature. This behavior is especially obvious for temperatures higher than 30°C (86°F). Since the heat input to the unit is almost independent of the outdoor temperature, the coefficient of performance (COP) shows the same behavior. The cause for this decrease in capacity is the fact that with increasing outdoor temperature the heat transfer from the condenser and air cooled absorber to the environment decreases significantly.

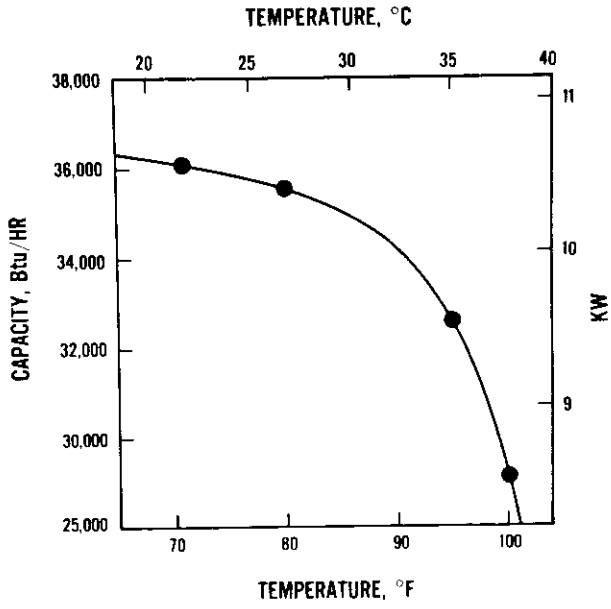


Fig. 3 The steady state capacity of the water chiller as a function of outdoor temperature

The steady-state performance of the heat pump was tested over a wide range of ambient and inlet water tempeatures. The coefficient of performance and the capacity are dependent on the outdoor temperature as shown in Figure 4. The performance approaches a constant value at both high and low outdoor temperatures. One

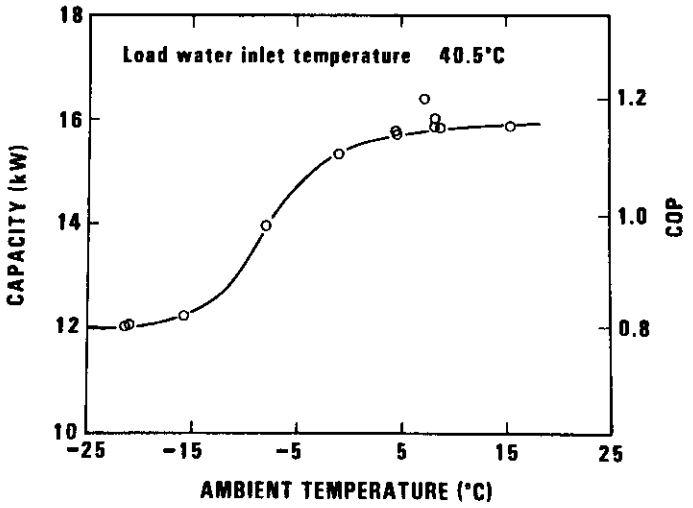


Fig. 4 The steady state performance of the absorption heat pump related to outdoor temperature

reason for the decrease of the performance at low temperatures is the fact that the limited heat transfer from the air permits a two-phase mixture to leave the evaporator, indicating that the size of the evaporator would have to be enlarged to improve performance at these conditions. Above 5°C (41°F), the generator is the limiting component in that it is unable to produce any more refrigerant to increase the capacity. The steady-state performance dependence on the water inlet temperature was also measured. Figure 5 shows the variation of the COP on the inlet water temperature. At high outdoor temperatures, the COP is insensitive to the changes of the water temperature, while at low temperatures (-21°C and -9°C) a significant decrease of the COP with increasing water temperature can be observed due to the limitation on refrigerant flow, resulting from the absorber being less effective at higher temperatures and lower pressures. Figure 6 shows the COP vs. outdoor temperature for various amounts of ammonia in the system. The water flowrate was not changed. It is obvious from this figure that at high outdoor temperatures the higher ammonia charge yields the better results; while the situation is just the opposite for low temperatures. This inverse relationship between charge and COP (capacity) at low temperatures is a result of the solution having a higher concentration of

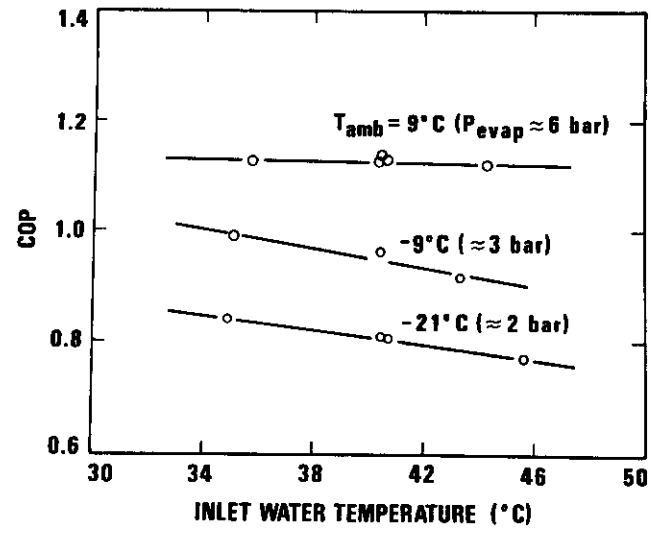


Fig. 5 Variation of COP of the heat pump with inlet water temperature

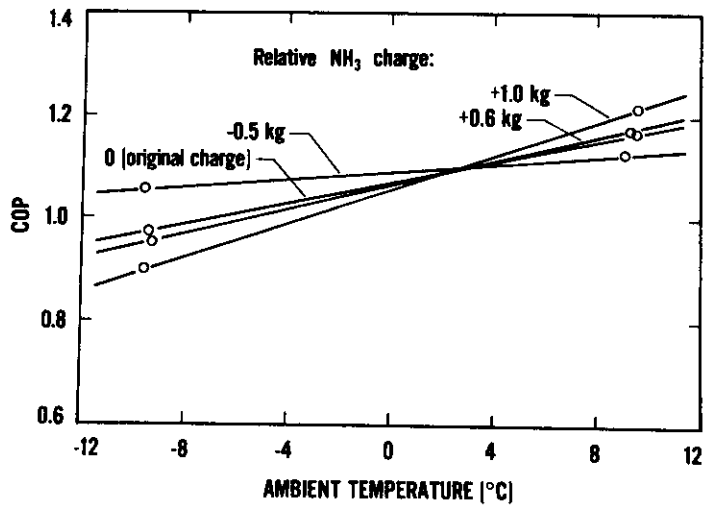


Fig. 6 The effect of the amount of ammonia charge on the COP for various outdoor temperatures

ammonia and thus not being able to absorb as much of the returning refrigerant as the evaporator produces.

In cyclic tests the thermostat's operation was simulated by turning the unit on and off for periods likely to exist in the field where a compromise between environmental comfort and equipment cycling is required. Except for the thermostat, the units were allowed to operate under their own controls. Most tests were performed at 20% and 50% on-time (i.e., the burner is in operation) with 1 cycle/hour and 1.5 cycles/hour, respectively.

The results are reported in terms of the Part Load Factor (PLF) and the Heating (or Cooling) Load Factor HLF (or CLF) for the

heating (or cooling) equipment. The PLF is defined by the following equation:

$$PLF = COP(cycle)/COP(steady\text{-}state)$$

Therefore, the PLF is a dimensionless number expressing the part-load performance in parts of the steady-state performance. The HLF (CLF) is defined as:

$$HLF = Q(cycle)/Q(steady\text{-}state)$$

Therefore, the heating load factor is a dimensionless number expressing the fraction of capacity supplied in terms of the steady-state capacity. The CLF is defined in exactly the same way, by replacing the heating capacities by the according cooling capacities.

Figure 7 shows the PLF vs. the CLF for the absorption chiller. The full dots show the experimental values for the absorption chiller. The size of the symbols in Figure 7 is indicative of the uncertainty in the measured values. The degradation factor, C_D, is commonly defined as the slope of a straight line passing through a point (CLF, PLF) = (1,1) and the measured value of the PLF at 20% on-time. The C_D value for this absorption chiller was 0.74.

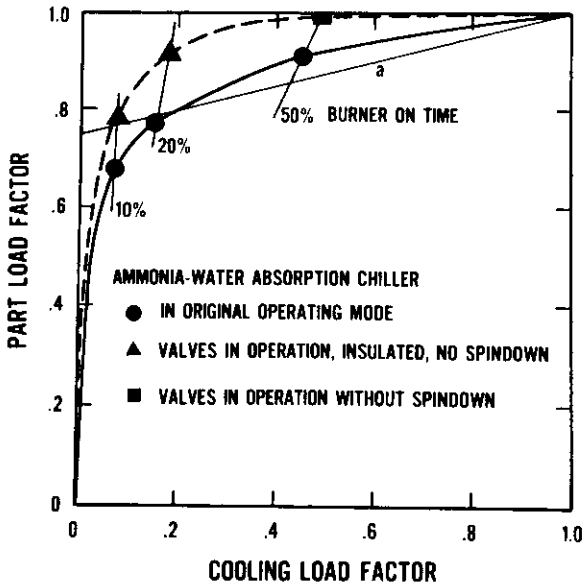


Fig. 7 Part-load performance of the absorption chiller

Figure 8 displays the experimental values of the PLF vs. HLF for the absorption heat pump under various cycling conditions. The uncertainty of the measured values is about ±2%. The degradation factor for the absorption heat pump was about 0.57.

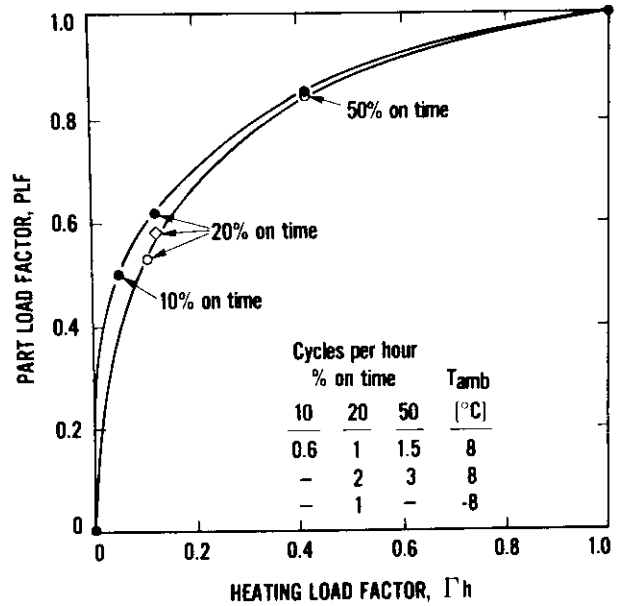


Fig. 8 Part load performance of the absorption heat pump

Reasons for the Degradation of Performance During Part Load Operation

In order to investigate the causes for the degradation due to part-load operation it is necessary to analyze a continuous plot of data vs. time obtained during an entire cycle. Figures 9 and 10 show the change of the capacity for the chiller and the heat pump respectively during the on-time, including the "spindown period" (a period during which the burner is turned off while pumps and blowers are kept operating in order to retrieve the residual capacity throughout the unit). It is obvious from both graphs that the capacity does not rise immediately after the start of the burner on-time. In case of the heat pump there is an initial peak, but this peak does not contribute significantly to the overall performance capacity of a particular cycle.

This period of variable capacity for both units suggests that at least some components of the units are not capable of steady-state performance until the effects of the uncontrolled equipment dynamics of the off-cycle period are rectified. Although the heat pump and the chiller serve different purposes, the thermodynamic processes and the cycle employed are the same.

In addition, both units were equipped with a similar falling film type absorber which was thought to be the limiting component. This would result in a direct restriction on the performance of the heat pump by not expelling as much heat into the coil waters as in steady-state operation. In the case of the chiller, a poor performing absorber results in less absorbed refrigerant and thus a lower amount of

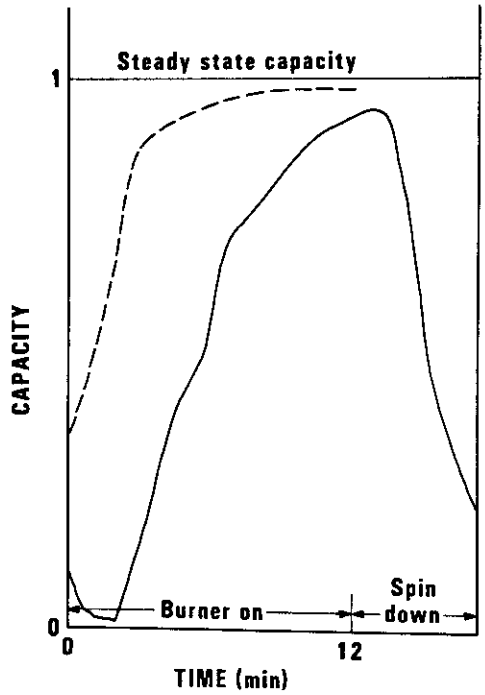


Fig. 9 Chiller performance as a function of time from start-up including a spin-down period when heating was stopped but pumps and blowers left running

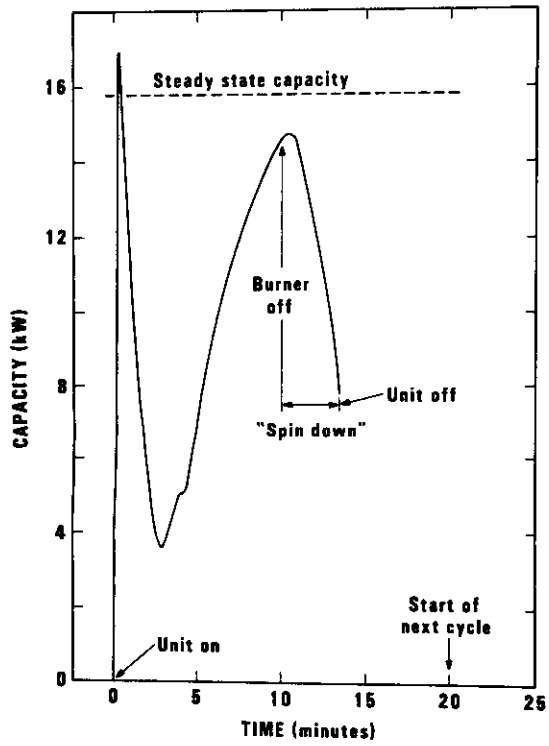


Fig. 10 Heat pump performance during one cycle

circulating charge with a respective decrease in cooling capacity.

Also, both the chiller and the heat pump had the same type of generator. This component allows for the weak ammonia solution to be at the bottom; thus, when either unit is turned off at the end of the spin-down period, the still existing pressure difference between generator and absorber forces the solution located in the generator into the absorber. An estimation of the volume ratio of the two components suggests the solution fills the absorber almost entirely. If this is true then no absorber heat and mass transfer area is available at the beginning of the on-time period until after the operating solution is evacuated by the solution pump.

Further data obtained show that for the tests with 20% on-time the high- and low-side pressure converge to a common value. Therefore, in addition to the migration due to the initial pressure difference further migration occurs due to the fact that the working fluids at locations of higher temperatures evaporate, migrate to, and condense in locations of colder temperatures. This effect not only causes dislocation of the working fluids but also a very effective but undesired heat transfer. During these processes the internal system eventually comes to an equilibrium. Due to this "heat-pipe effect" the conditions within the unit deviate even more from conditions existing during steady-state operation. Thus causing further degradation.* In addition to the processes described above there exist heat losses due to gradients existing within the absorption unit and between the absorption unit and the surroundings.

In order to substantiate some of the conclusions developed above, the following tests were performed on the absorption chiller only. Two quick closing valves were installed in the locations indicated in Figure 1 by "V". By this measure, the migration of solution from the generator into the absorber and from the condenser into the evaporator could be eliminated and the high and low pressure regions of the unit were separated because the solution pump acted as a check valve. These valves were air-actuated and controlled in a manner that they closed when the unit turned off and opened when the unit came back on. The cyclic tests were then repeated with the valves in operation. The results are shown in Figure 7 by the triangles. It is obvious that the part-load performance was significantly improved. The PLF was .89 instead of .75.

Therefore, by preventing the migration of the working fluids the losses due to part-load operation are reduced by more than 50%. A more

*Off-cycle refrigerant migration due to internal pressure and temperature differences, has been measured in vapor compression heat pumps as well, 3. Recovery period in these machines is strongly dependent on the heat transfer effectiveness of the component in which the liquid refrigerant resides at the start-up of the on-period.

detailed investigation indicated that the spin-down period can be eliminated resulting in an additional slight increase in performance due to savings in electrical energy. Also, the cyclic tests were repeated after all components of the unit were thermally insulated. During these tests the valves were in operation and the spin-down period was eliminated. It was found that the losses were reduced by another 5%. Although not all components were completely insulated, due to lack of accessibility to some spots, it is felt that the remaining degradation is mainly due to the heat and mass transfer by the "heat-pipe effect".

Although this investigation was done on equipment of specific design, the results can still be generalized. The contribution of each single cause to the degradation may differ for different designs. Although the application of automatic valves as described here increases the part-load performance significantly, they do not eliminate the degradation entirely. This is not surprising because the valves still allow parts of the unit to communicate through the so-called "heat-pipe effect". To eliminate this effect entirely it would be necessary to separate all components of the unit by valves. However, this may lead to excessive manufacturing costs and problems of charge leakage or valve sticking which increases maintenance. It is advisable to eliminate or reduce the migration of fluids by other means. One possible way could be to arrange the connecting tubes between components in such a manner that large amounts of liquid working fluids cannot pass during the off periods. This design criteria may be in conflict with that which requires draining of fluids in certain spots of a component where it is meaningful from a thermodynamic point of view during steady-state operation. That is where it is necessary in order to obtain a good steady-state performance with the optimum composition. Of course this measure only reduces the migration due to the pressure differential and thus the temperature difference migration would still occur.

Summary

In order to estimate the seasonal operating costs of an absorption heat pump (whether in the heating or cooling mode) is is necessary to determine both the full-load steady-state performance and the part-load cyclic performance. The latter is generally degraded in capacity due to the equipment requiring a considerable fraction (if not all) of the on-time period to recover from off-cycle solution migration. The current popular absorption equipment design allows for intercomponent fluid communication during the off-cycle; thus as the system tends toward equilization of internal pressures and temperatures solution and refrigerant quantities in each component are altered from that which exists during the normal operation period. Since reestablishment of steady-state conditions significantly penalizes the overall performance prevention of off-cycle migration would seem to be a logical design criteria.

References

1. Investigation of the Part Load Performance of an Absorption Chiller, Radermacher, R., Klein, S.A., Didion, D.A., ASHRAE Transactions 1983, Vol. 89, Part 1, Paper 2749.

2. A Laboratory Evaluation of the Steady-State and Cyclic Performance of a Prototype Air-to-Water Absorption Heat Pump, McLinden, M.O., Radermacher, R., and Didion, D.A., IIR Conference, Paris, Session E2, Paper 472, 31 August - 7 September, 1983.

3. A Laboratory Investigation of Refrigerant Migration in A Split Unit Air Conditioner, Mulroy, W. and Didion, D., NBSIR 83-2756, National Bureau of Standards, Washington, D.C. 20234.

4. Laboratory Evaluation of the Steady-State and Part Load Performance of Absorption Type Heating and Cooling Equipment, Radermacher, R, Mclinden, M., Klein, S., and Didion, D., NBSIR 84-2816, National Bureau of Standards, Washington, D.C. 20234.

Absorption Fluids Data Survey

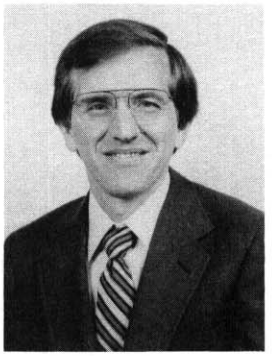
Robert A. Macriss and Thomas S. Zawacki

Institute of Gas Technology

Chicago, Illinois 60632

Robert A. Macriss directs IGT's research activities in the area of energy utilization and conservation. This includes R&D activities on appliance development, indoor air quality, fuel combustion, fluid properties, pollution control, energy conservation systems development, heat pumps, and heating and cooling systems.

Macriss' specific areas of responsibilities include programs covering the development of new, high-efficiency appliances, upgrading the efficiency of furnace and water heater installations in existing homes, the development of IGT's Solar-MEC heating and air-conditioning device (powered by solar energy), the development of new solar-absorption cooling concepts, heat pumps, studies that emphasize the interrelationship between appliance emissions and efficiency and the characteristics of the residences in which used, indoor air quality, health effects research, and environmental control. He has authored over 50 publications on these subjects and holds a number of related U.S. patents.



Thomas S. Zawacki is responsible for the activities of the IGT group working on energy conservation research for the residential and commercial sector. His work covers gas-fired appliances, space-heating systems, and the building structure. He is directly responsible for studies to improve the efficiency of space-heating equipment and installations, the development of data and models to describe the operation of gas furnace installations, the development of new high-efficiency gas furnaces and water heaters, and the assessment and control of nitrogen oxide emissions from such equipment.

Dr. Zawacki also supervises the development of novel absorption cooling concepts (for solar energy utilization) and is personally involved in the computer modeling of flames, space-heating installations, and IGT's Solar-MEC system.

ABSORPTION FLUIDS DATA SURVEY

Robert A. Macriss and Thomas S. Zawacki
Institute of Gas Technology
Chicago, Illinois

Objectives

The overall objective of this study is to develop improved data for the thermodynamic, transport and physical properties of absorption fluids. A specific objective of this phase of the study is to compile, catalog and coarse-screen the available U.S. data of known absorption fluid systems and publish it as a first edition reference document to be distributed to manufacturers, researchers and others active in absorption heat pump activities.

Introduction

Over the past 50 years, a wealth of physical and thermodynamic property data has been developed in the U.S. for fluid systems potentially applicable to the absorption refrigeration/heat pump cycle. A significant portion of the data is available in the literature, some still remain confined to in-house files for various reasons and other are proprietary.

Generally, published data have been generated in regions of temperature, pressure and composition of interest to designers of absorption chillers or coolers. Recent research has focused on developing efficient cycle configurations, using multi-effect or multi-stage concepts, for both space heating and cooling. These new designs operate at temperatures and pressures significantly higher than encountered in the conventional absorption cooling cycle. Other new concepts are based on the addition of a third component to the binary solutions normally used in the conventional cycle.

In an effort to define the availability and quality of existing data and, thereby, assist in the determination of needs for new data useful to advanced concepts, the Institute of Gas Technology (IGT) has performed a comprehensive literature search and a coarse screening of existing literature data in order to identify potential conflicts or gaps in the data, with particular emphasis on certain key fluid systems. In addition, IGT has carried out liaison activities with researchers and manufacturers known to possess other unpublished data that may be useful to this program and with several European researchers active in this field.

These activities have helped to further demonstrate the importance of a centralized, well documented source of the available data on absorption fluids to the timely development and introduction of advanced absorption heat pump technology for space conditioning and temperature boosting. Clear benefits are thought to be the identification of unresolved conflicts in data of known, important fluid systems, of data gaps for key fluids and properties, and of the total lack of data for new, potentially useful fluid systems.

This paper summarizes the methodology and findings of the compilation, cataloguing and coarse screening of the available U.S. data on absorption fluid properties and presents current status and future work on this project.

Methodology

Both in-house-file and literature searches were undertaken to obtain available U.S. publications, with several dating back to the 1920's, with pertinent physical, thermodynamic and transport properties data for absorption fluids. Cross-checks of literature searches were also made, using available published bibliographies and literature review articles, to eliminate secondary sources for the data and only include original sources and manuscripts.

Fluid Properties and Data

The properties of these fluids relate to the liquid and/or vapor state, as encountered in normal operation of absorption equipment employing such fluids, and to the crystallization boundary of the liquid phase, where

applicable. The actual data were systematically classified according to the type of fluid and property, as well as temperature, pressure and concentration ranges over which data were available. Data were sought for the 14 different properties listed below:

Mixture Properties

1. Vapor-Liquid Equilibria
2. Crystallization Temperature
3. Corrosion Characteristics
4. Heat of Mixing
5. Liquid-Phase-Densities
6. Vapor-Liquid-Phase Enthalpies
7. Specific Heat
8. Stability
9. Viscosity
10. Mass Transfer Rate
11. Heat Transfer Rate
12. Thermal Conductivity
13. Flammability
14. Toxicity

The type of available data was of importance, with raw experimental data considered to be highly desirable for use in actual process and equipment design and sizing, and for process design correlation development. For this reason, raw tabulated experimental data were given the highest possible ranking, with other forms of data following in importance, as shown below:

Data Rank

Form of Data	Symbol
RAW EXPERIMENTAL	(E)
• Tabular	(T)
• Graphical	(G)
SMOOTHED EXPERIMENTAL	(S)
• Tabular	(T)
• Graphical	(G)
EMPIRICAL POLYNOMIAL	(P)
EQUATIONS OF STATE	(C)

Empirical polynomial relationships based on raw or smoothed experimental data are often reported. Such correlations are at best as good as the experimental data and can help facilitate the use of the data in performance and design calculations through automation (computer software & terminals). In the absence of data, reliable methods of prediction were sought (equations of state, generalized correlations). These methods are expected to possess greater uncertainties than the previous forms of data.

Coarse-Screening of Literature Data

Several approaches have been used in carrying out quality review and coarse screening of data from multiple sources, for each of several key or highest priority binary and ternary fluids. These alternative approaches include —

- Consideration of the authors' statements concerning data quality
- Assessment of the measurement technique and apparatus used in obtaining the data
- Consideration of a given authors' reputation for measuring data, where a single authors' data were the only available for a fluid system.

A more reliable method used to compare data from various sources, for a given fluid and property, has been the plotting of data on a common basis. Such plots have also served to reveal the amount of scatter or internal consistency of a given author's data and, therefore, its quality. Another approach was based on general trends of the plotted data, as inferred from data for similar or homologous series fluids.

Accomplishments

Over 150 different U.S. publications were identified with data relating to properties of binary, ternary and multicomponent absorption fluids. After elimination of duplication through cross-checking, a total of 67 primary sources of data were selected (see references) and the actual documents acquired, dating as far back as 1925, with nearly 80% of the documents published between 1950 and 1983.

The absorption fluids covered in the 67 documents are combinations of 25 different "refrigerant" compounds with 67 single, 30 binary and 4 ternary "absorbent" compounds. Generally, the 25 refrigerants are divided among the following categories of chemical compounds —

Refrigerants

INORGANIC	3
ORGANIC	22
• Amines	3
• Alcohols	3
• Halogenated	16

Likewise, the single absorbent compounds are generally subdivided as follows —

Absorbents

INORGANIC	31
ORGANIC	48
• Alcohols	6
• Ethers	5
• Alcohol-Ethers	2
• Amides	8
• Amines	3
• Amine-Alcohol	1
• Esters	9
• Ketones	5
• Acids	3
• Aldehyde	1
• Others	4

The binary and ternary absorbents are various mixtures of 2 or more single absorbent compounds. Figure 1 presents examples of absorption fluid systems and the number of references that contain vapor-liquid equilibrium data for each fluid system. Similar charts have been prepared to cover other fluids, properties and relevant references.

In terms of absorption fluids properties, for which data are available in the 67 primary documents, the list shown below reveals that vapor-liquid equilibrium is the predominant property, reported in almost 60% of the documents. Flammability of fluids, on the other hand, is reported in only one document.

Number of References for Thermodynamic, Transport and Other Data

Fluid Property	Number of References	
Vapor-liquid equilibrium	46	
Enthalpy	4	
Density	7	
Corrosion (and inhibitors)	5	
Stability	6	
Transport Properties	13	(Viscosity, surface tension, additives, heat and mass transfer, etc.)
Heat of Mixing	10	
Crystallization	14	
Toxicity	2	
Flammability	1	

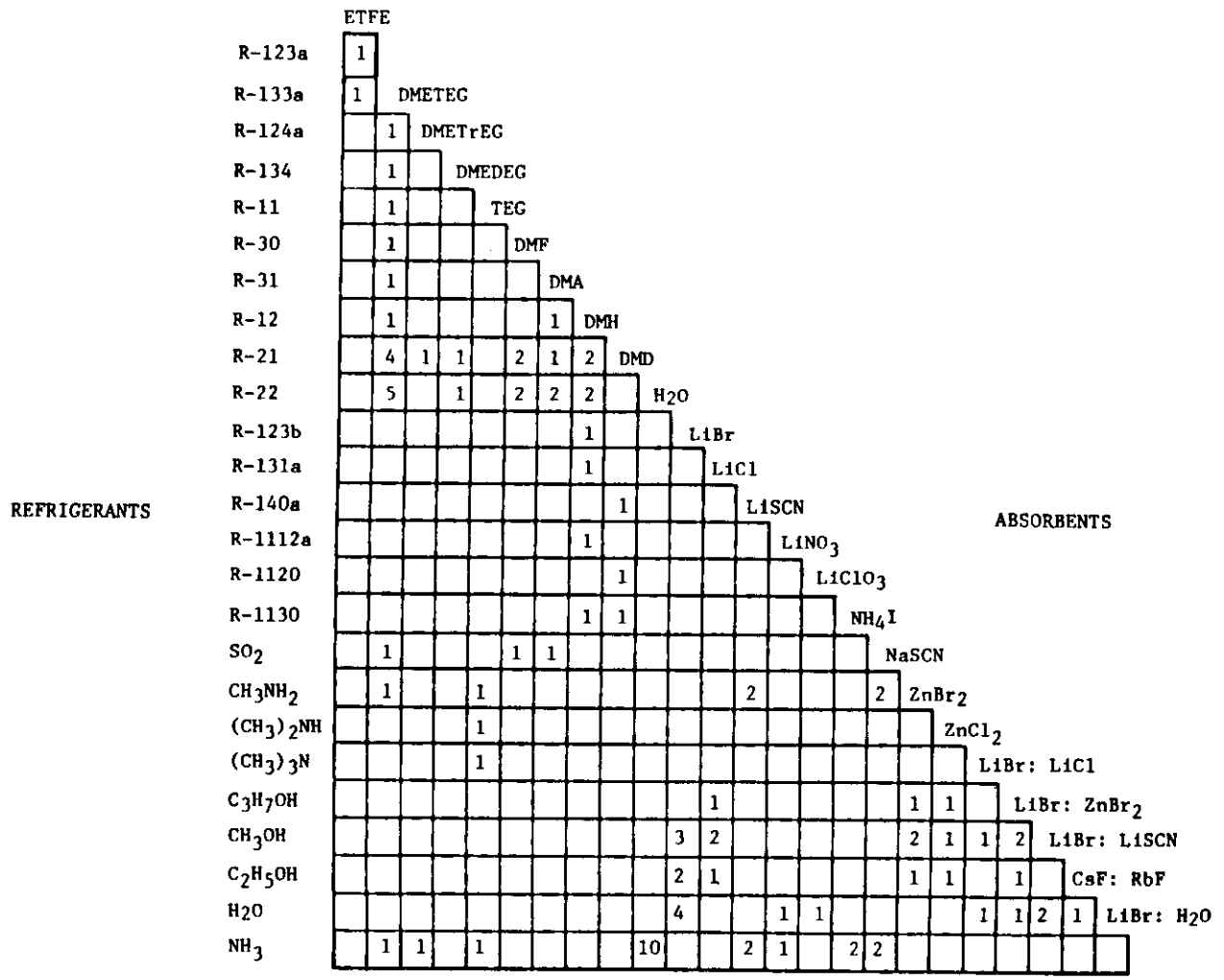


Figure 1. NUMBER OF REFERENCES FOR VAPOR-LIQUID EQUILIBRIUM DATA

For each fluid and property in this study, the type of data and the temperature, pressure and concentration ranges, to which the data correspond, and the reference access number, first author and year of publication, are all included in tables, as shown in the example of Table 1.0 attached.

It must be noted here that a given document may contain data relating to more than one property and it is this reason that the sum of all above references is larger than 67. For several key absorption fluids, currently of interest to researchers of advanced absorption heat pumps, the number of references available are shown below —

Key Fluid	Number of Data References
NH_3/H_2O	15
H_2O/LiBr	20
RI23a/ETFE	1
R22/DMETEG	11
NH_3H_2O + LiBr	0
CH_3NH_2/H_2O + LiBr	0

One would correctly conclude the absolute lack of data for the latter 2 fluids but would be drawn to incorrect conclusions for the RI23a/ETFE fluid (one reference). In fact, while only one organization has a proprietary interest in this fluid, the single reference covers adequate amounts of data for most properties of interest.

Data coarse-screening and evaluations were carried out for the selected or key absorption fluids. The data screening indicated that the plethora of references for fluids NH_3/H_2O and R22/DMETEG may be misleading for the following reasons —

- Most current research with the NH_3/H_2O fluid is concentrated in advanced cycle designs, with operating conditions (temperature, pressure) beyond the limits of currently available data.
- Data for several important properties (heat and mass transfer rates, stability), and for operating conditions beyond present limits for other properties, are not available.

1.0. Vapor-Liquid Equilibrium

Fluid System	Data Type	Temp. Range, °F	Press. Range, psia	Mol. Frac. X (1) Y (2)	Reference No.	Reference Author	Reference Year
$NH_3 + H_2O$	TE	32 - 196	0.3 - 169.3	0.09 - 0.85 0.616 - (250)[c]	61	Wilson	(1925)
$NH_3 + H_2O$	TE	141 - 242	222 - 521	0.962 - 0.995	34	Macriss	(1964)
$NH_3 + H_2O$	GTS	-105 - 470	1 - 500	0.0 - 1.0 0.0 - 1.0	35	Macriss	(1964)
$NH_3 + H_2O$	TE	195 - 296	14.7, 65	0.00092 - 0.0382 0.079 - 0.319	51	Polak	(1975)
$NH_3 + H_2O$	TC	-60 - 370	0.26 - 340	0.1 - 0.9 0.448 - (0.62)[c]	55	Scatachard	(1947)
$NH_3 + H_2O$	GC	60 - 170	NA	NA	21	Edwards	(1978)
$NH_3 + H_2O$	GC	-50 - 240	0.3 - 500	0.1 - 0.9 0.1 - (1)[c]	60	Won	(1980)
$NH_3 + H_2O$	CP	22 - 45, 110 - 132	50 - 80, 250 - 350	NA	29	Jain	(1971)
$NH_3 + H_2O$	TC	-58 - 417	6 - 300	0.0015 - 0.995 0.0114 - (1.35)[c]	14	Bogart	(1980)
$NH_3 + H_2O$	TS	-105 - 417	1 - 300	0.0 - 1.0 0.0 - 1.0	31	Jennings	(1965)

c - ppmv absorbent in vapor
X - Liquid phase
Y - Vapor phase
NA - Not available
(1) - Refrigerant
(2) - Absorbent

Table 1.0. AMMONIA BINARY AND MULTICOMPONENT ABSORPTION FLUIDS PROPERTIES DATA

Unresolved conflicts in important data of various fluids and data gaps for key fluids were summarized. Figure 2 is an example of conflicting data for the vapor phase composition of the $NH_3 + H_2O$ system at 140°F.

Data conflicts, lack of data for known key fluids at conditions beyond the limits of present data, and new fluids without property data constitute these summaries. Figure 3 presents an example of absorption fluid systems with prioritized gaps in vapor-liquid equilibrium data.

With extremely few exceptions among key fluids and their properties, and with emphasis toward the current development activities dealing with advanced absorption heat pump concepts, most all candidate fluids have data gaps that need to be addressed in the near future.

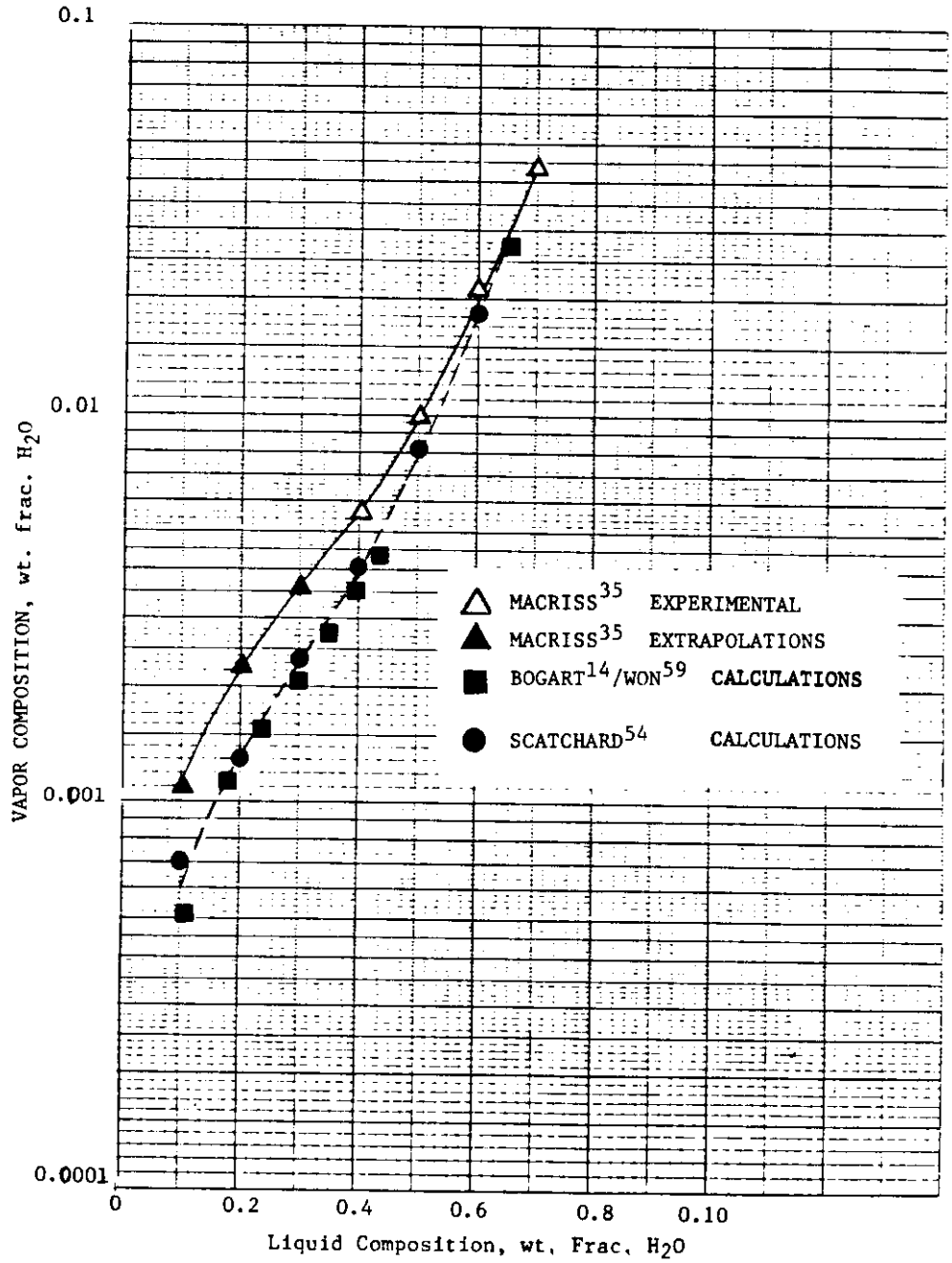


Figure 2. COMPARISON OF MEASURED AND PREDICTED VAPOR COMPOSITION OF THE $NH_3 + H_2O$ PAIR AT 140°F

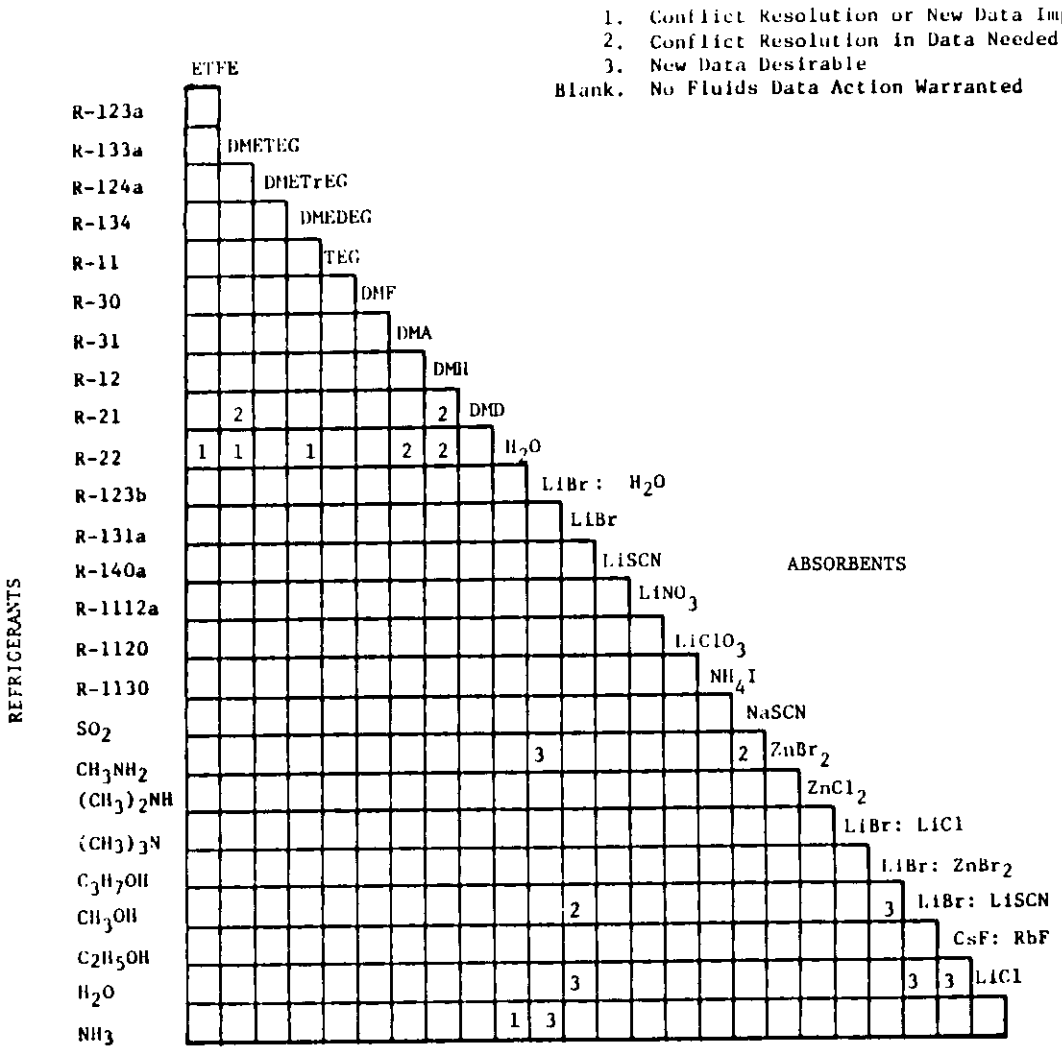


Figure 3. GAPS IN VAPOR-LIQUID EQUILIBRIUM DATA

Future Work

In the near term, additional data for known systems will be added to the U.S. reference document. Some of these data, provided by individual researchers, have been held confidential until now and have been developed in support of manufacturers past developmental activities. Other data, to be made part of the U.S. document, are presently in a final stage of development, prior to publication by the authors. For the longer term, potential inclusion of data developed abroad has not been precluded.

References

1. Albright, L. F., Buclez, P. C., Pluche, C. R., and Doody, T. C., "Solubility of Refrigerants 11, 21, and 22 in Organic Solvents Containing an Oxygen Atom," ASHRAE Transactions, 66, 423-433, 1960.

2. Albright, L. F., Shannon, P. T., Terrier, F., and Chueh, P. L., "Solubility of Chlorofluoromethanes in Nonvolatile Polar Organic Solvents," A.I.Ch.E.J. 8 No. 5, 668-672 (1962) November.

3. Albright, L. F., Shannon, P. T., Yu, S.N., and Chueh, P. L., "Solubility of Sulfur Dioxide in Polar Organic Solvents," Chem. Eng. Prog. Symp. Series 59, No. 44, 66-74, 1963.

4. American Gas Association (A.G.A.), "Symposium on Absorption Air-Conditioning Systems," Institute of Gas Technology, Chicago, Illinois June 1968.

5. Aker, J. E., Squires, R. G., and Albright, L. F., "An Evaluation of Alcohol-Salt Mixtures as Absorption Refrigeration Solutions," Paper presented at the ASHRAE Semiannual Meeting, Chicago, January 25-28, 1965.

6. Allen, R. A. and Murphy, K. P., "Development of a Residential Sized Gas Fired Absorption Heat Pump," Annual Report

prepared for Gas Research Institute by Allied Chemical Corporation, Morristown, N.J., November 1979.

7. Aronson, D. (assigned to Worthington Corp.), "Absorption Refrigeration System," U.S. Patent 3,478,530 (1969) November 18.

8. Biermann, W. J., "Candidate Chemical Systems for Air Cooled Solar Powered, Absorption Pairs," Unpublished Report Prepared for Department of Energy, Contract No. EG-77-C-03-1587, Carrier Corp., Undated.

9. Biermann, W. J., "Candidate Chemical Systems for Air Cooled Solar Powered, Absorption Air Conditioner Design, Part II - Solid Absorbents, High Latent Heat Refrigerants," Unpublished Report Prepared for Department of Energy, Contract NO. EG-77-C-03-1587, Carrier Corp., April 1978.

10. Biermann, W. J., "Candidate Chemical Systems for Air Cooled, Solar Powered, Absorption Air Conditioner Design, Part III - Lithium Salts with Anti-Freeze Additives," Unpublished Report Prepared for Department of Energy, Contract No. EG-77-C-03-1587, Carrier Corp, June, 1978.

11. Blytas, G. C., "Physicochemical Properties of Concentrated Solutions of Sodium Thiocyanate in Liquid Ammonia," unpublished Ph.D. Thesis, University of Wisconsin, 1961.

12. Blytas, G. C., Kertisz, D. J., and Daniels, F., "Concentrated Solutions in Liquid Ammonia: Solubility of $NaNO_3$ and KBr and other Salts; Vapor Pressure of $LiNO_3$-NH_3 Solutions, Journal of the American Chemical Society, 84 No. 6, 1083-1085 (1962) April 5.

13. Blytas, G. C., and Daniels, F., "Concentrated Solution of NaSCN in Liquid Ammonia. Solubility, Density, Vapor Pressure, Viscosity, Thermal Conductance, Heat of Solution and Heat Capacity", Journal of the American Chemical Society, 84, No. 7, 1075-1082, (1962) April 5.

14. Bogart, M. J. P., Ammonia Absorption Refrigeration in Industrial Processes. Houston: Gulf Publishing Co., 1980.

15. Boryta, D. A., "Solubility of Lithium Bromide in Water Between 50° and +100° C (45-70% Lithium Bromide)," J. Chem. and Eng. Data, 15, No. 1, 142-144, 1961.

16. Buclez, P. C., "The Solubility of Some Easily Liquefiable Gases in High Boiling Point Solvents," unpublished Master's Thesis, Purdue University, June 1959.

17. Burnett, J. C., and Himmelblau, D. M., "The Effect of Surface Active Agents on Interphase Mass Transfer," A.I.Ch.E.J. 16, No. 2, 185-193, (1970) March.

18. Cohen, A., and Jelinek, R. V., "Corrosion Rates of Mild Steel in Alkaline Lithium Bromide Solutions by the Polarization Resistance Method," Corrosion, 22, 39-47 (1966) February.

19. Deemer, A. R., "Absorption of Water Vapor by an Aqueous Lithium Bromide Solution," unpublished Master's Thesis, Illinois Institute of Technology, Chicago (1972) December.

20. Eding, H. J., and Brady, A. P., "Refrigerant-Absorbent Systems," SRI Project No. S-3372 Final Report prepared for Roy C. Ingersoll Research Center, Borg-Warner Corporation, Menlo Park, California: Standford Research Institute, (1961) March 31.

21. Edwards, T. J., Newman, J., and Prausnitz, J. M., "Thermodynamics of Vapor-Liquid Equilibrium for the Ammonia Water System," Ind. Eng. Chem. Fund, 17, No. 4, 264-269, 1978.

22. Eiseman, B. J., Jr., "Why Refrigerant 22 Should Be Favored for Absorption Refrigeration," ASHRAE J., 45-50 (1959) December.

23. Ellington, R. T., Kunst, G., Peck, R. E., and Reed, J. F., "The Absorption Cooling Process," Institute of Gas Technology Research Bulletin, No. 14, Chicago, 1957.

24. Ewing, W. W., and Fisher, H. M., "Studies in the Vapor Pressure - Temperature Relations of the Primary System Zinc Nitrate-Water," J.A.C.S. 59 1046-1048 (1937) June.

25. George, J. H. B., and Birkett, J. D., "Study of Liquid Ammonia-Salts Systems and Their Potential Application to the Absorption Cooling Process," Projects ZF-72 and ZF-72a Final Report prepared for A.G.A. New York: American Gas Association, 1965 (Catalog No. M10090).

26. Hensel, W. E., Jr., "Harlowe, I. W., Jr. (assigned to Arkla Industries, Inc.): "Compositions for Absorption Refrigeration System," U.S. Patent 3,643,455 (1972) February 22.

27. Hesselberth, J. F., and Albright, L. F., "Solubility of Mixtures of Refrigerants 12 and 22 in Organic Solvents of Low Volatility," unpublished report, Purdue University, undated.

28. "The International Critical Tables," Vol. III, 369. New York: McGraw-Hill Book Company, Inc., 1928.

29. Jain, P. C., and Gable, G. K., "Equilibrium Property Data Equations for Aqua-Ammonia Mixtures," AHRAE Transactions, No. 2180, 1971.

30. Jennings, B. H., "Ammonia-Water Properties (Experimentally-Determined P, V, T, x Liquid Phase Data)," Paper presented at ASHRAE Semiannual Meeting in Chicago, January 25-28, 1965.

31. Jennings, B. H., and Shannon, F. P., "The Thermodynamics of Absorption Refrigeration. Tables of Properties of Aqua-Ammonia Solutions," J. of ASRE, 333-336 (1938) May.

32. Ledding, A., Borg-Warner Corporation, private communication with R. A. Macriss of IGT, 1965.

33. Lee. C. Y., "An Evaluation of Ammonia-Salt Mixtures as Absorption Refrigeration Solutions," Unpublished Master's Thesis, Purdue University (1965) June.

34. Macriss, R. A., and Eakin, B.E., "Thermodynamic Properties of Ammonia-Water Solutions Extended to Higher Temperatures and Pressures." Paper presented at the ASHRAE Meeting, Cleveland, July 1964; published in ASHRAE Trans. 70, 319-27, 1964.

35. Macriss, R. A., Eakin, B. E., Ellington, R. T., and Huebler, J., "Physical and Thermodynamic Properties of Ammonia-Water Mixtures," Institute of Gas Technology Research Bulletin, No. 34, Chicago, 1964.

36. Macriss R. A., and Mason, D. McA, "Physical and Thermodynamic Properties of New Refrigerant-Absorbant Pairs," Project ZB-85 Final Report prepared for the Air Conditioning and Prime Mover Research Committee of the A.G.A. Institute of Gas Technology, (1964) October.

37. Macriss, R. A., Weil, S. A., and Rush, W. F. (assigned to the American Gas Association), "Absorption Refrigeration System Containing Solutions of Monomethylamine With Thiocyanates," U.S. Patent 3,458,445 (1969) July 29.

38. Macriss, R. A., Punwani, D., Rush, W. F., and Biermann, W. J., "Thermodynamic and Physical Properties of Monomethylamine-Lithium Thiocyanate System," J. Chem. Eng. Data, 15 No. 4, 466-470 (1970) October.

39. Macriss, R. A., and Rush, W. F., (assigned to the American Gas Association), "Lithium Bromide-Lithium Thiocyanate-Water Composition for an Absorbent-Refrigeration System," U. S. Patent 3,541,013 (1970) November 17.

40. Macriss, R. A., Punwani, D., Rush, W. F., and Biermann, W. J., "Thermodynamic and Physical Properties of the Ammonia-Lithium Thiocyanate System," J. Chem. Eng. Data (1971) December.

41. Macriss, R. A., "Selecting Refrigerant - Absorbent Fluid Systems for Solar Energy Utilization," Paper presented at the ASHRAE Semiannual Meeting, Dallas, Texas, February, 1976.

42. Mastrangelo, S. V. R., "Solubility of Some Chlorofluorohydrocarbons in Tetraethylente Glyco Dimethyl Ether," ASHRAE J. 64-68 (1959) October.

43. McNeely, L. A., "Thermodynamic Properties of Agueous Solutions of Lithium Bromide," ASHRAE Transactions, 85, Part 1, 1979.

44. Mink, W. H., and Filbert, R. B., Jr., "Design and Development of a Refrigeration Cycle Employing the Principle of Azeotropic Mixtures," Final Report prepared for A.G.A. Columbus, Ohio: Battelle Memorial Institute, July 29, 1955.

45. Morrone, J. W., "Solubility of Ammonia in Low-Volatile Organic Solvents," Unpublished Thesis, Purdue University, (1962) February

46. Murphy, K. P., and Allen, R. A., "Development of a Residential Gas Fired Absorption Heat Pump," Interim Report prepared for Oak Ridge National Laboratories and Gas Research Institute: Allied Chemical Corporation, Morristown, N.J. (1982) March.

47. Murphy, K. et al., "Development of a Residential Gas Fired Absorption Heat Pump, Physical and Thermodynamic Properties of R123a/ETFE, System Development and Testing, Economic Analysis," Final Report ORNL/Sub/79-24610/5 Allied Corportation, Morristown, N.J. (1983) November.

48. Pennington, W., "How to Find Accurate Vapor Pressures of Lithium Bromide Water Solutions," Refrig. Eng. 63, 57-61 (1955) May.

49. Pluche, C. R., "Solubility of Various "Freons" in Esters and Alcohols," Unpublished Master's Thesis, Purdue University, (1959) June.

50. Podoll, R. T., Gray, T. E., Rolon, C. E., and Sabo, K. A., "Determination of Properties of Fluids for Solar Cooling Applications," Prepared for Department of Energy: SRI International, Menlo Park, CA, 1982.

51. Polak, J., and Lu, B. C. Y., "Vapor-Liquid Equilibrium in System Ammonia-Water at 14.69 and 65 Psia," Journal of Chemical and Engineering Data, 20, No. 2, 1975.

52. Roberson, J. P., "Solubility of Ammonia in Solvent Mixtures of Low Volatility," Unpublished Master's Thesis, Purdue University, (1965) January.

53. Roberson, J. P., Lee, C. Y., Squires, R. G., and Albright, L. F., "Vapor Pressure of Ammonia and Methylamines in Solutions for Absorption Refrigeration Systems," ASHRAE Trans. 72, Part 1, 198-208, 1966.

54. Rush, W. F., Macriss, R. A., and Weil, S. A., "A New Fluid System for Absorptiion Refrigeration," Paper presented at the Fourth International Congress of Heating and Air Conditioning, Paris, May 17-23, 1967.

55. Scatchard, G., Epstein, L. F., Warburton, J., Jr., and Cody, P.J., "Thermodynamic Properties - Saturated Liquid and Vapor of Ammonia-Water Mixtures," J. of ASRE (1947) May.

56. Sims, W. H., O'Neill, M. J., Reid, H. C., and Bisenius, P. M., "Evaluation of Absorption Cycle for Space Station Environmental Control System Application," Final report prepared for NASA Marshall Space Flight Center, HREC-5986-2, LMSC-HREC D306225: Lockheed Inc., Hunstville, Alabama, November 1972.

57. Thieme, A., and Albright, L. F., "Solubility of Refrigerants 11, 21, and 22 in Organic Solvents Containing a Nitrogen Atom and in Mixtures of Liquids," ASHRAE J. 3, 71-75 (1961) July.

58. Weil, S. A., "Thermodynamic Properties of Lithium Chloride-Lithium Bromide-Water System," Unpublished report, IGT Project No. S-153 for the Foote Mineral Co. and A.G.A. Chicago: Institute of Gas Technology, (1960) February.

59. Weil, S. A., and Ellington, R. T., "Corrosion Inhibition of Lithium Bromide-Water Cooling Systems," Project ZB-29 Final Report for the General Research Planning Committee of the A.G.A. Chicago, Institute of Gas Technology, (1956) December.

60. Won, K. W., Selleck, F. T., and Walker, C. K., "Vapor-Liquid Equilibria of the Ammonia-Water System," presented at the II International Symposium on Phase Equilibrium and Fluid Properties in the Chemical Industry, Berlin, March 17-21, 1980.

61. Wilson, T. A., "The Total and Partial Vapor Pressures of Aqueous Ammonia Solutions," Bulletin No. 146 of the University of Illinois Engineering Experimental Station. Urbana, Illinois: University of Illinois, February 1925.

62. Zawacki, T. S., Macriss, R. A., and Rush, W. F., "The Effect of Additives on the Level of Instability of Gas/Liquid Interfaces: The Absorption of Water Vapor by Concentrated Lithium Bromide Solutions in Falling-Film and Open-Channel Absorbers," Paper presented at the 75th National Meeting of the A.I.Ch.E., Detroit, June 3-6, 1973.

63. Zawacki, T. S., Leipziger, S., and Weil, S. A., "Inducement of Convective Motion in Static Absorbers," Paper presented at 4th Joint Chemical Engineering Conference, A.I.Ch.E.C.S.CH.E., Vancover, September 9-12, 1973.

64. Zellhoeffer, G. F., "Solubility of Halogenated Hydrocarbon Refrigerants in Organic Solvents," Ind. and Eng. Chem. 29 (1937) May.

65. Zellhoeffer, G. F., "Copley, M. J., and Marvel, C. S., "Hydrogen Bonds Involving the C-H Link. The Solubility of Haloforms in Donor Solvents," J.Am. Chem. Soc. 60, 1337-43 (1938) June.

66. Zellhoeffer, G. F., and Copley, M. J., "The Heats of Mixing Haloforms and Polythylene Glycol Ethers," J. Am. Chem. Soc. 60 1343-45 (1938) June.

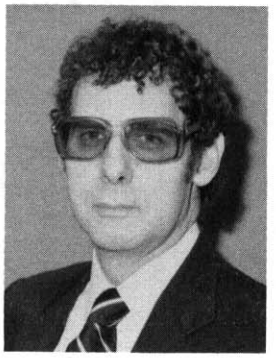
67. Zellhoeffer, G. F., "Commercial Refrigeration by Low Pressure Steam," Journal of the A.S.R.E., 317-320, (1937) May.

The GRI Gas Heat Pump Program—An Update

Arthur R. Maret

Gas Research Institute

Chicago, Illinois 60631



Arthur R. Maret received his undergraduate education in the United Kingdom and a Ph.D. in chemical physics from Case Western Reserve University in Cleveland, Ohio. He is currently program manager for the residential and commercial gas heat pump project at the Gas Research Institute (GRI). Before joining GRI, Dr. Maret was president of Teram Corporation, a consulting and manufacturing company in the energy sciences.

THE GRI GAS HEAT PUMP PROGRAM - AN UPDATE

by
Arthur R. Maret
Manager, Gas Heat Pumps
Gas Research Institute
Chicago, Illinois 60631

Introduction

The Gas Research Institute (GRI) is a private, not-for-profit organization that plans, manages, and develops financing for a comprehensive cooperative research and development program related to natural gas and equivalent substitutes. The goal of the program is to provide near-and long-term benefits for the natural gas industry and its present and future customers in all end-use sectors.

Since its beginning in 1977, GRI has placed a high priority on supporting an aggressive research program aimed at the development and commercialization of gas heat pumps (GHP) which could compete in the U.S. residential and commercial heating, ventilating, and air conditioning (HVAC) markets. On the basis of a percentage of the total GRI R&D budget, funding for gas heat pump development has been, and continues to be, among the larger project areas (Fig. 1).

Currently GRI is funding six GHP hardware projects as shown in Table I. Three, quite different technologies are under development: internal combustion engine-driven, stirling engine-driven and absorption. As recently as four years ago, it was believed that GHP COP's of 1.5 (heating) and 0.5 (cooling) could be competitive. It now appears that a much higher level of performance is needed, while maintaining a low first cost. These two requirements represent a formidable challenge. Table II summarizes the performance targets of the generic technologies under development at GRI.

As a prelude to this status report on each GRI GHP hardware project, it will be helpful to provide an overview of the current characteristics of the residential HVAC market.

Recent Trends in the Residential HVAC Market

The U.S. HVAC marketplace is highly competitive, and has traditionally emphasized high reliability and low first cost. The major competitors of the GHP are expected to be gas furnaces with electric air conditioners and electric heat pumps. The efficiency of each of these two system alternatives has improved significantly in recent years, and consumers have tended to emphasize high performance as well as reliability and cost in this era of rising energy prices. In all probability the successful GHP must demonstrate: high performance (especially in cooling), low first cost, and high reliability to be competitive in the late 1980, early 1990 time frame.

Another factor affecting the HVAC marketplace has been a trend towards smaller, better insulated houses requiring HVAC equipment with smaller capacities. At the same time, the retrofit market requires units of larger capacity. A substantial proportion of new residential construction is in the Southern region of the U.S., where air conditioning is now considered a necessity. Thus, a successful GHP must be able to be produced in a range of sizes, cost-effectively, to cover the broad range of the marketplace. It may also be attractive to have the capability to modulate output capacity to meet the seasonal variations of house heating demands.

A major marketing problem for the GHP is the variation in selection criteria by various potential purchasers. In the new construction market, the builder is likely to make the purchase decision, and initial cost rather than operating cost or efficiency is generally the most important criteria. In the replacement market (replacement of worn-out equipment), the installing contractor is the principal decision maker. The retrofit market (replacement of functioning equipment with a more efficient unit) is when the consumer probably plays the largest role in the purchase decision.

The above factors indicate the need for products which satisfy a large range of markets, which in turn exacerbates the already difficult task of developing a cost-effective GHP. The results of recent economic assessments suggest that the GHP shows potential in specific geographic regions having high building heating loads, but generally faces stiff competition from an advanced gas furnace/electric air conditioner combination or an advanced electric heat pump.

GRI Residential Heat Pump Project Area Strategy

In recent years, GRI has pursued a variety of technical approaches for development of a residential GHP, including Rankine, single-effect absorption, double-effect absorption, free-piston Stirling, and duplex Stirling cycles. In 1983 and 1984, GRI R&D expanded to include evaluation of several additional GHP concepts. These include (a) a dual cycle absorption concept, in which the use of enhanced-surface heat exchangers together with a novel thermal coupling of two cycles improves the performance of conventional refrigerant fluids and (b) IC-engine-driven, Rankine-cycle gas heat pumps. Based on this experience, four concepts have been judged most feasible for continued R&D: double-effect absorption, 'advanced' absorption technology, the IC-engine GHP and the free-piston Stirling/Rankine technology.

As shown in Figure 2, current GRI strategy calls for market entry of an absorption GHP or an IC-engine-driven GHP (or both) by 1990. If currently planned GRI R&D on free-piston Stirling/Rankine technology is successful, a heat pump of this type may also be available by late 1990. Additional advanced engine-driven and absorption GHP concepts are scheduled for study beginning in 1986; a heat pump system based on such concepts would be expected to enter the marketplace during the mid-1990s. It is also possible that prime movers developed for residential GHP use may prove suitable for other applications, such as cogeneration. R&D on these technologies will be supplemented by comprehensive analyses of manufacturing costs, potential applications, and thermodynamic cycles.

A qualified HVAC manufacturer must participate in any GRI GHP project before prototype field testing will be undertaken. Any participating manufacturer will be expected to provide an increasing portion of the project funding support as the uncertainties and risks associated with concept development diminish. GRI support will be reduced after the full-scale field test, with the manufacturer eventually assuming total support.

Double-Effect Absorption Heat Pump (Contractor: Columbia Gas)

OBJECTIVE: Develop a double-effect, residential size absorption heat pump with a three-ton cooling capacity and COPs of 1.55 (heating) and 0.85 (cooling).

ROLE IN PROJECT AREA STRATEGY: This concept represents the use of new absorption technology which can be used for either residential or small commercial applications with commercialization expected in the late 1980's. State-of-the-art components and controls are heavily emphasized in this system. This technology should be very useful in future advanced absorption projects.

CONTRACT RESULTS TO DATE AND STATUS: Columbia Gas Service Corp. undertook a preliminary effort to develop this concept in the mid-1970's. In 1980, GRI joined Columbia in initiating a jointly funded program to develop a heat pump utilizing this absorption technology. In Phase I, the cycle thermodynamics were demonstrated in a non-gas-fired breadboard system, and COPs of 1.55 heating and 0.80 cooling were achieved. In Phase II (1982-1983), an advanced gas-fired breadboard system was assembled using near-commercial components. Construction of a test facility for the advanced breadboard was completed and five critical issues were investigated: corrosion, solution pump design, energy recovery, system weight/cost and heat exchanger optimization. Work is continuing in 1984 on the five issues identified above, in addition to a performance evaluation of the advanced breadboard system. The results of these efforts will be used to develop a prototype heat pump design that can be utilized in discussions with potential heat pump manufacturing partners.

To date, the advanced breadboard system operating on natural gas has achieved COPs of 1.54 heating and 0.82 cooling with good stability characteristics. Several candidate construction materials have demonstrated corrosion rates of less than 1 mil/yr in 2400 hour tests. One of the solution pump concepts under consideration (a pump developed by Graco Co.) has maintained the required specifications through 5000 hours of testing under simulated heat pump conditions.

Preliminary component designs for a completely packaged prototype heat pump have been completed. By utilizing enhanced surface heat exchanger technology and good design practices, the current estimated system weight is below 700 lbs (the target weight is 800 lbs).

A fourteen-month Phase III effort was initiated on October 1, 1984 with the goal of the construction and evaluation of a packaged prototype.

Dual Cycle Absorption Heat Pump (Contractor: Battelle-Columbus)

OBJECTIVE: Develop a residential size gas heat pump with a 1-1/2 to 3 ton cooling capacity and COP of 1.80 (heating) and COP of 0.94 (cooling).

ROLE IN PROJECT AREA STRATEGY: This concept uses existing, well-characterized, fluids in conjunction with a novel cycle, thus should require a relatively short development period.

CONTRACT RESULTS TO DATE AND STATUS: In this concept, two single-effect absorption cycles are thermally coupled to provide the improved performance of a double-effect absorption system operating at moderate temperatures and pressures. The principle deliverable from Phase I of the contract effort is to design, build, and test a breadboard prototype system meeting the performance goals of 1.8 COP heating and 0.94 COP cooling. An advanced version of the breadboard prototype will be tested by a collaborating HVAC manufacturer.

One of the key components of the heat pump system is the Absorption Power Module (APM), a patented heat exchanger. Flow visualization testing was utilized to determine a reliable and reproducible method of applying the absorption fluid to the APM. A preliminary configuration heat pump system has been built to evaluate the APM performance under simulated heat pump conditions. Initial testing has given a cooling COP of 0.90 at an APM capacity of 15,000 Btu/hr.

Several system design modifications to the basic dual cycle configuration were modeled, resulting in analytical projections of COPs as high as 2.5 (heating) and 1.5 (cooling).

Free-Piston Engine Heat Pump (Contractor: MTI, Inc.)

OBJECTIVE: To improve the efficiency, reduce the hardware cost, increase the life, demonstrate the reliability necessary for a field test, and enhance the dynamic stability and control characteristics of an existing free-piston Stirling engine.

ROLE IN PROJECT AREA STRATEGY: Development of a reliable, efficient and long life natural gas fueled engine is essential to the development of small, mechanically driven heat pumps and cogenerators.

CONTRACT RESULTS TO DATE AND STATUS: The major effort to date has focused on technology development to advance the power, efficiency and reliability of the existing free-piston Stirling engine (FPSE) to the full extent of its design potential

for subsequent product evaluation. During 1983 and the first half of 1984, several technical goals have been achieved. A new combustion air preheater with increased area was developed and tested, resulting in an external heat system efficiency of 80% (HHV). The laboratory test engine's indicated mechanical efficiency was increased from 24% to 32%. The mechanical efficiency is based on PV power which is approximately 20% to 40% greater than shaft power. A modified displacer motor was designed and developed which resulted in the increased engine efficiency at higher displacer strokes and lower parasitic power losses. In addition, 1983 saw the successful operation of the laboratory test engine for over 1000 hours, including operation of part load, full load, and duty cycling.

In 1984, an internally supported gas bearing system has been designed, built, and is presently undergoing laboratory tests. In addition, an external gas combustor capable of operating on low pressure (4 inches water column) natural gas has been designed. Companion compressor development for the Stirling Heat Pump System is being supported by DOE in a coordinated and parallel program.

Reciprocating I.C. Engine Heat Pump (Contractor: Battelle Columbus)

OBJECTIVE: The overall objective is this multi-phase program is to develop a reciprocating engine-driven heat pump with cooling capacity of three tons, a heating COP of 1.7 and a cooling COP of 1.0. The objective of the Phase I effort is to establish an optimum engine/compressor configuration, fabricate three engines and obtain efficiency and durability characteristics.

ROLE IN PROJECT AREA STRATEGY: This project will utilize state-of-the-art engine technology in conjunction with well established manufacturing techniques and should provide a relatively low risk approach to a near to mid-term engine-driven heat pump.

CONTRACT RESULTS TO DATE AND STATUS:

The scheduled start date of 07/01/84 was delayed to 09/01/84 due to staffing problems at the contractor. Preliminary engine performance curves were provided to the cooperating HVAC manufacturer who then developed a series of scenarios for matching the engine to existing compressor and heat exchanger designs. Initial data portends to the use of a 4-cycle 4.9 horsepower engine to achieve a 3 RT capacity.

GRI Commercial Heat Pump Project Area Strategy

R&D efforts by GRI and others in the last several years have led to the development of three GHP concepts for the commercial sector. One concept would use a Brayton engine to power a Rankine-cycle compressor, the second would use a kinematic Stirling engine, the third an internal-combustion engine. Though the Brayton type is based on an established technology, GRI-supported R&D to achieve required improvements in performance and cost has not been successful. The Stirling concept appears more promising because it offers potentially higher COPs than the Brayton and may be smaller in size. R&D is needed, however, to address materials, heat transfer, endurance, and cost issues. The IC-engine GHP also involves the use of a well established technology, but maintenance and durability are major concerns.

High first cost could limit wide use of the GHP in the commercial sector. However, it is expected that a substantial number of commercial owner/operators would be willing to accommodate this added first cost if payback time is reasonable. One approach to improve payback involves the use of a GHP to heat water, either as a dedicated water-heating unit or as a "dual function" system that provides both space conditioning and hot water. Year-round demand for hot water can significantly improve overall GHP economics.

GRI's current 5 year plan for this area is shown in Figure 3.

Kinematic Stirling Heat Pump (Contractor: SPS)

OBJECTIVE: Develop a high efficiency, cost-effective, natural gas-fired, kinematic Stirling/Rankine heat pump for commercial applications.

ROLE IN PROJECT AREA STRATEGY: This concept represents a near-term market entry. As a fourth generation prototype, the Stirling engine used in this effort is well-characterized with many thousand operating hours of experience. Integration of this engine with a compressor and heat pump system components is considered appropriate at this time. Proof-of-concept for this GHP is expected in late 1985.

CONTRACT RESULTS TO DATE AND STATUS: The goals of this Phase of the project are to adapt the SPS V160 Stirling engine for natural gas use and develop a complete packaged prototype of a 10-ton commercial gas heat pump for laboratory testing. Using natural gas fuel, SPS has operated several engines for periods up to approximately 7000 hours. A net engine efficiency of 30% has been achieved at an engine shaft power output of 11.7 kW. The proof-of-concept goal for engine efficiency, set for late 1985, is 32%. During 1983, an HVAC manufacturer joined the SPS team in developing a kinematic Stirling engine heat pump. The manufacturer will help select components, test the engine, test the breadboard heat pump, and design and construct a fully operational heat pump prototype.

Two V160 engines have been installed and instrumented at the HVAC manufacturer's facility. Shakedown tests are under way prior to one engine being tested for the two year duration of the project and the second engine being mated to a compressor to construct a research prototype heat pump. Endurance testing and cycling testing of the remaining engines is continuing at SPS. Projected engine performance figures were verified by actual engine dynometer test results at SPS.

Brayton/Rankine Heat Pump
(Contractor: Airesearch)

OBJECTIVE: Develop a high efficiency, long life, cost-effective, natural gas-fired Brayton/Rankine heat pump for commercial applications.

ROLE IN PROJECT AREA STRATEGY: The Brayton/Rankine system represented one of the earliest of the commercial GHP concepts investigated by GRI. Large size Brayton engines have demonstrated the ability to operate many thousand hours without repair. The long life characteristics of this technology is very important for GHP applications.

CONTRACT RESULTS TO DATE AND STATUS: The long life and high reliability characteristics of gas bearing turbine technology was a very important element in the initiation of the Brayton/Rankine gas engine heat pump GHP development. Reasonable costs of the Brayton technology were anticipated in view of the emerging turbine technology for mass produced auto turbo chargers. In addition, engine efficiencies of 30% were expected to be achieved by 1980.

Unfortunately, prototype testing of the Brayton heat pump at two collaborating manufacturers was unsuccessful. Both test units failed within days of initiating testing. Unexpected corrosion in the sink heat exchanger fouled the rotating machinery. In addition, the low efficiency engines used in the testing were unable to operate in a stable mode.

A detailed Project Review meeting, held during the third quarter of 1983, concluded that the project should not be continued. Low-engine performance (14 to 18% efficiency), increased complexity of the GHP system, projected cost increases, and the inability of the system to effectively follow typical space conditioning loads were cited as the major decision criteria.

Rotary I.C. Engine Heat Pump
(Contractor: Teledyne)

OBJECTIVE: To develop a 10 RT commercial heat pump, using state-of-the-art rotary I.C. technology.

ROLE IN PROJECT AREA STRATEGY: This concept, which will involve the performance optimization of an existing rotary engine, represents a relatively low risk route to a near-term commercial heat pump.

CONTRACT RESULTS TO DATE AND STATUS: Work on Phase I was initiated on 11/1/84. Rotary engines for long-term testing have been acquired. Construction of the engine test facility was initiated.

TABLE I

GRI - SUPPORTED GAS HEAT PUMP R & D

Residential Concepts	Contractor	Target Size Range	Current Key Issues	Projected Dates Field Test	Projected Dates Commercialization
Dual-Cycle Absorption	Battelle	1-1/2 - 3 Ton Cooling 70,000 Btu per hour Heating	Solution Pump Corrosion Heat Exchanger Cost	Late '87	Early '89
Double-Effect Absorption	Columbia Gas	3 Ton Cooling 70,000 Btu per hour Heating	Solution Pump Parasitic Power Heat Exchanger Cost	Late '87	Early '89
Free Piston Engine Driven	MTI	3 Ton Cooling 70,000 Btu per hour Heating	Engine Life Maintenance Compressor Efficiency	Late '88	Late '90
Internal Combustion Engine Driven	Battelle	3 Ton Cooling 70,000 Btu per hour Heating	Engine Life Manufacturing Constraints Maintenance	Late '88	Late '90
Commercial Concepts					
Internal Combustion Engine Driven	Teledyne	10 Ton Cooling 250,000 Btu per hour Heating	Engine Life Maintenance Compressor Efficiency	Early '87	Early '88
Kinematic-Stirling Engine Driven	Stirling Power Systems	10 Ton Cooling 250,000 Btu per hour Heating	Engine Life Maintenance Compressor Efficiency	Early '87	Early '88

TABLE II

COP PERFORMANCE TARGETS OF GRI-SUPPORTED GENERIC TECHNOLOGIES

	Commercial Target		Achieved to Date	
	(H)	(C)	(H)	(C)
I.C. Engine-Driven	1.70	1.00		
Stirling Engine-Driven	1.80	1.10	1.40	1.00
Absorption	2.00	1.10	1.80	0.80

Figure 1

GRI RESIDENTIAL/COMMERCIAL HEAT PUMP FUNDING

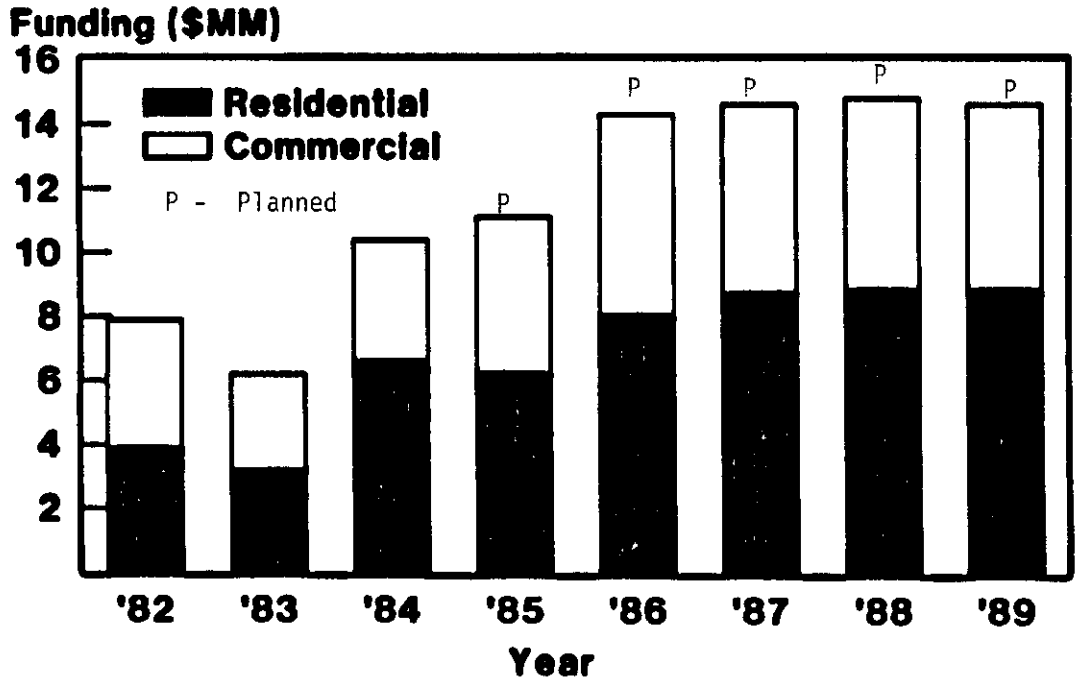


Figure 2

Flow Diagram for Residential Heat Pumps and Prime Movers Project Area (4.1.3)

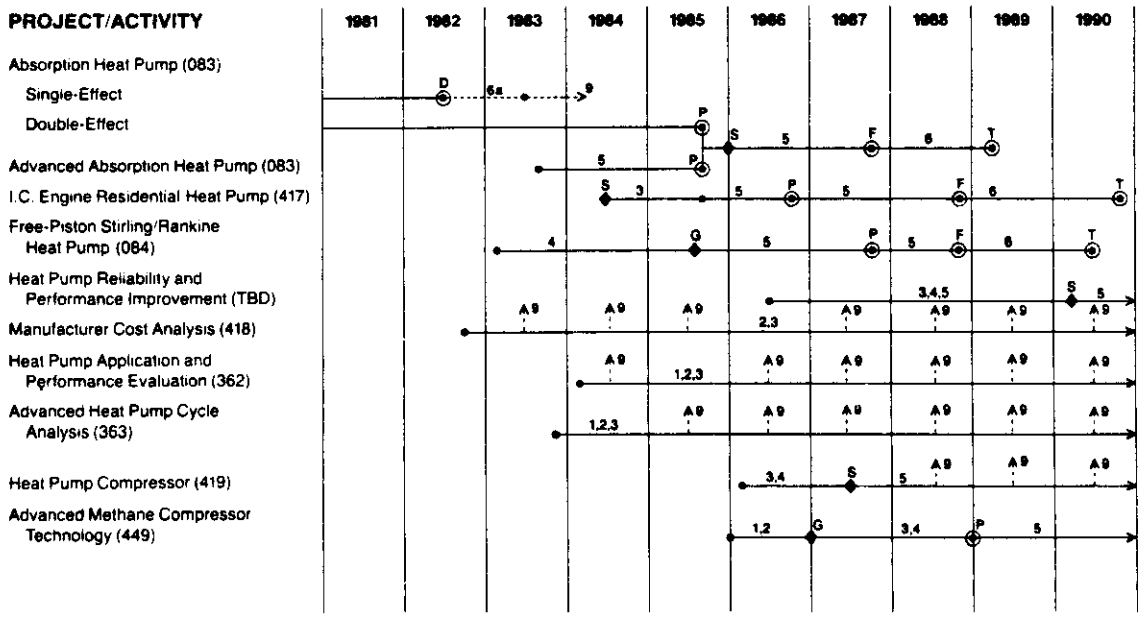
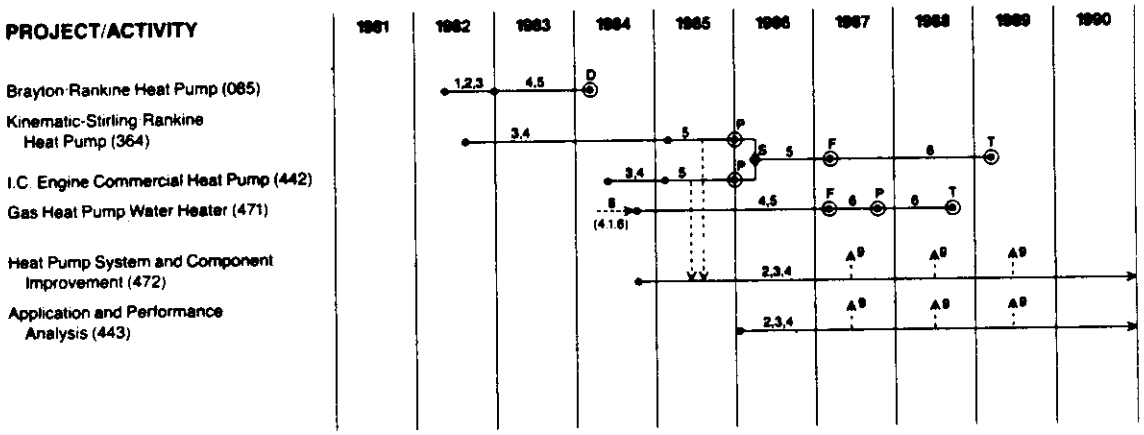


Figure 3

Flow Diagram for Commercial Heat Pumps and Prime Movers Project Area (4.1.4)



HEAT-ACTUATED HEAT PUMPS—III

Advanced Absorption Heat Pump Cycles

Robert C. Reimann

Carrier Corporation

Syracuse, New York 13221

Robert C. Reimann received B.S. and M.S. degrees from Syracuse University and is presently a senior engineer in the Research Division of the Carrier Corporation. He has worked in the absorption field for the past 8 years, primarily responsible for the development of a family of prototype low-temperature water and air-cooled absorption chillers ranging in size from 3 to 200 tons. Reimann is an active member of the American Society of Heating, Refrigeration, and Air-Conditioning Engineers Committee on Heat-Activated Machinery and is the holder of five U.S. patents.

ADVANCED ABSORPTION HEAT PUMP CYCLES

Robert C. Reimann
Carrier Corporation
Syracuse, New York

Objectives

The main goal of this project is the development of significantly improved absorption machinery. This was to be directed specifically towards the development of a direct natural gas-fired absorption heat pump with both heating and cooling efficiencies advanced beyond current commercial offerings. These were to be accomplished through the use of novel and superior cycles or by improved component performance. To keep the focus on machinery, the stipulation was agreed that candidate absorbent-refrigerant combinations were to be selected from among those whose properties were adequately known to permit cycle calculations and preliminary machine layouts.

Description of Phase I Work

This project began by undertaking two basic tasks:

1. An assembling of as broad a selection of possible candidate absorption heat pump cycles as our ingenuity permitted. Input to this included known single- and two-stage machine designs, cycles suggested on theoretical bases by E. Altenkerk and others, and cycles deduced by the application of the methodology of G. Alefeld. Some screening was possible on the basis of estimated potential gains and the increased complexity (closely related to cost) necessary to achieve them.

Eight possible cycles were identified as candidates for screening.

2. An assembling of possible absorbent-refrigerant combinations applicable to heat pump service. In general, two types of information are needed for a cycle calculation. First, and generally more readily available, are the thermodynamic data, vapor pressure-composition-temperature relationships, enthalpy, and specific heat capacity. Second, one needs heat and mass transfer rate information as well as physical properties such as viscosity and density.

We were somewhat disappointed in the availability of absorbent-refrigerant information but did arrive at seven pairs for which enough information was available to justify consideration in the screening process. After screening, two cycle-chemical pair combinations were selected as "most promising" for a machine design, construction, and test phase.

Method of Screening

The possible combinations of fluid pair and cycle were quite numerous, but fortunately, a great number could be eliminated by inspection; i.e., water cannot be used in an ambient-heated evaporator. The combinations which survived were then screened by superimposing each prospective cycle on the appropriate Duhring plot (vapor pressure-composition-temperature diagram) on which crystallization limits were also drawn. Non-operational and marginal cycles were easily identified, and those that remained, scrutinized based on the following considerations:

1. Peak generator temperature for ambient temperatures of -8.3 C (17 F) (lower ARI heat pump rating point) and -23.3 C (-10 F). The latter temperature was considered desirable for industrial applications but potential systems were not eliminated for poor performance at this point. Peak temperature is a key comparative factor because most two-stage absorption machinery is limited by materials restrictions in the high-side generator. This peak generator temperature is strongly affected by the selection of the temperature at which heat is delivered to the load. Therefore, performance data for compression heat pumps in the commercial size range was used as a guide in screening estimates and cycle calculations.

2. Chemical stability of the system at the upper generator temperatures.

3. Absorbent solubility -- to safely avoid crystallization throughout the cycle temperature and composition range.

4. High-side pressure of the system -- in order to avoid problems of pump design and power, vessel wall strength, codes, etc.

5. Low absorbent temperatures -- some cycles considered require absorbent solution operation at near-ambient temperatures.

6. Increased problems of mass transfer and non-condensible accumulation in the absorber if a high boiling point refrigerant is used at low evaporator temperatures.

7. Physical layout and cost -- an assembly sketch was made for each of the final combinations considered to establish mechanical and cost feasibility and to disclose any unusual or unexpected design or control problems.

Final Qualification

Initial planning had envisioned a computer simulation program to carry out performance mapping of the few systems which would survive the screening process and appear to show merit. Experience proved, however, that both thermodynamic and heat transfer data were so lacking that detailed computer simulations would yield results of very limited value. Cycle calculations were therefore done only to a limited extent, pending the availability of numerical design data.

The final result of the selection process resulted in the choice of a somewhat novel "dual-loop" cycle which double-couples two independent single-stage machines to produce two-stage efficiencies. Shown in Fig. 1, the cycle is characterized by the coupling of both the high-side condenser and absorber to the low-side generator, and producing heat to the load as the sum of the low-side condenser and absorber minus that required by the high-side evaporator. Since the two loops are isolated from each other with respect to any mass transfer, different absorbent-refrigerant pairs can be used in each loop thus permitting considerable freedom in selection of suitable pressures, in ability to utilize pairs of limited thermal stability, isolation of flammable refrigerants or absorbents from the direct-fired generator and independent selection of pairs for optimum efficiency of the cycle.

Lithium bromide-water will be used in the upper temperature loop because of the demonstrated chemical stability in excess of 150 C (300 F), high latent heat refrigerant and proven experience. Methylamine will be the lower temperature loop refrigerant, dissolved in a 60% lithium bromide-water solution absorbent. This choice was made on the basis of anticipated chemical stability and an acceptable set of pressures.

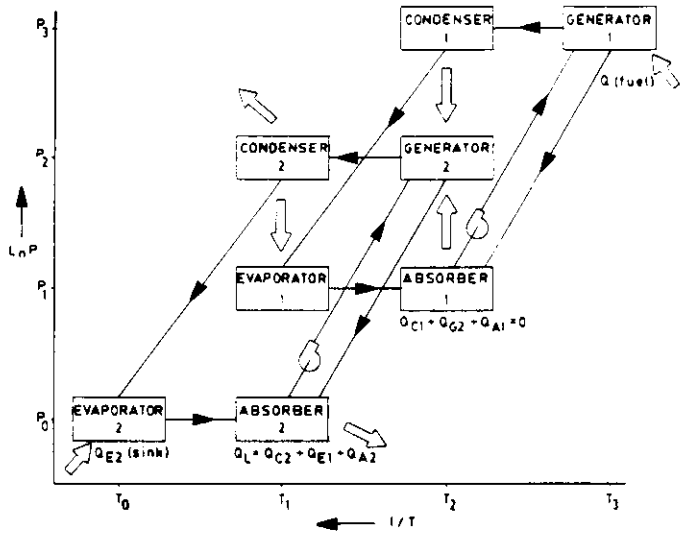


Fig. 1 Dual Loop Cycle

Calculated Performance

In the heating mode, at an ambient of -8.3 C (17 F) and 32 C (90 F) supply air to the load, a high side generator solution temperature of 146 C (295 F) is predicted with a thermal COP (including combustion) of 1.95.

In the cooling mode, with 29.4 C (85 F) tower water and 7.2 C (45 F) chilled water to the load, a thermal COP for cooling (including combustion) of 1.49 at a generator solution temperature of 137.2 C (280 F) is calculated. With further utilization of flue gas heat (as in the heating mode), the thermal COP in cooling could be raised to about 1.7.

These results, even allowing for a fair amount of uncertainty in the methylamine-LiBr-H20 thermal data, appear to be safely within our design goals and are not alarming with respect to corrosion or thermal decomposition.

The absorption heat pump design following was sized for 58.5 kW (17 tons) heating at the -8.3 C (17 F) ambient air rating point.

Machine Design

The physical layout, consisting of five basic shells is shown in Fig. 2. Heat required by the low-temperature generator is supplied, in part, by condensing the high-temperature loop refrigerant inside the generator tubes. The remainder is furnished by the high-temperature absorber by means of a heat pipe, thus negating the need for a secondary fluid loop pump.

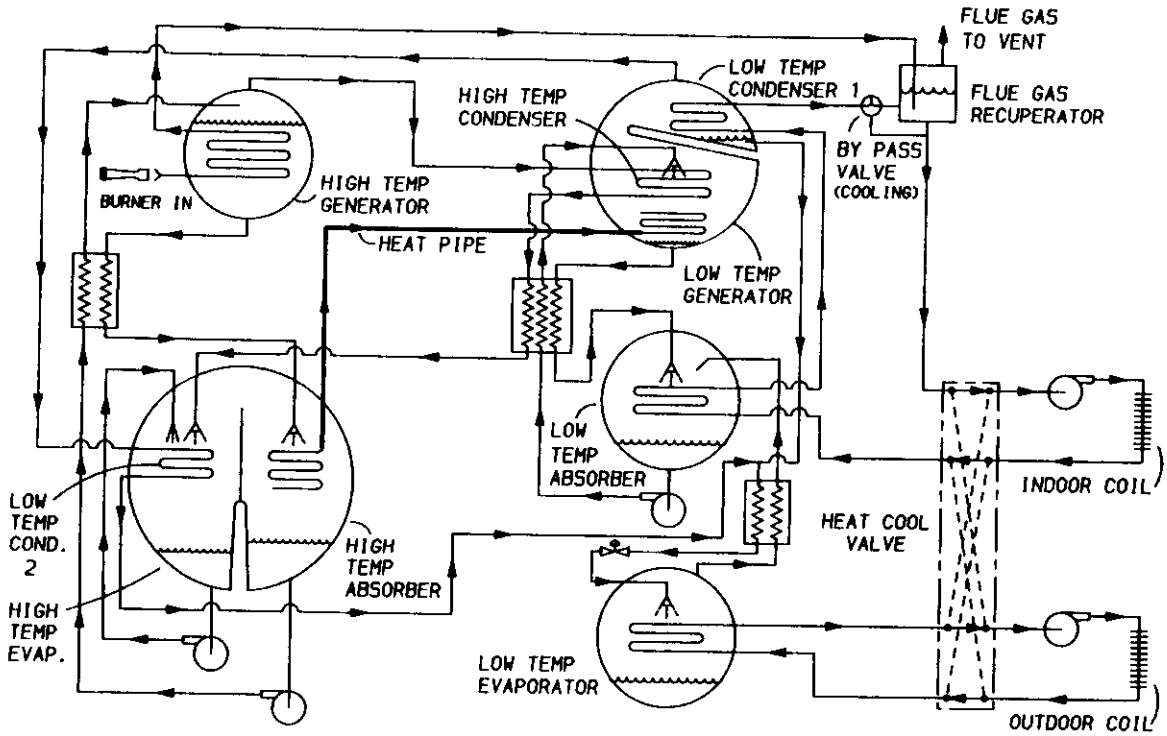


Fig. 2 Dual Loop Heat Pump Schematic

Heat required by the high-temperature evaporator is furnished by condensing the necessary portion of the low-temperature refrigerant vapor, supplied by the low-temperature generator, inside the evaporator tubes.

As in a conventional water-to-water heat pump, secondary fluid loops and pumps communicate with indoor and outdoor heat exchangers through a reversing valve. A condensing flue gas recuperator cools the flue gas and provides additional heat to the load in the heating mode.

Future Activities

Anticipated activities in Phase II include:

1. Testing and evaluation of fluids with respect to chemical stability and transport properties as well as testing of potential "backup" candidates.

2. Construction and testing of a single-stage machine as an intermediate step in order to reduce the risks and unknowns of the assembly of too many unproven technologies simultaneously. This direct-fired machine would employ a 1:1 mole ratio of zinc bromide/lithium bromide as the absorbent and methanol as the refrigerant and would establish the direct-fired generator design, test the flue gas recuperator, and serve as a test bed for data collection. Preliminary cycle calculations indicate that this system's performance, though marginal in comparison with the dual loop system, has potential in meeting the project goals as well as use in other applications.

3. Design, construction and testing of the advanced dual-loop machine employing data from 1. and 2. above.

A possible third phase is anticipated in which a packaged, proof-of-concept prototype would be constructed from the results of Phase II.

Publications

"Advanced Absorption Heat Pump Cycles", Wendell J. Biermann and Robert C. Reimann, Phase I Final Report, Contract No. DE-AC05-840R21400, U.S.D.O.E. (1984).

Analyses of Advanced Residential Absorption Heat Pump Cycles

Benjamin A. Phillips

Phillips Engineering Company

St. Joseph, Michigan 49085

Benjamin A. Phillips received the B.S. degree in chemical engineering and the M.A. and Ph.D. degrees in physical chemistry from the University of Texas. His professional experience has been primarily in research and product engineering, refrigeration, air conditioning, and heat pumps (mainly absorption, but compression as well).

Prior to his present affiliation, he worked with Servel, Inc. (gas and electric refrigerators), White Sands Proving Grounds (U.S. Army missiles, propulsion, structures, and warheads), and Whirlpool Corporation (absorption refrigerators, absorption and compression air conditioners, and heat pumps). He is presently working with absorption heat pumps, with both organic and inorganic working fluids.

ANALYSES OF ADVANCED RESIDENTIAL ABSORPTION HEAT PUMP CYCLES

B. A. Phillips

Phillips Engineering Co.

St. Joseph, Michigan 49085

Introduction

This report refers to work done primarily as Phase I of Oak Ridge Contract #86X-17497C, Development and Proof Testing of Advanced Absorption Refrigeration Cycle Concepts.

The overall purpose of this project is to develop a gas fired absorption heat pump for residential and small commercial applications that will produce a coefficient of performance of at least 1.6 Btu heating and .7 Btu of cooling per Btu of gas input. Being for residential operation, these goals refer to air-to-air heat pump operation at the rating conditions of 47°F outdoor ambient in heating mode and 95°F ambient in cooling mode.

For year-round residential and small commercial use other requirements were added. One was that the heat pump itself be able to provide all the heating and cooling required by the building over the majority of temperatures experienced in the United States (from -10°F to 110°F) without resource to supplemental heat. It should also be potentially competitive with existing space conditioning products in cost, reliability, operating life, etc.

The heat pump should be capable of meeting all national codes for residential installation. Due to the limitations of existing absorption fluid pairs, this requirement has made it necessary that air cooled absorption systems be outdoor products and that the heating and/or cooling effect be transferred to the conditioned space by means of a suitable secondary fluid. The additional temperature drops required for that heat transfer reduce the system coefficients of performance somewhat.

The means of making major improvements in gas fired absorption systems include the following categories:

1. Higher efficiency cycles
2. Better fluids
3. Higher flue efficiencies

The Phase I investigations related primarily to item 1. Development of new fluid pairs is not a part of this project. It is only required that a fluid pair with well established physical and thermodynamic properties be picked. Item 3 was left for later phases, except that the flue and insulation loss objective was established at 10% (from the 25% of past systems). That objective determined that the minimum cycle COP's should be 1.78 heating and .78 cooling.

Background

The Basic Single Stage cycle and the Absorber Heat Exchange (AHE) cycle have been used in production air-cooled, residential air conditioners at gas-fired, cooling COP's ranging from 0.3 for the single stage cycle to 0.5 for the AHE cycle. At a 25% flue and insulation loss, the 0.5 AHE COP represented a cycle COP of 0.67. The AHE cycle has also been used at that cycle COP in past experimental gas heat pumps of residential size, with organic fluids (1) as well as with ammonia/water (2).

Phase I Investigations

The purpose of Phase I was to evaluate high efficiency absorption cycles and to select the cycle and working fluid combination preferred for development of an advanced absorption test unit of residential size in Phase II. The preferred cycle was to meet or exceed the minimum COP's and to be potentially capable of evolving into a reliable, marketable product.

Fluid Selection

A primary step in the Phase I study was the selection of the fluid, or fluids, to be used in the comparative evaluation of the cycles. The original plan was to make a first screening using one pair of fluids, and the same operating conditions, for all cycles. Fluids that might match the cycles better could then be used in subsequent comparisons.

The following factors entered into the choice of fluids for the screening process.

a. As an air-to-air system, the refrigerant had to be suited to evaporation temperatures as low as -10°F or -20°F. Pairs using water as the refrigerant, H2O/LiBr, H2O/LiBr-Glycol, etc., and others that might freeze at those temperatures were thus not applicable.

b. To take maximum advantage of the Carnot potential, operation to high boiler temperatures was desired. Operability to temperatures of about 400°F was specified because those temperatures have been used in long lived absorption refrigerators. That temperature eliminated most organic fluids. Most, including fluorocarbons have been found to exhibit some decomposition on operation at 350°F (3, 4).

c. Potential flammability and toxicity of fluids were also objectionable, both from the standpoints of codes and of manufacturers' willingness to use them in residential applications.

d. Availability of adequate thermodynamic and physical property data was necessary.

No absorption pair was found to meet the requirements completely. The ammonia-water combination come closest, and was well ahead of the next best. It is well suited to the total temperature range contemplated. Ammonia has a flammability that is quite limited. It is also difficult to ignite. Its acute toxicity limits are well known, it appears to have no long term toxicity. Codes for its use in residential and related applications have been in place for many years. Thermodynamically, water and ammonia are, individually, probably the best characterized of fluids. Solution data are not as complete as often wished, but, with water/lithium bromide, are probably more complete than those of any other pairs.

However, even ammonia/water places constraints on cycle potentials. Depending on the cycle, the constraints may include: the temperature limit of the water saturation curve, the limits of the temperature spread between the ammonia and water vapor pressure lines, the critical pressure and temperature of ammonia, and generally, the need for outdoor installation and for a secondary fluid to transfer heating and cooling to the house air.

Property improvements needed by the individual cycles will be mentioned in the cycle discussions.

Cycles Investigated

The cycles chosen for evaluation in Phase I represented a slice out of the total spectrum of absorption cycles. The slice was bounded on one side by the minimum COP's, and on the other by the cost and complexity limits placed by the residential application. The latter boundary was, of necessity, a hazy one and became, to some extent, arbitrary. Many of the potential multistage, multieffect cycles which, as outlined by Alefeld (5), could have higher COP's than those considered, had to be left out because the cost could be foreseen to be excessive for the residential market.

This limit was placed, partly because even simple absorption cycles have tended to be more expensive than compression systems, and also because, as Figure 1 shows, the energy saving advantages of high COP's improve slowly beyond the 1.78 level.

Cycles Evaluated

Within those limits, the cycles chosen for evaluation were the following:

a. Double Effect Cycle
b. Resorber Augmented Cycle
c. Generator-Absorber Heat Exchange Cycle
d. Double Effect Regenerative (2R) Cycle
e. Variable Effect Cycle
f. Two Stage Generator-Absorber Heat Exchange Cycle

Double Effect Cycle

The Double Effect cycle utilizes the concept of multi-effect distillation used in the chemical industry, water desalination, etc. In multiple effect

operation the heat of condensation at a high pressure provides the boiler heat for a lower pressure effect. Because the solubilities of absorption fluids result in the boiler temperature being much higher than the condensation temperature, the number of effects possible in absorption is generally limited to two.

Double effect cycles can use three types of solution circuits. The three differ in the first effect condensing pressure, in pumping power, and in their COP's. All were checked in the preliminary heat balances. The cycle shown in Figure 2 was selected for the evaluation because it produced the best COP at an intermediate first effect pressure. Depending on the second effect pressure, solution circulation rate, absorber outlet temperature, etc. the first effect pressure can be lower or higher than that shown in the diagram. In these evaluations the second effect conditions were made equal to those of the other cycles.

In the cycle diagram of Figure 2, the heat input to the system occurs at line DF, the first effect generator. First effect condensation at A must occur at a temperature higher than that of the second effect generator D'F'. Second effect condensation occurs at the normal condenser temperature, A'. The two condensate streams are joined, and flow to the evaporator. All evaporation occurs at B, and all absorption at CE. The heat output of the cycle thus occurs at A' and at CE.

As seen in the diagram, the first effect and the second effect circuits are individually single stage cycles. The concentration of the weak liquid is not much below that of the rich liquid because the condensing temperature and pressure of the first effect is determined by the boiling temperature of the weak solution in the second effect.

In these evaluations, the designs and operations of the cycles were not limited to the basic concepts of the cycles. Wherever possible, means of increasing the COP's were incorporated. Mainly, heat available in the cycle was recuperated, being used within the cycle at the highest temperature possible. As an example, in the double effect cycle, the rectification heat from the first effect was used to aid the weak liquid in heating the rich liquid in a Triple-X manner. Since not all the rectification heat could be so used, the excess was added to the condenser heat as input to the second effect generator. The sensible heat of the first effect condensate was used to help heat the rich liquid of both circuits from the absorber temperature to the temperature of the second effect generator. AHE and other types of recuperation were also utilized wherever possible, so that, in the double effect cycle, no heat losses occurred other than those intended as the heat output from the absorber and the second effect condenser.

In this approach of maximum recuperation the other cycles gained less than the double effect cycle.

Resorber Augmented Cycle

In the Resorber Augmented cycle (6), a resorption-desorption circuit is added to a single stage cycle, to increase evaporation cooling. The cycle includes a normal condenser and evaporator, shown at A and B in Figure 3. The refrigerant vapor from the single stage evaporator is absorbed in a resorber which has a high absorbent solution circulation rate, making the average refrigerant concentration in that solution relatively high. The enriched solution from the resorber is then cooled, in a liquid heat exchanger, to the temperature of the normal evaporator. The pressure of the solution is reduced through an expansion device so the refrigerant can desorb from the solution at, or below, the temperature of the evaporator. The refrigerant is thus evaporated twice. The weak solution flows back to the resorber through the liquid heat exchanger while the refrigerant vapor flows to the main absorber at desorber pressure. The desorber-resorber circuit requires a second solution pump, as well as the addition of the resorber, desorber and liquid heat exchanger.

The Generator-Absorber Heat Exchange Cycle

The Generator-Absorber Heat Exchange (GAX) cycle (7, 8) utilizes recuperation of absorber heat beyond that possible in the Absorber Heat Exchange (AHE) cycle. In the AHE cycle, the recuperation of absorber heat is limited to the sensible heat required to raise the rich solution from the absorber outlet temperature to the generator feed temperature. Similarly, recuperation heating of the generator is limited to the sensible heat in the weak solution leaving the generator. In the GAX cycle, shown in Figure 4, those heat recuperation steps are used, but, by broadening the absorption and generation

temperature ranges, absorber heat is also transferred directly to the generator. This direct transfer can occur whenever the spread between the rich and weak solution concentrations is sufficient to provide a temperature overlap between the cool end of the generator temperature range and the hot end of the absorber temperature range. The heat of absorption at the overlap temperatures can all be utilized by the generator, but the overlap range is narrowed by the temperature differentials required for the heat transfer.

In Figure 4, the usable overlap between the absorber and generator temperature ranges is marked by the dashed lines enclosing the central arrows indicating the heat transfer. The single arrows outside that range represent AHE transfer.

Variable Effect Cycle

The Variable Effect cycle (9) also recuperates absorber heat to the generator, but in two steps. The cycle utilizes an intermediate pressure between the evaporator and condenser pressures. A cycle diagram is shown in Figure 5. The rich solution from the absorber is pumped to the intermediate pressure, at which pressure the rich liquid is heated in a desorber, GH, by overlap heat from the absorber. The vapor generated flows to a resorber, JK, at a higher temperature, but slightly lower pressure. The partially-stripped liquid from the desorber is then pumped along HD to the generator (operating at normal condenser pressure). Resorption of the desorbed vapor into the weak liquid enriches the weak liquid and produces heat, which is transferred to the generator. The enriched weak liquid then flows along JE, to the absorber. It flows through a liquid heat exchanger, partly heating the intermediate liquid from H to D.

In the heat balances made for this cycle, the heat of rectification was used to supply added heat input to the desorber as well as the extra heat needed by the intermediate liquid flowing from H to D. The rich solution was heated from C to G by absorber heat. The sensible heat of the weak liquid from F to K was transferred to the generator.

Due to the intermediate pressure operation of this cycle, the fraction of the absorber heat that falls within the overlap range is greater than in the GAX cycle at the same operating conditions, producing a potential COP advantage.

Two Stage GAX Cycle

The Two Stage GAX cycle (10) also recuperates absorber heat, transferring it to the generator. As shown in Figure 6, the cycle differs from the basic GAX in that the absorber-generator circuit is separated into two circuits, making the cycle equivalent to two single stage cycles, one at a higher generator temperature than the other. The absorber heat of the high temperature circuit supplies all of the heat input to the low temperature generator.

An advantage of the split arrangement is that it provides freedom to control the solution flow rates in the two circuits independently. This might allow a closer approach to Carnot conditions. It may also provide opportunities to simplify the heat transfer from absorber to generator. Obvious disadvantages are that two pumps and two relatively large heat exchangers are required.

As in other cycles, recuperation was used when possible. The rectifier from the high temperature generator was used as heat input to the low temperature generator and in heating the rich solution of the high temperature circuit.

Double Effect Regenerative (2R) Cycle

The 2R cycle (11) is also a type of two stage recuperative cycle. This cycle, and the Two Stage GAX Cycle, can be considered double effect cycles which use absorber heat instead of condenser heat for the second effect generator. As in the Two Stage GAX cycle, the condenser and evaporator are common to both stages, and there are independent generator-absorber circuits, but the high temperature circuit of this cycle is quite different. A simplified cycle diagram is shown in Figure 7.

The lower temperature circuit is a single stage circuit. The upper temperature circuit is a solution enrichment circuit. The rich solution is enriched by resorption processes at a series of pressure levels between the evaporator pressure and the generator pressure. The vapor to be resorbed is obtained from desorption stages in the weak liquid circuit that correspond to the resorption stages (but at slightly higher pressures so the vapor will flow to the resorbers). The enrichment used is sufficient to lower the temperature of the generator to below the temperature of the absorber. The heat of absorption then

serves to provide all the heat required by the generator. The external heat input to the cycle is the heat input to the desorber stages. The heat outputs from the resorber stages serve as the heat input to the low temperature generator, as indicated by the arrows.

Cycle Evaluation Criteria

The criteria used in the cycle evaluations included:

a. Coefficient of Performance
b. Pumping Power
c. Total Heat to be Transferred
d. Operational Range and Convertibility
e. Complexity and Probable Cost

Coefficient of Performance - While the .78 cooling COP and 1.78 heating COP were the lower limits, the objectives were to achieve the highest COP's possible within the requirements of the other criteria. The COP's were determined under a single set of conditions for all cycles.

Pumping Power - Parasitic power costs can be a significant fraction of the operating cost. The small solution pumps and motors for residential size systems tend to have low efficiencies (10% to 20% in past pumps). While solution pumps can be powered by the returning fluids, no commercially usable pump/hydraulic motor combination has yet been developed. Until that is accomplished, excessive pumping requirements can be significant factors in operating costs. The pumping powers were calculated as the theoretical pumping power.

Total Heat to be Transferred - The total heat to be transferred in a cycle is a first order indication of the heat transfer surface requirements, and thus also represents relative system costs.

Operational Range and Convertibility All refrigeration cycle and fluid combinations have limited ranges over which they can operate effectively. If the usable range of a cycle is less than the range of -10°F to 110°F proposed for operation without supplemental heat source, the operating range of the cycle, i.e. the low temperature cut-off point, would be a limiting factor. However, if a cycle with a cut-off temperature within the range were readily convertible to a cycle which would continue operation to low temperatures, the high cut-off point might not be as negative a factor.

Complexity, Probable Cost, etc. This category was to include, if necessary, other factors affecting the marketability of the ultimate heat pump, including unusual cost items, manufacturability, size, appearance, installation problems, probable life, maintenance and service requirements, etc.

Procedure

Component-by-component material and heat balances were made around each cycle at 47°F ambient conditions. The refrigeration and heating COP's were calculated from those results. Using a heat output of 80,000 Btu/hr as a basis, the theoretical pumping power of each cycle was calculated from the operating pressures and the rich liquid flows. On the same 80,000 Btu/hr output basis, the total heat quantities to be transferred in the operation of the cycle were calculated. Those results are tabulated together with the operating pressure levels, number of pumps and number of major components.

Determinations were also made of the low temperature limit of operation of the cycles in terms of the evaporator temperature limit. If that limit was above -15°F, the possibility, and/or difficulty, of converting the cycle to one capable of operating to lower temperatures was investigated. The determination of the low temperature limit was made by heat balances in some cases, but most commonly by means of PTX diagrams at low temperatures. (In a number of cases, it became evident that increasing pumping power would probably make the cycles impractical at temperatures higher than the cycle cut-off temperatures thus determined.)
Based on the results of these analyses it was found possible to make a tentative selection of the preferred cycle. Further investigations were then made. Operation over the full operating range was reviewed for each cycle, the potentials for improving the cycle performance by variations on the cycle design and by use of fluids specific to the cycle, and means of eliminating cycle disadvantages were investigated. The final selection of the cycle for Phase II development was made from the combination of all the factors investigated.

Results

A tabulation of the results of the analyses is shown in Figure 8.

The COP's determined at the 47°F ambient conditions are listed both as heating COP's and as refrigeration COP's. Being on a net, or cycle, basis, the two COP's differ by 1.0. Since the refrigerating COP's are better measures of performance in upgrading low temperature heat than are the heating COP's, they will be used in much of the following discussion.

The Resorber-Augmented cycle had a COP of .83, above the .78 requirement, but the other five cycles all had significantly higher COP's.

That COP of .83 became the primary basis for elimination of the Resorber Augmented cycle, but, in addition, the cycle was not among the best in heat transfer surface or pumping power.

The cooling coefficients of the other five cycles were clustered together in the 1.03 to 1.11 range, providing little basis for preference of one over the others. Further investigations relative to COP potentials did not greatly change things, showing that the COP's of most, if not all, of the cycles could be improved by 20% - 30% by better fluids, cycle modifications, etc., but that the COP's would still remain close enough together to be a secondary factor in the selection. The choice therefore had to be made on the basis of the other criteria. Considerations that affected the standing of each of the cycles follow.

The Double Effect cycle is of major importance in large tonnage, water-cooled lithium bromide air conditioning. In this ammonia/water cycle, the air-cooled COP, though the highest in the group, could perhaps have been higher if the critical temperature of ammonia were not so close to the temperature of the first effect condenser. There were also uncertainties in the calculations, in that the fluid property data for the first effect calculations, at pressures of 1250 psia, were extrapolations. Most experimental ammonia/water data extends only to 300 psia with some PTy and PTx data to 500 psia and 700 psia respectively. Therefore the Double Effect calculations in Figure 8 are listed in parenthesis. It is planned to review the extrapolation equations before the final draft of the project report.

Major problem areas were encountered with this cycle in relation to the other criteria however. The disadvantages included: high pumping power, the large amount and complexity of heat transfer surface used to achieve the COP, the high cutoff temperature (which in practice would be raised by the rapid increase in pumping power as the evaporator temperature was reduced) and the difficulty in converting the cycle to an AHE or single effect cycle for operation at low temperatures.

Lower pressure refrigerants like the methylamines, trifluoro ethanol or R123a would reduce some of these problems, but they did not meet all the fluid requirements. A solution pump powered by a weak liquid hydraulic motor would also provide a partial solution. However, all these factors plus the developments required, together with expected system costs, caused this cycle to be eliminated, in spite of having the leading COP.

It appears that, for residential use, this cycle requires working fluids specially suited to the cycle and the operating temperatures. Requirements for the refrigerant include: a low freezing point, a low vapor pressure, a high critical temperature relative to the first effect condenser temperature, and a high heat of vaporization. A broad range of solubility in the absorbent might be used to reduce the cutoff temperature, perhaps eliminating the need for convertibility.

The other four cycles have much in common. They all recuperate absorption heat to the generator, and can all be improved by higher temperature fluids and other means. The Two Stage GAX cycle is equivalent to a GAX cycle, operating over the same temperature range, but divided into two solution circuits. The performance gains resulting from flow optimization were cancelled to a great extent by increased liquid heat exchange and rectification losses, leaving the COP only some 3% higher than that of the GAX cycle. The cycle requires two solution pumps and its heat transfer surface requirements are on the high side. Its cutoff point was within the operating range. Convertibility to a single effect cycle would be as difficult as for the Double Effect, requiring a group of three-way valves for interconnections, which could be expected to produce service problems. This cycle was therefore also eliminated.

The 2R cycle was found to be fundamentally more efficient, relative to Carnot, than the other cycles, because its heat input temperature was lower than the rest.

As indicated on Figure 7, the heat input temperatures, i.e. the temperatures of the stepped desorbers, peaked at slightly above the absorber peak temperature, being limited by the water vapor pressure line. Thus, this cycle was also limited by the working fluids, but at a temperature almost 100°F lower.

For gas firing the actual COP rather than the relative COP was the operative factor, however. The bases for selection again had to be the practical criteria rather than the relative COP. The cycle was found to require a relatively high heat transfer surface, two pumps, one multistage, and to be relatively complex. The most serious problem for the residential heat pump application was the high cutoff temperature and the virtual impossibility of converting to a single stage cycle unless a stand-by generator were added to the single stage part of the cycle. Thus, while very good for solar heating as intended, it was not suited to full range heat pumping.

The two cycles remaining, the Variable Effect cycle and the Generator Absorber Heat Exchange cycle, had the lowest, and equal, pumping powers. The COP's were found to be essentially the same. The potential for COP advantage of the VE cycle, that the overlap range is a greater fraction of the absorption temperature range, had been cancelled out by heat exchange losses in the two step process of transferring heat from the absorber to the generator, which also caused added rectification and liquid heat exchange losses. The large overlap did lower the cut off temperature of the VE cycle to -15°F, and below, depending on the intermediate pressure. It was thus the only one of the advanced cycles able to operate to the -10°F objective. The 5°F cutoff temperature of the GAX cycle required that it be convertible. It can be converted to an AHE cycle automatically by shutting off the fluid circulation circuit transferring heat from the absorber to the generator.

The VE cycle required 20% more heat and mass transfer surface and one more solution pump than the GAX cycle. The cycles had equivalent requirements for development of the absorber to generator heat transfer system. However, an area of special concern relative to the VE cycle, which did not apply to the GAX, was the controlling of the intermediate solution flows and of the intermediate pressure. Both the second solution pump (pumping the solution from the desorber to the generator), and the second weak liquid flow control valve (controlling the solution flow from the resorber to the absorber), must control those flows very accurately, matching the requirements of every operating condition. Otherwise, the operation of the desorber or resorber, or both, and hence, of the cycle as a whole, can be upset.

The optimum intermediate pressure was found to be a function of: solution concentrations, heats of absorption and desorption, heat transfer Δt's, etc. Operating at the optimum intermediate pressure was found to be specially important at the lowest evaporator temperatures. Consideration of these flow and pressure requirements led to the conclusion that development of the means of controlling the flows, and of establishing operation at the correct intermediate pressure at start up and of subsequently adjusting the pressure to changing conditions, were likely to be major problems.

As a result of these analyses the GAX cycle emerged as the cycle preferred for the Phase II development. It had development problems also, but the major problems, the heat and mass transfer problems of assuring that all of the available heat is recuperated into the cycle, and problems in developing reliable pumps and flow control valves, etc., would be part of developing the other cycles as well.

Additional data on the GAX cycle performance are shown in Figures 9 and 10. The effect of peak boiler temperature on cycle COP at 47°F ambient operating conditions is shown in Figure 9. As can be seen, the curve has a high slope, which emphasizes the importance of using high boiler temperatures in the GAX type cycles. Fluids allowing boiler temperatures above 400°F would be expected to cause the GAX cycles to cross the Double Effect curve.

In Figure 10, the COP's of the GAX cycle are shown as a function of the evaporator temperature and corresponding outdoor ambient temperature. The reduction in performance as the refrigeration work increases is evident. In the past, the performance curves of absorption cycles have commonly been less steep than those of compression. However, as seen here, when the absorption COP's are increased the slope of the curve increases toward those of compression systems. The COP's of the AHE cycle at low temperatures are shown. The AHE cycle can operate to evaporator temperatures of -40°F or lower.

Potential air conditioning performance is also shown. The COP at 82°F, at which data for SEER's is taken, can be seen to be about 1.26.

Better Fluids

A limited investigation of the use of the ternary combination of ammonia with 60% lithium bromide-water was made. The higher boiler temperatures made possible by that combination should raise the COP limits substantially. The PTX cycle diagram of Figure 11 shows the greater overlap possible. The diagram in Figure 12 also shows that the higher boiler temperatures also reduce the cut-off point to below -15°F.

It was not possible, during Phase I, to convert the enthalpy data (12) for this combination so that they could be used in heat balances in combination with ammonia/water. Therefore, accurate COP calculations could not be made. Calculations based on estimates believed to be conservative were made however. Those results are shown in Figure 13, indicating an important gain over the full range of temperatures.

Cost Estimate

A preliminary design for a 3RTon heat pump was made, using as a basis the past AHE cycle products. Drawings were made in sufficient detail for costing. A factory cost estimate of that system was made by Pioneer Engineering and Manufacturing Co. Those figures, based on production of 50,000 heat pumps per year, were used to estimate the installed cost, which fell within the range of installed costs of the various combinations of existing gas furnaces plus electric air conditioners.

PHASE II

The selection of the GAX cycle for Phase II development of a breadboard test unit using ammonia/water was accepted by DOE and ORNL. It was also proposed that if that program was successful, it should be followed by trials of the ammonia/lithium bromide-water combination.

The Phase II work was started by designing the breadboard test unit for operation with electric heat at 3RTon capacity. The unit has been built, and testing and development are starting.

Figure 14 is a picture of the bare absorption unit. It shows all the main components except the pump, which, will be an external pump used on past air conditioners. Figure 15 shows the insulated unit on the test stand. And Figure 16 the instrumentation side while in operation.

REFERENCES

1. J. Marsala, E. Whitlow, B. Phillips, Development of a Residential Gas Fired Absorption Heat Pump, ORNL/SUB/79-24610/3 (1983).

2. Unpublished Heat Pump Development, Phillips Engineering Co (1983).

3. B. Eiseman, Jr., ASHRAE Journal, 45, Dec. (1959).

4. K. Murphy et al, Development of A Residential Gas Fired Absorption Heat Pump, ORNL/SUB/79-24610/5 (1983).

5. G. Alefeld, Brennst-Warme-Krafft, V. 34, 64-74 February (1982).

6. K. Richter, Journal of Refrig. V.5 (5), 105, (1962).

7. R. Plank, Handbuch der Kaltetechnik, V. 7, Springer-Verlog. (1959).

8. B. Phillips, ASHRAE Transactions, V. 82, Part 1, 966, (1976).

9. K. Kaufman, Proc. 1981 Ann. Mtg. AS-ISES, V 4.1, 515.

10. A. Johnston, Solar Energy, V. 25, 243, 1980.

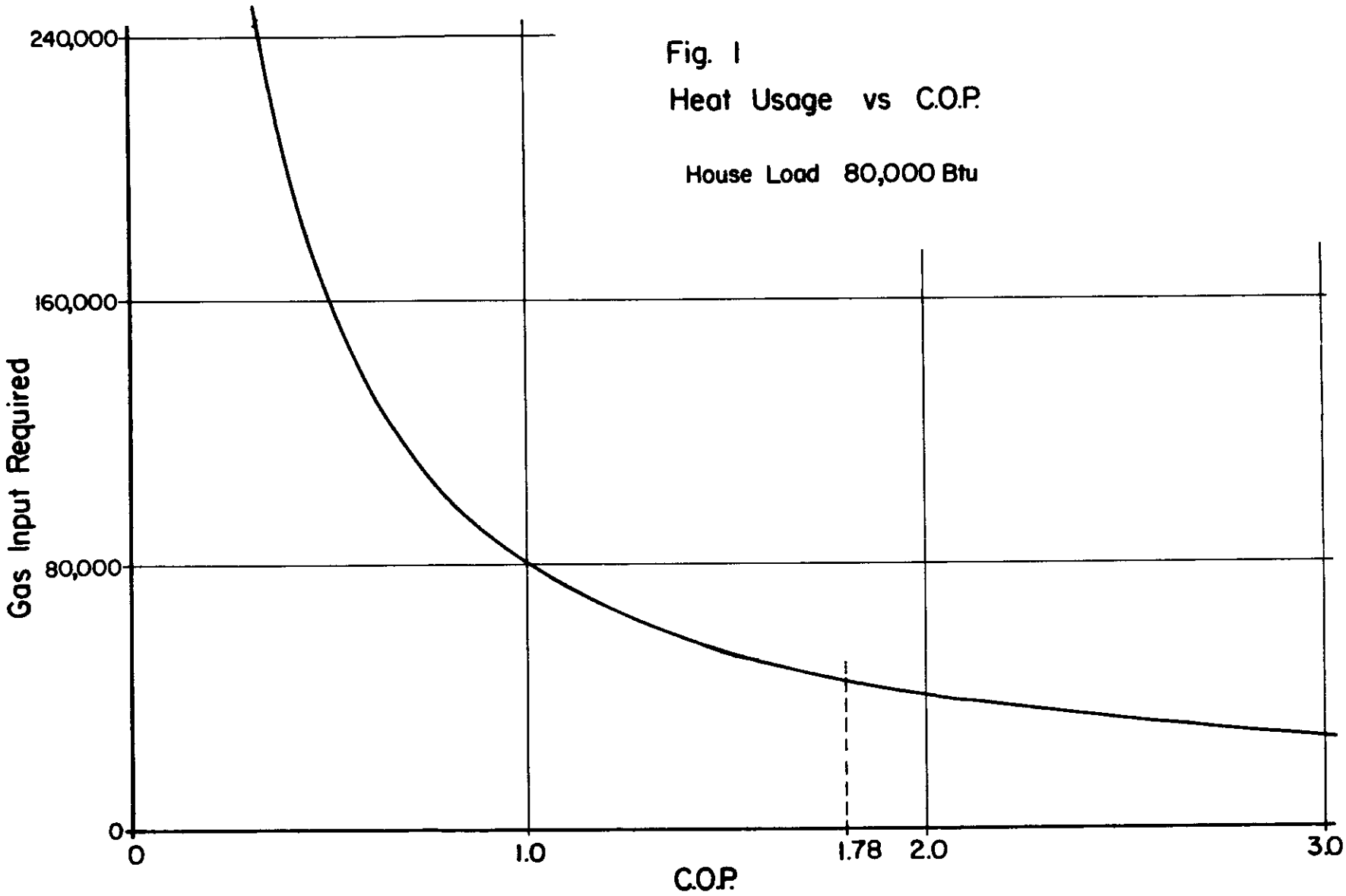
11. K. Dao, Conceptual Design of an Advanced Absorption Cycle, LBL-8405 (1978).

12. R. Radermacher, Working Substance Combinations for Absorption Heat Pumps, Ph.D. Thesis, Tech. Univ. of Munich, (1981).

Fig. 1

Heat Usage vs C.O.P.

House Load 80,000 Btu



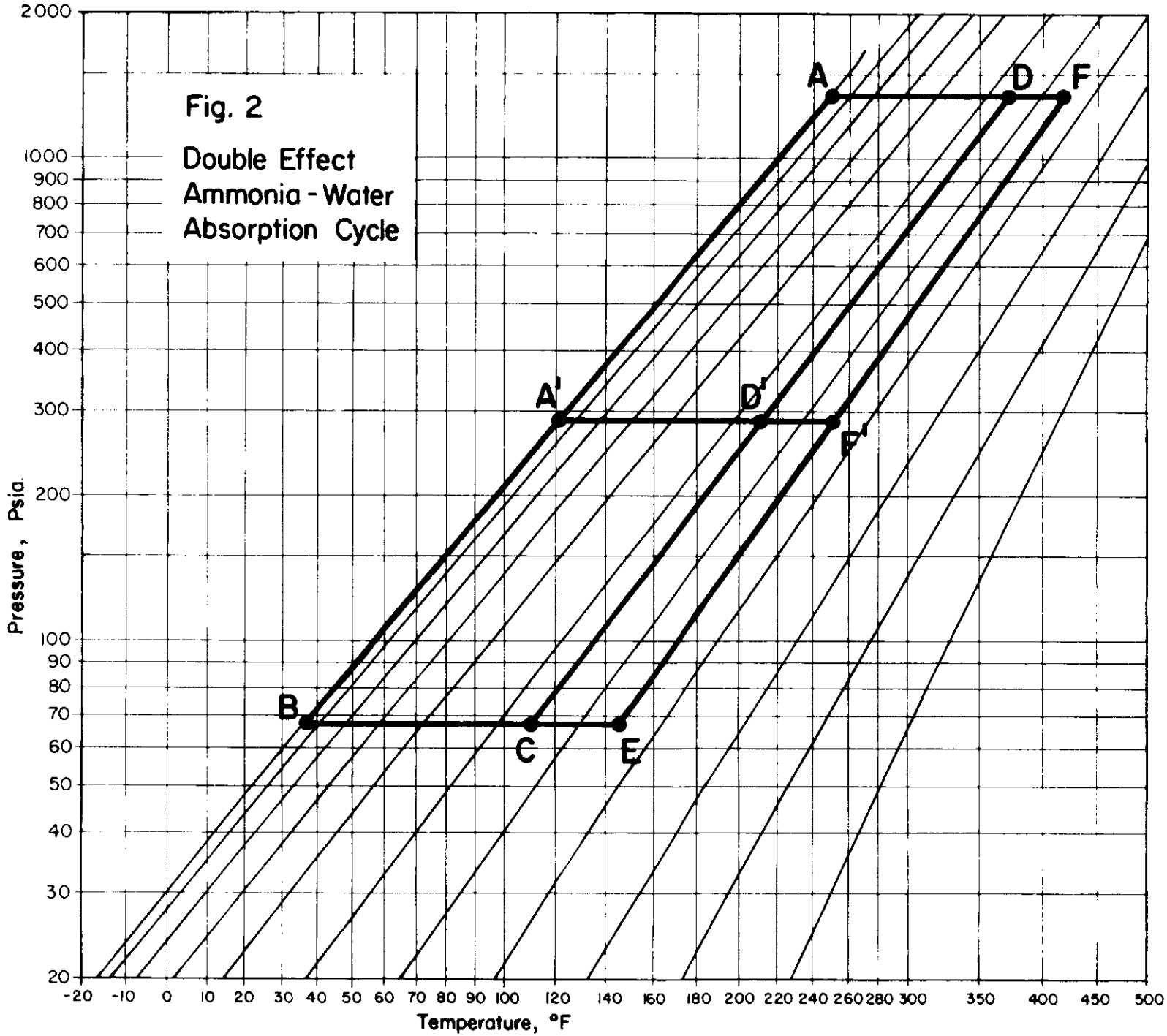


Fig. 2
Double Effect
Ammonia - Water
Absorption Cycle
A
D
F
A'
D'
F'
B
C
E
Pressure, Psia
2000
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
-20 -10 0 10 20 30 40 50 60 70 80 90 100 120 140 160 180 200 220 240 260 280 300 350 400 450 500
Temperature, °F

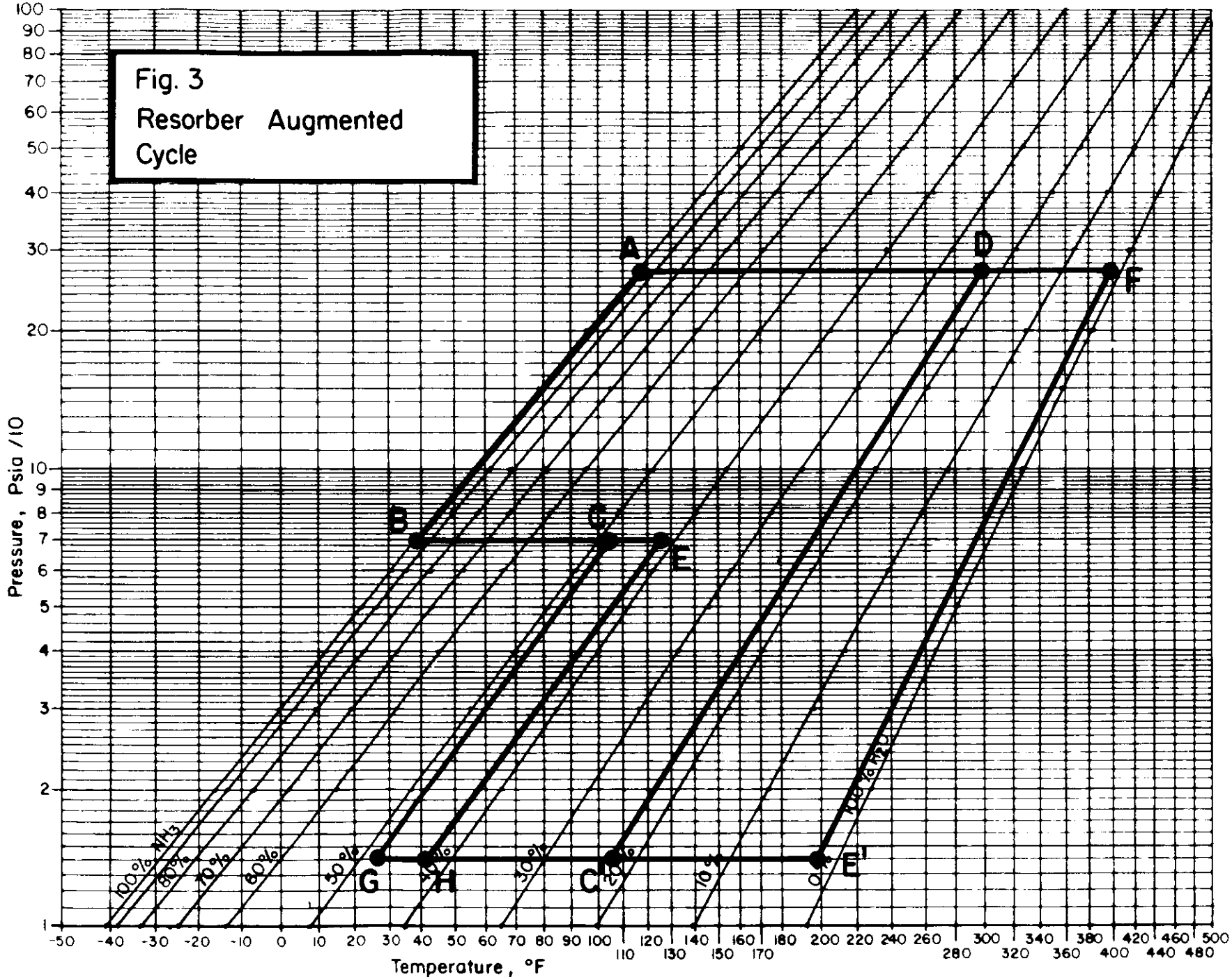


Fig. 3
Resorber Augmented Cycle
Pressure, Psia /10
Temperature, °F
A
B
C
D
E
F
G
H
C'
E'
100% NH3
80%
70%
60%
50%
40%
30%
20%
10%
0%

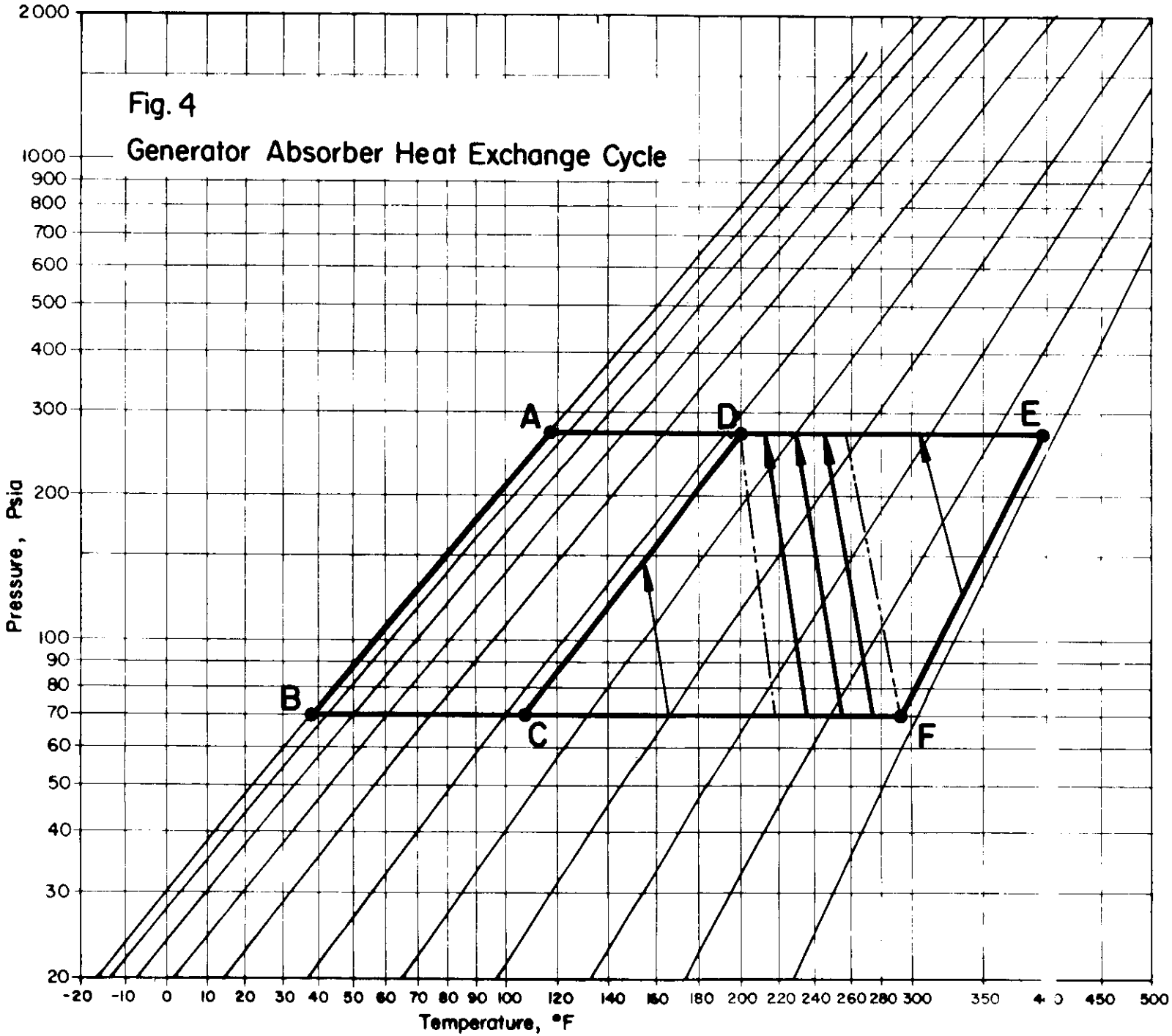


Fig. 4
Generator Absorber Heat Exchange Cycle
Pressure, Psia
2000
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
A
B
C
D
E
F
-20 -10 0 10 20 30 40 50 60 70 80 90 100 120 140 160 180 200 220 240 260 280 300 350 450 500
Temperature, °F

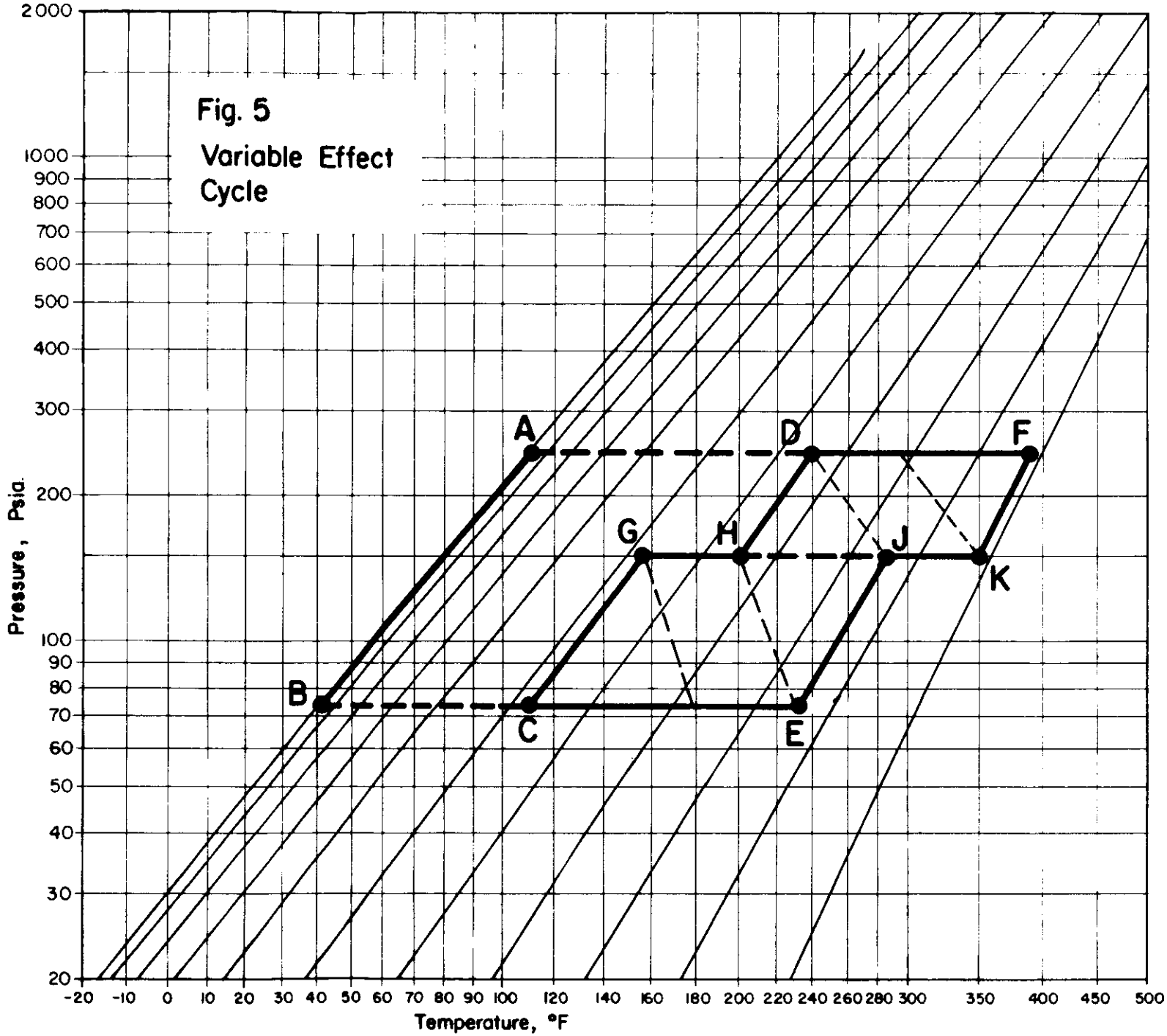


Fig. 5
Variable Effect
Cycle
Pressure, Psia.
2000
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
A
B
C
D
E
F
G
H
J
K
-20 -10 0 10 20 30 40 50 60 70 80 90 100 120 140 160 180 200 220 240 260 280 300 350 400 450 500
Temperature, °F

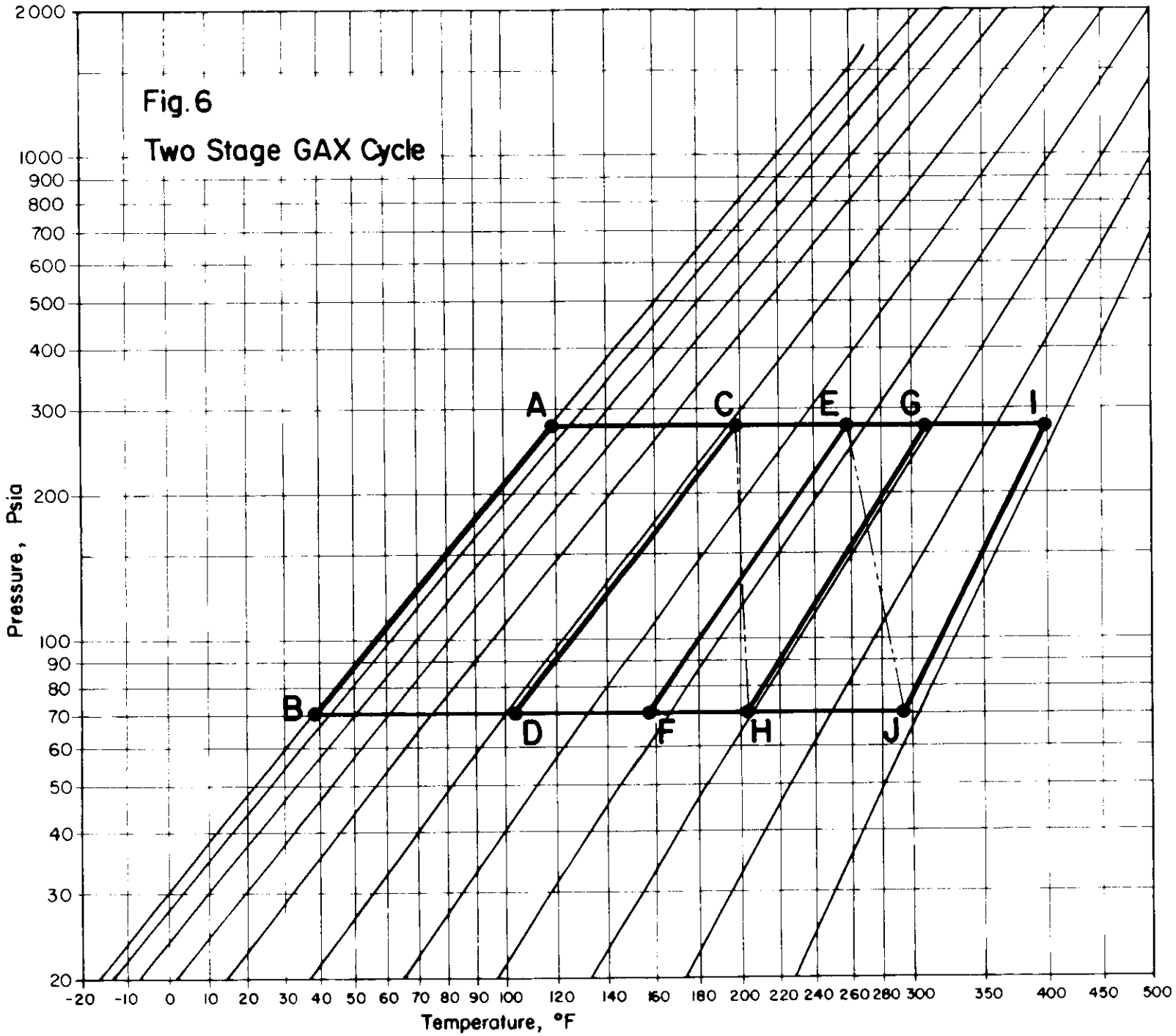


Fig. 6
Two Stage GAX Cycle
Pressure, Psia
2000
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
A
B
C
D
E
F
G
H
I
J
-20 -10 0 10 20 30 40 50 60 70 80 90 100 120 140 160 180 200 220 240 260 280 300 350 400 450 500
Temperature, °F

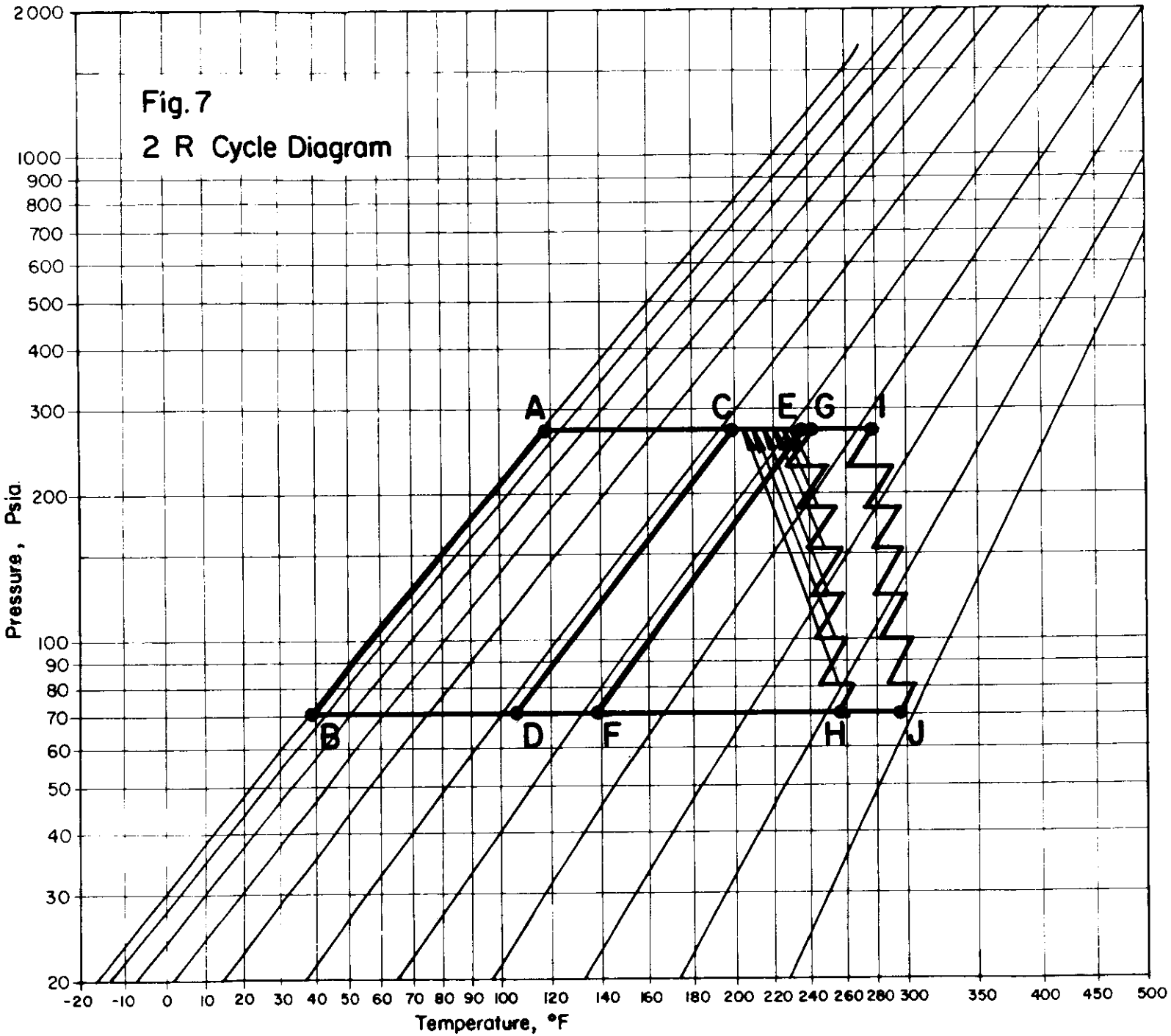
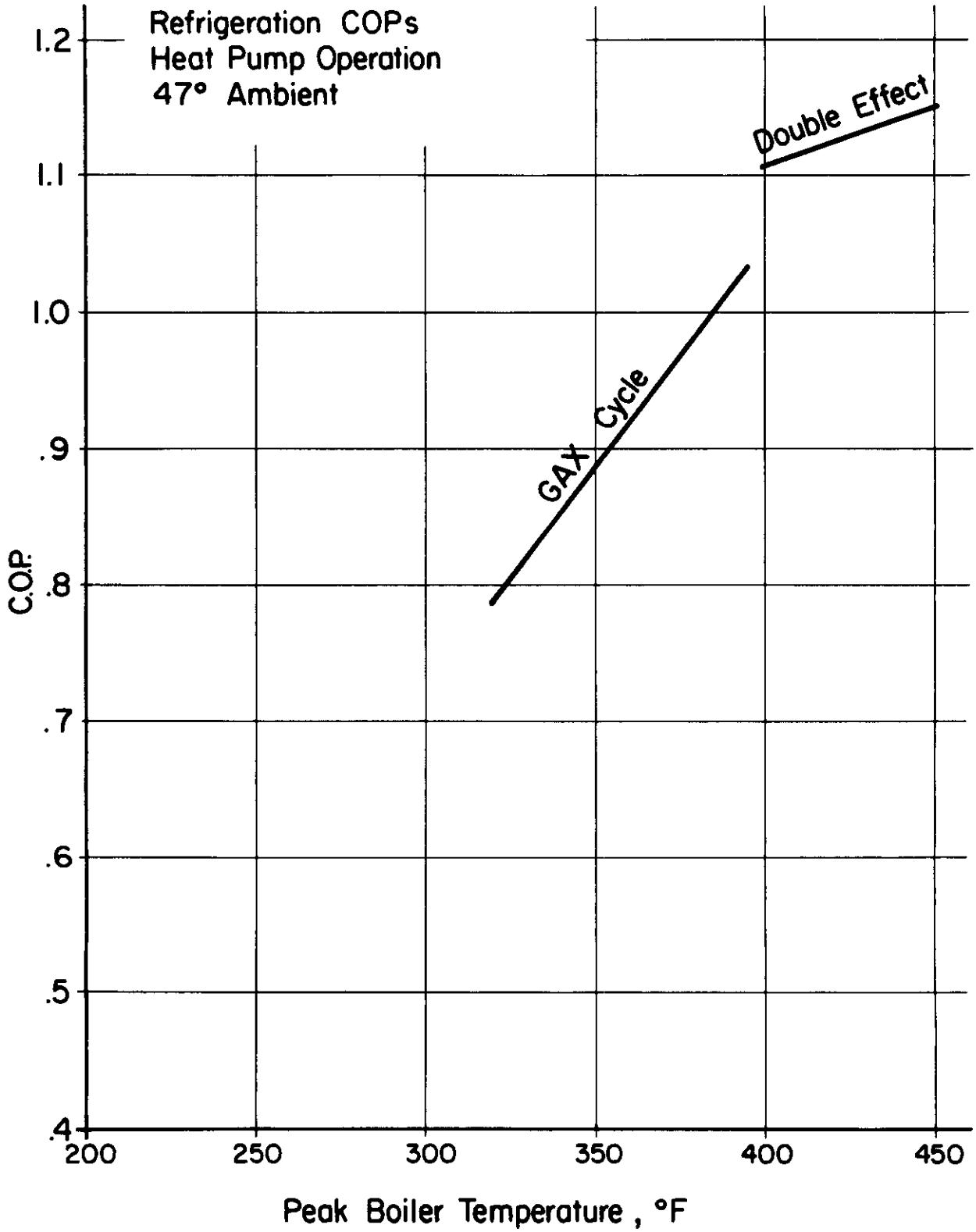


Fig. 7
2 R Cycle Diagram
Pressure, Psia.
2000
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
A
B
C
D
E
F
G
H
I
J
-20 -10 0 10 20 30 40 50 60 70 80 90 100 120 140 160 180 200 220 240 260 280 300 350 400 450 500
Temperature, °F

Fig. 8 CYCLE ANALYSES RESULTS (80,000 BTUH OUTPUT)

CYCLE	DOUBLE EFFECT	RESORBER AUGMENTED	GENERATOR ABSORBER HEAT EXCHANGE	VARIABLE EFFECT	TWO STAGE GAX	2R
COP REFRIGERATION	(1.11)	.83	1.03	1.04	1.06	1.03
COP HEATING	(2.11)	1.83	2.03	2.04	2.06	2.03
THEORETICAL PUMPING POWER WATTS	(388.4)	87.5	36	35	45	110
TOTAL HEAT TRANSFER BTUH	(391,550)	288,904	220,033	263,512	293,608	275,339
PRESSURE LEVELS PSIA	(67/285/1250)	15/68/265	68/265	68/135/265	68/265	68/265
NO. PUMPS	2	2	1	2	2	2
NO. MAJOR COMPONENTS	8	7	5	6	7	8
CUT OFF LIMIT °F	(20)	10	5	-15	10	20
CONVERTIBILITY	COMPLEX	COMPLEX	AUTOMATIC	NO NEED	COMPLEX	COMPLEX

Fig. 9


Refrigeration COPs
Heat Pump Operation
47° Ambient
Double Effect
GAX Cycle
C.O.P.
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
200
250
300
350
400
450
Peak Boiler Temperature, °F

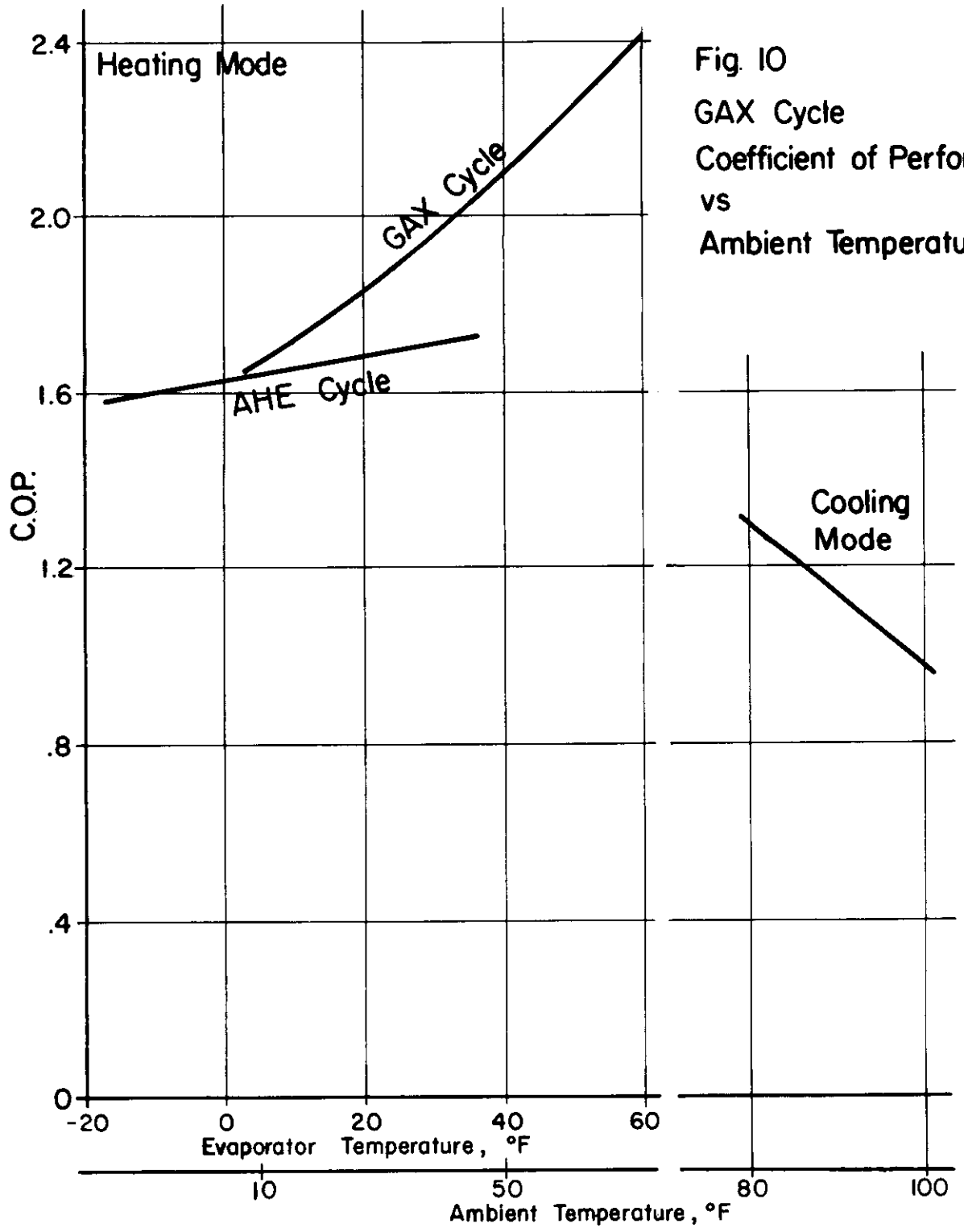


Fig. 10
GAX Cycle
Coefficient of Performance
vs
Ambient Temperature

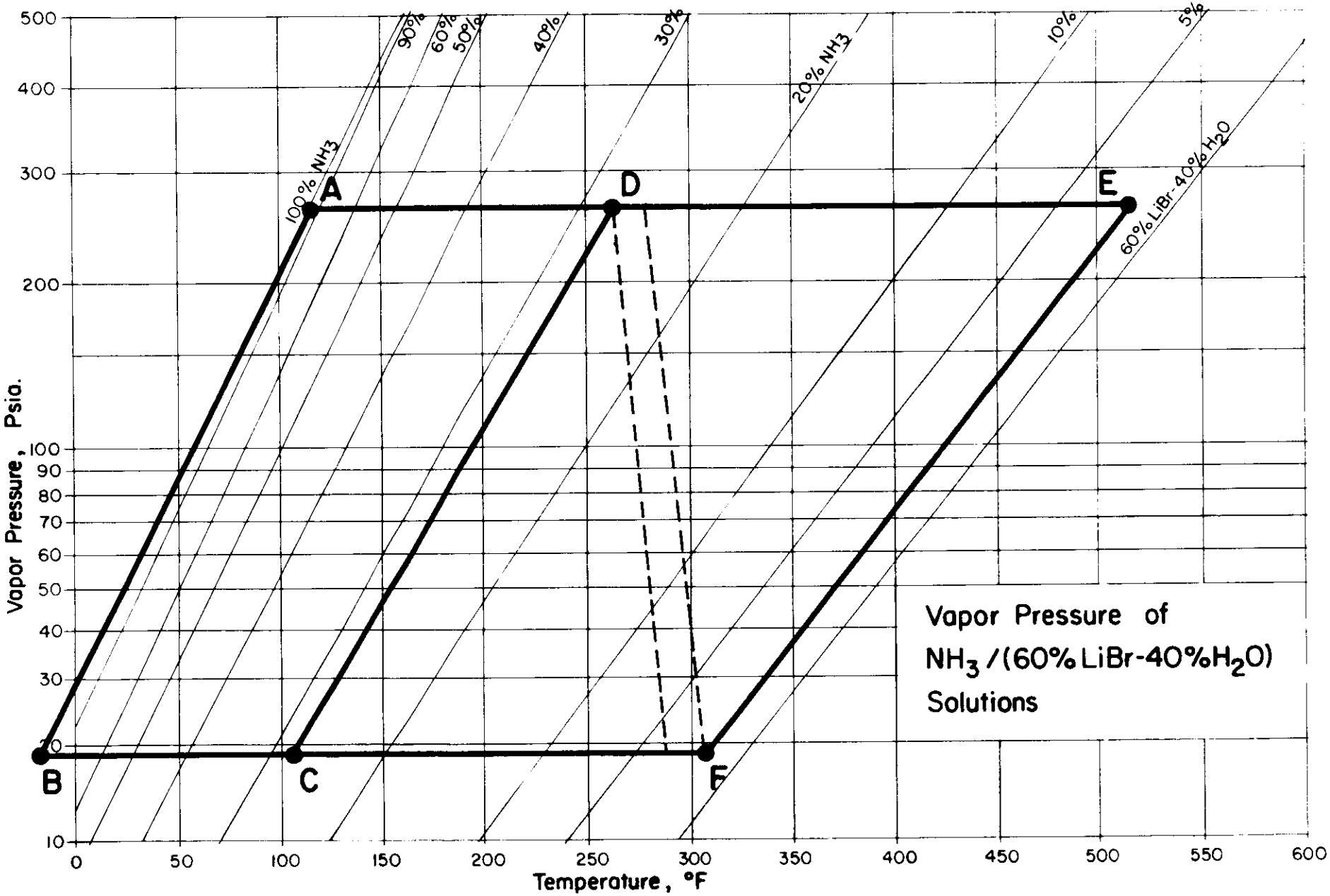


Fig. 11 GAX Cycle, Using NH_3/H_2O-LiBr at -10° F Ambient

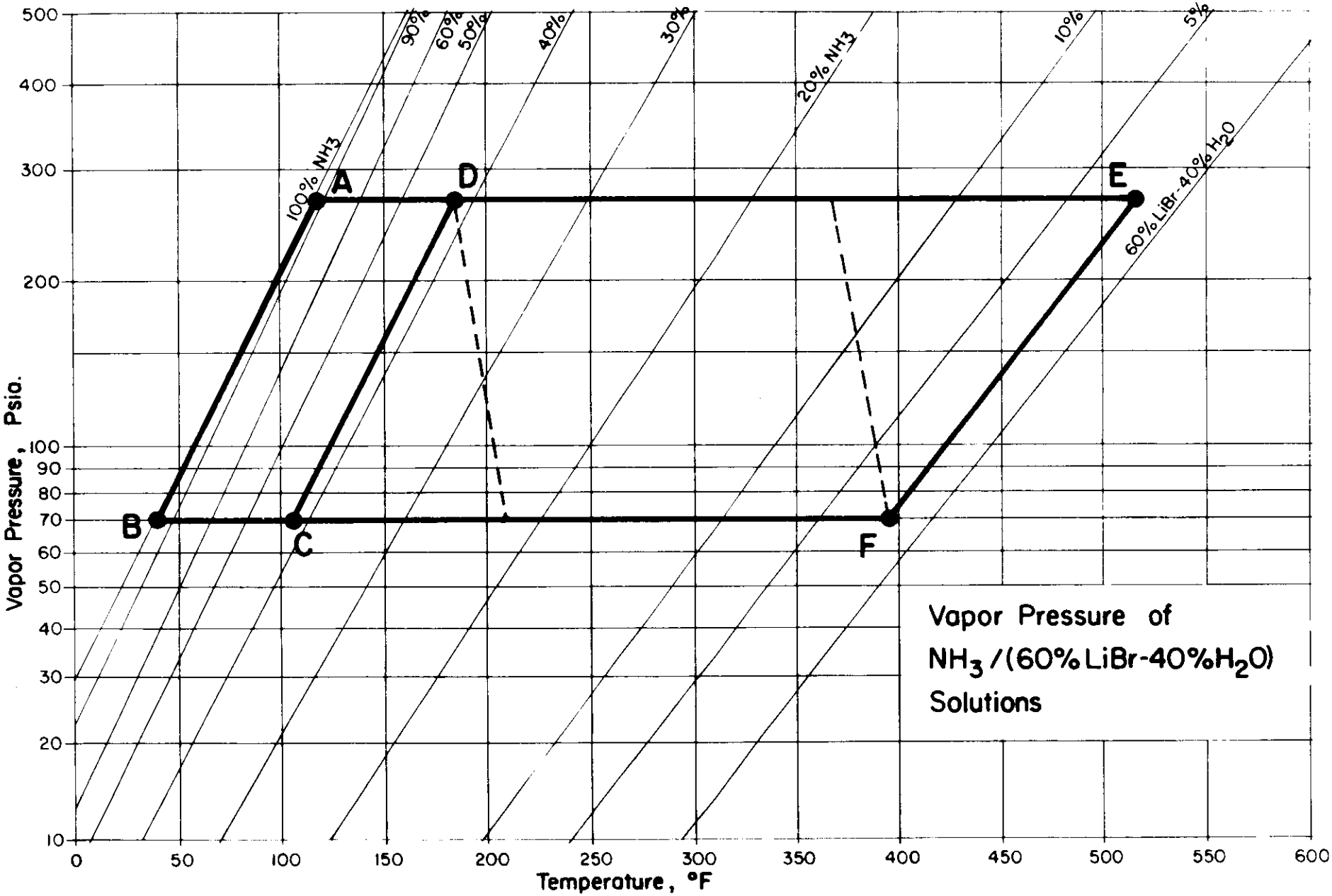


Fig. 12 GAX Cycle, Using NH_3/H_2O-LiBr at 47°F Ambient

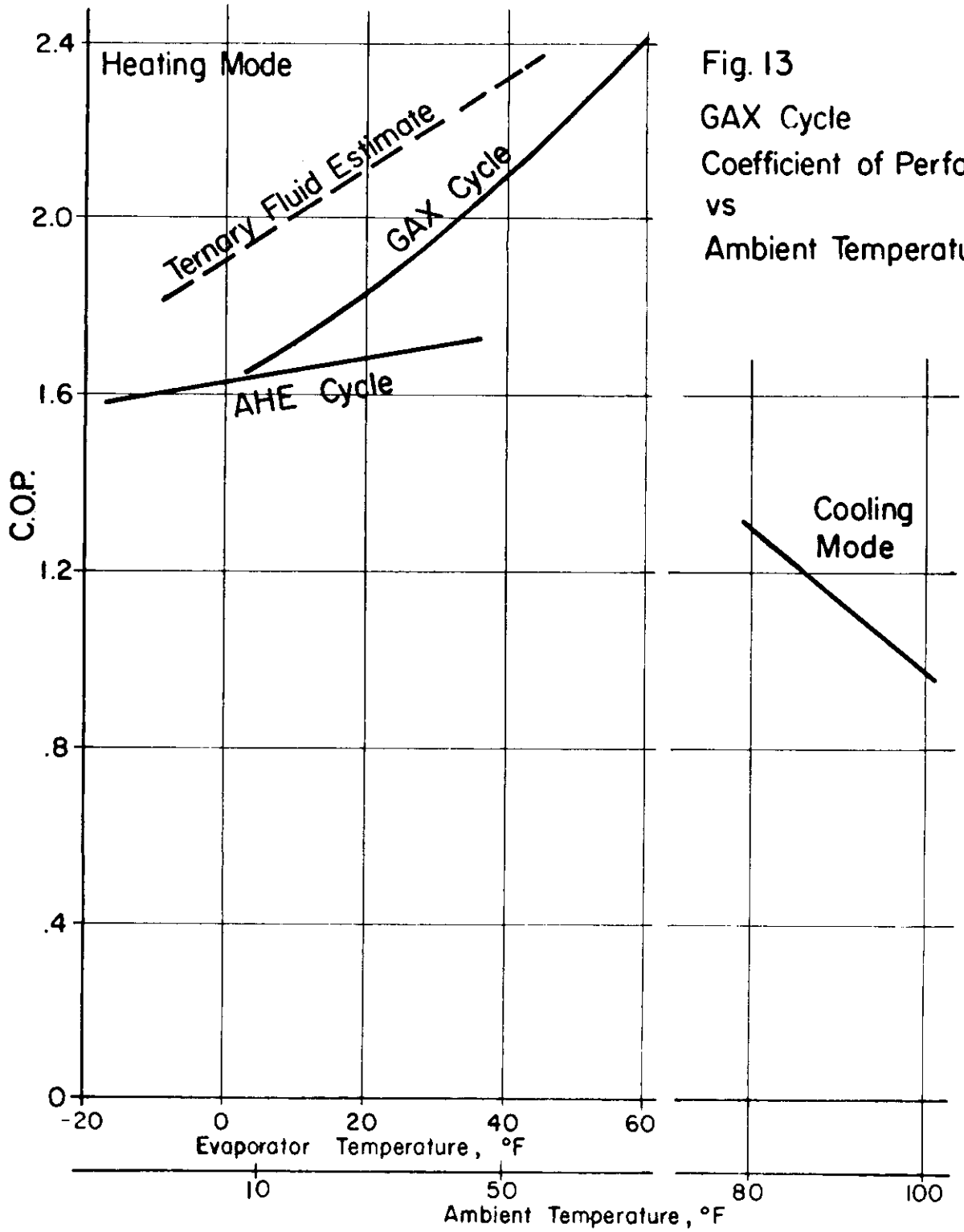


Fig. 13

GAX Cycle
Coefficient of Performance
vs
Ambient Temperature

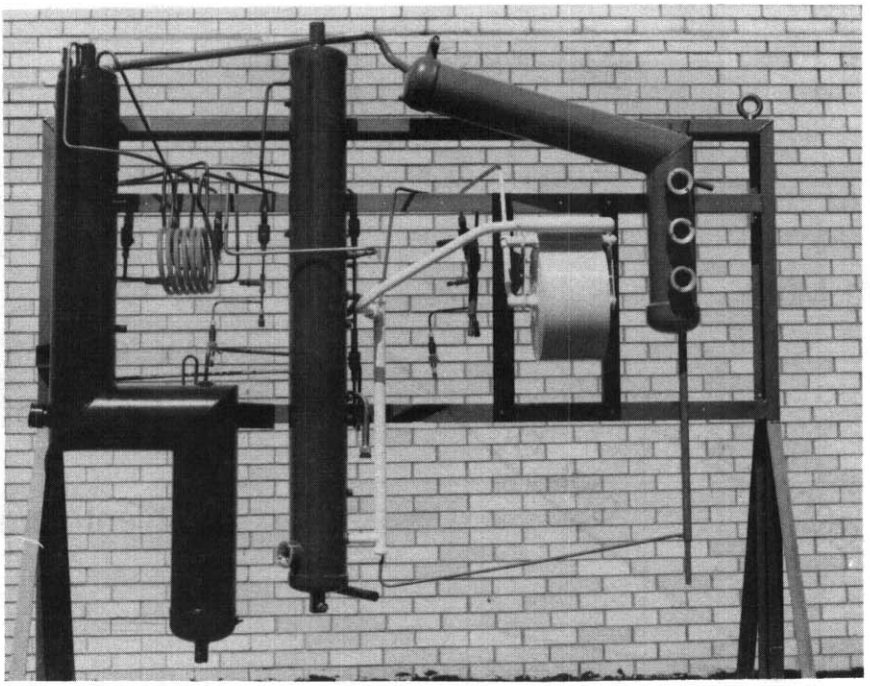


FIGURE 14

ABSORPTION TEST UNIT, UNINSULATED

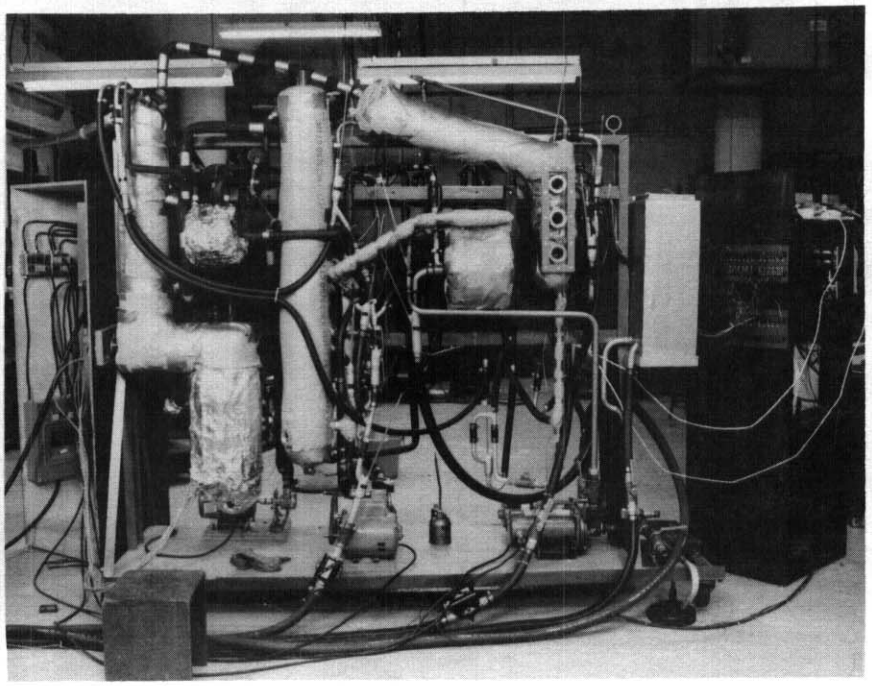


FIGURE 15

ABSORPTION UNIT ON TEST STAND

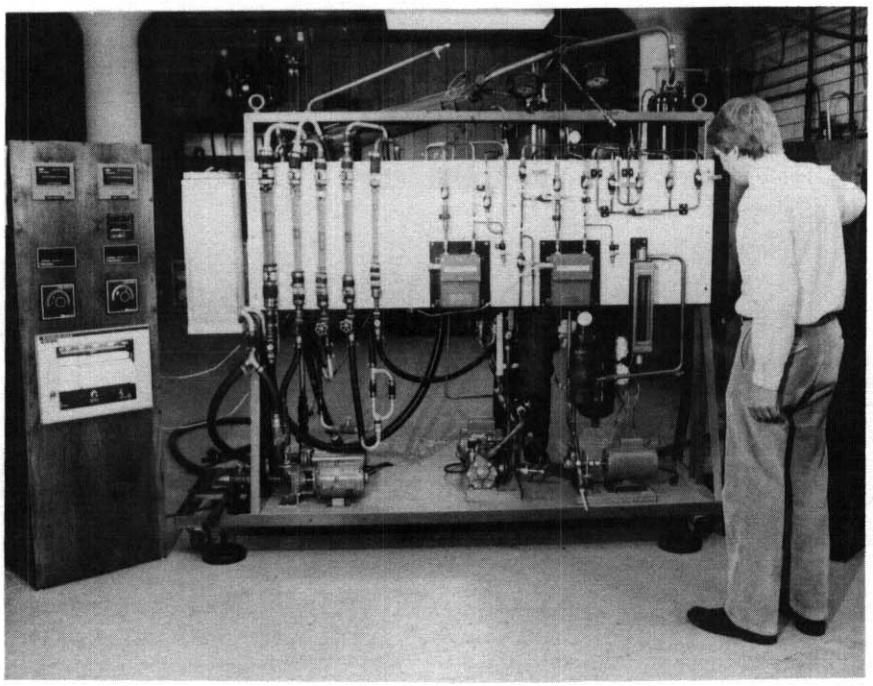


FIGURE 16

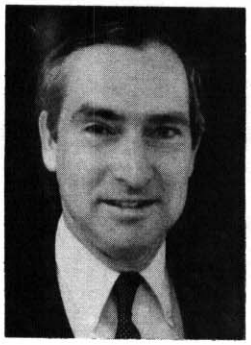
TEST STAND, UNINSULATED UNIT IN OPERATION

Evaluation of Advanced Design Concepts for Absorption Heat Pumps

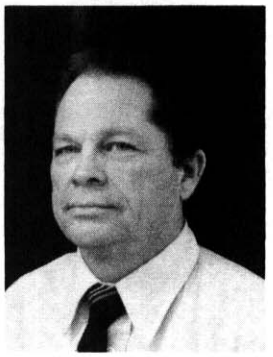
Floyd C. Hayes and Robert J. Modahl

The Trane Company

La Crosse, Wisconsin 54601



Floyd C. Hayes joined The Trane Company in 1968 after receiving the Ph.D. degree in mechanical engineering from the University of Illinois. His fields of specialty include thermodynamics, heat transfer, and fluid mechanics. He is currently manager of Thermal Systems Research at Trane. He has coauthored seven technical papers in the area of fluid mechanics and heat transfer and is the co-holder of one patent on heat pumps.



Robert J. Modahl received the B.S. degree in chemical engineering from the University of Wisconsin in 1958. He joined The Trane Company as a laboratory test engineer soon after graduation. He was promoted to supervisor of the Laboratory Analytical Chemistry Section in 1961. In 1963 he joined the company's Research Department and was assigned project responsibility in absorption refrigeration research involving absorbent-refrigerant combinations, corrosion inhibitors, and surfactants for the enhancement of absorption rate. In 1973 he was assigned project responsibility for distillation in brazed aluminum exchangers and nucleate boiling heat transfer, serving in that capacity until 1983. He is presently principal engineer in Thermal Systems Research.

EVALUATION OF ADVANCED DESIGN CONCEPTS FOR ABSORPTION HEAT PUMPS

Floyd C. Hayes
Robert J. Modahl

The Trane Company
La Crosse, Wisconsin

Introduction

This paper deals with the status of an advanced absorption heat pump program being conducted by The Trane Company, with support from the U.S. Department of Energy. The overall objective of the program is the evaluation, development, and proof testing of advanced absorption refrigeration cycles which are applicable to residential and commercial space conditioning systems. Phase I of the program involved the analytical evaluation of cycles and fluids, and the selection of a preferred concept for breadboard demonstration in Phase II. This paper summarizes our findings in Phase I.

Project objectives included direct firing with natural gas, and to use working fluids whose properties are reasonably well known. Target COP's were 1.6 at 47°F and 1.2 at 17°F in the heating mode, and 0.7 at 95°F in the cooling mode, including the effect of flue losses. In addition, Trane chose to concentrate on an air cooled system for commercial applications.

Advanced Cycles

Before proceeding to a discussion of the cycles investigated in this study, a few comments are in order which should clarify terminology and summarize the basic screening process which was used to select the most promising cycles for more detailed computer modeling. Table 1 summarizes the basic cycle alternatives which were considered, alone or in combination, to achieve the project goals. Also shown are the approximate ideal COP's which might be expected.

For convenience, in our preliminary discussion of advanced cycles, we will be using simplified Duhring plots such as shown in Figure 1. Basically the simplification introduced is that all lines of constant concentration will appear as 45° lines when plotted as refrigerant temperature vs. solution temperature, realizing of course that real fluids depart from this to varying degrees. In the final evaluation of advanced cycles, real fluid properties were used.

Single Effect (with liquid/liquid heat exchange)

The conventional single effect absorption cycle is illustrated in Fig. 1. External heat is added in the generator FG to boil off refrigerant. The refrigerant is condensed at C and evaporated at E with external heat input. The evaporated refrigerant is absorbed back into the solution in the absorber AB. Heat input is conserved by transferring heat from the strong solution to the weak solution, so that the only external heat input is at FG.

Each unit of refrigerant vaporized at FG provides one unit of cooling at E. Thus, the ideal cooling COP is 1. Similarly, in the heating mode, each unit vaporized at FG provides one unit of heating at C and one unit at AB. Thus, the ideal heating COP is 2.

Double (or Multiple) Effect

The most widely known advanced absorption cycles are double effect cycles. There are several types of double effect cycles, with the most common being double effect generator/common condenser, double effect generator/common absorber, and double evaporation. If the solution properties permit the use of double effect cycles, ideal cooling COP's twice as large as the conventional single effect cycle can be obtained.

A double effect generator/common condenser cycle is shown in Fig. 2. One unit of refrigerant is produced in the first stage generator FG as before, using an external source of heat. Then, another unit of refrigerant is produced in the second stage generator F'G' using heat from the first stage absorber. Thus, for one unit of heat input,

TABLE I: TYPES OF ADVANCED CYCLES

		IDEAL COP	
		Cooling	Heating
1. Single Effect (with liquid/liquid heat exchange)		1.0	2.0
2. Double (or multiple) Effect	n=2	2.0	3.0
	n=3	3.0	4.0
3. Generator Heat Exchange/		$\frac{1}{1-\epsilon}$	$\frac{2-\epsilon}{1-\epsilon}$
Absorber Heat Exchange	ϵ =1/3	1.5	2.5
4. Absorber/Generator Heat Exchange		$\frac{1}{1-\delta}$	$\frac{2-\delta}{1-\delta}$
Binary: δ = 1/4		1.33	2.33
Ternary: δ = 1/2		2.0	3.0
5. Absorber/Generator Heat Exchange Generator Heat Exchange/ Absorber Heat Exchange		$\frac{1}{(1-\delta)(1-\epsilon_1)}$	$1+\frac{1}{(1-\delta)(1-\epsilon_1)}$
Binary: δ = 1/4, ϵ_1 = 1/3		2.0	3.0
Ternary: δ = 1/2, ϵ_1 = 1/3		3.0	4.0
6. Regenerative Cycles		$\left(\frac{1}{1-\delta}\right)+\left(\frac{n-1}{n}\right)$	$1+\left(\frac{1}{1-\delta}\right)+\left(\frac{n-1}{n}\right)$
Binary: δ = 1/4,	n=2	1.83	2.83
	n=3	2.0	3.0
Ternary: δ = 1/2,	n=2	2.5	3.5
	n=3	2.67	3.67
with Generator Heat Exchange/ Absorber Heat Exchange		$\left(\frac{1}{1-\epsilon_1}\right)\left(\frac{1}{1-\delta}+\frac{n-1}{n}\right)$	$1+\left(\frac{1}{1-\epsilon_1}\right)\left(\frac{1}{1-\delta}+\frac{n-1}{n}\right)$
Binary: δ = 1/4, ϵ_1 = 1/3,	n=2	2.75	3.75
	n=3	3.0	4.0
Ternary: δ = 1/2, ϵ_1 = 1/3,	n=2	3.75	4.75
	n=3	4.0	5.0
7. Resorption/Desorption Cycles			
a. With Double Evaporation		2.0	3.0
b. With Double Effect Generator		2-X	3-X
For X = 1/2		1.5	2.5

two units of cooling are produced, resulting in an ideal cooling COP of 2. The corresponding ideal heating COP is 3.

The concept of Fig. 2 can be extended to three or more effects if the solution field is wide enough, in which case the ideal cooling COP equals the number of effects.

Generator Heat Exchange/Absorber Heat Exchange

Efficiency improvements can be made on the conventional single effect cycle without going to the complication of double effect machines. The simplest improvements are generator heat exchange and absorber heat exchange. The effects are illustrated in Fig. 3.

Basically this concept reduces losses associated with temperature differences in the liquid/liquid heat exchanger of the conventional cycle. As shown in Fig. 1, there is a significant temperature difference between the strong solution GA and the weak solution BF throughout the length of the heat exchanger. Cycle efficiency can be improved

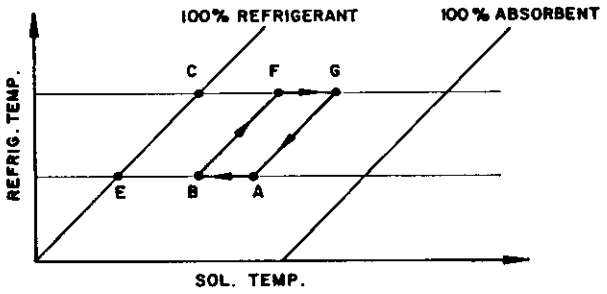


FIG. 1: CONVENTIONAL SINGLE EFFECT ABSORPTION CYCLE

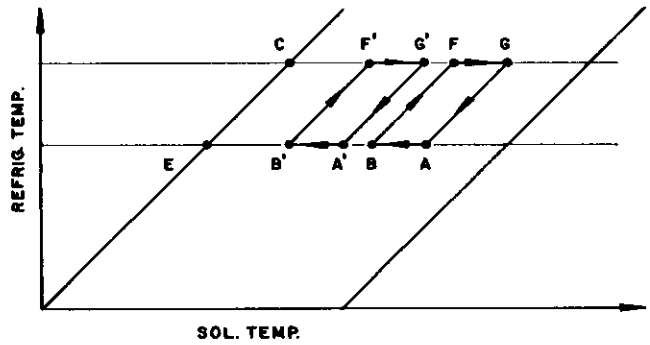


FIG. 2: DOUBLE EFFECT GENERATOR / COMMON CONDENSER CYCLE

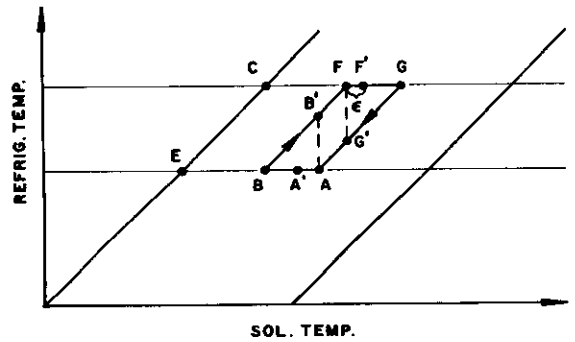


FIG. 3 : GENERATOR HEAT EXCHANGE / ABSORBER HEAT EXCHANGE

by using heat from the highest temperature strong solution (GG') to boil off refrigerant in the first part of the generator (FF'). The maximum fraction, ϵ , of generator heat which can be supplied in this manner is about 1/3 for the NH_3/H_2O binary which has a high ratio of refrigerant latent heat to solution specific heat. For fluids with a low ratio of refrigerant latent heat to solution specific heat the ratio may be in the range of 3/5, but these fluids usually are not the best absorption fluids for other reasons.

Liquid/liquid heat exchange is now between the strong solution (G'A) and the weak solution (B'F), but in this ideal case the temperature difference for heat transfer is zero. To supply the additional heat for the weak solution between B and B', part of the high temperature absorber heat (AA') is used.

What has been achieved is that the external heat required has been reduced to $1-\epsilon$ for each unit of cooling. The ideal cooling COP is then

$$COP_c = \frac{1}{1-\epsilon}$$

For $\epsilon = 1/3$, $COP_c = 1.5$.

Absorber/Generator Heat Exchange

Absorber/generator heat exchange requires significant overlap of solution temperatures at the high temperature end of the absorber and the low temperature end of the generator, as shown in Fig. 4. External generator heat input to the generator is reduced by an amount δ by utilizing absorber heat AA'. The principle is the same as the double effect generator/common condenser cycle in which the first stage absorber supplies heat for the second stage generator, except that now the effect is achieved in a single effect cycle.

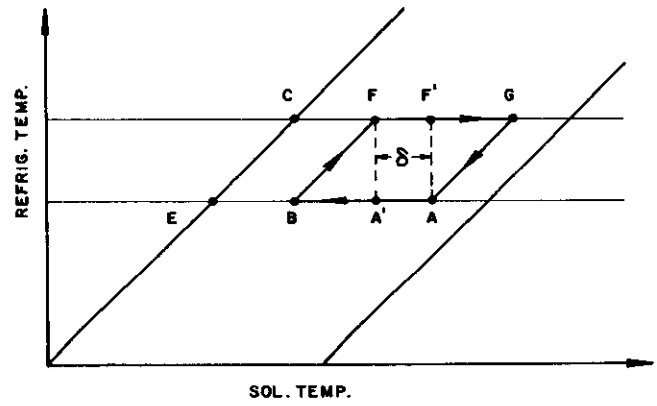


FIG. 4: ABSORBER / GENERATOR HEAT EXCHANGE

Absorber/generator heat exchange looks best with a wide solution field. The maximum fraction of generator heat that can be supplied with absorber heat is typically 1/4 for binary fluids, but may be as high as 1/2 for ternary fluids. The simplified relation for the ideal cooling COP is as follows:

$$COP_c = \frac{1}{1-\delta}$$

For binary fluid, $\delta = 1/4$: $COP_c = 1.33$
For ternary fluid, $\delta = 1/2$: $COP_c = 2.0$

Absorber/Generator Heat Exchange Combined with Generator Heat Exchange and Absorber Heat Exchange

Generator heat exchange and absorber heat exchange can be utilized with other single and double effect efficiency improvement concepts. An attractive combination is to combine them with absorber/generator heat exchange. The resulting simplified relations for the ideal cooling COP is as follows:

$$COP_c = \frac{1}{(1-\delta)(1-\epsilon_1)}$$

where $q_{ex} + \delta + \epsilon = 1$

q_{ex} = fraction of generator heat supplied externally

δ = max. fraction of generator heat supplied by absorber

ϵ = max. fraction of generator heat supplied by strong solution (generator heat exchange),

and $\epsilon_1 = \frac{\epsilon}{q_{ex} + \epsilon}$

Binary fluid: $\delta = 1/4$, $\epsilon_1 = 1/3$ $COP_c = 2$
Ternary fluid: $\delta = 1/2$, $\epsilon_1 = 1/3$ $COP_c = 3$

Regenerative Cycles

Regenerative processes involve using vapor transfer at one or more intermediate pressures (refrigerant temperatures) in order to achieve a larger overlap for absorber/generator heat exchange. Fig. 5 shows a possible regenerative cycle with one intermediate pressure compared to the one-step absorber-generator heat exchange cycle FGAB with overlap δ. In this cycle, vapor is generated at intermediate pressure between B' and H using absorber heat A'A". This vapor is absorbed at intermediate pressure between G' and I and this absorber heat is used in the high pressure generator between F' and F". As Fig. 5 shows for the example chosen, the fraction of generator heat supplied by absorber heat exchange has increased from 1/2 for the one-step process to 3/5 for the two-step process. The simplified analysis yields the following expression for the ideal cooling COP:

$$COP_c = \frac{1}{1-\delta} + \frac{n-1}{n}$$

where n = number of pressure levels at which vapor is generated.
(n = 1 for conventional cycle).

δ = max. fraction of generator heat supplied by absorber for n = 1.

If generator heat exchange and absorber heat exchange are also utilized,

$$COP_c = \left(\frac{1}{1-\epsilon_1}\right)\left(\frac{1}{1-\delta} + \frac{n-1}{n}\right)$$

Typical values for binary and ternary fluids are shown in Table 1.

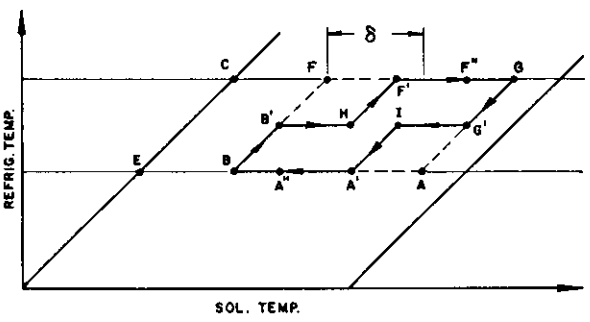


FIG. 5 REGENERATIVE CYCLE

Resorption/Desorption Cycles

A resorption/desorption cycle is defined as a cycle in which the flow is reversed from the conventional absorber/generator cycle. In the resorption/desorption cycle, refrigerant is desorbed (generated) at the low pressure and resorbed (absorbed) at the high pressure, just the reverse of the conventional absorber/generator cycle.

The resorption/desorption cycle can be used in a variety of ways in combination with a conventional absorber/generator cycle. The combined effects might range from a double effect machine to a system with reduced operating pressures. Following are three examples which illustrate the benefits of resorption/desorption cycles.

1. Double effect evaporator cycle - The resorption/desorption cycle allows double effect evaporation to be achieved. For example, in the double effect evaporation cycle shown in Fig. 6, one unit of refrigerant is generated with external heat in the generator/absorber loop FGAB, producing one unit of cooling at E. The refrigerant evaporated at E is resorbed at A'B' of the

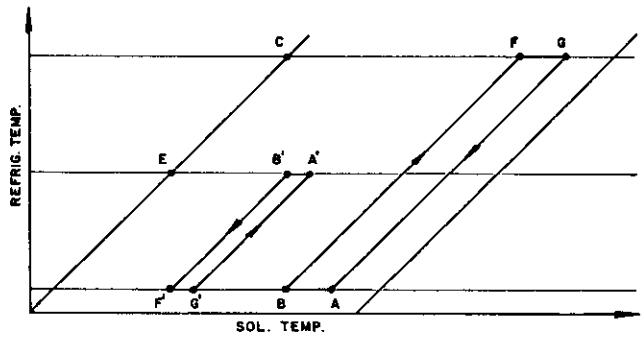


FIG. 6 DOUBLE EFFECT EVAPORATION EMPLOYING RESORPTION/DESORPTION CYCLE

resorption/desorption loop, and desorbed at F'G', producing a second unit of cooling and thus allowing an ideal cooling COP of 2 to be attained. The refrigerant desorbed at F'G' is absorbed at AB. Thus, in this case the resorption/desorption cycle was crucial in allowing double effect evaporation to be achieved at the same temperature.

2. Double effect generator with resorption/desorption cycle - Another application of resorption/desorption cycles lies in achieving intermediate COP's if the solution stability and refrigerant vapor pressure do not allow the benefits of a full double effect to be attained. This concept is illustrated in the double effect generator cycle shown in Fig. 7.

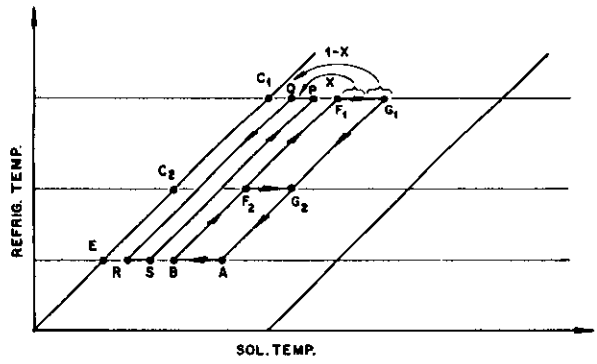


FIG. 7 : DOUBLE EFFECT GENERATOR WITH RESORPTION / DESORPTION CYCLE

One unit of refrigerant is boiled off in the first stage generator F_1G_1 using an external source of heat. But a fraction of this refrigerant, X, is resorbed at high pressure in a resorber cycle PQRS, while the rest goes to the first stage condenser, C_1.

Another unit of refrigerant is boiled off in the second stage generator F_2G_2 using heat from the first stage condenser C_1 and the resorption process PQ. This refrigerant is condensed in the second stage condenser C_2. Part of the heat from the second stage condenser is used to generate vapor at RS in the low pressure section of the resorption cycle. This vapor as well as vapor from the evaporator, E, is absorbed at AB.

The simplified expression for the ideal cooling COP of this cycle is as follows:

$$COP_c = 2 - X$$

where X is the fraction of refrigerant generated in the first stage generator which is resorbed in the resorption/desorption cycle. Because this refrigerant provided no useful cooling effect the ideal COP of the cycle is less than 2, even though it is a double effect cycle.

3. Resorption/desorption circuit substituted for evaporator/condenser circuit - A final example of the application of a resorption/desorption circuit is replacing the conventional condenser/evaporator circuitry of the system. This would not affect ideal cycle COP, but it would lower the operating pressure of the system and might reduce auxiliary power and system cost.

The purpose of the preceeding exercise in geometry was primarily to clarify terminology and to develop a first order summary of the efficiency potential of the various cycle concepts. There are many variations and combinations of these concepts. While it was not possible to consider every variation and combination in this investigation, a representative number were evaluated using real fluids and representative design conditions.

While one normally first thinks of double or multiple effect cycles when considering advanced absorption concepts, Table I shows that there are a number of alternatives for achieving high efficiencies. The choice of cycle must therefore include consideration of other factors including system cost and complexity, manufacturability, reliability, and auxiliary power requirements.

Fluids

The fluid mixtures considered in this investigation are listed in Table II. Initially our selection of fluids was based on binary mixtures whose properties were reasonably well known and whose stability and corrosion characteristics were thought to be reasonably good in the region of interest. The list of fluids was later expanded to include ternary mixtures when it was determined that binary mixtures showed marginal prospects for achieving project efficiency goals. This introduces more uncertainty into property data and stability and corrosion characteristics, and will require that more consideration be given to these questions in Phase II of the program.

TABLE II: FLUID PROPERTIES

MIXTURE	DATA SOURCE
NH_3/H_2O	Macriss, R.H., et.al, "Physical and Thermodynamic Properties of Ammonia-Water Mixtures", Research Bulletin No. 34 prepared for A.G.A. Chicago: IGT, Sept., 1964.
R22/DMETEG	Macriss, R.A., Mason, D., "Physical and Thermodynamic Properties of New Refrigerant-Absorbent Pairs", Project ZB-85 Final Report prepared for the Air Conditioning and Prime Mover Research Committee of the A.G.A. Chicago, IGT, Oct., 1964.
R123/ETFE	Allied Chemical Data Book
$LiBr/H_2O$	ASHRAE Guide
$NH_3/H_2O/LiBr$, $CH_3NH_2/H_2O/LiBr$	Rademacher, R., "Working Substance Combinations for Absorption Heat Pumps", Dissertation, U. of Munich, Jan., 1981.
$CH_3OH/LiBr/ZnBr_2$	Aker, J.E., et al, "An Evaluation of Alcohol-Salt Mixtures as Absorption Refrigeration Solutions", ASHRAE Transactions, 1965.

Cycle/Fluid Combinations

A large number of cycle/fluid combinations were considered utilizing the cycle concepts and fluids shown in Tables I and II, respectively. Those selected for computer modeling and more rigorous analysis are summarized in Table III.

Steady state computer models were successfully developed for most of the cycles shown in Table III. These models involved writing energy balances, flow balances, concentration balances and transfer rate equations for the various components. This resulted in 23 to 54 equations for each cycle to be solved simultaneously in order to determine cycle efficiencies , component capacities, and fluid conditions throughout the system.

The various cycle/fluid combinations were compared under the following set of conditions:

Maximum solution temperature = $400^{\circ}F$

Minimum absorber solution temperature = $110^{\circ}F$

Minimum internal heat exchanger approach temperature = $10^{\circ}F$

Condenser saturation temperature = $120^{\circ}F$ (130-110 for resorber type)

Evaporator saturation temperature = $37^{\circ}F$ (25-43 for desorber type)

These conditions were selected to be representative of conditions that might be experienced in an air cooled unit at the heating rating point of 47/43/70 as well as the cooling rating point of 80/67/95.

Input for the computer model consisted of refrigerant flow rate and concentration,

TABLE III: CYCLE/FLUID COMBINATIONS

CYCLE/FLUID DESIGNATION	FLUID	CYCLE
ABS1	NH_3/H_2O	Absorber Heat Exchanger/Generator Heat Exchange
ABS2	R22/DMETEG	"
ABS3	R123/ETFE	"
ABS4	NH_3/H_2O	Absorber/Generator Heat Eschange, Absorber Heat Exchange/ Generator Heat Exchange
ABS5	R22/DMETEG	"
ABS6A	R22/DMETEG	Regeneration, Abs./Gen. Heat Exchanger Abs. Heat Exch./Gen. Heat Exch.
ABS6B	NH_3/H_2O	"
ABS7A	R22/DMETEG	Two-stage absorption, Partial Flash regeneration, Resorp./Desorp. cycle Abs/Gen. Heat Exch., Abs. Heat Exch./Gen. Heat Exch.
ABS7B	NH_3/H_2O	"
ABS8	LiBr/H_2O	Double Effect Generator, Cascaded Evap.
ABS9	NH_3/H_2O	Use of Resorp./Desorp. cycle to achieve double effect evaporation.
ABS10A	R123A/ETFE	Double effect generator with Resorp./Desorpt. cycle
ABS10B	R123A/ETFE	"
ABS11	NH_3/H_2O	Double Effect Generator, Common Condenser, Abs. Heat Exch., Gen. Heat Exch.
ABS12	NH_3/H_2O/LiBr	Abs./Gen. Heat Exch., Abs. Heat Exch./ Gen. Heat Exch.
ABS13	NH_3/H_2O/LiBr	Same as ABS12 plus absorbent recirculation.
ABS14	NH_3/H_2O/LiBr	Same as ABS13 plus regeneration.
ABS15	CH_3NH_2/H_2O/LiBr	Same as ABS12
ABS16	"	Same as ABS13
ABS17	"	Same as ABS14
ABS18	NH_3/H_2O/LiBr	Same as ABS12 except Resorp./Desorp. cycle substituted for conventional evap./cond.
ABS19	"	Same as ABS13 except Resorp./Desorp. cycle substituted for conventional evap./cond.
ABS20	"	Same as ABS14 except Resorp./Desorp. cycle substituted for conventional evap./cond.
ABS21	CH_3OH/LiBr/ZnBr	Abs. Heat Exch./Gen. Heat Exch.

saturated condensing and evaporating temperatures, generator and absorber fluid leaving temperatures, reflux ratio, and heat exchanger approach temperatures. Output consisted of thermodynamic properties and flow rates at each point in the cycle, all external and internal heat transfers, pump power, and cycle COP.

A summary of output from the computer model calculations is given in Table IV.

Selection of Preferred Cycle/Fluid Combinations

The results shown in Table IV are difficult to interpret. Basically COP_H and theoretical pump power are representative of operating cost while internal heat exchange relates to the first cost of the unit. In order to make a rigorous comparison of the various systems including the effects of the three parameters, a rating procedure based on penalty points was

TABLE IV: COMPUTER MODEL RESULTS

CYCLE FLUID DESIGNATION	COP_H*	THEO. PUMP POWER, WATTS**	INTERNAL HEAT EXCHANGE, BTU**
ABS 1	1.67	6.5	6,000
ABS 2	1.77	33.3	16,200
ABS 3	1.75	3.9	15,300
ABS 4	1.74	5.9	7,200
ABS 5	1.81	32.4	18,600
ABS 6A	1.72	54.6	30,100
ABS 6B	1.94	8.3	14,000
ABS 7A	1.77	33.0	16,600
ABS 7B	1.59	6.0	6,100
ABS 8	1.32	.3	29,100
ABS 9	1.80	14.7	17,000
ABS 10A	1.73	21.0	29,300
ABS 10B	1.63	12.0	23,100
ABS 11	1.99	20.1	18,900
ABS 12	2.08	6.6	9,100
ABS 13	2.19	7.8	10,800
ABS 14	2.32	10.0	14,300
ABS 15	1.87	2.9	8,000
ABS 16	2.09	3.9	11,200
ABS 17	2.18	3.9	14,400
ABS 18	2.00	7.2	12,800
ABS 19	2.29	8.5	16,000
ABS 20	2.45	9.9	22,000
ABS 21	1.58	2.8	10,900

* Neglecting flue losses.
** Per 12,000 Btu/Hr Heating.

developed. The method of assigning penalty points is shown in Table V.

The ranking of the various cycle/fluid combinations based on the penalty point system described in Table V is shown in Table VI. The following observations are made concerning the ranking:

1. The 9 highest ranking cycle/fluid combinations are ternary fluid systems.
2. The 4 highest ranking systems utilize the ternary mixture $NH_3/H_2O/LiBr$.
3. The 3 highest ranking binary mixture systems utilize the mixture. NH_3/H_2O.
4. The highest ranking binary system utilizes regeneration, absorber/generator heat exchange, absorber heat exchange, and generator heat exchange.

TABLE V: DEFINITION OF PENALTY POINTS

PENALTY POINTS	COP 3	THEO. PUMP POWER, WATTS*	INTERNAL HEAT TRANSFER MBH*
0	2.0	0	0
1	1.9	10	12
2	1.8	20	24
3	1.7	30	36
4	1.6	40	48
ETC.	ETC.	ETC.	ETC.

* Per 12,000 Btu/hr heating

VI: RANKING OF CYCLE/FLUID COMBINATIONS

RANK	CYCLE/FLUID DESIGNATION	COP	PENALTY POINTS PUMP POWER	IHE	TOTAL	SENSITIVITY ANALYSIS*
**1	ABS 20	-4.5	1.0	1.8	-1.7	.1
2	ABS 14	-3.2	1.0	1.2	-1.0	.2
3	ABS 19	-2.9	.8	1.3	- .8	.5
4	ABS 13	-1.9	.8	.9	- .2	.7
5	ABS 17	-1.8	.5	1.2	- .1	1.1
6	ABS 16	- .9	.4	1.0	.5	1.5
7	ABS 12	- .8	.7	.8	.7	1.5
8	ABS 18	0.	.7	1.1	1.8	2.9
9	ABS 15	1.3	.3	.7	2.3	3.0
**10	ABS 6B	.6	.8	1.2	2.6	3.8
11	ABS 11	.1	2.0	1.6	3.7	5.3
**12	ABS 4	2.6	.6	.6	3.8	4.4
**13	ABS 3	2.5	.4	1.3	4.2	5.5
**14	ABS 1	3.3	.7	.5	4.5	5.0
15	ABS 9	2.0	1.5	1.4	4.9	6.3
16	ABS 7B	4.1	.6	.5	5.2	5.7
17	ABS 21	4.2	.3	.9	5.4	6.3
18	ABS 5	1.9	3.2	1.6	6.7	8.3
19	ABS 10B	3.7	1.2	1.9	6.8	8.7
20	ABS 2	2.3	3.3	1.4	7.0	8.4
21	ABS 7A	2.3	3.3	1.4	7.0	8.4
22	ABS 10A	2.7	2.1	2.4	7.2	9.6
23	ABS 8	6.8	0.	2.4	9.2	11.6
24	ABS 6A	2.8	5.5	2.5	10.8	13.3

* Total with double penalty points for Internal Heat Exchange.
** Selected for more detailed economic analysis.

5. The second highest ranking binary system is a double effect generator, common condenser system with absorber heat exchange and generator heat exchange. However, while the system looks good at the 47/43/70 rating condition, it would not be able to operate in the double effect mode at the 17/15/70 condition.

6. The third highest ranking binary system is a somewhat simpler system utilizing absorber/generator heat exchange, absorber heat exchange and generator heat exchange.

Sensitivity Analysis

Questions might be raised in regard to the validity of the penalty point method of ranking systems as defined in Table V. Concerning the relative importance of heating COP and theoretical pump power, it can be shown that a 1 penalty point differential translates into approximately the same operating cost differential for gas and electric usage when burner efficiency, pump efficiency, motor efficiency, and typical electricity and gas prices are used. The biggest question is how the penalty points assigned to operating cost (COP and pump power) compare with the penalty points assigned to first cost (internal heat transfer).

A first order analysis shows that a 1 penalty point differential in internal heat transfer will give a 5 to 10 year payback if an efficiency improvement can be achieved comparable to a 1 penalty point differential in heating COP and pump power. Since this is higher than is generally accepted in commercial and residential products, the ranking was repeated with the penalty points for internal heat transfer doubled. This would translate to a 2-1/2 to 5 year payback. The results are shown in the extreme right hand column of Table VI.

Doubling the penalty points for internal heat transfer had no effect on the ranking of the top 10 cycle/fluid combinations. The most significant effect was to modify the relative ranking of the second through the fifth highest ranking binary fluid systems, with the double effect system moving from the second to

TABLE VII.
SUMMARY OF PRELIMINARY SYSTEM PERFORMANCE AND INSTALLED COST

(Size basis for all systems is 100 ton cooling capacity @ 95F)

Cycle Designation	ABS1	ABS3	ABS4	ABS6B	ABS20
Cooling (45^{o}F glycol solution circulated to conditioned space)					
Capacity, (MBH) @ 95F	1,200	1,200	1,200	1,200	1,200
(MBH) @ 115F	1,060	--	960	1,020	9,000
COP (including 85% comb. eff)					
@ 95F	.56	.65	.80	.91	1.21
@ 115F	.51	--	.64	.78	.90
Heating (120^{o}F glycol solution circulated to conditioned space)					
Capacity, (MBH) @ 47F	2,880	2,880	2,380	2,170	2,050
@ 17F	2,740	--	2,060 (@ 20.4F)	1,870	1,570
COP (including 85% comb. eff.)					
@ 47F	1.35	1.55	1.61	1.65	2.08
@ 17F	1.29	--	1.40 (@ 20.4F)	1.41	1.59
Parasitic Power					
Fanpower, outdoor, KW	15	15	15	15	15
indoor, KW	13	13	13	13	13
Glycol sol'n pump outdoor, KW	4	4	4	4	4
indoor, KW	4	4	4	4	4
Solution Pump, KW	6	3.5	5.3	7.4	9
Installed Cost	60,900	143,200	78,500	79,500	90,500

the fourth highest ranking binary fluid system.

The five systems selected for further economic analysis are identified in Table VI. Four of these were selected from the top 5 binary fluid systems, primarily because this work was done before the scope of the project was expanded to include ternary fluids. The other system chosen was the highest ranking ternary fluid system. The second ranking binary system, the double effect generator, was dropped from consideration because of its operational problems at low ambients and because the sensitivity analysis showed a lower ranking when more emphasis was placed on first cost in the ranking procedure.

Preliminary System Designs and Installed Cost Estimates

Preliminary system designs were developed for each of the 5 systems selected for further economic analysis. The design concept selected was to use a secondary fluid loop for both the indoor and outdoor coils. System reversal was accomplished by switching secondary fluid flows.

Parametric studies were conducted using the computer models to investigate efficiency/cost trade-offs for each design. It was determined impractical to design the ABS1 and ABS3 systems to meet or exceed project COP goals, and designs were selected which did not meet the goals. The other three systems met or

exceeded project COP goals, although ABS4 required very small solution side heat exchanger approach temperatures and relatively large air side heat exchangers.

Cost estimates were developed for a 100 ton unit of each of the five preliminary designs. Cost estimates were based on component costs of similar equipment in this size range.

A summary of the performance and installed cost estimates of the five preliminary designs is given in Table VII. The high installed cost of the ABS3 system should be noted. This is due in large part to our attempt to utilize brazed aluminum construction in the design. However, it was found that large frontal area brazed exchangers were required to avoid flooding in the rectifier and absorber sections, resulting in higher than expected exchanger costs. Although costs of a steel ABS3 design were not estimated, it would be higher than the ABS4 or ABS6B designs because of lower heat and mass transfer coefficients for the R123A/ETFE vs. NH_3/H_2O and because of higher internal heat transfer requirements. Thus, the ABS3 was dropped from further consideration at this point.

User Economics

The criteria used to measure the economic feasibility of the absorption heat pump systems were simple pay-back perid (SPB) and the after tax internal rate of return (IRR). Cash flows for the IRR calculations include energy costs, taxes, insurance, maintenance, replacement costs, salvage value, depreciation effects and tax credits. A 20 year period was assumed for the IRR analysis.

TRACE, Trane's computer program for building system energy and economic analysis, was used to compare the various system alternatives. An apartment building with a design cooling load of 155 tons was used. System alternatives were compared to a reference system which included an air cooled reciprocating water chiller and a gas fired boiler. The primary analyis was conducted using Chicago weather data, 1983 Commonwealth Edison electricity prices, and $.55/therm gas prices. In addition, the effects of geographic region and electricity prices were investigated by including New York and Atlanta weather data and Con-Edison and Georgia Power electricity prices.

Typical results from the TRACE program are shown in Table VIII for the Chicago location. As indicated previously the ABS3 system was dropped from consideration because of its high first cost, and ABS1 results are not shown because the IRR was negative. SPB and IRR

TABLE VIII

RESULTS OF USER ECONOMICS ANALYSIS

BASIS: Apartment building in Chicago climate with Commonwealth Edison rate structure. All costs in 1983 dollars.

INPUT

System	Conventional	ABS4	ABS20	ABS6B
Size, tons cooling	155	155	155	155
Installed cost, $/ton	490	678	780	687
Heating COP, 47F	.8	1.61	2.08	1.65
, 17F	.8	1.37	1.59	1.41
Cooling COP, 95F	3.0	.80	1.21	.91
, 115F	-	.64	.90	.78
Parasitic power, KW/Ton	.18	.283	.319	.304

OUTPUT

Ann. elec. consump., 10^3 KWH	1022	1017	1074	1044
Ann. gas consump., 10^3 therms	83.1	64.5	48.1	63.7
Elect cost, energy, 10^3 $	50.5	49.3	51.5	50.5
, demand, 10^3 $	15.0	12.9	13.3	13.1
Gas cost, 10^3 $	45.7	35.5	27.7	35.0
Total utility cost, 10^3 $	111.2	97.7	91.7	98.7
SPB, yrs.	-	2.2	2.3	2.5
IRR, %	-	37	30	34.2

values look quite good for the ABS4, ABS6B, and ABS20 systems when compared to the reference system. There is not a great deal of difference among the three systems, but ABS4 is best.

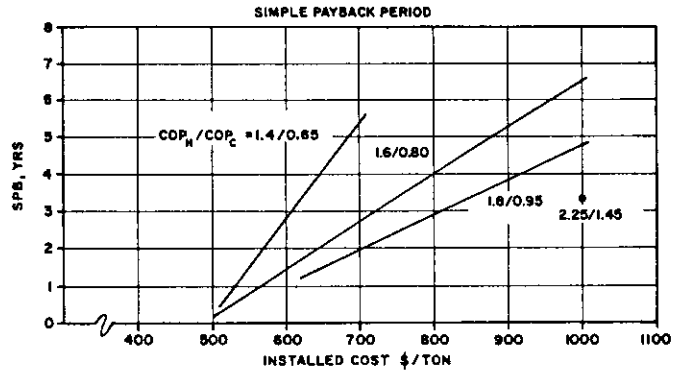
Other points to be noted concerning Table VIII are the following:

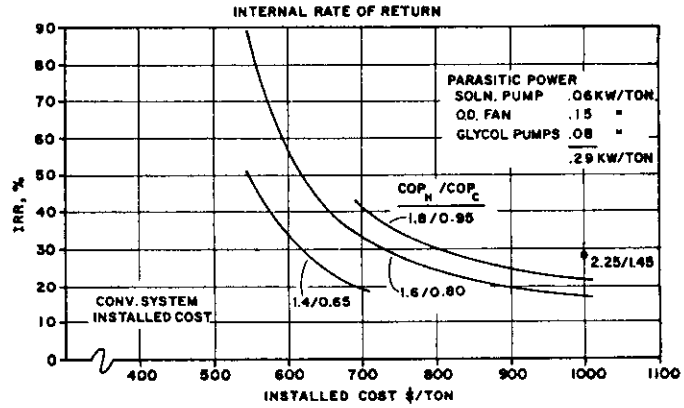
- The installed cost figures do not agree with those in Table VII because $/ton decreases with increasing tons.
- The annual electric consumption numbers include everything for the building, not just HVAC equipment.
- Two of the three absorption systems require more annual electrical energy than the conventional system. This is due to the fact that the absorption system pumps were assumed to run continuously while gas flow was modulated.
- The absorption systems all show slightly lower electric energy costs because of lower demand charges. Demand charges are higher today than in 1983 and are on an increasing trend.
- The primary operating cost savings resulting from absorption heat pumps are due to less gas consumption during the heating season. Thus, the economics of their use will improve as the cost of gas increases and will be primarily related to the cost of gas rather than on the relative cost of gas and electricity.

The results of a parametric study for the Chicago location are shown in Fig. 8. SPB and IRR are plotted vs. installed cost in $/ton for various efficiency levels. These curves can be used to set cost goals for a given efficiency level.

The effect of geographic location on the economics of the ABS4 system are illustrated in Table IX. Table IX shows that the economics are marginal in New York and probably not widely acceptable in Atlanta. This is due to the fact that in the warmer climates more gas is burned during the summer and less gas is saved during the winter. This suggests that a system with a higher cooling COP will likely be required in warmer climates. Although a TRACE run was not made for the ABS20 system in Atlanta, an approximate calculation indicated the SPB for the ABS20 system would be less than 5 there.

FIG. 8: PARAMETRIC STUDY RESULTS
APARTMENT BUILDING
COMM. ED. ENERGY PRICES
DESIGN SIZE: 155 TONS COOLING





Conclusions

The primary conclusions of this program to date are as follows:

1. The gas fired absorption heat pump can be an attractive heating/cooling alternative in the commmercial market. The economics look best in areas which have high heating loads.
2. The economics of the commercial gas fired absorption heat pump will improve as the cost of gas increases, at least up to the point where an all electric system shows lower operating costs in Northern climates.
3. Compared to the conventional electric chiller/gas boiler system used as the reference system in this study, the gas fired absorption heat pump accomplishes DOE's goal of conserving energy and provides much more balanced loads for both electric and gas utilities.
4. The preferred absorption cycle for commercial gas fired heat pumps

TABLE IX

SUMMARY OF TRACE RUNS

APARTMENT BUIDING IN VARIOUS LOCATIONS, ALL WITH THE SAME (COMMONWEALTH EDISON) ENERGY PRICE STRUCTURE.

Input Data

Run No.	9-1	9-2	10-1	10-2	11-1	11-2
Weather(location)	Chicago	Chicago	N.York	N.York	Atlanta	Atlanta
Utility	Commonwealth Edison					
System	Conv.	ABS4	Conv.	ABS4	Conv.	ABS4
Heating COP, 47F	.8	1.61	-	1.61	-	1.61
, 17F	.8	1.37	-	1.37	-	1.37
Cooling COP, 95F	3.0	.80	-	.80	-	.80
, 115F	-	.64	-	.64	-	.64
Cost, $/ton	490	678	490	678	490	678
Parasitic power, KW	-	.283	-	.283	-	.283
Sol'n pump	-	.053	-	.053	-	.053
O.D. fan	.14	.15	-	.15	-	.15
Glycol pumps	.04	.08	-	.08	-	.08
Size, tons cooling	155	155	163	163	161	161

Output

Ann. elec. consump., 10^3 KWH	1022	1017	1045	1029	1071	1021
Ann. gas consump., 10^3 therms.	83.1	64.5	66.6	56.3	52.2	52.3
Elec. Cost, Energy, 10^3 $	50.5	49.3	51.5	49.8	52.6	49.5
, Demand 10^3 $	15.0	12.9	14.4	13.0	15.7	12.8
Gas Cost, 10^3 $	45.7	35.5	36.6	31.0	28.7	28.8
Total Utility cost, 10^3 $	111.2	97.7	102.5	93.8	97.1	91.2
SPB, yrs	-	2.2	-	4.3	-	6.5
IRR, %	-	37.2	-	22.6	-	15.2

contains the following efficiency enhancement features:

- absorber heat exchange and generator heat exchange,
- absorber/generator heat exchange,
- regeneration, and
- a resorption/desorption circuit to replace the conventional condenser/evaporator circuit.

5. The preferred fluid at this point appears to be the ternary mixture $NH_3/H_2O/LiBr$, with $CH_3,NH_2/H_2O/LiBr$ a possible alternative.[3] The preferred binary mixture is NH_3/H_2O.

   During the next eighteen months the program will be in the hardware demonstration phase. The objective is to determine if the performance which has been predicted can be achieved in a practical system.

AUTHOR INDEX

B

Baugh, Robert N. 121
Baxter, Van D. 37
Braun, A. Tom 215

C

Catan, Mark A. 111
Chen, Fang C. 155
Chen, Norbert C. J. 181
Creswick, Frederick A. 149

D

Didion, David A. 89
Domingo, Norberto 99

F

Fairchild, Phillip D. 17
Fischer, Steven K. 55
Fiskum, Ronald J. 3

G

Griffin, Frederick P. 181
Griffith, William A. 121

H

Hayes, Floyd C. 289

J

Johnson, William S. 121

L

Lannus, Arvo 13

M

Macriss, Robert A. 233
Maret, Arthur R. 245
Marusak, Thomas J. 161
McGraw, Bernie A. 121
Mei, Vince C. 131
Miller, William A. 47
Modahl, Robert J. 289
Moyers, John C. 67
Mulroy, William J. 89

N

Nephew, Edmund A. 67

P

Phillips, Benjamin A. 263
Privon, George T. 209

R

Radermacher, Reinhard 225
Reimann, Robert C. 257
Rice, C. Keith 55
Ryan, John D. 141

S

Schreiber, Jeffrey G. 169
Stoecker, Wilbert F. 75

T

Teagan, W. Peter 193

V

Veyo, Stephen E. 25

W

West, Colin D. 181

Z

Zawacki, Thomas S. 233

AUTHOR AFFILIATIONS

Arthur D. Little, Inc.
20 Acorn Park
Cambridge, MA 02140
(617) 864-5770
W. Peter Teagan

Brookhaven National Laboratory
Building 701
Upton, NY 11973
(516) 282-2504
Mark A. Catan

Carrier Corporation
Research Division
P.O. Box 4808
Syracuse, NY 13221
(315) 432-7462
Robert C. Reimann

U.S. Department of Energy
1000 Independence Ave., SW
Washington, DC 20585
(202) 252-9130
Ronald J. Fiskum
John D. Ryan

Electric Power Research Institute
P.O. Box 10412
Palo Alto, CA 94303
(415) 855-2398
Arvo Lannus

Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631
(312) 399-8311
Arthur R. Maret

Institute of Gas Technology
4201 West 36th Street
Chicago, IL 60632
(312) 890-6444
Robert A. Macriss
Thomas S. Zawacki

Mechanical Technology, Inc.
Stirling Engine Systems Division
968 Albany-Shaker Road
Latham, NY 12110
(518) 438-7952
Thomas J. Marusak

NASA—Lewis Research Center
21000 Brookpark Road
Cleveland, OH 44135
(216) 433-4000
Jeffrey G. Schreiber

National Bureau of Standards
Building 226, Room B-126
Gaithersburg, MD 20899
(301) 921-2994
David A Didion
William Mulroy

Oak Ridge National Laboratory
P.O. Box Y
Oak Ridge, TN 37831
Van D. Baxter
Fang C. Chen
Norbert C.J. Chen
Frederick A. Creswick
Norberto Domingo
Phillip D. Fairchild
Steven K. Fischer
Frederick P. Griffin
Vince C. Mei
William A. Miller
John C. Moyers
Edmund A. Nephew
George T. Privon
C. Keith Rice
Colin D. West

Phillips Engineering Company
721 Pleasant Street
St. Joseph, MI 49085
(616) 983-3935
Benjamin A. Phillips

The Trane Company
3600 Pammel Creek Road
La Crosse, WI 54601
(608) 783-1925
Floyd C. Hayes
Robert J. Modahl

Tectonics Research, Inc.
9556 West Bloomington Freeway
Minneapolis, MN 55431
(612) 881-2617
A. Tom Braun

University of Illinois
1206 West Green Street
Urbana, IL 61801
(217) 333-0916
Wilbert F. Stoecker

University of Maryland
Mechanical Engineering Dept.
College Park, MD 20742
(301) 454-8876
Reinhard Radermacher

University of Tennessee
414 Dougherty Hall
Knoxville, TN 37996
(615) 974-5307
Robert N. Baugh
William A. Griffith
William S. Johnson
Bernie A. McGraw

Westinghouse Electric Corporation
Research and Development Center
1310 Beulah Road
Pittsburgh, PA 15235
(412) 256-1901
Stephen E. Veyo

CONFERENCE ATTENDEES

Ackermann, Robert B.
Mechanical Technology, Inc.
968 Albany-Shaker Road
Latham, NY 12110
(518) 785-2211

Albertson, Phil E.
The Charles Machine Works
P.O. Box 66
Perry, OK 73077
(405) 336-4404

Amalfard, M.
Air-Conditioning and
Refrigeration Institute
1501 Wilson Boulevard
Arlington, VA 22209
(703) 524-8800

Balinger, Richard K.
Washington Gas Light Co.
6801 Industrial Road
Springfield, VA 22151
(703) 750-4360

Ball, David A.
Battelle Columbus Laboratories
505 King Avenue
Columbus, OH 43201
(614) 424-4901

Baxter, Van D.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-2104

Benson, Glendon M.
ERG, Inc.
Lowell & 57th Streets
Oakland, CA 94608
(415) 658-9785

Biermann, Wendell
Consultant
45 Foxcroft Drive
Fayetteville, NY 13066
(315) 637-3048

Bledsoe, Jean L.
Oak Ridge National Laboratory
P.O. Box X, 4500N, B-214
Oak Ridge, TN 37831
(615) 574-2006

Braun, A. Tom
Tectonics Research, Inc.
9556 West Bloomington Freeway
Minneapolis, MN 55431
(612) 881-2617

Bryant, Ralph S.
Heil-Quaker Corporation
635 Thompson Lane
Nashville, TN 37204
(615) 244-0450

Calm, James M.
Electric Power Research Institute
P.O. Box 10412
Palo Alto, CA 94303
(415) 855-8949

Catan, Mark
Brookhaven National Laboratory
Building 701
Upton, NY 11973
(516) 282-2504

Chappell, Robert N.
DOE Idaho Operations Office
550 Second Street
Idaho Falls, ID 83401
(208) 526-0085 FTS 583-0084

Chaump, Louis E.
TRANE Dealer Products Group
6200 Troup Highway
Tyler, TX 75711
(214) 581-3512

Chen, Fang C.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-0712

Crawford, James G.
TRANE Dealer Products Group
6200 Troup Highway
Tyler, TX 75711
(214) 581-3603

Creswick, Frederick A.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-2009

Crowder, R. Scott III
Battelle Pacific Northwest Laboratories
P.O. Box 999
Richland, WA 99352
(509) 376-5595

Dao, Kim
Lawrence Berkeley Laboratory
University of California
1 Cyclotron Road
Berkeley, CA 94720
(415) 486-5786

De Bellis, David R.
Battelle Pacific Northwest Laboratories
P.O. Box 999
Richland, WA 99352
(509) 375-2649

De Lima, Henry
Applied Management Sciences
962 Wayne Avenue
Silver Spring, MD 20910
(301) 495-8636

Denny, Richard J.
Air-Conditioning and
Refrigeration Institute
1501 Wilson Boulevard
Arlington, VA 22209
(703) 524-8800

De Vault, Robert
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-0738

Didion, David A.
National Bureau of Standards
Building 226, Room B-126
Gaithersburg, MD 20899
(301) 921-2994

Domingo, Norberto
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-1083

Dudenhoefer, James E.
NASA—Lewis Research Center
21000 Brookpark Road
Cleveland, OH 44135
(216) 433-4000

Erickson, Donald C.
Energy Concepts Company
627 Ridgely Avenue
Annapolis, MD 21401
(301) 266-6521

Fairchild, Phillip D.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-2020

Ferris, Richard
Southern California Gas Company
9240 East Firestone Boulevard
Downey, CA 90241
(213) 803-7718

Fischer, Harry C.
Private Consultant
P.O. Box 445
Maggie Valley, NC 28751
(704) 926-9200

Fischer, Steven K.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-2017

Fiskum, Ronald J.
U.S. Department of Energy
Energy Conversion Equipment
1000 Independence Avenue, SW
Washington, DC 20585
(202) 252-9130

Freedman, Steven
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631
(312) 399-8390

Gerstmann, Joseph
Advanced Mechanical Technology
141 California Street
Newton, MA 02158
(617) 964-2042

Goldschmidt, Victor W.
Ray W. Herrick Laboratories
Purdue University
West Lafayette, IN 47907
(317) 494-2132

Goldwater, Bruce S.
Mechanical Technology, Inc.
Stirling Engine Systems Division
968 Albany-Shaker Road
Latham, NY 12110
(518) 785-2299

Griffin, Frederick P.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 576-4684

Griffith, Russell W.
Copeland Corporation
1675 West Campbell Road
Sidney, OH 45365
(513) 498-3579

Harnish, James R.
Borg-Warner Central
Environmental Systems
P.O. Box 1592
York, PA 17405
(717) 771-7165

Hastings, Charles A.
The Trane Company
2020 14th Street North
Arlington, VA 22201
(703) 525-4015

Hayes, Floyd C.
The Trane Company
3600 Pammel Creek Road
La Crosse, WI 54601
(608) 783-1925

Hickman, Kenneth E.
Borg-Warner Air
Conditioning, Inc.
P.O. Box 1592
York, PA 17405

Hughes, Patrick J.
W. S. Fleming & Associates
5802 Court Street Road
Syracuse, NY 13206
(315) 437-1789

Hundt, Roger C.
Lennox Industries, Inc.
1600 Metrocrest Drive
Carrollton, TX 75006
(214) 245-2525

Huntley, Peter
Borg-Warner Research
1200 South Wolf Road
Des Plaines, IL 60018
(312) 827-3131

Hurwitz, Anna
Consumer Energy Council
of America
Washington, DC
(202) 659-0404

Ingle, William D.
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631
(312) 399-8337

Izawa, Howard S.
Northern Natural Gas Company
2223 Dodge Street
Omaha, NB 68102
(402) 633-4741

Johnson, William S.
University of Tennessee
414 Dougherty Hall
Knoxville, TN 37996
(615) 974-5307

Kaplan, Stephen I.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-5819

Kapus, Ted E.
U.S. Department of Energy
Building Equipment Division
1000 Independence Avenue, SW
Washington, DC 20585
(202) 252-9123

Khan, Nasim H.
Northern Natural Gas Company
2223 Dodge Street
Omaha, NB 68102
(402) 633-4843

Khattar, Mukesh, K.
Florida Solar Energy Center
300 State Road 401
Cape Canaveral, FL 32920
(305) 783-0300

Kuliasha, Michael A.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-0330

Lannus, Arvo
Electric Power Research Institute
P.O. Box 10412
Palo Alto, CA 94303
(415) 855-2398

Leslie, Neil P.
Gard Division of
Chamberlain National
7449 North Natchez
Niles, IL 60648
(312) 647-9000

Lewis, Gerald P.
Argonne National Laboratory
9700 South Cass Avenue
Argonne, IL 60439
(312) 972-5332

Lewis, Pamela J.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-2012

MacCracken, Calvin D.
Calmac Manufacturing
150 South Van Brundt Street
Englewood, NJ 07631
(201) 569-0420

Macriss, Robert A.
Institute of Gas Technology
4201 West 36th Street
Chicago, IL 60632
(312) 890-6444

Maret, Arthur R.
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631
(312) 399-8311

Marusak, Thomas J.
Mechanical Technology, Inc.
968 Albany-Shaker Road
Latham, NY 12110
(518) 438-7952

McLinden, Mark O.
National Bureau of Standards
Building 226, Room B122
Gaithersburg, MD 20899
(301) 921-3844

Mei, Vince C.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 576-4945

Michel, John W.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-5271

Miller, William A.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-2013

Millhone, John
U.S. Department of Energy
Office of Building Energy
1000 Independence Avenue, SW
Washington, DC 20585

Mills, John
Idaho National Engineering Laboratory
P.O. Box 1625
Idaho Falls, ID 83415
(208) 526-9362

Minturn, Robert
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-5213

Modahl, Robert J.
The Trane Company
3600 Pammel Creek Road
La Crosse, WI 54601
(608) 787-2711

Modera, Mark P.
Lawrence Berkeley Laboratory
Building 90, Room 3074
Berkeley, CA 94720
(415) 486-4678

Mortazavi, Saied
Waterfurnace International
4307 Arden Drive
Fort Wayne, IN 46804
(219) 432-5667

Murphy, Kevin P.
Allied Corporation
P.O. Box 1087R
Morristown, NJ 07960

Nephew, Edmund A.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-2015

Peck, Richard G.
Peck Engineering Company
7 Water Street
Mystic, CN 06355
(203) 572-0311

Percival, Worth H.
United Stirling, Inc.
Huntington Beach, CA
(714) 896-4460

Perez-Blanco, Horacio
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-0419

Peterson, Stephen R.
National Bureau of Standards
Gaithersburg, MD 20899
(301) 921-3701

Phillips, Benjamin A.
Phillips Engineering Company
721 Pleasant Street
St. Joseph, MI 49085
(616) 983-3935

Privon, George T.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-2
Oak Ridge, TN 37831
(615) 574-1013

Radermacher, Reinhard
University of Maryland
Mechanical Engineering Dept.
College Park, MD 20742
(301) 454-8876

Raymond, Glendon A.
Carrier Corporation
P.O. Box 4800
Syracuse, NY 13221
(315) 432-7462

Reid, Edward A., Jr.
Columbia Gas System
Service Corporation
1600 Dublin Road
Columbus, OH 43215
(614) 481-1278

Reimann, Robert C.
Carrier Corporation
P.O. Box 4808
Syracuse, NY 13221
(315) 432-7462

Reinhold, Herbert E.
Energy Concepts Company
627 Ridgely Avenue
Annapolis, MD 21401
(301) 266-6521

Rice, C. Keith
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-2016

Ryan, John
U.S. Department of Energy
Energy Conversion Equipment
1000 Independence Avenue, SW
Washington, DC 20585
(202) 252-9130

Saunders, James H.
Brookhaven National Laboratory
Upton, NY 11973
(516) 242-3154

Savitz, Maxine L.
Consultant
5019 Lowell Street, NW
Washington, DC 20016
(202) 966-4456

Schreiber, Jeffrey G.
NASA—Lewis Research Center
21000 Brookpark Road
Cleveland, OH 44135
(216) 433-4000

Schultz, Gary W.
Zoneaire Corporation
P.O. Box 3517 CRS
Johnson City, TN 37601
(615) 282-2881

Shelton, Sam V.
Thermax, Inc.
296 14th Street, NW
Atlanta, GA 30318
(404) 874-5214

Sollman, Larry
Energy Systems Engineering
Bloomington, IN 47401

Spauschus, Hans O.
Research Institute
Georgia Institute of Technology
Atlanta, GA 30332
(404) 894-3530

Stoecker, Wilbert F.
University of Illinois
1206 West Green Street
Urbana, IL 61801
(217) 333-0916

Studstill, William
Southern Company Services
64 Perimeter Center East
Atlanta, GA 30346
(404) 457-5987

Teagan, William P.
Arthur D. Little, Inc.
20 Acorn Park
Cambridge, MA 02140
(617) 864-5770

Thomas, Stephen
National Center for
Appropriate Technology
815 15th Street, NW
Washington, DC 20005
(202) 347-9193

Vaughn, David
U.S. Army
Belvoir R&D Center
Ft. Belvoir, VA 22060

Vaughn, Thomas
Intertherm, Inc.
3345 Morganford Road
St. Louis, MO 63116

Veyo, Stephen E.
Westinghouse R&D Center
1310 Beulah Road
Pittsburgh, PA 15235
(412) 256-1901

Vineyard, Edward A.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 574-0576

Voorhis, Roger
Carrier Air Conditioning
P.O. Box 4808
Syracuse, NY 13221

West, Colin D.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9108
Oak Ridge, TN 37831
(615) 574-0370

Wright, Leslie R.
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631

Wurm, Jaroslav
Institute of Gas Technology
3424 South State Street
Chicago, IL 60616

Zimmerman, Kay H.
Oak Ridge National Laboratory
P.O. Box Y, Bldg. 9102-1
Oak Ridge, TN 37831
(615) 576-4921