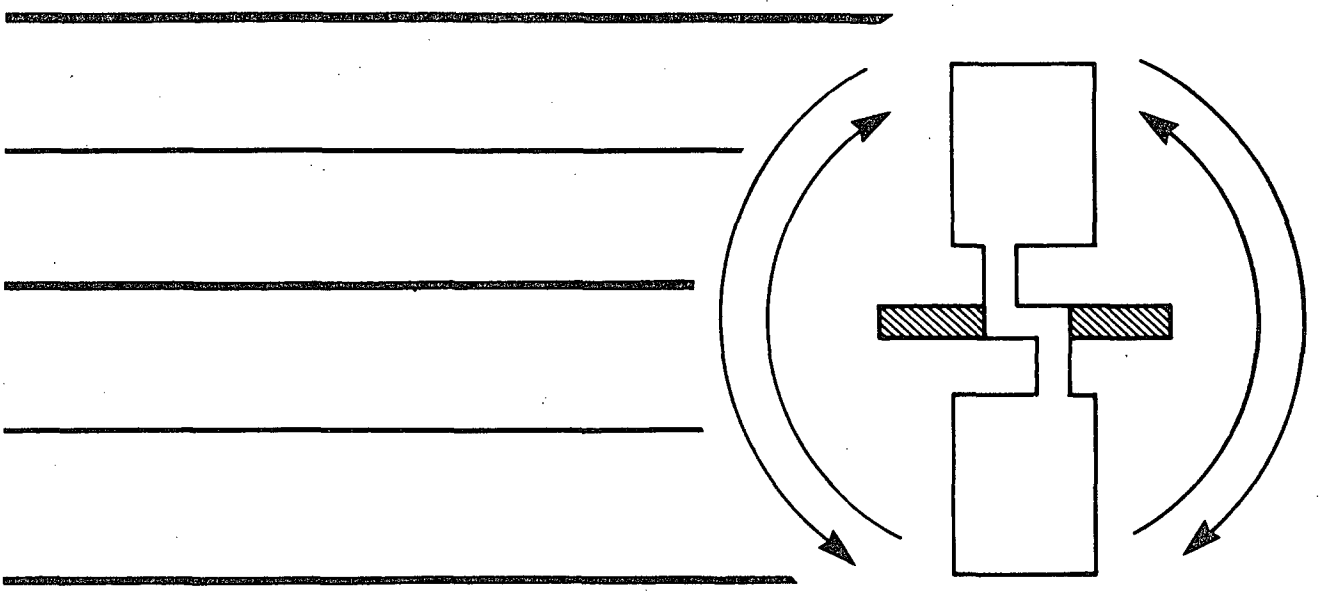
PROCEEDINGS
THE 7TH
HEAT PUMP TECHNOLOGY CONFERENCE



A University Extension Program of the
OSU Division of Engineering, Architecture and Technology

OCTOBER 15-16, 1984
TULSA, OKLAHOMA

Sponsored by
The School of Mechanical and Aerospace Engineering and
OSU Engineering Extension

OKLAHOMA STATE UNIVERSITY

PREFACE

This publication is a compilation of most of the papers presented at the Seventh Heat Pump Technology Conference, held at the Hilton Inn in Tulsa, Oklahoma, October 15 and 16, 1984.

For the travel convenience of the participants, the conference is being held for a second time away from the Oklahoma State University campus. We hope that the Tulsa location is satisfactory for the majority of the participants.

In the two years that have passed since the Sixth Conference was held, exciting developments continue to take place in the heat pump field. This conference is an opportunity for each of us to share our knowledge of these developments, to describe our experiences, and to consider solutions that may be beneficial to consumers, to utilities and manufacturers.

I hope that each of you will find some useful information for the areas in which you have concern. You will note that a very large number of papers deal with ground coupled heat pumps. My colleague, Dr. James E. Bose, has accomplished some particularly interesting and useful results with ground coupled heat pumps. If you have opportunity, you may want to visit Dr. Bose's laboratory on the Stillwater campus or to attend one of the design briefings, and/or workshops sponsored periodically by OSU's School of Engineering Technology.

I wish to thank the conference speakers for their willingness to participate in this conference. I especially appreciate the able assistance of my secretary, Dee Meier; Program Coordinator, Jacklyn Price; and Dr. Bill Cooper, Director of Engineering Extension. Dr. Karl N. Reid, Head of the School of Mechanical and Aerospace Engineering, continues to give his deeply appreciated support.

Jerald D. Parker, P.E.
Professor of MAE
Conference Director
October 1984

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SCHEDULE
7th Heat Pump Technology Conference

Hilton Inn — Tulsa, Oklahoma

MONDAY October 15, 1984

8:30 - 9:00	REGISTRATION (Coffee and Rolls)	
9:00 - 9:30	PAPER I	CARL C. HILLER
9:30 -10:00	PAPER II	RONALD J. FISKUM
10:00 -10:30	BREAK	
10:30 -11:00	PAPER III	NANCE C. LOVVORN
11:00 -11:30	PAPER IV	ALAN F. KESSLER
11:30 -12:00	PAPER V	M.C. ROBERTO BEST, C. DISPENZA, S. MERCADO, H. FERNANDEZ AND L. PIGNATO
12:00 - 1:00	LUNCH	(On Your Own)
1:00 - 1:30	PAPER VI	THEODORE L. SHELDON
1:30 - 2:00	PAPER VII	KAY H. ZIMMERMAN
2:00 - 2:30	PAPER VIII	EDWARD A. VINEYARD
2:30 - 3:00	BREAK	
3:00 - 3:30	PAPER IX	THOMAS J. MARUSAK AND J. MICHAEL CLINCH
3:30 - 4:00	PAPER X	W. T. HANNA, W. H. WILKINSON, D. A. BALL, AND A. R. MARET
4:00 - 4:30	PAPER XI	JAMES B. SEARS
6:30	MEXICAN FIESTA DINNER (Hosted by OSU)	

TUESDAY October 16, 1984

9:00 - 9:30	PAPER XII	U. GINO DI REZZE
9:30 -10:00	PAPER XIII	RAM S. NAGRANI AND R. DONALD DEXHEIMER
10:00 -10:30	BREAK	
10:30 -11:00	PAPER XIV	ROBERT D. FISCHER, GEORGE H. STICKFORD AND FRANK E. JAKOB
11:00 -11:30	PAPER XV	V. D. BAXTER, AND M. A. CATAN
11:30 -12:00	PAPER XVI	ALVIN C. BIERBAUM
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1:00 - 1:30	PAPER XVII	BERNIE MCGRAW, ROBERT N. BAUGH, WILLIAM S. JOHNSON, AND WILLIAM GRIFFITH
1:30 - 2:00	PAPER XVIII	HARRY J. BRAUD
2:00 - 2:30	PAPER XIX	JAMES R. PARTIN
2:30 - 3:00	BREAK	
3:00 - 3:30	PAPER XX	STEVE KAVANAUGH
3:30 - 4:00	PAPER XXI	RICK J. COUVILLION
4:00 - 4:30	PAPER XXII	JAMES E. BOSE AND JERALD D. PARKER
	ADJOURN	

FIELDS PERFORMANCE OF NOVEL, HIGH EFFICIENCY HEAT PUMPS

by Carl. C. Hiller

Carl C. Hiller

Recived BSME From the University of Michigan in 1971, graduating Summa Cum Laude.

Received SMME from the Massachusetts Institute of Technology in 1972.

Received the Mechanical Engineer's Degree from the Massachusetts Institute of Technology in 1974.

Received PhD in Mechanical Engineering from the Massachusetts Institute of Technology in 1976. PhD Dissertation: "Improving Heat Pump Performance via Compressor Capacity Control-Analysis and Test."

Employed with Sandia Laboratories, Livermore, California, from 1976 to present.

Paper not available.

Presented at the 7th Heat Pump Technology Conference at Tulsa, Oklahoma, October 15-16, 1984

DOE-RESIDENTIAL AND COMMERCIAL HEAT PUMP R&D PROGRAM

by Ronald J. Fiskum

Ronald J. Fiskum

Ronald J. Fiskum has been with the Department of Energy, Washington, DC, since 1980. As Program 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 BS in Engineering Science in 1968. Previous work has been in the thermal Design of Satellite with the Naval Research Laboratory and Grumann Aerospace Corporation.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

DOE - RESIDENTIAL AND COMMERCIAL HEAT PUMP R&D PROGRAM

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Program Manager
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INTRODUCTION

DOE research activities related to residential and commercial heat pumps technology are supported by the Office of Building Energy Research and Development, Building Equipment Division under the DOE Assistant Secretary of Conservation and Renewable Energy. The goals of the Building Equipment Division are to conduct long range, high risk technology R&D to provide the private sector with the technical basis for developing and testing high-efficiency equipment utilized in the operation of residential and commercial buildings and to carry out the intent of P.L. 94-163 as amended by P.L. 95-619 including development of test procedures, support to the Federal Trade Commission (FTC) Appliance Labeling Program, energy efficiency standards and consumer education.

BACKGROUND

The Energy Conversion Equipment Branch, formerly known as Technology and Consumer Products Branch, within the Building Equipment Division, in carrying out its mission in Heat Pump research has since 1977 contracted Oak Ridge National Laboratory (ORNL) to perform the technical management of the Heat Pump Research and Development Program and the Residential and Commercial Appliance Program. The objective of these programs is to develop a technology base for future development by the private sector which will result in improved energy utilization. The objective of these programs is to concentrate on understanding the inherent efficiency limits of conventional building equipment and establish the technical feasibility of advanced efficiency improvements or options and to develop the concepts to a state of readiness for application by the private sector to building HVAC and auxiliary equipment. Program efforts are focused on generic technology base development. This research is generic in nature, high risk and long-term efforts that the private sector is not likely to engage in or does not have the facilities or resources to accomplish. Programs are coordinated with various National Laboratories, the Gas Research Institute, Electric Power Research Institute and Universities.

PROGRAM SIZE

Operating with a budget of $6.3 million per year, the Heat Pump Program is divided into two segments. One segment deals with thermally activated heat pumps. The other segment deals with Electric driven refrigeration systems and appliance. Major emphasis is being placed on thermally activated systems and funding in the amount of $4.0 million is allocated to this research program.

THERMALLY ACTIVATED HEAT PUMP PROGRAM

Three principal areas of research have developed under this program: (1) absorption heat pumps and chillers; (2) Free Piston Stirling engine-driven heat pumps; and (3) Internal Combustion Engine driven heat pumps. The efficiency targets for these three technologies for the near term and advanced systems are given in the following table. All heating COP's listed include utilization of waste heat as shown. The COP's listed do not include parasitic losses. The steady state COPs for heating and cooling are at 47°F and 95°F out door air temperature.

TECHNOLOGY	NEAR TERM HCOP*	NEAR TERM CCOP*	FUTURE HCOP*	FUTURE CCOP*	WASTE HEAT USE
Absorption	1.6	0.7	2.2	1.6	85%
IC Engine	1.8	1.1	2.1	1.5	75%
Stirling	1.9	1.1	2.3	1.4	80%

*Steady-state heating and cooling COPs at 47°F and 95°F, out door air temperature respectively

Current Projects:

A. Absorption Heat Pump Technology:

Absorption heat pump projects currently underway, are subcontracts to Carrier, Trane and Phillips Engineering to identify and evaluate advanced absorption cycles. Each subcontract consists of two parts: (1) selection and evaluation of a preferred cycle utilizing a fluid pair with well-established properties; and (2) identification and evaluation of an innovative new cycle using a ternary fluid combination. The target COPs for these near term and future accomplishments are given in the previous table. In each project, provision is made for analyses necessary to select a preferred cycle and fluid

corrosion and stability problems, and for the construction, instrumentation and operation of a breadboard system to establish proof-of-concept.

The remaining on-going projects are in base support of the technology. Of these is the development of analytical tools for advanced cycle research. A computer model for use with binary fluids in conventional cycles is being developed under subcontract to Technion University in Israel, and it may be extended to alternate cycles and fluids and will include capacity modulation and system optimization. The model will be validated against experimental data.

The remaining on-going support projects are in absorption fluids. An investigation into the sensitivity of cycle efficiency calculations has been completed. This showed that accurate vapor pressure data were needed in order to make reliable calculations for advanced cycles. An experimental/analytical project into the dynamics of heat and mass transfer in absorption systems is proposed. The objective of this is to provide basic information on heat and mass transfer of absorption fluids over the range of conditions for the advanced cycles. Currently, design specifications for the equipment and its instrumentation are being developed. The Institute for Gas Technology is conducting an international survey and compilation of existing fluids property data. Missing data critical to the technology development will be identified so that steps may be taken to make them available.

A project in the development of advanced fluids is being planned for near-, intermediate- and long-term absorption fluids development for advanced absorption concepts. Chemical support studies to elucidate the corrosion and stability characteristics of existing absorption fluids in support of the advanced cycles hardware development activities is in thee planning stage.

B. IC Engine-driven Heat Pump Technology:
The purpose of this project is the evaluation of Internal Combustion Engine for heat pump use. Subproject areas identified are technology assessment, component development and systems development. A technology assessment, carried out by A.D. Little, has been completed and concluded that IC engine technology can achieve substantial savings and has strong economic potential at the present time in larger sizes (50HP). The key areas of uncertainty were identified as first costs,noise isolation and emissions. The primary work under system development has been the development and testing of a linear engine designed by Tectonics Research, Inc. The demonstration of system proof-of-concept and the utility of a hermetic bellows seal has been established. Work now underway, with the objective of verifying by breadboard test, is focused on high performance of a heat pump system driven by a fuel-injected linear engine of demonstrated high efficiency. Extension of the work to the design and cost analysis of a residential sized model is anticipated.

The research continues for new engine concepts, which, if properly research and developed, could lead to substantial improvements in durability, efficiency and carefree maintenance. It is also anticipated that as highly-efficient gas-fired engines through technology advancements, it will be necessary to develop advanced refrigerant compressors capable of high speed operation, so that full advantage can be taken of the modulation features of the IC engine.

C. Stirling Engine Driven Heat Pump Technology.
Heat Pump research in the Stirling area is concerned with systems development, component studies, supporting research and technology assessment.

Systems development: Two systems development projects are underway. The largest of these is the development of a Free Piston Stirling Engine-Driven, diaphragm-coupled compressor for residential heat pump. The objectives of this research are to demonstrate the technical feasibility of the concept, to characterize performance improvement. DOE and the Gas Research (GRI) are cosponsoring this work, and Mechanical Technologies, Inc., is the subcontractor. GRI has funded engine development, and DOE has funded the research in the compressor and in the engine/compressor coupling.

At the present time, satisfactory engine performance has been attained, and the engine has been coupled, to a refrigerant compressor. Efficiency losses, however, occur in the coupling, and efforts are underway to identify and reduce those losses.

The second concept evaluation study effort has as its objectives to demonstrate the performance potential of a Stirling cycle heat pump, to investigate its controllability, and to develop concepts for a compact, high-performance duplex Stirling (Stirling-Stirling) heat pump. At the present time, analytical efforts are underway to estimate the feasibility of the Stirling cycle heat pump, and this, coupled with results from supporting research described below, will determine future levels of effort.

Components study: The objectives of this effort are to obtain reliable heat transfer data for use in design analysis and to test novel heat exchanger concepts for Stirling cycle machines. Under subcontract, ERG is testing the heat transfer enhancement made possible by a novel heat exchanger design, in an effort to make the compression stroke in a Stirling cycle machine more nearly isothermal. It is anticipated that the heat transfer characteristics of a number of existing heat exchangers for Stirling engines will be tested in the oscillating flow test facility at Argonne National Laboratory.

Supporting research: The objectives of this task are to obtain Stirling engine test data for the validation of computer codes for engine design and to develop an improved design methodology. NASA, under subcontract, has collected performance data during a 1000-hr engine endurance test, and continued endurance testing to 10,000 hours total is underway with NASA and GRI funding. Under DOE funding, NASA is using the experimental data to validate the RE-1000 computer model, and is monitoring the design and fabrication of a hydraulic dynaometer to facilitate the test of engine operating characteristics under a variety of load conditions. At ORNL, a small computer code

based on simple harmonic analysis has been developed to facilitate the identification of loss mechanisms in Stirling engines.

Technology assessment:

The objective of this subcontracted work (A.D. Little) is to provide an independent assessment of the prospects for technical and economic feasibility of free pistonStirling engine-driven heat pumps, and to identify a range of opportune research and development options. Issues to be addressed include costeffectiveness, durability, efficiency and reliability.

As a result of this program, it is anticipated that more efficient and economically viable fossil-fired systems for heating and cooling buildings will be available. These will be attractive alternatives to conventional oil and gas furnaces and boilers as the fossil-fired system of choice. These alternatives point toward significant reduction in future use of gas and oil for building space conditioning.

REFRIGERATION AND APPLIANCES TECHNOLOGY

The objectives of the research in this area are to develop the technology to bring heat pumps, refrigeration systems and appliances to the practical limits of efficiency, usefulness and effectiveness, including consideration of utility load impact to identify new applications of the technology which can yield nationally significant energy savings.

Ground-coupled heat pump projects are currently underway. Of these, three are field experiments, two involve ground heat-and-mass transfer, one is a system-design project and technology-transfer activity, and one is a technology assessment.

The basic approach being followed on ground-coupled systems is to develop reliable, analytically based design methods for optimum coil configuration and to demonstrate thermal performance and cost effectiveness in field experiments. Some laboratory experiments will be used to provide detailed validation data. A secondary activity is to develop a design for a heat pump mechanical package that is well suited to ground coupling.

The analytical models of the ground coil heat transfer and related soil moisture effect are being developed by ORNL. The program-supported field experiments, are being conducted at the local TECH site near Knoxville, TN. Two of the experiments involve horizontal ground-coil installations in two different soil types. Another experiment involves a vertical-coil installation. 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 these analytical models. The laboratory experimental work to examine moisture migration and soil freezing phenomena under the IEA project will be conducted by the National Research Council - Canada.

The secondary objective related to development of a water - source heat pump mechanical package design optimized for ground-coupled application is a joint effort of ORNL, BNL, and Frederick (the industry contractor). This is, in part, a technology transfer activity to work closely with a manufacturer in utilizing the heat pump model and other related analytical tools and techniques previously developed at ORNL.

The technology assessment work has been conducted under subcontract by Battelle and has included a worldwide state-of-the-art survey of the technology plus technology and economic assessments of several different ground-coil configurations.

Dynamic Losses:

Projects in this area may be viewed as basic supporting research for all heat pump technology in the sense that they address 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 some performance data from prior equipment development projects supported by DOE is utilized to help establish advanced technology equipment characteristics.

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

Major projects in this are include field experiments, laboratory study, data survey, and analytical efforts.

Field experiments are being conducted by ORNL near knoxville and by Westinghouse in a residence near Pittsburgh. This work has focused on state-of-the-art (SOA) single-speed residential air-to-air heat pumps for the past few years, but is currently changing to a SOA two-speed unit. The Westinghouse preprototype dual-stroke advanced heat pump developed under a previous DOE/ORNL cost-shared contract in its second year of testing.

ORNL laboratory studies are aimed at detailed measurement of refrigerant migration with varying on-off cycles, frost formation with varying relative humidities, and other such factors to develop an understanding of effect and cause of dynamic losses. The study will cover both single-speed and variable-capacity equipment.

To utilize the dynamic loss data design analysis tools and techniques are needed to 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. 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. The computer programs have been developed and current efforts are on validation, technology transfer and documentation. Cooperative working arrangements with industry have been implemented which will 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:

Ongoing projects involve scouting experimental work with nonazeotropic mixtures in actual hardware systems, basic supporting research, and a technology assessment project. Two new projects are parallel concept generation and evaluation projects on advanced cycles for composition/capacity control. The objectives in this research area are: to develop an understanding of nonazeotropic refrigerant mixture behavior in heat pump and refrigeration equipment and to use this information to (a) develop an analytical base for component development and system design, including reliable techniques for predicting mixture properties; (b) determine the applications in which nonazeotropic mixtures are most promising and the potential equipment performance improvements and related energy savings benefits; and (c) identify and characterize new, improved refrigerant mixtures (e.g., ternary mixtures). The present focus is on capacity modulation and leveling in heat pumps. Other potential applications include two-evaporator refrigerator-freezers, two-condenser heat pumps, water-source heat pumps and heat-pump water heaters using counterflow heat exchangers.

The basic supporting research project at the University of Illinois involves condensing heat transfer studies of mixtures and generalized system simulation work.

The technology assessment is being done as part of an ongoing international project under IEA, with Chalmers University of Technology in Sweden and the University of Technology in Graz, Austria, leading the technical effort.

Commercial Building Systems Research:

Work in this area presently consists of an effort conducted by ORNL to establish R&D needs and to organize a national R&D program which will involve both the Federal Government and the private sector. An advisory group will be organized to help identify R&D needs. Coordination efforts have been initiated with other organizations working in this area which include other DOE laboratories (Lawrence Berkeley Laboratory, Pacific Northwest Laboratory, Argonne National Laboratory), the U.S. Army Construction Engineering Research Laboratory (CERL), NBS, EPRI, and Gas Research Institute.

Building Equipment Occupant Interaction:

This project area is currently in the planning stage. A Request for Proposals (RFP) has been issued which will gather information on work already completed or underway. The data from this survey will be used to formulate future research objectives and plans.

Private sector participation is needed to make this program a well balanced, non-redundant effort. Participation of manufacturers, consultants, universities and other institutions will be solicited to fulfill the needs of this program.

Advanced Appliance Insulation:

Advanced appliance insulation activities presently involve investigation of a single material concept--compressed ultrafine silica powders havein interstitial pore sizes maller than the 0.07mm mean free path of air at atmospheric pressure and room temperature. Both analytical and experimental efforts are involved, including the evaluation of various candidate powders. The ultimate objectives of this work are to demonstrate R/in. values of 15 ft^{o} x h x ^{o}F/Btu x in. (k=0.0096 W/m x K) in air and 20 (k=0.0072) in vacuum, and to transfer this technology to one or more industrial organizations that would be potential suppliers. Attainment of these objectives would provide the technology for reducing the energy consumption of refrigerators, ovens, and water heaters by up to one half.

Recent experiments have demonstrated that the conductivity of ultrafine silica powders in air can be reduced below the thermal conductivity of air, but results are not close to the level that theory indicates is attainable. Efforts in the near future will be aimed at determining separately the radiative and conductive heat transfer components in the material.

CONCLUSION

The Energy Conversion Equipment Branch in its quest for achievement of increased energy conservation in the residential and commercial building sector will continue the development of technologies for the improvement of efficiencies in building equipment. Planned research will focus on high risk long term generic activities which lead to development of the most promising concepts and demonstrate that the technical risk have been reduced.

The transfer of these technologies to the private sector will be achieved through workshops, reports and industrial interaction.

LIST OF PUBLICATIONS

1. "Development of a Residential Gas Fired Absorption Heat Pump - Final Report," Allied Corporation, ORNL/SUB/79-24610/3-5 (In Press)

2. "Engineering Development and Demonstration of an Ammonia-Water Absorption Heat Pump," Arkla Industries, Inc., ORNL/SUB/80-61603/1 (In Press)

3. "Development and Proof-testing of Advanced Absorption Refrigeration Cycle Concepts-Phase I Final Report," Carrier Corporation, ORNL/SUB/81-17499/1 (DRAFT)

4. "Development and Proof-testing of Advanced Absorption Refrigeration Cycle Concepts-Phase I, Final Report," Phillips Engineering Company, ORNL/SUB/81-17497/1 (DRAFT)

5. "Development of a Brown Linear Engine-Driven Heat Actuated Heat Pump-Final Report," July 1984," Honeywell, Inc./Tectonics Research, Inc., ORNL/SUB/80-61619/1

6. "Development and Demonstration of a Stirling/Rankine Heat Actuated Heat Pump - Final Report," GE Document No. 83-AEP-004, Volume I, II, III, April 1984

7. "Free Piston Stirling Engine Diaphram Coupled Heat-Actuated Heat Pump Component Technology Program-Technical Status Report," MTI Report No. 83FPSE10 (September 1983)

8. "State-of-the-Art Survey of Existing Knowledge for the Design of Ground-Source Heat Pumps," Battelle Columbus Laboratories, ORNL/S013/80-7800/2&06

9. "Technical and Economic Feasibility of Horizontal, Multiple Shallow Well and Deep-well Ground Coupling for Residential Heat Pump Applications," Battelle Columbus Laboratories, ORNL/SUB/80-7800/3&06, April 1983

10. "A Theoretical and Experimental Analysis of Vertical Concentric-tube Ground-Coupled Heat Exchangers," ORNL/CON-153 (August 1984)

11. "New Approach for Analysis of Ground-Coil Design for Applied Heat Pump Systems," ORNL, ASHRAE Transactions," Volume 91, Part 1 (To be Published)

12. "Field Measurement of Cycling Frosting, and Defrosting Losses Experienced by a High-Efficiency Air-to-Air Heat Pump," ORNL/CON 150 (July 1984)

13. "Advanced Electric Heat Pump Dual-Stroke Compressor and System Development-Final Report," Westinghouse, ORNL/SUB/79-24712/3 (December 1983)

14. "The Oak Ridge Heat Pump Model, A Steady-State Computer Design Model for Air-to-Air Heat Pumps," ORNL/CON-80/R1 (August 1983)

15. "Development of Advanced Thermal Insulation for Appliances," ORNL/CON-159 (July 1984)

16. "RD&D Opportunities for Large Air Conditioning and Heat Pump Systems-Final Report, "TRW, Inc., ORNL/SUB-80/13817C/1 (October 1981)

HEAT PUMP RELIABILITY

by Nance C. Lovvorn

Nance C. Lovvorn

Mr. Lovvorn holds a BS Degree from the School of Engineering at Auburn University and an MBA from Samford University. He joined Alabama Power Company in 1967.

Mr. Lovvorn is a member of ASHRAE and has served on several committees of ACCA. He is also currently serving as a consultant to EPRI on heat pump reliability.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

HEAT PUMP RELIABILITY

Nance C. Lovvorn
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Alabama Power Company has been involved with heat pump reliability through its Assured Heat Pump Service Program since 1967. With units installed as early as 1964, some 18,000 units have been under this ten year service program. The purpose of this program was to help overcome buyer resistance to heat pumps which existed due to lack of confidence in reliability. Many improvements including equipment and dealer installation and service practices have had a positive impact on this trend toward improving customer acceptance. Today, about one-third of the new home buyers in Alabama have heat pumps and almost all people who have heat pumps replaced, use another heat pump.

Since participating in Oklahoma State University's first Heat Pump Technology Conference in 1975, Alabama Power has increased its data base significantly. New insights into trends have led APCo to identify areas of emphasis for the future.

In 1975, compressor failure rates on package, remote, and combination were discussed by age. At that time, reports were available only on units of 5 years age or less. This paper will address compressor reliability, cost to service, and service frequency, as described below and is based on data available March 30, 1984.

- Compressor Failure Rates By Age
  - Package
  - Remote
  - Total
- Cost To Service By Age
  - Package
  - Remote
  - Total
- Units Serviced By Age
  - Package
  - Remote
  - Total
- Service Frequency By Year
- Cost To Service Per Service Call
- Cost To Service Per Unit Per Year

Compressor Failure Rates

Table #1 shows the compressor data by year of installation rather than year of failure. Taking the five year figures, for example: of the package units installed in 1979, 14 compressors have failed on 708 unit years (172 units) for a 1.9% annual compressor failure rate. These failures could have occured during any year (1st, 2nd, 3rd, 4th, or 5th). On remotes, 54 failures on 1,890 unit years (464 units) for a 2.8% failure rate. For a combination of both packages and remotes, figures are 68, 2,598, and 636 respectively for a 2.6% rate.

TABLE 1
Compressor Failure Rates

YEAR INSTALLED	AGE	PACKAGE F.R.(%)	REMOTE F.R.(%)	TOTAL F.R.(%)
1983	1	0.9	0.3	0.5
1982	2	1.1	0.8	0.9
1981	3	1.2	1.9	1.1
1980	4	0.4	2.1	1.6
1979	5	1.9	2.8	2.6
Cum.	1-5	1.2	1.9	1.7
1978	6	2.8	2.6	2.6
1977	7	3.6	3.5	3.5
1976	8	3.6	4.0	3.8
1975	9	5.2	5.3	5.3
1974	10	4.0	4.4	4.2
Total		3.4	3.5	3.5

Source: March 30, 1984, ASP, Alabama Power Co.

Table #2 shows the 1974 data which was used at the 1975 OSU Conference. The older years were not included at that time.

TABLE 2
Compressor Failure Rates

YEAR INSTALLED	AGE	PACKAGE (%)	REMOTE (%)	TOTAL (%)
1973	1	2.8	2.1	2.5
1972	2	3.2	4.2	3.8
1971	3	3.7	5.2	4.8
1970	4	5.2	6.7	6.1
1969	5	5.1	7.8	6.8
Cum.	1-5	4.6	6.3	5.[illegible]

Source: September 30, 1984, ASP, Alabama Power Co.

A quick glance at Tables 1 and 2 shows significant differences in every year and dramatic improvement in the 1-5 cumulative categories as follows:

	1974	1984	% Change
Package	4.6	1.2	74
Remote	6.3	1.9	70
Total	5.6	1.7	70

What factors led to the improvements? Can the figures be improved even more? There are some manufacturers who have made tremendous individual improvements in their products, others who have remained at an acceptable level, and some who have lost ground. We also know "heat pump" compressors and associated refrigerant components have entered the market during this time frame. Some believe further improvements can be made to these rates and have set goals accordingly. The high SEER race and factory cost reduction efforts aimed at increasing share of market with the builders have had a strong influence on others. Both are important considerations.

What happens beyond the fifth year? Do failures increase with age? Table #1 gives a clearer picture to both questions for all manufacturers; however, a closer look is required. For example, for units installed in 1974, or 10 year old units, the failure rates were 4.0%, 4.4%, and 4.2% for package, remote, and total, respectively. This means a compressor failure rate of 4.2% has been experienced on units installed in 1974 but the failure may have occurred in any year, first through tenth. One scenario could be that units manufactured in a particular year experienced different results from others. This case has proven accurate with some manufacturers. A more likely scenario, however, is that longer life to the unit gives use to additional mechanical problems. Tables #3 and #4 support the latter theory.

Charts #1 and #2 show four manufacturers who have been active over the past ten years and have significant sample sizes. Earlier points made about different levels of improvement being made are graphically displayed.

Cost to Service is a prime consideration to individual homeowners and to Alabama Power Company. Table #3 shows cumulative cost to service units that were installed between 1974 and 1983, with the cost shown representing cost that may have occurred in any year. Labor rates are currently averaging $30/hour in 1984 versus approximately $15 in 1974. The closeness in package and remote figures on cost and compressor failure rates are significant also; because that was not the case ten years ago. Previously we had seen this happen on only a few manufacturers (1 or 2); remotes being generally higher on some and significantly higher on others. The prime factor in this cost consideration is the compressor cost.

TABLE 3
Cost of Service/Unit Year ($)

YEAR INSTALLED	AGE	PACKAGE	REMOTE	TOTAL
83	1	$11.51	$ 2.48	$ 5.01
82	2	11.60	11.87	11.78
81	3	21.83	17.65	19.20
80	4	18.29	29.18	25.76
79	5	33.87	32.57	32.92
78	6	31.86	32.05	32.00
77	7	34.49	37.33	36.54
76	8	43.63	49.47	47.31
75	9	55.33	54.75	54.97
74	10	54.14	63.32	59.55

Source: March 30, 1984, ASP, Alabama Power Co.

Units Serviced by Age

Service requirements for different age units are shown in Table #4. Again with age more units require service. If data were available on other mechanical equipment, it would likely show similar results. This particular chart shows those units which have been serviced one or more times during Alabama Power Company's contract period. Some manufacturers equipment has been serviced more than others, of course; however, about 9% of the units go 10 years without any service calls being made. The number requiring service increases about 10% per year. Also, overall there is little difference in package versus remote equipment.

TABLE #4
Contracts Serviced One or More Times

YEAR INSTALLED	AGE	% SERVICED PACKAGE	REMOTE	TOTAL
83	1	7.4	2.3	3.7
82	2	12.7	11.7	12.0
81	3	28.7	29.8	29.4
80	4	39.2	46.1	44.0
79	5	57.0	62.1	60.7
78	6	64.6	61.8	63.8
77	7	66.7	73.6	71.7
76	8	80.1	82.6	81.7
75	9	86.5	83.0	84.3
74	10	89.8	92.5	91.4

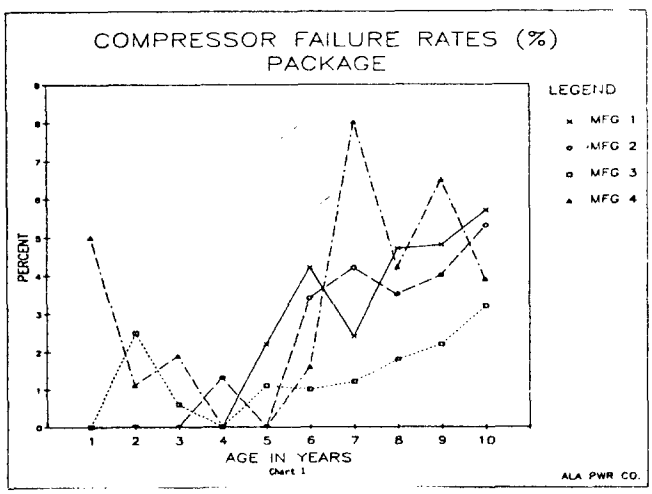
Charts #3 and #4 compare the same four (as in #1 and #2) manufacturers but little difference is noted except variations between manufacturers. As each gets older, more service is required; however, a slight variance can be noted between manufacturers.

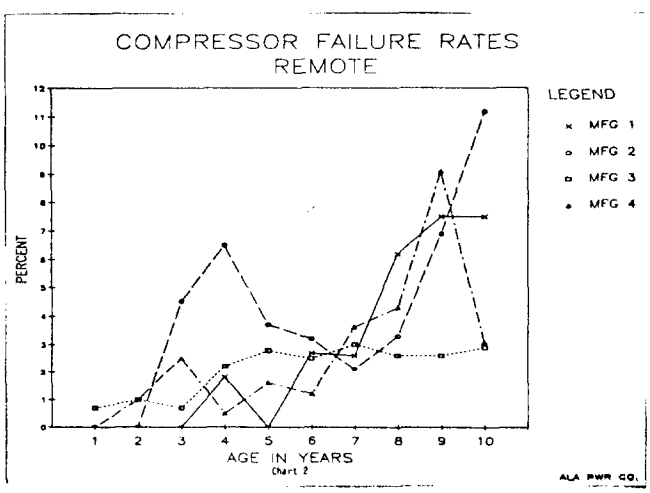
The final three charts relate to frequency of service required, cost per service occurrence, and overall cost to service by year. These are on an individual year basis, hence not cumulative. Also, on each exhibit yearly data has been re-calculated in 1983 dollars to reflect the effects on inflation. These include all manufacturers on the ASP.

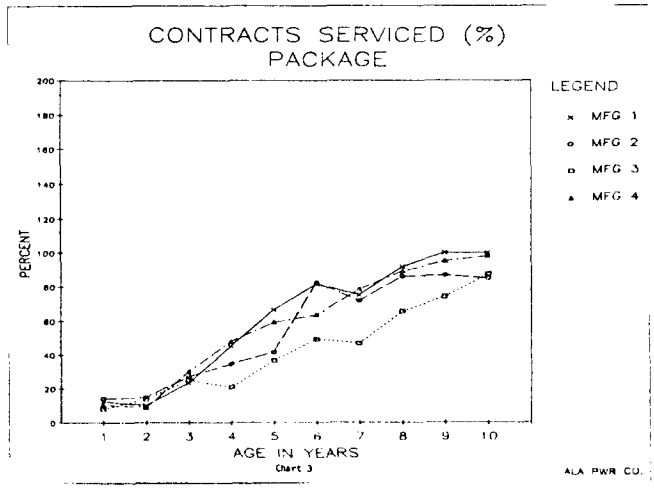
Chart #5 gives clear evidence that heat pumps required less service in 1983 than in 1971. Fortunately, manufacturers have made units more reliable.

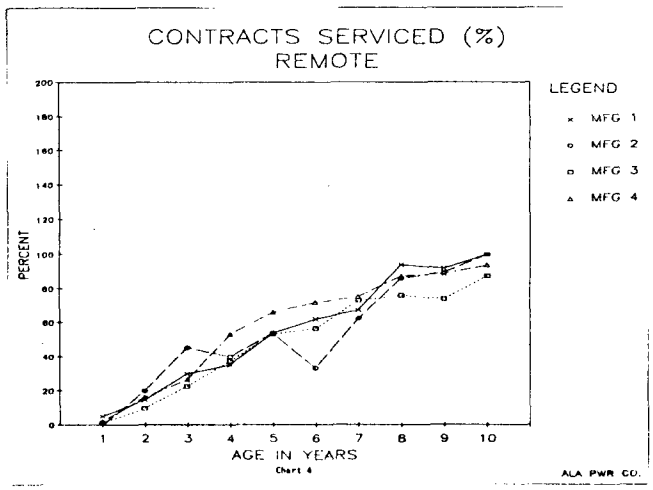
The cost per service occurrence (Chart #6) is a function of labor rates, parts cost, equipment age, and serviceability. When the labor and parts inflationary trends are eliminated, age and serviceability are significant contributors. Culmination of contractual commitments on manufacturers whose units are no longer active helped influence this trend also.

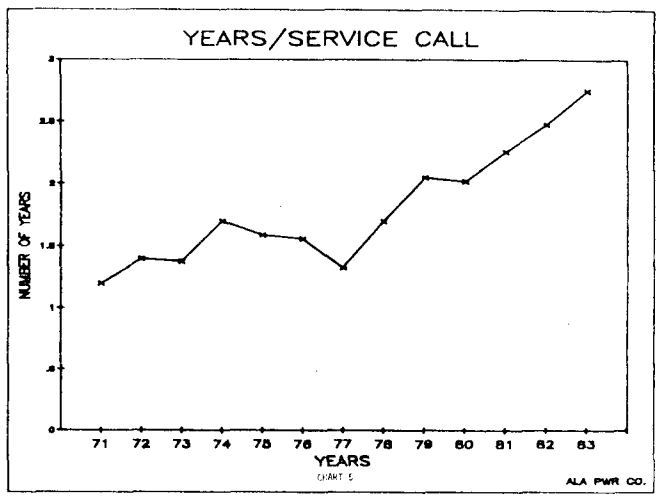
The total cost to service (Chart #7) is influenced by the above plus the increased average unit age. The average age in 1973 was 3.1; in 1976, 4.6; in 1983, 5.2. With the constant dollar factors applied to all considerations, it is rather obvious to an unbiased observer that heat pumps are more reliable, require less service, and certainly operate more efficiently than in the past years of this report.

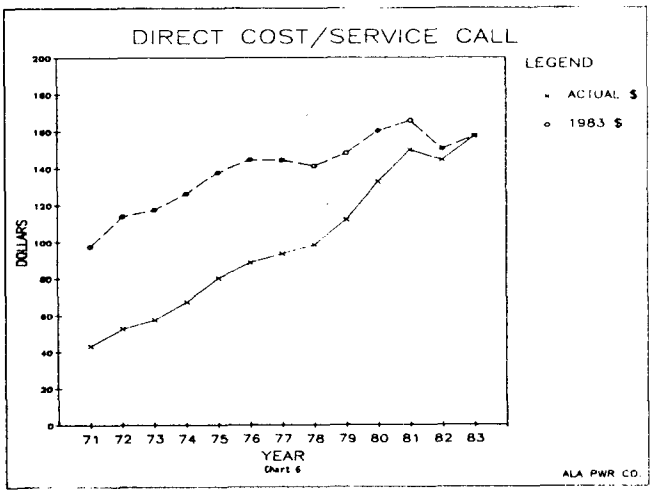


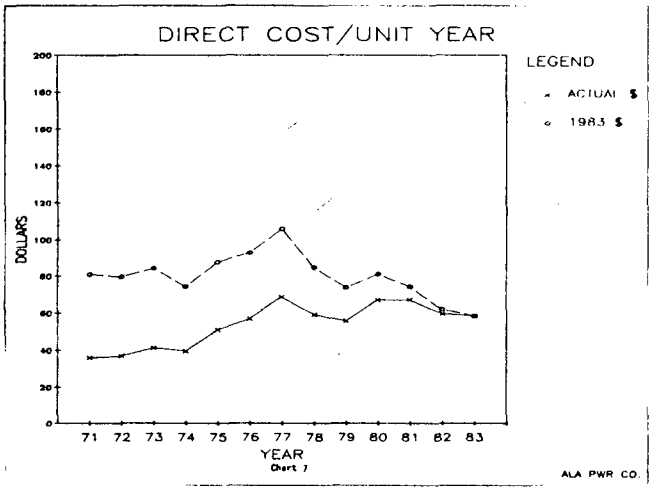









DIRECT COST/SERVICE CALL
LEGEND
ACTUAL $
1983 $
DOLLARS
200
180
160
140
120
100
80
60
40
20
0
71 72 73 74 75 76 77 78 79 80 81 82 83
YEAR
Chart 6
ALA PWR CO.


DIRECT COST/UNIT YEAR
LEGEND
ACTUAL $
1983 $
DOLLARS
200
180
160
140
120
100
80
60
40
20
0
71 72 73 74 75 76 77 78 79 80 81 82 83
YEAR
Chart 7
ALA PWR CO.

WEATHERTRON III - WATER HEATING AND SPACE CONDITIONING HEAT PUMP

by Alan F. Kessler

Alan F. Kessler

Alan F. Kessler, P.E., is Manager, Product Development Engineering, and Design of new residential comfort systems for The Trane Company, Dealer Products Group. He has been with The Trane Company since 1970 in various engineering design and management positions responsible for the design and development of residential and commercial air conditioning systems and components.

Mr. Kessler received a BSAgE (Machine Design) from Kansas State University and MSE (Mechanical Engineering) from Purdue University. He is a member of ASHRAE and a licensed professional engineer in several states.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

WEATHERTRON III
WATER HEATING AND SPACE CONDITIONING HEAT PUMP

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This paper discusses the design, features and operation of the recently introduced Trane Weathertron III Heat Pump Water Heating System and includes a comparison of performance to other domestic water heating systems.

Water heating makes up approximately 20% of the total energy usage in a home, while space heating and cooling are 60% and 10%, respectively depending on location. Therefore, reducing water heating costs by heating with a heat pump at a COP of 2.5 would reduce total home energy usage by 10% to 12%.

Domestic water is generally provided through fossil fired heating units or electric resistance water heaters. When electric resistance water heaters are utilized there are three alternative electric heating concepts available to reduce home energy consumption. They are desuperheaters, dedicated heat pump water heaters and the new Trane system, which is an integrated home air conditioning, heating and hot water heat pump.

The Weathertron III, an integrated air conditioning, heating and domestic hot water heating system. The basic components of the Trane Weathertron III system are the outdoor unit, indoor air handler, water heating unit and computerized control center.

The hot water desuperheater system heats water only when the air conditioning system is cooling or heating. It is low in cost and has good payback. The dedicated heat pump water heater heats water on demand, has moderate cost and payback. It provides year round localized cooling which is a disadvantage in the winter, and it requires only plumbing skills for installation. The Weathertron III system heats water on demand, has a good payback, excellent recovery capability, utilizes either indoor or outdoor air as a source, provides free cooling in the Summer and utilizes conventional available HVAC dealer skills.

Unique aspects of system design and operation are described in United States Patent Nos. 4,299,098, 4,399,644 and others pending. A significant feature of the invention lies in the fact that during operation in any particular mode, the inactive heat exchanger of the circuit is vented to the suction side of the compressor, so as to afford proper refrigerant control within the system. This will be seen as we view the various operating modes.

In space cooling, heat is transferred from the indoor coil to the outdoor unit. Figure 1 shows the refrigerant operation with dark arrows being high side discharge progressing from the compressor to the water reversing valve (WRV) to the refrigerant reversing valve (RV) to the outdoor coil. In the outdoor coil, it is condensed to a liquid and passes through the water heating unit to the indoor coil, where it is expanded and absorbs heat. The low pressure suction gas returns through the reversing valve and the accumulator to the compressor. The unused water heating coil is vented to suction pressure through the water reversing valve back to the compressor suction line.

In space heating, heat is transferred from the outdoor unit to the indoor coil. The circuiting shown in Figure 2 is reversed from that shown in Figure 1 by switching the reversing valve (RV) and one of the solenoids in the water heating module. There again, the water heating module remains vented to the compressor suction line.

When heating water during the spring, fall and winter using the outdoor source, heat is transferred directly from the outdoor unit to the indoor water heating unit. In this case, the water reversing valve (WRV) is shifted as shown in Figure 3 and discharge gas is directed to the water heating coil to heat water. It is returned through an open solenoid valve as a liquid to the outdoor coil where it is expanded and absorbs heat. It passes through the normal reversing valve and accumulator back to the compressor. When the water reversing valve was switched, it also evacuated through the refrigerant reversing valve, the indoor coil as shown.

In the summer water heating and space cooling mode, heat is transferred from the indoor conditioned space to the hot water, utilizing the outdoor unit only for the compressor function. Figure 4 shows the refrigerant circuit to accomplish this. The refrigerant reversing valve is used to switch between indoor and outdoor coils as a source along with appropriate solenoid valve changes in the water heating module. Again, you will note that the unused outdoor coil is evacuated to suction pressure to manage refrigerant and maintain proper system charge and operation.

As noted earlier, this system is more complex than desuperheaters, dedicated heat pump water heaters or standard space conditioning heat pumps. As a result, microcomputer controls are utilized to integrate the operation and decision making require-

ments of the system. Utilizing microcomputer controls allows additional features and benefits to be added at minimal incremental cost for software development. The system benefits provided by the Weathertron III Control Center are water temperature set point changes, water heating mode decisions, intelligent night and day setback or setup programming over a true seven day week cycle, automatic heating cooling changeover and trip programming, all of which are easily entered by the user. In addition, service diagnostics are available.

The Control Center interface includes a "banker's display" format, which alternately flashes time, indoor temperature, and outdoor temperature. The display can be turned off or on. An annunciator in the middle of the display indicates AM or PM time, while top and bottom annunciators indicate various other features. These annunciators are "WARMER" and "COOLER", which indicate a temporary set point adjustment has been made by depressing the WARMER or COOLER keys. When operating in an economy or set-back mode, the ECON annunciator is lighted. Economy can be initiated or terminated early by depressing the appropriate START or END ECON keys. A FAIL light indicates a failure mode has been encountered and the system is operating on backup electric resistance heat in either air heating or water heating. The BAT annunciator indicates a weak battery which would cause customer programming to be lost in the event of a power interruption of several minutes duration. The UNIT ON annunciator indicates normal system operation is programmed. UNIT OFF results in shifting of the set points to 50°F for heating and 90°F for cooling. The DISHWASHER annunciator indicates that the DISHWASHER key has been depressed and the water heating set point temporarily raised to 140°F for one hour. This feature is available to provide extra hot water for dishwashing and other applications only when required and utilize lower temperature water typically 120°F for other domestic applications, and thus achieve a sizeable energy savings. The FAN ON annunciator indicates whether the continuous fan or automatic fan mode of operation has been selected.

Customer programming is entered through programming cards which change the key function with a switch on the face of the Control Center. This also changes the annunciator functions, which also provide guidance in directing the customer through the programming steps. In addition to setting the CLOCK time with the programming card, there are the COMFORT, SLEEP, WORK and TRIP programs. The COMFORT program is for the normal temperatures, heating and cooling, desired during the occupied hours. The SLEEP program is for the setback/-economy mode desired during the night time hours. The WORK program provides energy conservation during the unoccupied time. The TRIP program utilizes the WORK program settings and a 100°F water temperature whenever it is programmed to set back for a specific number of days and recover at a certain time on the last one. As noted, these programs can be overridden temporarily by use of the end economy keys or they may not be utilized at all due to customer lifestyle simply by programming start and end times the same or temperatures the same.

Optimum advantage of these economy modes with a heat pump, is obtained with the intelligent or smart setback and recovery. Significant details of smart heat pump set-back are described in U. S. Patent No. 4,266,599. In addition to setback and recovery time, temperature set points, and actual indoor and outdoor temperatures, it is necessary to have EQUIPMENT CONSTANTS relative to the building structure. There are four EQUIPMENT CONSTANTS that must be addressed for good comfort control and energy conservation. These constants are set by the user along with the NORMAL WATER TEMPERATURE desired. The COOLING equipment constant is the outdoor temperature at which the cooling system capacity balances the cooling load requirements. The HEAT PUMP equipment constant is the balance temperature point where the heat pump capacity equals the structure load. The HEATING equipment constant is the temperature where the total heating system (heat pump plus auxiliary heat) balances against the structure load. A $ LOCK equipment constant is provided for dual fuel systems, i.e. heat pumps mounted on natural gas or more typically oil furnaces where economics dictate switching to all alternate fuel below a certain temperature as the electric heat pump efficiency declines. There isa RECOVERY FACTOR which is a customer input variable to account for the building design, structure mass, infiltration, etc. It is factory set at an average value and adjusted experimentally based on actual recovery time vs. scheduled recovery time by the customer and need not be changed once determined for a particular installation.

Figure 5 is a graphic presentation summarizing the logic and operating modes of the heat pump water heating decisions. Depending on the outdoor ambient and the water temperature setting, either the heat pump water heater system is enabled or backup electric heat utilized. Below 10°F outdoors backup electric heat is used at all conditions. This is done because of the low heat pump water heating capacity and the need for space heating capacity. Between 10°F and 25°F, backup electric heat is utilized at water temperature settings above 130°F and in all cases above 140°F water temperature setting, backup electric heat is used due to code/control requirements and system design/reliability considerations. Above the heat pump system balance point water heating has priority over air heating. This priority continues below the heat pump balance point until a second stage air heating call is initiated. Then air heating has priority until the load is satisfied. Under normal comfort conditions, a false 35°F heat pump balance point is set to prevent conditioned space droop during water heating. This is done to prevent situations in the morning above the balance point when all family members rise and shower coincidentally which can produce large demands for hot water. (This false balance point also prevents conditioned space droop under high wind conditions and severe icing conditions when the balance point would actually shift upward.) This demand, while met with the heat pump water heater, could allow the condition space to droop unacceptably. Under these conditions, when the droop reaches a second stage call for heat, the system switches to air heating with backup electric water heating. The free cooling switchover from outdoor to indoor source is based on a 70°F outdoor ambient and the indoor condition space temperature.

As noted earlier, the Weathertron III system incorporates microcomputer controls. This also allows service diagnostics to be available for system monitoring. This service tool identifies the system temperature sensors which can be monitored along with heating, cooling and water temperature set points. The temperatures are compressor discharge, outdoor coil, heating suction line, water at the bottom of the conventional water storage tank, the hot water supply leaving the unit to the conventional storage tank, the outdoor ambient, the liquid line and indoor conditioned space temperature.

A failure mode detection is available to determine which failure mode occurred, if a sensor has fail ed or if an intermittent failure has occurred but not caused the system to lock out and go to backup heating modes.

To assure the proper system operation in all modes at startup or diagnose abnormal conditions during or after occurrence of a failure, a drive override-/monitor mode is available. In this mode, the system may be operated in any of the normal operating modes, regardless of settings, external temperatures and demand requirements. This allows typical system operating pressures and temperatures to be checked while operating. The various annunciators on the display identify which system components have been energized by the control.

Figure 6 shows a typical operating cost comparison of just the comfort conditioning and water heating costs used in an average home. The annual operating cost savings for the three systems vary from 27% to 36% with the Weathertron III system having the most.

A detailed heat pump system savings comparison is shown in Figure 7 for Atlanta. The base system has a 9.2 SEER in cooling. In the heat pump mode the HSPF is 7.0 with a COP of 2.7 at 47°F. Figure 8 compares the seasonal application efficiency in Atlanta of a typical electric resistance air and water heating system with a conventional heat pump system and the new Weathertron III system. The steady state water heating efficiency is 1.0 COP for the conventional resistance type with the Weathertron III having a COP of 3.0 utilizing the indoor source and a COP of 2.8 utilizing the outdoor source providing 120° hot water.

In summary, the design and features of the integrated space conditioning and water heating system Weathertron III have been presented. This system yields substantial economic and performance benefits compared to conventional water heating methods such as desuperheaters and self-contained dedicated heat pump water heaters.

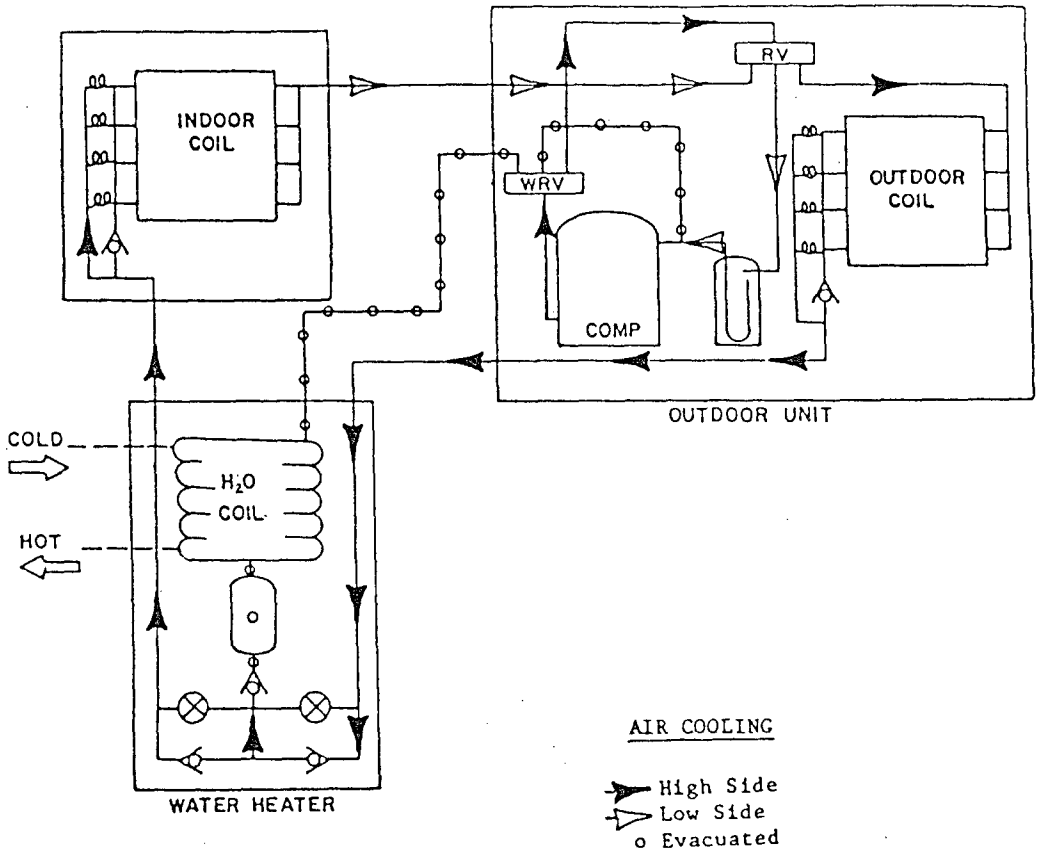


Figure 1

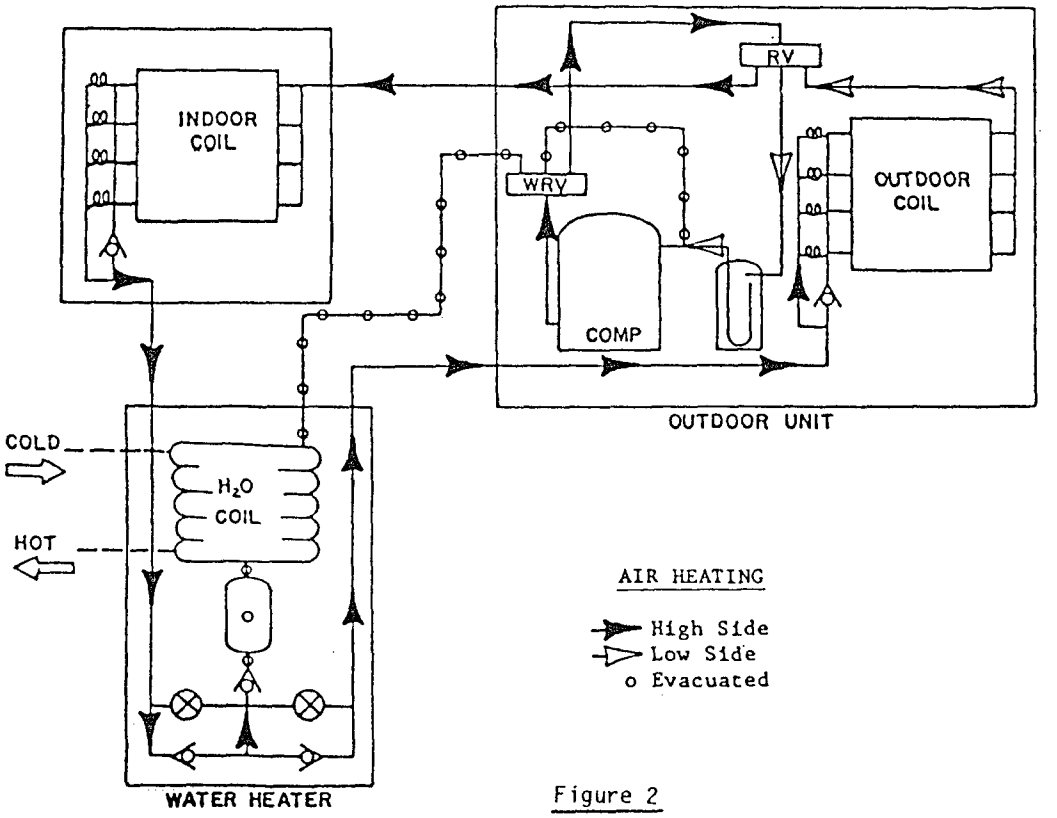
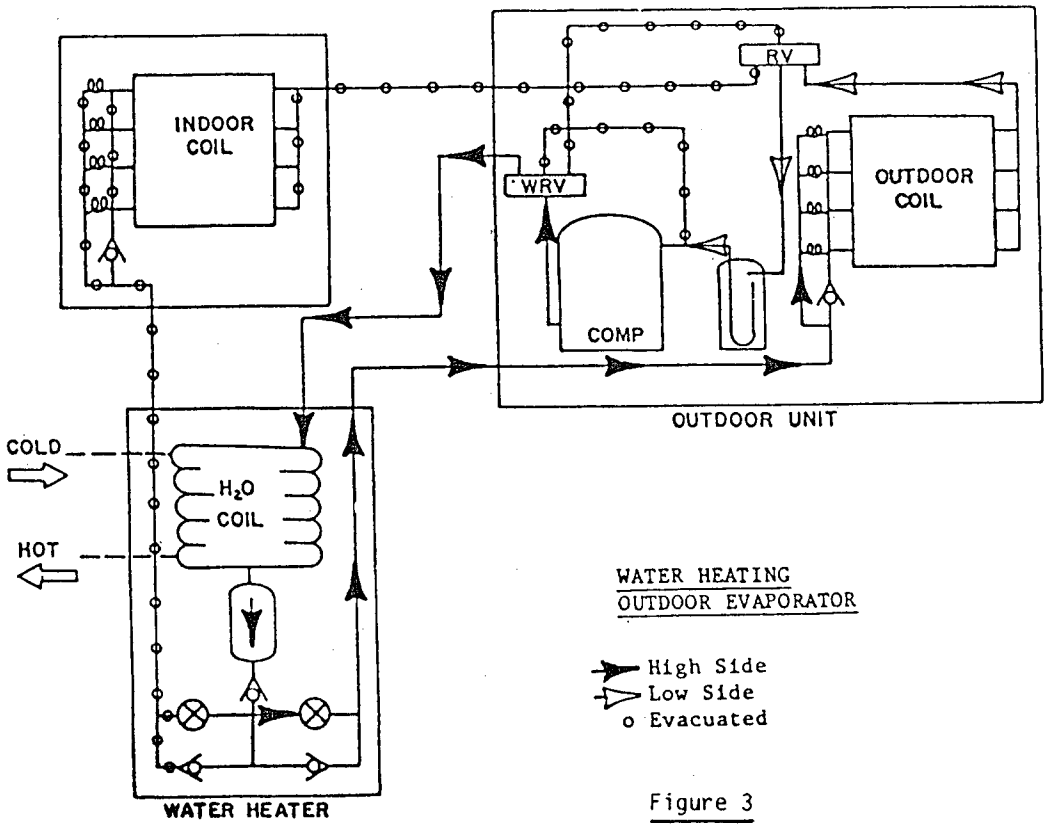


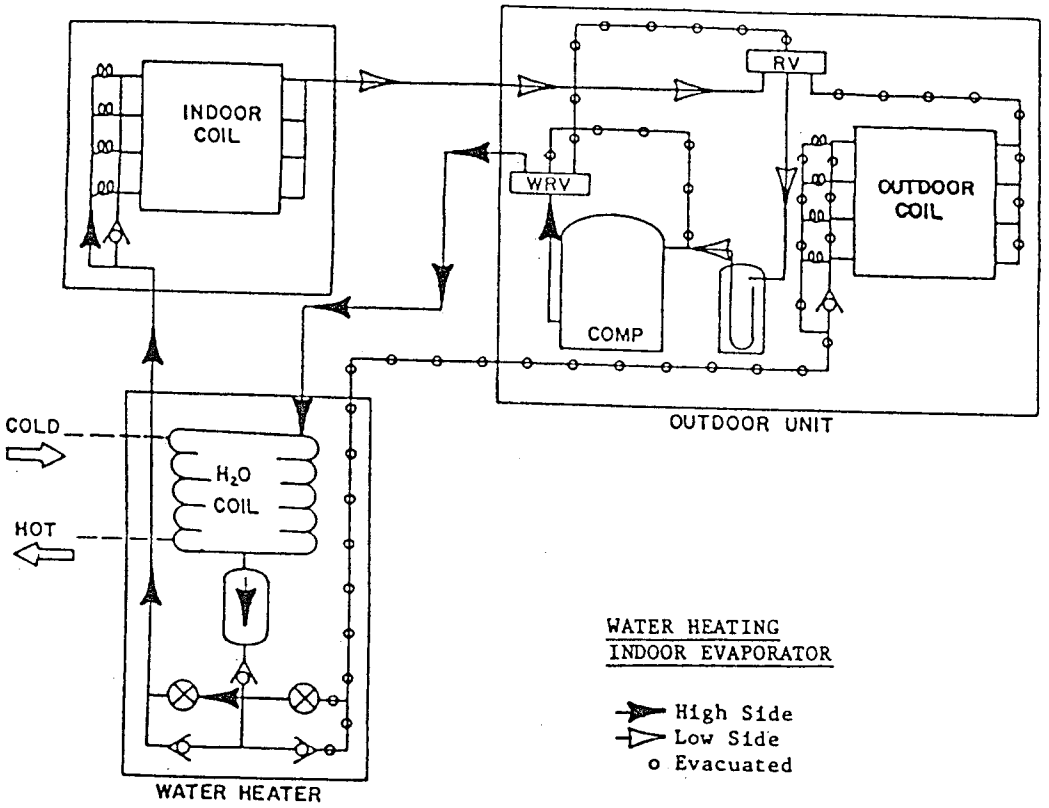
Figure 2



WATER HEATING
OUTDOOR EVAPORATOR

High Side
Low Side
o Evacuated

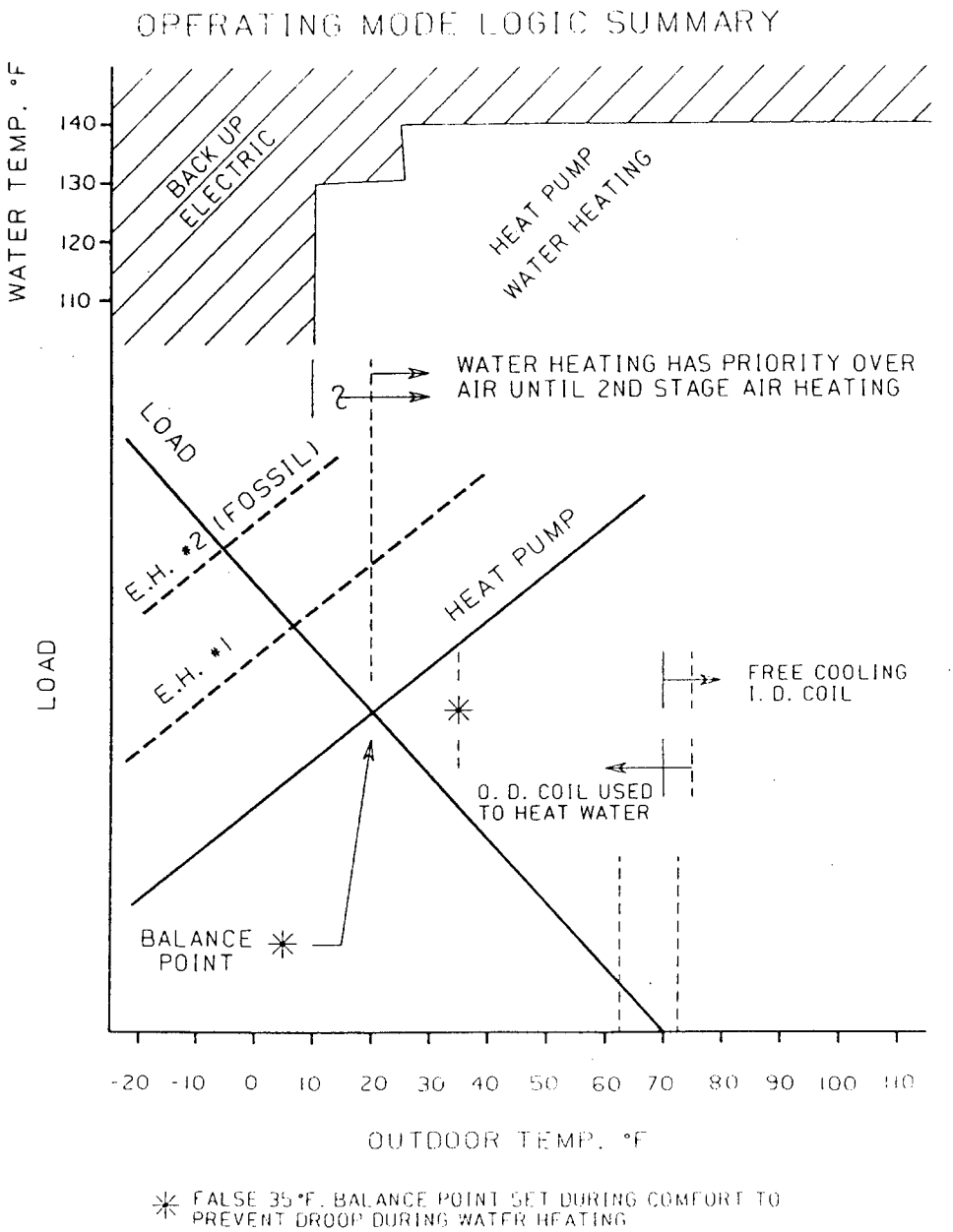
Figure 3



WATER HEATING
INDOOR EVAPORATOR

High Side
Low Side
o Evacuated

Figure 4



✳ FALSE 35°F. BALANCE POINT SET DURING COMFORT TO PREVENT DROOP DURING WATER HEATING

Figure 5

TYPICAL OPERATING COST COMPARISON

Base System Heat Pump With Electric Resistance Water Heater	With Desuperheater	With Dedicated Water Heater	Weathertron III
$1250	$913	$850	$788
	-27%	-32%	-36%
Annual Savings	$337	$400	$462

Figure 6

ENERGY CONSUMPTION COMPARISON

(ATLANTA)

Mode	Electric Resistance Water Heater: Conventional Cooling Electric Furnace	Electric Resistance Water Heater: Conventional Heat Pump	Weathertron III
Cooling (Kwh)	3009	3009	2741
Heating (Kwh)	9245	5228	4047
Water Heating (Kwh)			
Cooling	1791	1791	646
Heating	2964	2964	1174
Total (Kwh)	17,009	12,992	8608

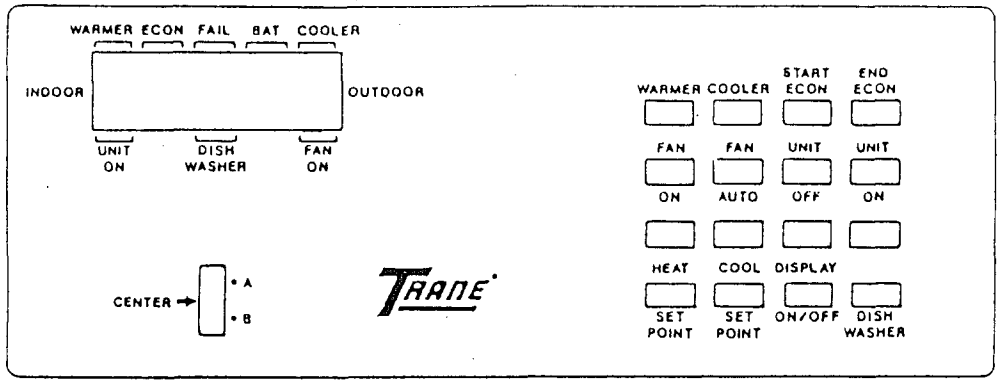
Figure 7

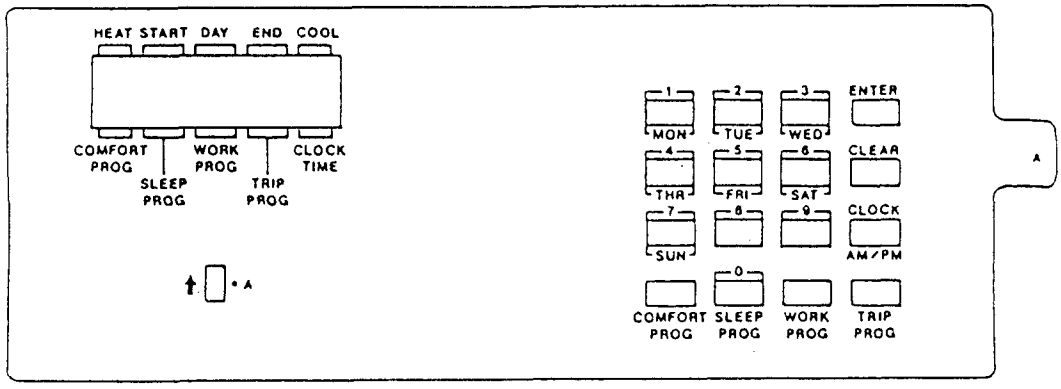
TYPICAL SEASONAL APPLICATION EFFICIENCY INCLUDING WATER HEATING

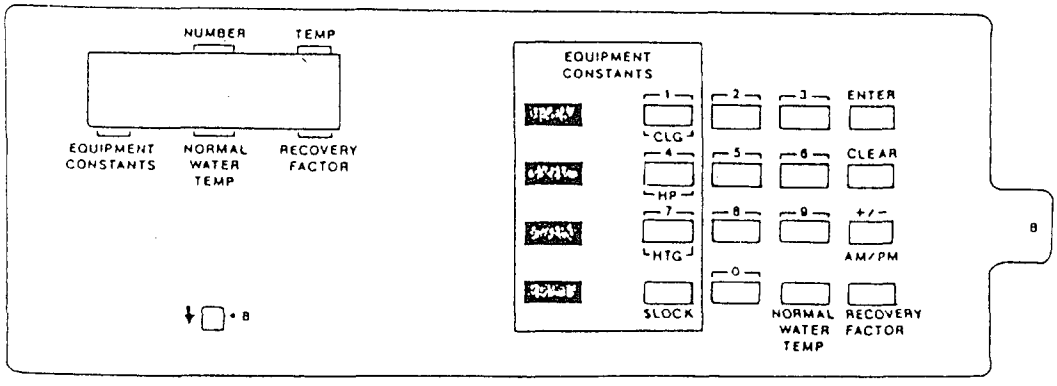
(ATLANTA)

System	Cooling (BTU/W)SEER	Heating (BTU/W)HSPF	Annual (BTU/W)APF
All Electric Resistance Heat	7.0	3.4	4.4
Heat Pump Air Heating	7.0	5.5	6.1
Weathertron III	9.6	7.5	8.2

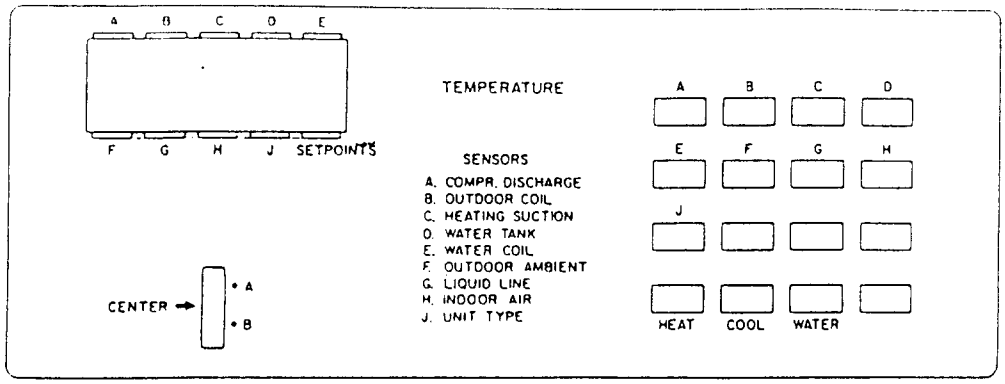
Figure 8

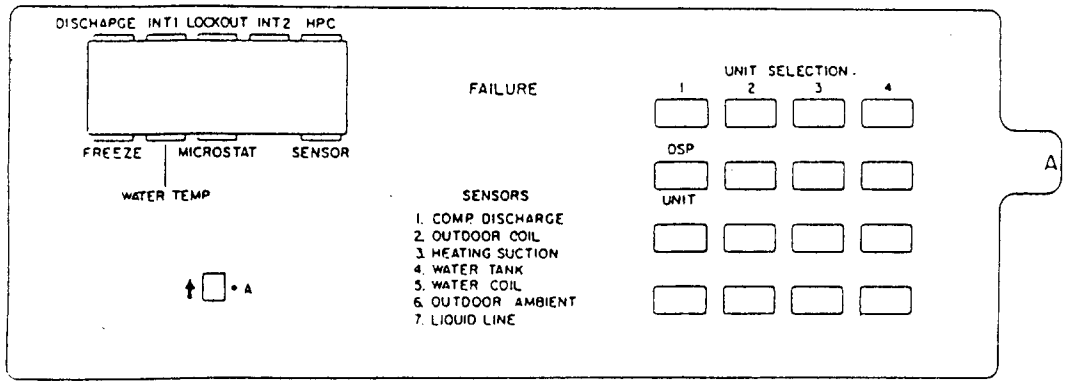


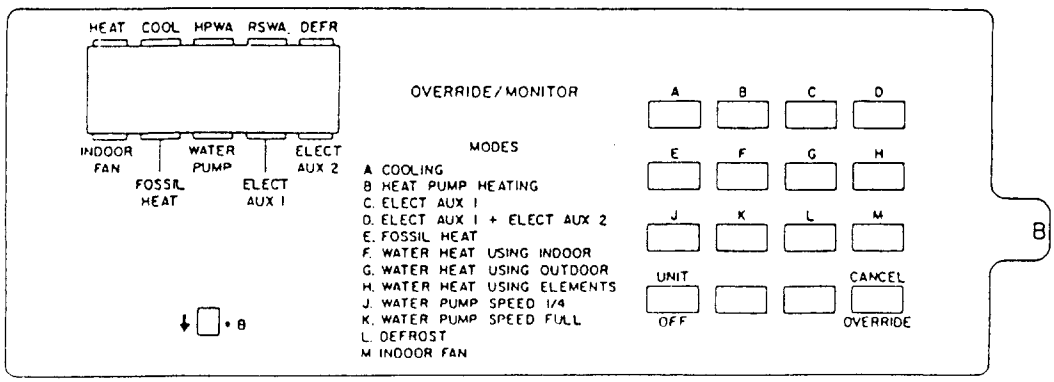




Weathertron III Control Center







Control Center
service diagnostics

FIELD EXPERIMENTS WITH A HEAT PUMP IN THE GEOTHERMAL FIELD OF LOS AZUFRES, MICHOACAN, MEXICO

by M. C. Roberto Best, C. Dispenza, S. Mercado, H. Fernandez, and L. Pignato

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Graduate in Nuclear Engineering, Associate Professor of Heat Transfer Technology at the University of Palermo-Italy; Secretary of the Associazione Termotecnica Italiana. Consultant in the field of heat and mass transfer equipment, energy management and oil and gas pipeline engineering. Has published a number of research papers in heat and mass transfer.

S. Mercado

Bio not available.

H. Fernandez

Bio not available.

L. Pignato

Bio not available.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

FIELD EXPERIMENTS WITH A HEAT PUMP IN THE GEOTHERMAL FIELD OF LOS AZUFRES, MICHOACAN-MEXICO

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SUMMARY

The Instituto de Investigaciones Eléctricas is currently developing a project for the integral use of geothermal fluids with the main objective of electric generation.

A field pilot experiment is being conducted by the Geothermal Department of I.I.E. to determine the feasibility of using low temperature geothermal fluid as a heat source for heat pumps. A templifier heat pump is used in this experiment for the heating of some dwellings in the Comisión Federal de Electricidad field in Los Azufres, Michoacán. The heat pump has been working intermittently for the last four months.

In the paper the experimental set up is explained as well as the data obtained. The problems encountered and the need of heat pumps that work efficiently also at high heat source temperatures is discussed.

INTRODUCTION

In the framework of the IIE Program for the exploitation of intermediate and low enthalpy geothermal resources for direct energy use, the Instituto de Investigaciones Eléctricas (IIE-México) is carrying out on site experiments with pilot plants in the geothermal field of Los Azufres-Michoacán-México.

One of the field experiments concerns the testing of a water-to-water heat pump in which the evaporator section is fed with geothermal heat. This equipment, is to be employed in the heating of some dwellings of the Comisión Federal de Electricidad, camp of Agua Fría (Fig. 1).

At the present time, the heat pump in fact feeds the heating loop of dwelling No. 8 and an initial set of measurements has been made to verify the performance characteristics of the system and their variation with the most significant external parameters.

As the heat load delivered to dwelling No. 8 is only a fraction of the thermal power delivered heat pump, the tests were made with the aid of an auxiliary loop.

In the future, the heat pump should be connected to a circuit which takes all the heat delivered by the condenser section of the heat pump to meet heat loads of dwellings 5,6,7,8,9, and 10 (Fig. 1) and to heat sanitary water for the same users. With this plant configuration it should be possible to make medium and long run tests to ascertain both the performance characteristics and the reliability of the pilot plant. The most important parameters of the plant (of the evaporator loop, the heat pump, the dwelling heating loop and the hot sanitary water system) will be measured and registered at suitable time intervals with the aid of a data aquisition and processing system which will be equiped with a microcomputer with process control capability (*).

SELECTION OF THE HEAT PUMP

In order to evaluate the use of geothermal heat, a commercially available heat pump was purchased.

The templifier model TPB-20D was selected for the experiments because of the high delivery temperature at which it can work (up to 104°C), it is also the smallest of the series.

The system is equipped with two pairs of R114 vapor compressing units (Fig. 2). The evaporators are fed in series by the source water.

The condensers are coupled in parallel and cooled by the water to be heated. The appropriate source water temperatures lie in the range: 30 to 65 °C. The system is completely automatic.

The compressors are of the hermetic type manufactured by Copeland.

Two single step thermostats are used for temperature control of the return water (T_{cw}) heated in the condenser loop. The control is of the on-off type. Each thermostat has a differential of approximately 2°C.

The evaporator of the No. 2 unit, since it receives water from the evaporator of the No. 1 unit, works at a lower pressure. Then the No. 2 compressor works with a lower suction pressure and a lower line current than No. 1.

Initially compressor No. 1 starts, and if the temperature of the water which returns to the condensers is lower than the set point value, compressor No. 2 starts and operates up to the time when the required temperature is reached. Then, compressor No. 2 stops and, if the set point value is maintained, No. 1 unit stops also.

The cycle is repeated in the same way when the set-point temperature falls.

(*) This part of the program is developed in the framework of an International Collaboration between the IIE-México and the EEC.

EXPERIMENTAL SET UP

Due to the fouling problems which are being incurred with the geothermal brine, it is not possible with the present system to feed the heat pump evaporators directly with it (*).

The system now is fed with clean source water heated in a closed loop with a "pool heat exchanger". This is a submerged heat exchanger which consists of a series of 3/4" tubes in an S shaped configuration connected by two 2½" headers. The heat exchanger is connected to the evaporators of the templifier with a loop made of 1½" diameter tubes. It is placed inside a concrete sump 1 x 1 x 1.8 m inside dimensions.

Geothermal steam coming from well A-22, via a flash separator, is injected into the sump and condenses in a bath of the same geothermal brine. Alternatively, it is possible to feed the sump with hot brine rejected from the other pilot plants, which are located near the heat pump or directly from the well.

The heat exchanger has a good surface for the present requirements, as it is a multipurpose unit.

With a suitable circulator and a by-pass line it is possible to meet the appropiate flow rate for each test run.

The condensers are cooled with water coming from a storage tank where for each run, a preselected temperature is maintained.

A part of the heat rejected from the condensers goes to the heating loop of dwelling No. 8, while the rest is sent to an auxiliary circuit which rejects the heat into a cold sink. The flow rate for each run is selected with the aid of a by-pass line.

In the flow sheet (Fig. 2) the process instrumentation is also indicated.

PROBLEMS ENCOUNTERED IN THE IN-FIELD EXPLOITATION

The heat pump is designed to operate with clean water both in the evaporator and condenser. Utilizing geothermal fluids as a heat source with the inevitable problem of fouling and corrosion implies the usage of a secondary heating loop and an extra heat exchanger with an increase of the heat transfer temperature difference.

Working under the ambient conditions encountered near a geothermal well leads to corrosion of the electric controls. The reset buttons and the low pressure cut-out of the second system of the templifier had to be changed after a few months operation.

STEADY STATE TESTS

The steady state performance of the heat pump was tested with the aim to verify to some extent the influence of external parameters while the new experimental set up is ready with sophisticated regulations of the external as well as some internal cycle parameters(**)

(*) Some heat exchanger prototypes are currently being tested at the IIE with the purpose to transfer liquid fludized bed heat transfer technology to this field of application.

(**) With the microcomputer system and with the modification of the actual control circuit of the heat pump.

In Figures 3,4,5, the typical characteristics of the equipment are reported (the power delivered at the condenser section Q_c, the power input P* and the $(COP)_{shaft}$ with reference to this last parameters) as function of the leaving source water temperature and with the leaving hot water temperature as parameter The power input is for a standard condition, in a real situation of an "input Kw adjustment factor for source water variations" which is given in Table 1 of the Westinghouse performance data brochure INT-TP B-B1 it is necessary to account for an appropriate evaluation of the "effective power input" P. Then a series of in-field steady state tests have been made which are reported in the same figures as data points for some well selected situations from the experimental data bank.

Table 1. Input Kw adjustment factor for source water variations.

Source water temperature difference ΔTev (°C)	Adjustment factor $\eta^*(\Delta Tev) = (P/P^*)$
Over 8.3	0.90
5.5 to 8.3	0.92
Less than 5.5	0.94

Test Methodology

For a pair of values of the parameters T_{oe} and T_{co} on a curve of Fig. 3 it is possible to read the correspondent values of the power input P* and of the power delivered $\dot{Q}_c$ at the condenser outlet temperature Tco. To prepare a set of values of the operational parameters it is necessary to make some assumptions. A ΔTco temperature variation is fixed for the water which crosses the condenser sections and also for the ΔTev the temperature variation of the water across the evaporator sections. Now it is possible to calculate the water flow rates $\dot{M}_{co}$ and $\dot{M}_{ev}$ in condenser and evaporator sections respectively, with the well known equations:

$$\dot{Q}_c = C_p \dot{M}_{co} \Delta Tco \quad (1)$$

$$\dot{Q}_e = C_p \dot{M}_{ev} \Delta Tev = \dot{Q}_c - P \quad (2)$$

P*, is derived from Fig. 4 and P obtained with the aid of Table 1.

Then with the preselected values of Toe (which is maintained regulating the pool temperature of the brine in the sump Tp) Toc (which is maintained with the aid of an external auxiliary circuit which dissipates the power delivered at the condensers) of ΔTev and ΔTco it is possible to make measurements when steady state is reached (for the appropriate values of M_{co} and M_{ev}). When this happens the energy balance is verified for the selected values:

$$\dot{Q}_c = \dot{Q}_e + P = \dot{Q}_e + \eta^* (\Delta Tev)\ P^* \quad (3)$$

The value of $(COP)_{shaft}$ is then calculated from the equation

$$(COP)_{shaft} = \frac{\dot{Q}_c \quad \eta^* (\Delta T_{ev})}{\dot{Q}_c - \dot{Q}_e} \quad (4)$$

and the "effective" COP is derived with the aid of the measured values of the electric power input to the compressor system Pe:

$$(COP) = (COP)_{shaft} \frac{P^*}{Pe} \qquad (5)$$

In the present experiments only the steady state (COP) of the heat pump is derived, in the future the tests will be made taking into account the power absorbed by the circulation pumps as well as the efficiency of the pool heat exchanger.

Data Discussion

In the Figures 3,4, and 5 are reported some data points selected from the experimental data bank for a number of situations which refers to the case where the steady state energy balance has been well verified. It can be seen that the points fall in the proximity of the performance curves which are derived from the data on the Westinghouse Report INT-TPB-B1. In all cases the derived COP falls below the correspondent point on the performance curve.

The electrical power input to the compressors has been also measured. With the aid of this data, it has been possible to estimate to some extent the electrical efficiency of the drivers which falls in the range: 81÷87%.

Future Tests

With the future configuration it will be possible to make measurements for situations different from those represented on the performance curves of Figures 3,4, and 5. To make this, the modification of the actual control system of the heat pump is also necessary.

The future program aims also to make cyclic tests and other series of experiments which take into account the whole system composed of the geothermal feeding module , the heat pump and heat storage system interposed between the heat pump and the heat load.

CONCLUSIONS

At the present time it is not possible to feed the heat pump with geothermal brine. An auxiliary equipment (the "pool heat exchanger") has been interposed between the geothermal source and the heat pump.

The condenser side has been equipped with an auxiliary loop to dissipate the heat delivered in the measurement runs.

In the actual phase the aim has been to make some experiments to verify the performance characteristics of the equipment. This is necessary to schedule future works which will include long run tests on the whole system, composed of the heat pump and the heat load.

The heat load consists of a closed loop which feeds the space heating and the hot water production facility for the Agua Fría Camp of the C.F.E. (Comisión Federal de Electricidad de México) dwellings 5÷10. In the future the whole system will be controlled and the data automatically registered and processed with a microcomputer multipurpose equipment.

AKNOWLEDGEMENTS

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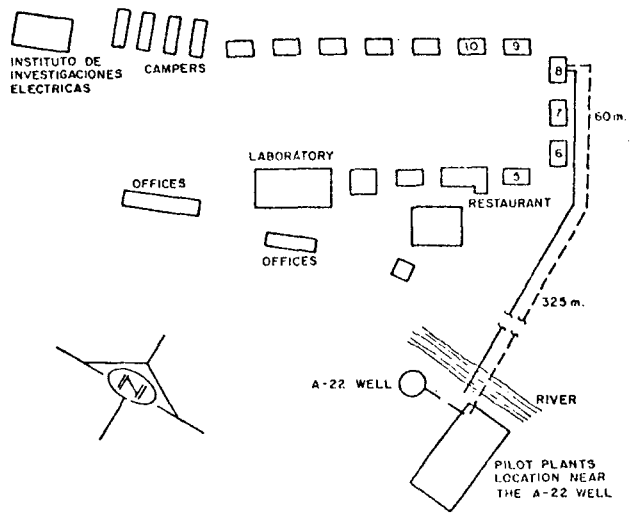
INSTITUTO DE INVESTIGACIONES ELECTRICAS
CAMPERS
10
9
8
7
6
60 m.
LABORATORY
OFFICES
5
RESTAURANT
OFFICES
325 m.
A-22 WELL
RIVER
PILOT PLANTS LOCATION NEAR THE A-22 WELL

FIG. 1

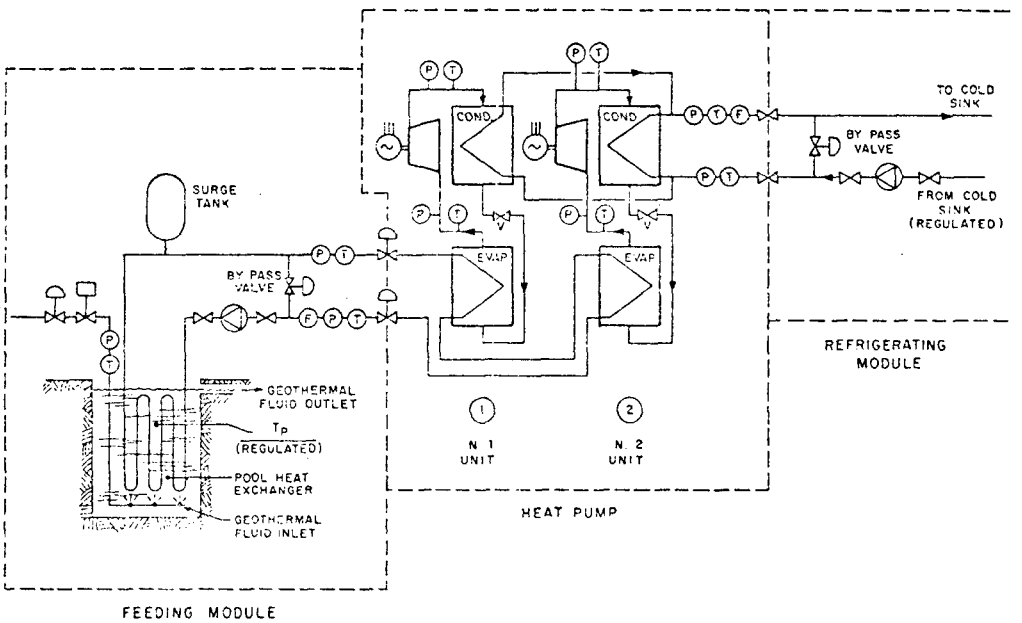
COND.
COND.
TO COLD SINK
BY PASS VALVE
FROM COLD SINK (REGULATED)
SURGE TANK
EVAP.
EVAP.
BY PASS VALVE
REFRIGERATING MODULE
GEOTHERMAL FLUID OUTLET
Tp
(REGULATED)
POOL HEAT EXCHANGER
GEOTHERMAL FLUID INLET
N. 1 UNIT
N. 2 UNIT
HEAT PUMP
FEEDING MODULE

FIG. 2

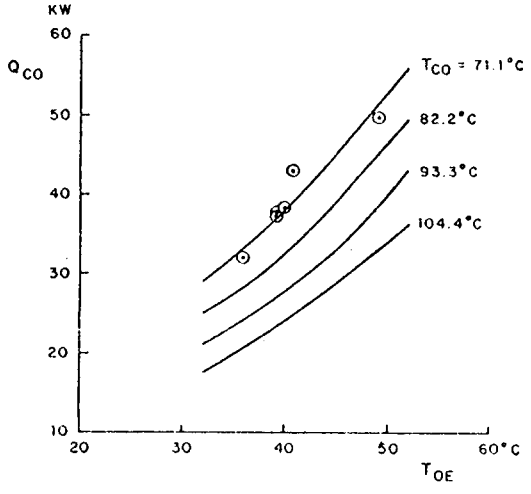

fig. 3

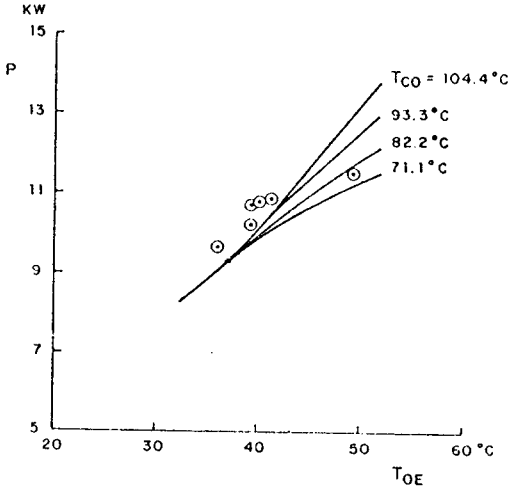

FIG. 4

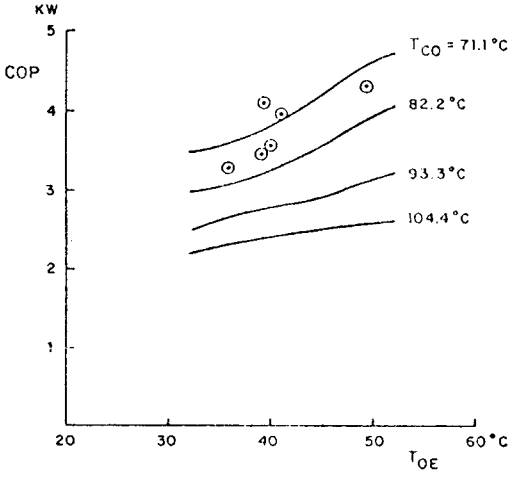

FIG.5

TVA'S OPERATIONAL EXPERIENCES WITH HEAT PUMP WATER HEATERS

by Theodore L. Sheldon

Theodore L. Sheldon

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Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

TVA'S OPERATIONAL EXPERIENCES WITH HEAT PUMP WATER HEATERS

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Summary

TVA's first operational experience with heat pump water heaters (HPWH) began in June 1979 with a 2-year field test of 9 units at Russellville, Alabama. The success of this test led TVA to further investigate the HPWH as a conservation device and the inclusion of the HPWH in TVA's conservation programs.

In October 1982, the HPWH was included in TVA's new conservation program known as The Energy Package™. However, the consumer did not respond as was anticipated and the program has progressed much slower than was expected. There are several factors that can be attributed to the lack of enthusiasm for HPWHs, but the major one seems to be a lack of public awareness of hot water usage and the HPWH as an alternate method of heating water.

Although the number of installations under TVA's HPWH Program has been less than predicted, there have been successes. The 284 installations have been performing well with no major complaints from the consumers. The number of installations is increasing each month, indicating HPWHs are gaining public recognition.

Russellville Test

Background

In 1976 Energy Utilization Systems (EUS) developed an integral HPWH utilizing a condenser tube positioned inside the storage tank and the heat pump mounted on top. At the same time, E-Tech, Inc., was developing a HPWH which pumped water from an existing tank to a heat exchanger and then back to the tank.

In May 1977 EUS was employed by the Department of Energy (DOE) to develop and test their integral HPWH. The project consisted of a 12-month test of 100 units and was to be carried out by 20 cooperating utilities across the country. TVA, 1 of 20 cooperating utilities, purchased 4 of the EUS integral units and 1 retrofit-type EUS unit. In addition, TVA also elected to purchase and test 5 retrofit-type units from E-Tech.

In order to determine the benefits, performance, reliability, and costs associated with heat pump water heater technology, TVA decided to conduct a 2-year field test of the E-Tech and EUS HPWH systems. The test was conducted by the Load Management Branch in the Division of Energy Conservation and Rates.

Test Description

Cooperating Distributor

The test commenced in June 1979 and was completed in May 1981. TVA selected a cooperating power distributor from a list of distributors expressing interest in the project. The selected distributor, the Electric Board of Russellville, Alabama, serves approximately 3,500 residential consumers and is located in northwestern Alabama.

Participating Consumers. The Russellville Electric Board conducted an informal survey to solicit interested participants for the field test. Participation was restricted to homeowners with electric heating and a suitable location for the HPWH. Ten homeowners who met the above criteria were selected to participate.

Data Collection. Submetering packages were installed in each participating consumer's home. Along with a TVA submetering package and water meter, each EUS home also had a metering panel provided by EUS.

The TVA submetering package consisted of a General Electric PDM-76 four channel magnetic tape demand recorder and two watthour meters equipped with pulse initiators. One meter monitored the total electrical consumption of the home, while the other measured energy used by the water heating system. A magnetic drive water meter was installed in the supply line to the water heater to provide data on the gallons of hot water used each month.

The EUS metering package consisted of five watthour meters, a timeclock, switching relays, and a strip chart recorder. This package provided the data from which EUS was to evaluate the HPWHs for DOE. At the end of the 12-month DOE/EUS test period, the EUS's packages were removed.

The magnetic tapes were changed each month. At that time, visual readings were taken from the watthour meters and the water meters.

Results

Equipment Reliability. The EUS retrofit unit did not function properly. After the homeowner complained of excessive energy consumption and the unit running constantly, it was removed 3 months after installation. The remaining EUS units were found to have malfunctions due to refrigerant leaks and improper charge. These problems were attributed to the units being prototypes, transportation, and compressor vibration. The leaks were repaired and the units recharged.

The attachment of the thermostat to the water tank posed a major problem for the E-Tech units. The attachment procedure relied solely on pressure from the surrounding insulation to keep the thermostat in contact with the tank. As a result, the thermostats became detached from the tank causing the units to run excessively. This was repaired by utilizing the mounting clips of the original thermostat.

Once these problems were resolved, the HPWHs in the test functioned with a reasonable degree of reliability. A certain number of prototype "bugs" were expected. All major complications were noted by the manufacturers and in each case, steps were taken to correct the problem in future production models.

Location Problem. In one home, the E-Tech unit was installed in a small utility room measuring 8 feet by 8 feet. The installation never provided consistent data which was attributed to a lack of air circulation when the door to the utility room was closed.

Energy Savings. After 23 months of data collection, the EUS units and the E-Tech units demonstrated an average savings of 45 percent and 52 percent, respectively. Table 1 provides a list of the average monthly savings for each type of unit.

Peak Load Reduction. Figures 1 through 4 depict the average electrical loads required by HPWH and resistance water heaters during each hour of a typical weekday. Although the sample size of 5 is small and may not represent the behavorial patterns of the general population, the graphs indicate that HPWHs provide load reductions over conventional water heaters during most time periods.

Consumer Acceptance. The Russellville Electric Board conducted informal discussions with the participating consumers concerning satisfaction and acceptance of HPWHs. Complaints and negative comments were traced to equipment malfunctions which were repaired. Excluding the consumer with the EUS retrofit unit, the 9 remaining participants voiced no complaints with respect to water temperatures, recovery rates, or operation of the units. Noise levels were acceptable.

Perhaps the best gauge of customer acceptance is the fact that 8 of the 9 full-term test participants elected to purchase their units at the conclusion of the test. In the case of the one E-Tech unit that was not purchased, the family moved to a new home where the location of the water heater was not suitable for a HPWH.

Further TVA Study

Heat Pump Application Unit Gets Responsibility for HPWH

The Russellville Test along with tests from other utilities were indicating that the HPWH was a viable conservation device that might be beneficial to include in TVA's conservation program, therefore, in January 1981 TVA's Heat Pump Application Unit was given the responsibility to further study the HPWH and possibly set up a demonstrational finance program.

Power Credit Analysis. TVA uses a computer model that forecasts the benefits of conservation measures as they apply to the consumer, the power distributor, and the TVA power system in order to evaluate the value of such measures. The program is constantly upgraded as changes take place in future load forecasting.

From the Russellville test and data from other utilities, along with manufacturers data, numbers were generated for kWh and kW savings. The power credit analysis at the time showed positive benefits for the consumer, the power distributor, and the TVA power system.

Manufacturer's Testing. Since there was not a standard for testing HPWHs, TVA developed a draw test to evaluate the various models of HPWHs. A TVA staff member visited several manufacturers, randomly picked a unit off the production line and tested it in the manufacturer's lab. These tests again verified the energy savings that were achieved in the Russellville Test.

ARI Standards Committee. In the spring of 1981, the Air-Conditioning and Refrigeration Institute (ARI), realizing a need for a testing standard and certification program for HPWHs, set up a Standards Writing Committee. TVA was invited to participate on the committee along with representatives of many of the HPWH manufacturers. TVA acted as a neutral party with no "ax to grind" as to the operating characteristics of any particular model HPWH. Much of TVA's draw test was incorporated into the ARI standard.

At the same time, the Gas Appliance Manufacturers Association (GAMA), which had a water heater certification program, also began work on a HPWH standard. Since GAMA was already certifying water heaters, they were able to offer a better financial proposal to the HPWH manufacturers and, therefore, secure the certifying program.

Demonstration Project. At first a proposal was worked up for a demonstration financing program for HPWHs, patterned after the Space Conditioning Heat Pump Program which was working successfully. The HPWH Program would be limited to three different locales in the Tennessee Valley. If successful in these areas, the program would be expanded to the whole Valley.

However, TVA's Board of Directors decided that with TVA's previous testing, along with other utilities' tests, the HPWH was a proven conservation device and should be included in a Valley-wide program. The major concern was whether the HPWH was commercially available with adequate dealer networks in all areas. The manufacturers assured TVA that there were adequate dealers in place, anxiously waiting for TVA to initiate a financing program.

The Energy Package™

Revised Home Insulation Program

The Options. TVA decided to revise its Home Insulation Program and add other conservation measures to the Home Energy Survey. The measures included: home weatherization, heat pumps, wood heaters, solar water heaters, and HPWHs. The new program was titled The Energy Package™ and was initiated October 1, 1982.

TVA wholesales electricity to 160 power distributors who in turn retail electricity to the residential consumer. Under The Energy Package™ the power distributor has the option to choose the conservation measures they would like to promote in their area. The distributors also have the option to choose the financing plan they use. Option 1 is $1200 at zero interest for a 7-year term, option 2 is $3800 at TVA's cost of money for a 10-year term, and option 3 is a combination of both option 1 and 2.

Under The Energy Package™, when a homeowner requests an energy audit, the energy advisor explains the particular options that the power distributor has contracted for. In the case of HPWH's, if the consumer is interested and the location of the water heater meets the criteria in TVA's installation standards, the homeowner is given a list of dealers authorized to install HPWHs in TVA's HPWH Program. The homeowner then chooses a dealer and has the HPWH installed (in October 1983 the program was changed to include the financing of "do-it-yourself" installations). Once the unit is installed, a heat pump advisor inspects the installation per TVA's standards. If the installation passes the inspection, the dealer is paid and the consumer is billed through the monthly electric bill.

TVA Installation Standards. TVA established installation standards to assure HPWH equipment was installed properly and in the correct location. This not only protected the consumer but would help secure TVA's investment over the term of the loan.

Manufacturers Requirements. The criteria for a manufacturer to receive TVA approval to participate in the program includes being UL, ETL, or ARL safety listed, GAMA certified with a minimum energy factor of 1.85 and have a 1-year labor, 5-year all parts warranty backed by the manufacturer.

Location Restrictions. Many manufacturers have a switch over device available for locations where the ambient temperature falls below 45°F, enabling the HPWH to switch to resistance heat until the ambient temperature raises above 45°F. However, TVA's concern was that there was a good chance that the period of time that the temperature would be below 45°F would occur during a peak period, consequently causing higher demand charges for the power distributor. Therefore, switch over devices were not allowed and locations such as garages and some crawlspaces were not qualified. To assure adequate air flow, the minimum room size for a HPWH is 1000 cubic feet unless the door is fully louvered.

Slow Start. As of July 1984, 284 HPWH have been installed in the TVA program. This is well below TVA's initial expectations and predictions. There are many possible reasons, but the most obvious is public awareness. First of all, few consumers are aware of how much hot water they use or what it costs. Secondly, they have never heard of a HPWH, or if they have, they confuse it with a space conditioning heat pump or a desuperheater. Other possible factors contributing to a low number of installations are first costs, lack of dealers in all areas, fear of heat pumps, and apathy on energy savings (energy prices have stabilized in the last couple years).

Sixty-three power distributors are participating in the HPWH Program. These include some of the larger distributors and represents approximately 45 percent of the residential customers in the Tennessee Valley.

Satisfied Consumers. Except for problems with a flow control valve on one particular brand, there have been no major problems or complaints with the TVA installations. The manufacturer has changed procedures and no longer uses the flow control valve, eliminating that problem.

Several of the installations were monitored for a short time. Energy savings on these installations were in the 60 percent range over resistance water heaters.

Conclusion

TVA believes the HPWH is a reliable conservation device that will allow homeowners to save at least 50 percent of their hot water bill. However, with TVA's low rates and the public being uninformed on hot water usage and the availability of HPWHs, it will take some time to really penetrate the market.

In order to promote the HPWH, TVA has stepped up advertising. Using television and radio spots, along with newspaper and magazine ads, TVA is beginning to make the public aware of the HPWH as an alternative to the resistance water heater. TVA has also set up HPWH displays at homeshows, energy fairs, and shopping malls to allow consumers to witness an actual working model.

TVA is also making changes to the program to allow more consumers to be eligible for HPWHs. For example, in areas that fall below 45°F, a switch-over device would be allowed if a cycling device is installed on the resistance elements to prevent them from being activated during peak periods.

Although the response to the HPWH has not been what TVA anticipated, we believe in time it will be a common appliance. The HPWH has proven itself; we now have to educate the public to its success.

Table 1

SUMMARY OF ENERGY SAVINGS

Period Ending	EUS Units kWh	EUS Units %	E-Tech Units kWh	E-Tech Units %
07/13/79	195	46	284	66
08/13/79	195	47	287	66
09/13/79	187	45	293	67
10/12/79	200	45	336	66
11/13/79	238	46	357	61
12/13/79	283	46	309	54
01/14/80	421	50	444	57
02/13/80	310	43	259	50
03/13/80	339	49	215	40
04/14/80	369	49	256	46
05/14/80	262	44	329	57
06/15/80	241	48	295	58
07/18/80	175	42	242	59
08/21/80	-	-	260	57
09/18/80	171	48	245	63
10/16/80	195	40	258	58
11/26/80	402	49	412	54
01/06/81	238	29	144	27
02/06/81	317	42	159	32
03/10/81	312	43	167	34
04/13/81	355	44	162	36
05/18/81	281	43	171	41
Average Savings/Month	271	45	267	52
Average Savings/Year	3,252	45	3,204	52

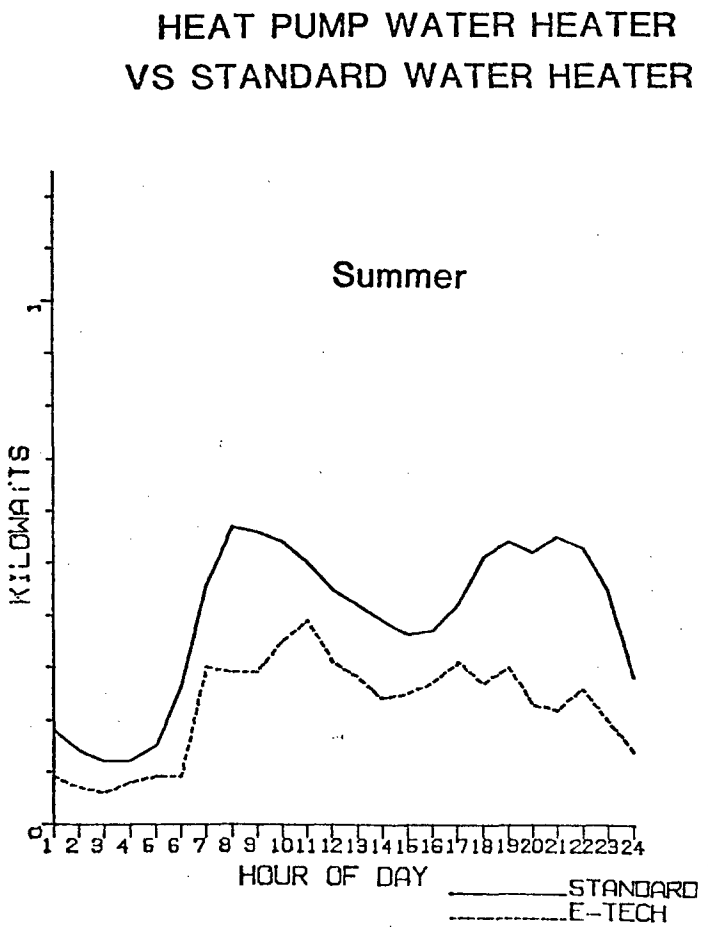

FIGURE 1

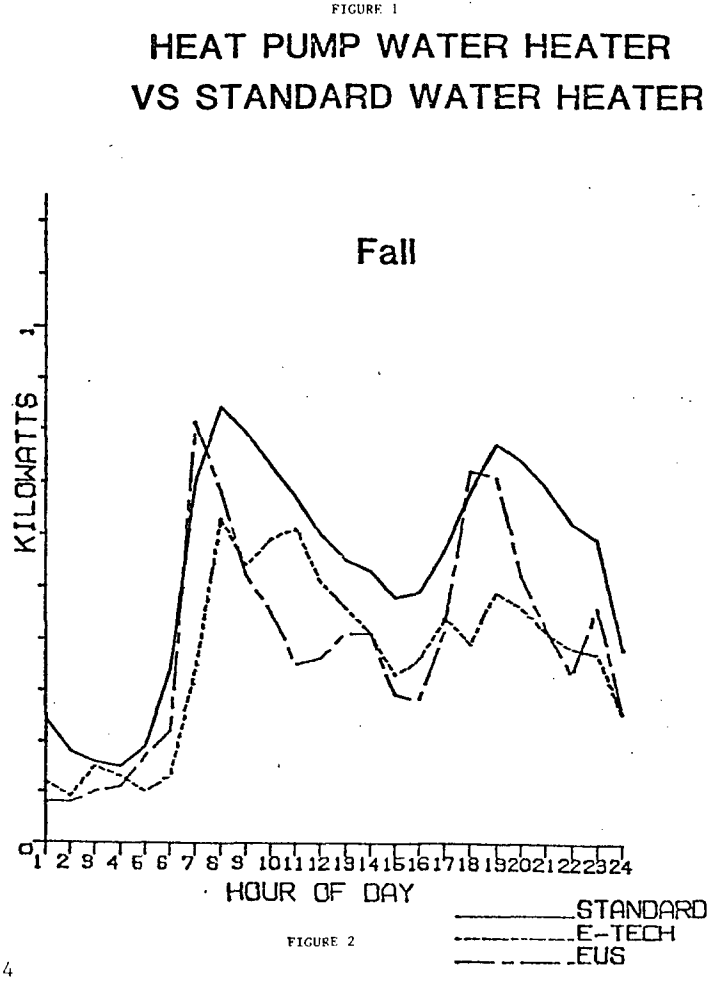

FIGURE 2

HEAT PUMP WATER HEATER
VS STANDARD WATER HEATER

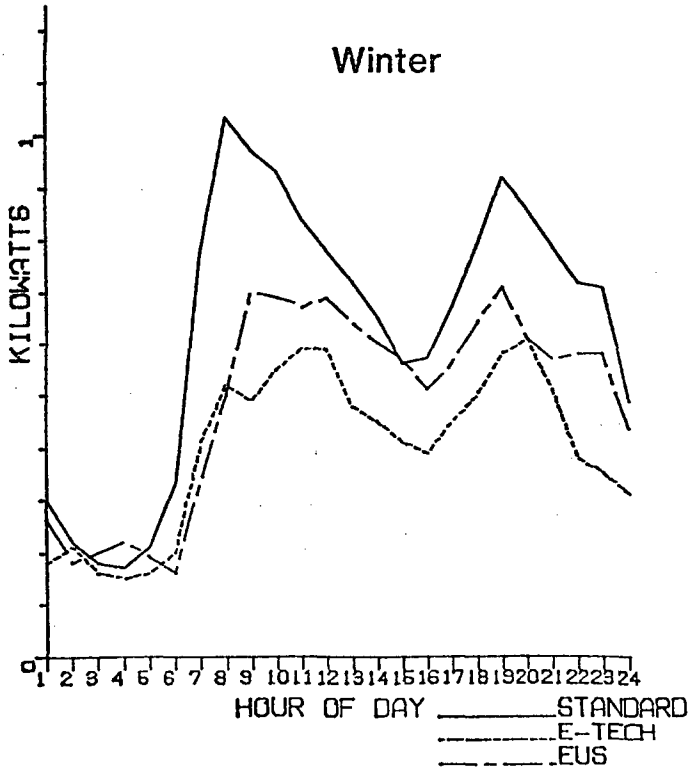

FIGURE 3

HEAT PUMP WATER HEATER
VS STANDARD WATER HEATER

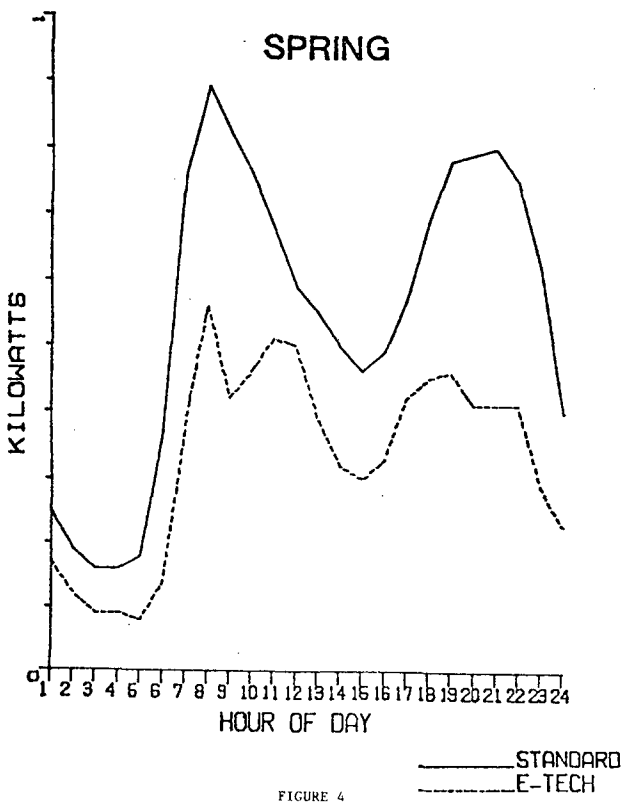

FIGURE 4

LABORATORY TEST, DESIGN MODEL VALIDATION, AND PARAMETRIC STUDY OF A HEAT PUMP WATER HEATER

by Kay H. Zimmerman

Kay H. Zimmerman

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LABORATORY TEST, DESIGN MODEL VALIDATION, AND PARAMETRIC STUDY OF A HEAT PUMP WATER HEATER*

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Introduction

Heat pump water heater (HPWH) units now available are first generation design, assembled with available components which were developed for other uses. Therefore, it is likely that the units can be improved.

The objective of the research activities described in this paper is to provide an analytical capability that can assist the industry in evaluating the effect of improved components or new design features on HPWH system performance. This experimental and analytical study began by collecting sufficient data to form a data base on the steady-state performance of a residential HPWH unit. The analytical steady-state model was formulated as a modification of an existing heat pump model by developing new subroutines. The experimental data was used to formulate and validate an analytical steady-state model. The model can be used to evaluate recommendations for improvements in component and system design. Early parametric study results are included to illustrate the model's potential usefulness in design studies.

Experimental Facility

The HPWH used in this study was representative of the separated-type units currently available on the market. The rotary compressor has a nominal rating of 10,200 Btu/h. R-12 refrigerant flow is controlled by a thermostatic expansion device. The water-to-refrigerant condenser is a coaxial tube-in-tube design. A schematic of the HPWH instrumented test loop is shown in Fig. 1.

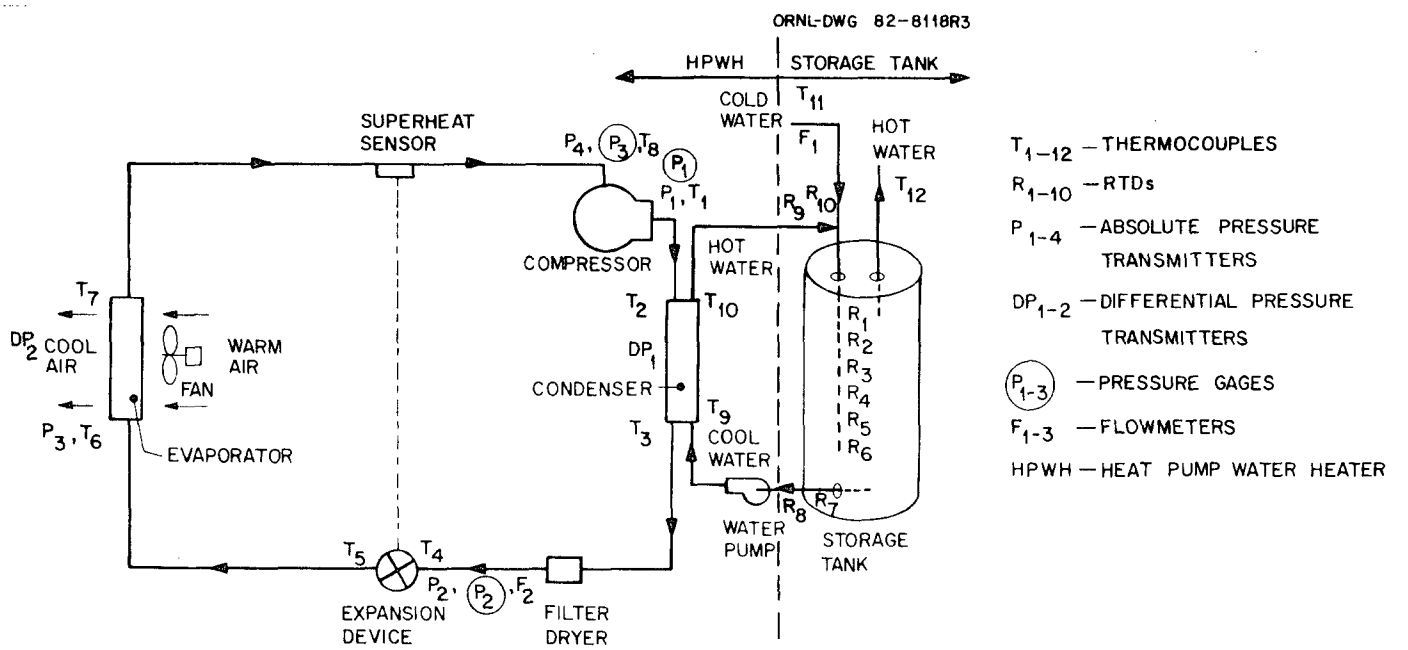

Fig. 1. HPWH Instrumented Test Loop

The HPWH unit was connected with copper tubing to an 82-gallon storage tank. The copper tubing was insulated to prevent recirculation of the water due to natural thermal convection between the storage tank and the water tubing. A closed water loop is formed by the HPWH, the water tubing, the water storage tank, and the water circulating pump.

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

To obtain data of sufficient accuracy from the experiment, it was necessary to house the experimental facility in an environmental chamber having controlled temperatures and humidity.

The data acquisition system (DAS) consisted of a Hewlett Packard (HP) desktop computer, an HP digital voltmeter, an HP multiprogrammer, an HP multiprogrammer interface, an HP digital clock, an HP printer, and a Fluke scanner.

Experimental Procedure

The experimental tests were conducted to study the effect of various ambient temperature and humidity levels and the entering water temperature on the HPWH system performance. Ambient temperature was varied from 50 to 100°F, relative humidity from 28 to 97%, and entering water temperature from 115 to 130°F.

A steady-state test began when the HPWH inlet water temperature reached 115°F. By this time, the HPWH unit had been operating approximately 4 hours. The entering water temperature was changing slowly enough (1F° per 3-5 minutes) to assume quasi-steady-state conditions.

Data was collected at every 1F° temperature rise of the HPWH inlet water temperature. The components were operating for the entire duration of the test. The water was continuously circulated around the closed loop between the water storage tank and HPWH unit at a constant rate of 1.4 gpm. The test continued until the condenser inlet water temperature reached 131°F, at which time the HPWH was turned off.

The water temperature range of 115 to 131°F chosen to represent steady-state conditions is the typical on and off points of the HPWH unit when in normal "as installed by the dealer" operation.

Experimental Results

Ambient Temperature Effect

The ambient temperature is the major driving force in the evaporator. As the ambient temperature changes from 50 to 100°F, the evaporator saturation temperature and pressure increase. This, in turn, causes the compressor inlet refrigerant vapor density to increase. As the density increases, the refrigerant mass flow rate increases proportionally. Although higher mass flowrates require more compressor work, the compressor work per unit mass decreases.

The heat rejected from the condenser into the water storage tank is a sum of the increased (by 62%) evaporator capacity and increased (by 27%) power input to the compressor that resulted from the 50 to 100°F ambient temperature change. From these combined effect the heat rejected from the condenser increases 67%.

The coefficient of performance (COP) for HPWH units is defined as the ratio of the condenser capacity to the total power consumption. This relationship can be written as Eq. 1.

$$COP = \frac{\Delta h_{cond}}{\dot{W}_{comp} + \frac{(W_{fan} + W_{pump})}{\dot{M}_r}} \qquad (1)$$

where

Δh_{cond} = enthalpy change through condenser,

$\dot{W}_{comp}$ = compressor power input per unit mass of refrigerant,

W_{fan} = fan motor power,

W_{pump} = pump power,

$\dot{M}_r$ = refrigerant mass flow rate.

The term $(W_{fan} + W_{pump})/\dot{M}_r$ decreases as the ambient temperature changes from 50 to 100°F because $\dot{M}_r$ increases. As previously discussed the compressor power input per unit mass declines. The enthalpy change through the condenser decreases slightly as the ambient temperature changes. Thus as the ambient temperature increases, the denominator of Eq. 1 decreases faster than the numerator decreases resulting in higher values of COP. As shown in Fig. 2 the COP

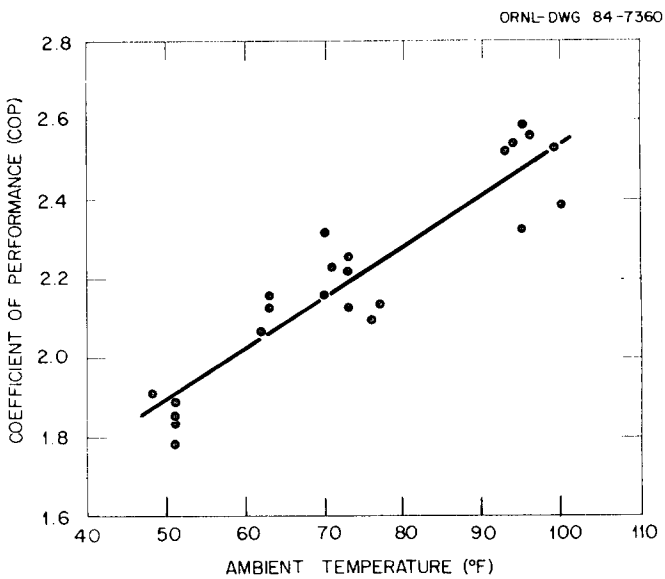

Fig. 2. Effect of ambient temperature on COP.

improves 34% from 1.89 to 2.54 for the ambient temperature change of 50 to 100°F.

HPWH Inlet Water Temperature Effect

The results in this section are from a steady-state test operated at 73°F ambient temperature and 60% relative humidity. The inlet water temperature is the major driving force in

the condenser. As the condenser inlet water temperature increases from 115 to 130°F the condenser exit saturation temperature increases causing the enthalpy change through the condenser to drop. The refrigerant mass flow rate increases slightly but overall the condenser capacity decreases slightly due to the decline in heat transfer between the water and refrigerant. More compressor work is required for the inlet water temperature increase. The COP is directly proportional to the decreasing (by 4%) condenser capacity and inversely proportional to the increasing (by 13%) total power consumption. Therefore the COP declines 14% from 2.21 to 1.89 for the 115 to 130°F water temperature change as shown in Fig. 3.

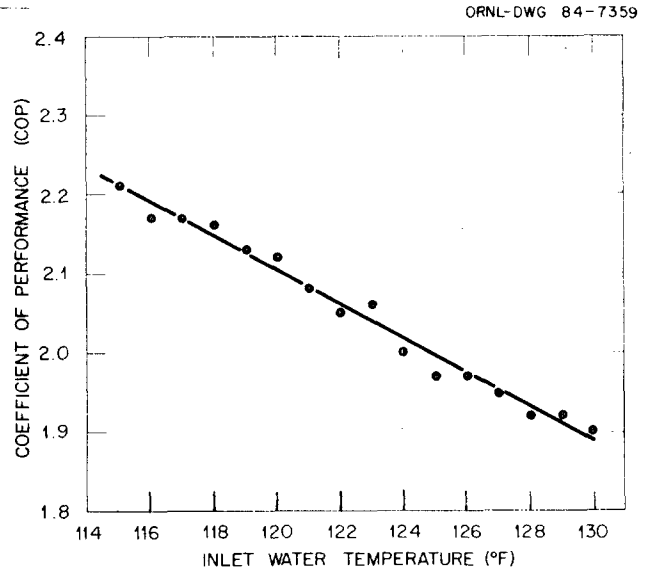

Fig. 3. Effect of HPWH inlet water temperatures on COP.

Relative Humidity Effect

The experimental tests were performed in a high humidity condition and in a low humidity condition for each ambient temperature. The high humidity tests listed in Table 1 produced a wet evaporator coil surface. A wet coil compared to a dry coil at the same dry bulb temperature has higher rates of mass transfer and increased latent heat. Thus the evaporator capacity increases and that results in improved condenser capacity and COP as indicated in Table 1. The effect is less apparent at higher ambient temperatures.

Table 1. Influence of Relative Humidity

Ambient Temperature °F	Humidity Level	Evaporator Capacity 10^3 Btu/h	Condenser Capacity 10^3 Btu/h	COP
51	Low	6.0	7.7	1.78
51	High	7.1	8.6	1.84
70	Low	8.2	10.4	2.16
70	High	9.2	11.6	2.32
93	Low	10.4	13.3	2.52
93	High	10.7	13.4	2.54

Computer Design Model

The HPWH computer design model is a modification of the Oak Ridge National Laboratory (ORNL) Heat Pump Design Model 1. The new subroutines added are (1) a water-to-refrigerant condenser model developed at Brookhaven National Laboratory 2 sponsored by ORNL and (2) a circulating water pump model developed at ORNL. The Heat Pump Design Model is a computer program designed to predict the steady-state performance of conventional vapor-compression, electrically driven, air-to-air heat pumps in heating and cooling modes. The heat pump design model was developed by using physical principles and generalized correlations where possible rather than empirical correlations obtained from manufacturer's literature. The water-to-refrigerant condenser model includes both a shell-and-tube and a tube-in-tube design. The tube-in-tube model was used in the validation and parametric studies. The condenser subroutine was developed by empirically fitting published heat transfer and pressure drop correlations to experimental data. The circulating water pump model has not been generalized for the present work. It is based on characteristic curves obtained from the pump manufacturer to predict water flow rate and power input to the pump as a function of the system and pump pressure head.

Model Validation

The HPWH Design Model was validated against the experimental runs prior to the addition of the circulating water pump model.

The compressor model used is based on biquadratic fits to performance curves obtained for a rotary compressor from the compressor manufacturer's calorimeter measurements. The performance curves pertain strictly to the superheat level at which they were generated (50F° superheat). For actual operating conditions small correction factors were applied for adjusting the compressor motor power input and refrigerant mass flow rate. These factors were based on reciprocating compressors but were used in this study because of the lack of information of the effect of superheat change in rotary compressors.

While some differences between compressor types would be expected, it is asumed here that these would be second order effects. Table 2 shows the comparison of the experimentally measured values of compressor power input and refrigerant mass flow rate with the unadjusted manufacturer's calorimeter data, the adjusted map fit, and the adjusted map fit model calculated value. The unadjusted manufacturer's data underestimates the compressor power input and the refrigerant mass flow rate. This is due to no superheat correction and to the manufacturer's data being within ±5% accuracy. The superheat adjusted map fit lowers the compressor power input value and raises the refrigerant mass flow rate.

Table 2. Comparison of Compressor Power Input and Refrigerant Mass Flow Rate at 68°F ambient and 54% RH.

Superheat °F	Measured Value	Manufacturer's Data Unadjusted	Map Fit Adjusted	Model Calculated Adjusted
		Compressor Power Input, watts (% Difference)		
13.5	1304	1259 (-3.5)	1232 (-5.5)	1206 (-7.5)
		Refrigerant Mass Flow Rate, lbm/h (% Difference)		
13.5	174	164 (-5.7)	174 (-0.5)	163 (-6.7)

This adjusted map fit value is the value used as input into the model. Therefore the compressor power input and refrigerant mass flow rate values have changed from their measured values when they are input into the model.

Table 3 shows the results of a typical validation run (excluding the circulating pump model). Any additional heat losses and gains or pressure drops in the refrigerant circuit found in the experimental setup were not accounted for in the validation. The evaporator and condenser capacity are underpredicted by 7.7% and 8.1% respectively but this is mainly because of the refrigerant mass flow rate difference. The predicted coefficient of performance is close to the measured value

Table 3. HPWH validation at 68°F ambient and 54% RH.

HPWH Performance	Measured	Calculated
Compressor Performance		
Refrigerant mass flow rate, lbm/h	175	163
Compressor motor power input, watts	1304	1206
Overall isentropic efficiency, %	47.8	49.2
Refrigerant-side Conditions		
Temperatures, °F		
Compressor shell inlet	58.7	53.6
Compressor shell outlet	201.1	197.3
Condenser inlet	191.9	187.4
Condenser exit	141.8	142.0
Condenser subcooling, F° *	7.0	7.0
TXV inlet	129.9	142.0
Evaporator inlet	54.1	46.2
Evaporator exit	56.4	51.2
Evaporator superheat, F° *	13.5	13.5
Pressures, psia		
Compressor shell inlet	54.4	51.7
Compressor shell exit	247.7	246.3
TXV inlet	241.1	244.8
Evaporator inlet	65.5	57.6
ΔP across condenser	1.8	1.3
ΔP across evaporator	11.3	5.6
Air- and Water-side Temperatures		
Evaporator inlet	68.0	68.0
Evaporator exit	54.7	56.4
Condenser inlet	126.6	126.6
Condenser exit	141.4	140.1
Evaporator Capacity, Btu/h	7907	7298
Condenser Capacity, Btu/h	10,075	9,256
COP	2.00	1.97

* Required to agree
Water flowrate was set at 1.4 gpm.

(-1.5% difference) because of the balancing out of the lower condenser capacity and lower compressor power input.

Parametric Study

The effect on the performance of the HPWH of varying certain parameters was analyzed to evaluate the design flexibility about the base case configuration. For these analyses the circulating water pump model was included. The following variables were chosen for this preliminary study:

- refrigerant subcooling at condenser exit,
- condenser length,
- evaporator tube diameter,
- number of evaporator tube rows,
- number of evaporator refrigerant circuits, and
- compressor displacement.

The remaining program input variables were fixed at the values used in the validation run base case. Plots were generated that showed COP as a function of two design parameters varied about their base case value. The base case results are indicated on the plots.

Constant condenser capacity lines were drawn as dashed lines on the plots for COP comparisons at the same capacity level. All parametric runs were executed at 68°F ambient and 54% relative humidity.

Figure 4 shows the effect of evaporator parallel equivalent refrigerant circuits and evaporator tube diameter on the HPWH COP. The rapid decline in COP for the smaller number of circuits is due to increased refrigerant pressure drops. This is even more apparent for the smaller evaporator tube diameters because of the larger amount of refrigerant pressure drop associated with them. The COP peaks at 2 to 3 circuits and then slowly decreases as the number of circuits increase because of the drop in refrigerant-side heat transfer coefficient as the mass flow is reduced in each circuit. At constant condenser capacity, a trade-off is possible between the number of circuits and the tube diameter size with minimal effect on COP.

In Fig. 5 the effect of evaporator tube rows and compressor displacement on COP is shown. The fan motor input power changed according to the air-side pressure drop which increased nearly proportionally to the number of tube rows. The larger number of evaporator tube rows is a direct relation to increased evaporator area. The changes in capacity and COP with increasing area

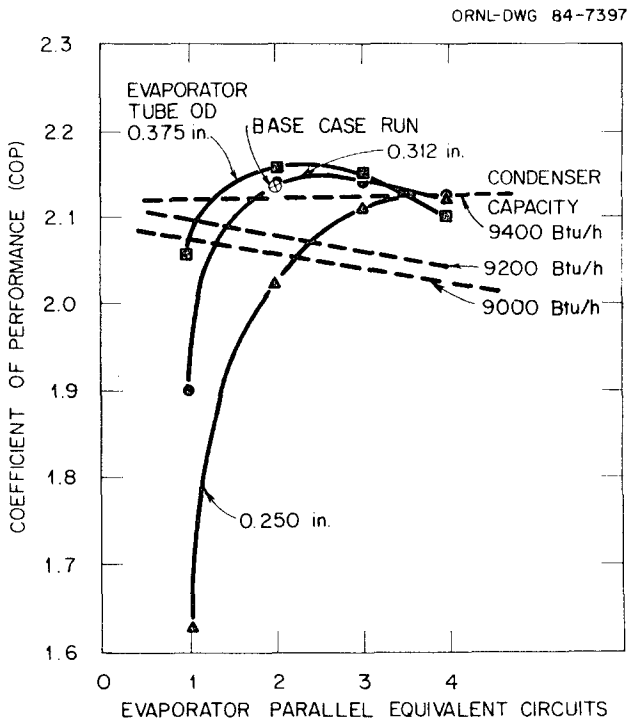

Fig. 4. Effect on COP of evaporator parallel equivalent refrigerant circuits and evaporator tube O.D. with constant condenser capacity lines.

and displacement are expected. The lines of constant capacity however show the amount the COP can be improved with larger areas at constant output and to what extent the compressor size must be reduced to achieve this.

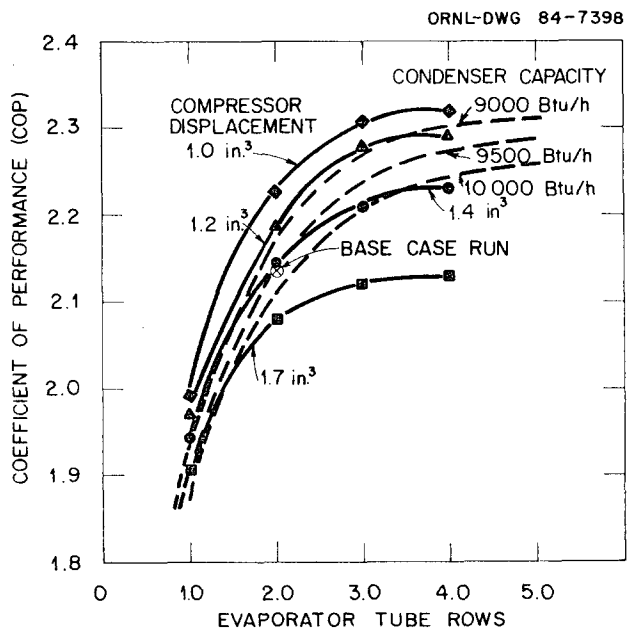

Fig. 5. Effect on COP of evaporator tube rows and compressor displacement with constant condenser capacity lines.

Figure 6 shows the effect of condenser length and subcooling on COP. The increase in compressor work due to higher condensing temperatures resulting from the extra subcooling outweighs the slight increase in capacity and results in a

decline in COP. The effect on COP of larger condenser areas diminishes as the condenser length increases.

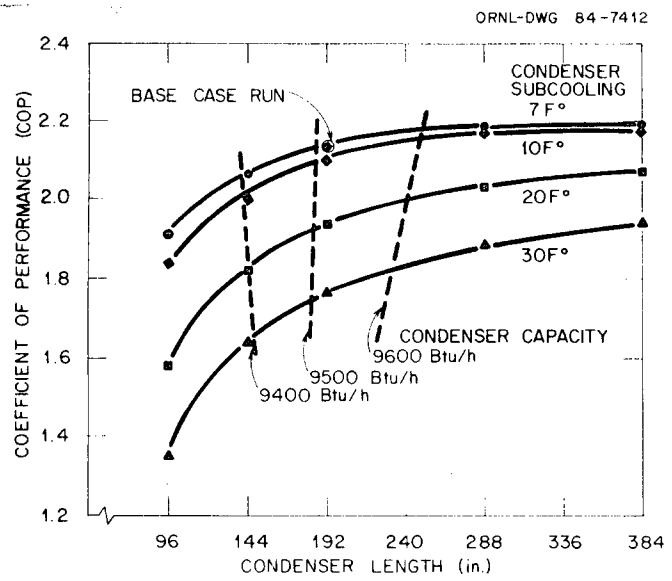

Fig. 6. Effect on COP of condenser length and condenser subcooling with constant condenser capacity lines.

Conclusions

1. The HPWH was experimentally tested under steady-state operation. The results showed the effect of varying ambient temperature and inlet water temperature on system performance.

2. The data base formed from the experiment was used to validate the HPWH steady-state design model. The model was proved capable of predicting the performance of the HPWH system.

3. The design model was used in a parametric study. Early parametric results show the design model's potential usefulness in design studies.

Recommendations

Early parametric results show the effect on performance of changing certain variables but no significantly large increase in COP was found. To see any substantial improvement the correct combinations of variables must be selected. Further studies involving contour-plots or optimizing programs should be undertaken. Both of these design techniques have been developed for use with the HPWH computer design model and are available at ORNL.

References

1. S. K. Fischer and C. K. Rice, "The Oak Ridge Heat Pump Models: A Steady-State Computer Design Model for Air-to-Air Heat Pumps," ORNL/CON-80/R1, Oak Ridge National Laboratory, August 1983.

2. M. Catan et al., "Optimized Ground Coupled Heat Pump Design Phase I Final Report," Brookhaven National Laboratory, (in publication process).

PREDICTED ENERGY SAVINGS FOR A RESIDENTIAL HEAT PUMP UTILIZING A FLUIDIZED-BED HEAT EXCHANGER

by Edward A. Vineyard

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Mr. Vineyard is presently involved in program planning in the area of commercial building systems at the Oak Ridge National Laboratory (ORNL). Prior to that assignment, he was a project manager in the ORNL appliance program. Before joining ORNL in 1980, he was employed at General Electric in Louisville, Kentucky, as a design engineer. He received a BS in Mechanical Engineering and an MS in Engineering Science from the University of Tennessee. Mr. Vineyard is a member of Pi Tau Sigma, Sigma Xi, and the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). He has chaired several symposiums on energy-efficent equipment at ASHRAE meetings and served as program chairman for two ASHRAE technical committees. In addition, he is an advisor for the Gas Research Institute and the New York State Energy Research and Development Administration.

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PREDICTED ENERGY SAVINGS FOR A RESIDENTIAL HEAT PUMP UTILIZING A FLUIDIZED-BED HEAT EXCHANGER*

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Abstract

The objective of this study was to determine if the energy efficiency of an optimized, state-of-the-art, residential heat pump could be increased by utilizing a fluidized-bed heat exchanger in place of a conventional heat exchanger on the outdoor side of the heat pump system. Using the Oak Ridge National Laboratory (ORNL) Heat Pump Model, comparisons were made between the advanced system (utilizing a fluidized-bed heat exchanger) and the base system (optimized, state-of-the-art) in eleven cities. The calculated annual energy savings for the advanced system ranged from 29 to 777 kWh/yr, with the largest savings occurring in areas where the frosting and defrosting losses of the base unit are greatest, such as in the northeast.

Background

In the design of air-to-refrigerant heat exchangers for residential heat pumps, the air-side heat transfer coefficient is one of the limiting factors in a better design. Typical values for this coefficient range from approximately 10 to 20 Btu/h-ft^2-°F.[9] As a means of compensating for these low values, more fins can be introduced to the outer tube surface to increase the area available for heat transfer. However, since this only partially compensates for the low heat transfer coefficient, other means of improvement are being investigated.

Recently, research has been performed using fluidized-bed heat exchangers as a means of improving the air-side heat transfer coefficient. It is believed that this is an area of great potential since air-side heat transfer coefficients as high as 280 Btu/h-ft^2-°F have been reported.[2] In experiments with a residential heat pump performed at Lehigh University, the heat exchanger on the outdoor side was replaced with a fluidized-bed heat exchanger resulting in an air-side heat transfer coefficient of approximately 97 Btu/h-ft^2-°F, which is eight to nine times that of conventional air-side heat transfer coefficients found in residential heat pumps.[9] The resulting coefficient of performance (COP) for heating was approximately 5.9, exclusive of fan losses.[9] If the pressure drop across the fluidized-bed heat exchanger, which was 2.2 in. H_2O, had been taken into account, the resulting fan losses would have lowered the COP considerably. For this reason, the analysis for the ORNL study concentrated on ways to minimize the fan losses.

Also included in the experiments at Lehigh were tests to determine frost prevention and deicing capabilities of the fluidized-bed heat exchanger. It was found that particle movement in the fluidized bed prevented ice crystals from forming on the tubes, thus eliminating the dynamic losses associated with frost build-up and defrosting.[9] For this reason, the ensuing analysis at ORNL considered these losses to be zero, whereas they are usually on the order of 5 to 10%.[4]

Annual Performance Factor

One basis for comparing the advanced system against the base system is the annual performance factor (APF). Following optimization of each system for a Nashville climate, the APFs were calculated for eleven climates using the ORNL Heat Pump Model, which provides steady-state COP and capacity data as a function of ambient temperature, along with

1. the average seasonal loads for each temperature bin along with the number of hours of occurrence, and

2. frosting, defrosting, and cycling degradation factors for COP and capacity based on research by Parken and Kelly.[6]

The curves of steady-state COP vs ambient temperature, to be used in determining the APFs, were generated by executing the model at four different ambient temperatures, two each heating and cooling, and assuming that steady-state performance was linear across the entire range of heating (-13°F to 62°F) and cooling (67°F to 97°F) loads.[8]

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

Utilizing the Parken and Kelly frosting degradation factors, calculations were made of the degradation in performance caused by the frost accumulation on the outdoor coil, energy required to clear the coil, and amount of heat removed from the conditioned space during defrost which, in turn, had to be replaced by resistance heat. The model simulated a defrost cycle 3 min long with occurrences every 60 min at 42°F up to every 150 min at 12°F (with linear extrapolations between the two temperatures).

Cycling loss factors, also based on research by Parken and Kelly, were used for the advanced and base case analyses to calculate the degradation of steady-state COP due to cycling.

Sensitivity Plots

The APF is a function of many parameters of the heat pump system design such as air flow rates, compressor displacement, heat exchanger areas, and pressure drops. The ORNL Heat Pump Model, used in conjunction with a plotting routine, makes it possible to determine the sensitivity of the APF to changes in design conditions.[7] Utilization of this procedure results in plots which show contours of constant values of the APF as a function of pairs of specified design parameters, while remaining system parameters are held fixed. A sample plot, shown in Fig. 1, reveals the sensitivity of the APF to changes in the indoor and outdoor air flow rates. Lines of concentricity are of constant APF, while the solid lines moving diagonally from the middle of the graph to the upper left-hand portion are of combinations of air flow rates that yield capacities of 30,000 and 30,170 Btu/h, the latter being the design heating capacity in a Nashville climate for an 1800-ft2 house with minimum insulation as required by the United States Department of Housing and Urban Development (HUD).

Typically, the combination of design parameters producing the best APF will be achieved where the required constraint (in this example, capacity) is tangent to a surface of constant APF. For the case shown in Fig. 1, the tangency point lies at the maximum unconstrained APF. There are

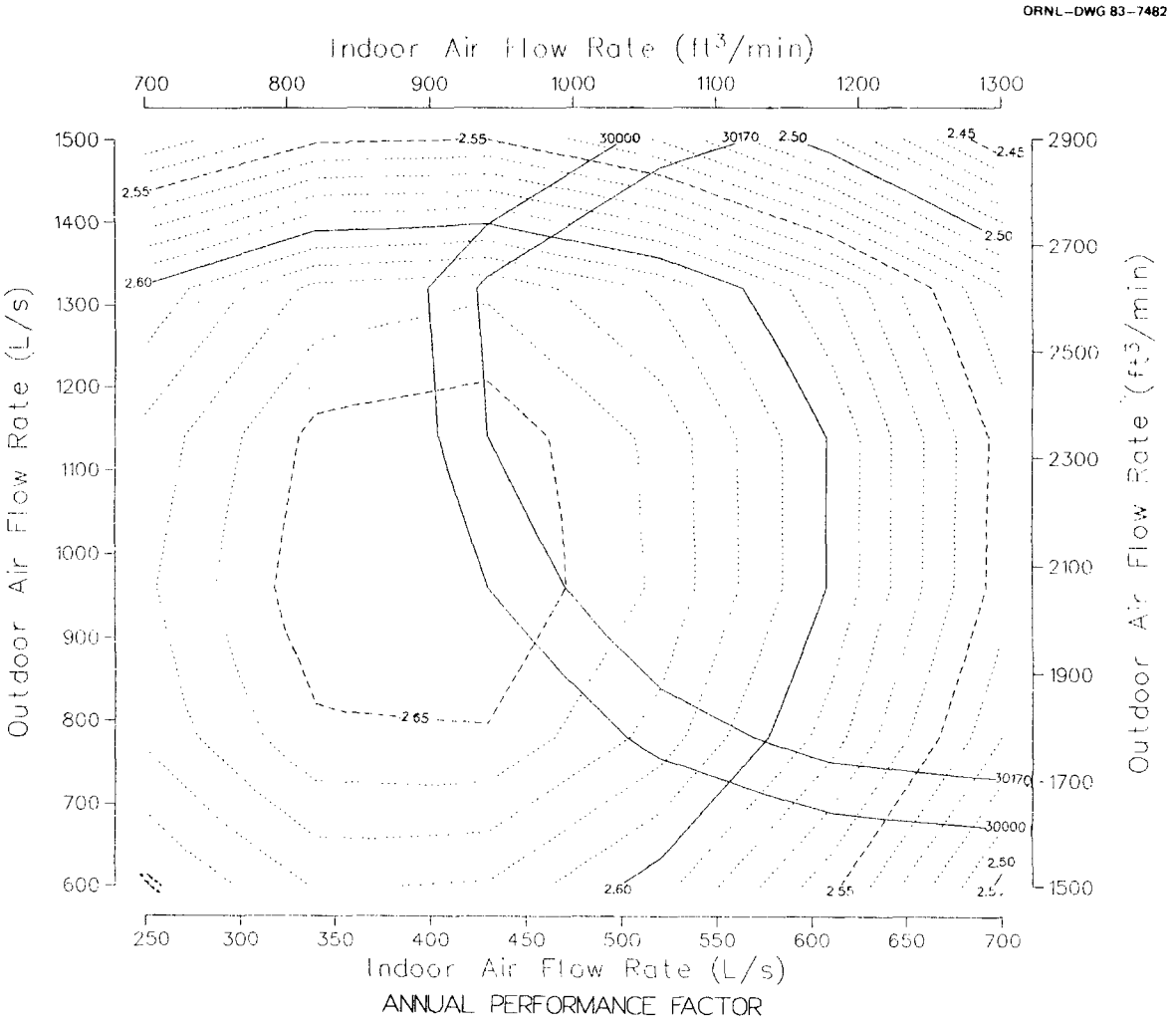

Fig. 1. APF as a function of outdoor and indoor air flow rates for a glass microsphere material with a particle diameter of 250 microns, density of 9 lbm/ft^3, and specific heat of 0.27.

instances, however, where the tangency point is at some point less than the maximum unconstrained value.

For purposes of determining the optimum APF of the advanced and base systems, plots were generated of the APF as functions of the outdoor air flow rate, compressor displacement, and indoor air flow rate. These three design parameters were selected since they appear to have the most influence on the APF based on previous analytical work at ORNL.[8] It is noted that the advanced and base systems were optimized for a Nashville climate rather than for each of the eleven individual climates because of the amount of time required for each optimization procedure. The Nashville climate was selected because it has characteristics closest to the average for all the climates.

Advanced System Design

Based on the results of the testing at Lehigh, the approach taken was to try to minimize the pressure drop and increase the efficiencies of the fan and motor in an effort to reduce fan power. In doing so, the probability of increasing the system APF over that of a conventional system is increased. No effort was made to lower the air flow rate since its optimal value was determined from sensitivity plots.

For the most part, the advanced system used the same design parameters as those used for the base system. For instance, the same total heat exchanger surface area is used in both designs to enable a fair comparison. There are parameters, however, that are not present in both designs or that dictated changing in order to arrive at a more optimal design for the advanced system. These are described in the following sections.

Configuration

A tiered arrangement, as shown in Fig. 2, was selected as the configuration for the fluidized-bed heat exchanger in order to minimize pressure drop and maintain an optimal volume to package the required heat transfer surface area. This configuration, credited to Mechanical Technology, Inc., was considered to be the best option when compared with other configurations, shown in Figs. 3 and 4.[2] The design in Fig. 3, one in which several tube rows are immersed in a single tray of bed material, has too large a pressure drop across the heat exchanger; this pressure drop prohibited further consideration. Figure 4 shows a design consisting of a large frontal area heat exchanger with a single row of tubes. This design, while having a low pressure drop, requires a lot of space and would probably have to be mounted on a rooftop. For this reason, it was not considered for further analysis.

ORNL-DWG 83-7447

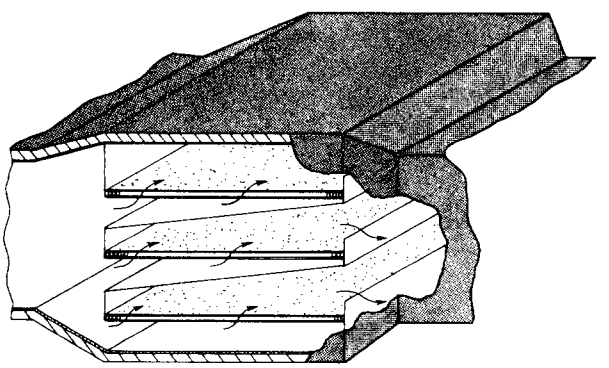

Fig. 2. Parallel arrangement.

ORNL-DWG 83-7443

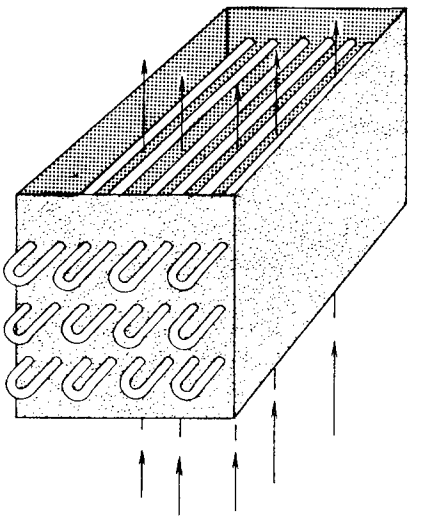

Fig. 3. Deep bed arrangement.

ORNL-DWG 83-7442

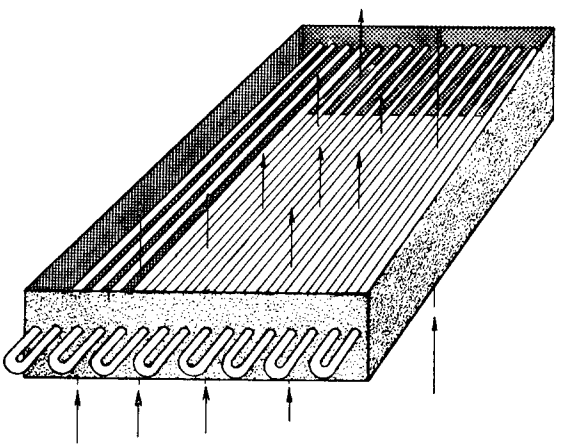

Fig. 4. Shallow bed arrangement.

Bed Material Selection

The main difference between a fluidized-bed heat exchanger and a conventional heat exchanger is that the fluidized-bed heat exchanger requires a material, such as sand, in the bed to transfer heat to the tube bundle, while a conventional heat exchanger relies only on air to transfer heat to

the tubes. Due to the necessity of suspending and mixing the bed material to achieve high rates of heat transfer, the fan power requirements are higher for a fluidized-bed heat exchanger than they are for a conventional heat exchanger. Since fan power is proportional to the density of the material in the bed, it is desirable to choose a material with the lowest density possible in order to achieve a low fan power. In contrast to this, the air-side heat transfer coefficient, which is directly proportional to the density raised to the 0.2 power, requires that density be as high as possible to achieve high rates of heat transfer.[3] Thus, the effects of increased density had to be evaluated to determine if the increase in the air-side heat transfer coefficient more than offset the increase in fan power.

A second material property that affects the air-side heat transfer coefficient is the particle diameter. Previous work determined that the air-side heat transfer coefficient is inversely proportional to the square root of the particle diameter.[3] Since a high air-side heat transfer coefficient is the intent of the project, small particle diameters are desirable.

Also affecting the air-side heat transfer coefficient is the specific heat. This property raised to the 0.2 power is directly proportional to the air-side heat transfer coefficient.[3] Therefore, it is desirable to choose a material with as high a specific heat as possible.

Figures 5, 6, and 7 show the discussed trends of the fan power and air-side heat transfer coefficient as functions of density, particle diameter, and specific heat. The graphs help to examine the

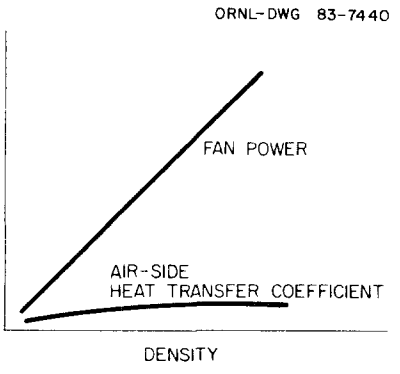

Fig. 5. Fan power and air-side heat transfer coefficient as functions of density.

overall effects of material property combinations and thus save time by eliminating unnecessary analysis of materials that are not practical. Summarizing the results of the three graphs, the most desirable material properties are low density, small diameter, and high specific heat. Based on the results of the graphs, materials were chosen with the following ranges of property values: the densities varied from 9 to 25 lbm/ft3; the particle diameters ranged from 50 to 250 microns; and the

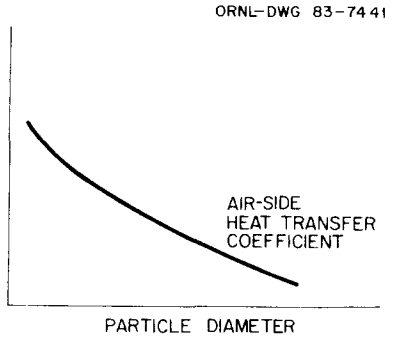

Fig. 6. Air-side heat transfer coefficient as a function of particle diameter.

specific heat values were from 0.20 to 0.27 Btu/lbm-°F.

The final material selection was based on results from sensitivity plots of the APF. The material selected was a glass microsphere with a

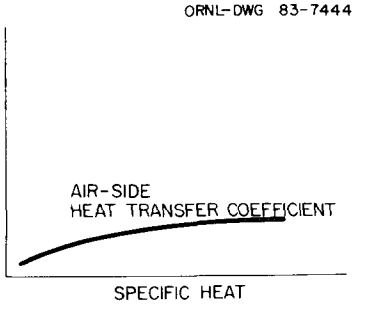

Fig. 7. Air-side heat transfer coefficient as a function of specific heat.

density of 9 lbm/ft3, a particle diameter of 250 microns, and a specific heat of 0.27 Btu/lbm-°F.

Outdoor Air Flow Rate

The outdoor air flow rate was determined from a sensitivity plot of APF as a function of outdoor air flow rate vs compressor displacement. From the plot, it was revealed that an outdoor air flow rate of 2,320 ft3/min yields the highest APF as determined from the point of tangency between the lines of constant APF and a capacity of 30,170 Btu/h.

Indoor Air Flow Rate

Once the outdoor air flow rate was determined, a sensitivity plot of APF as a function of indoor air flow rate vs outdoor air flow rate was generated to see if the optimal indoor air flow rate varied significantly from the initial value of 959 ft3/min used in determining the APF as a function of outdoor air flow vs compressor displacement. From the plot, it was determined that the optimal value was very close to the initial value of 959 ft3/min. Therefore, the 959 ft3/min value was used in the design.

Bed Height

The bed height (the static depth of the material in the fluidized bed), selected as 1 1/2 in., is based on work performed at Lehigh University.[9] There were no problems cited with this bed height, and the bed material completely covered the tubes. The reason for choosing as low a bed height as practical was to minimize pressure drop across the heat exchanger, which, in turn, lowered the fan power.

High-efficiency Propeller Fan

The propeller fan used in the design was developed by Westinghouse Electric Corporation as part of a program to improve the efficiency of an advanced electric heat pump. It is driven by a 0.2-hp, 8-pole fan motor which has an efficiency of 67.5% when utilizing a low-resistance rotor.[10] Characteristics of the propeller fan performance are shown in Fig. 8.[11] From these performance curves, a table was generated which is the basis for equations used in the ORNL Heat Pump Model to calculate the efficiency of the fan/fan-motor combination. Using the design conditions of 2,320 ft^3/min for the outdoor air flow rate, a fan speed

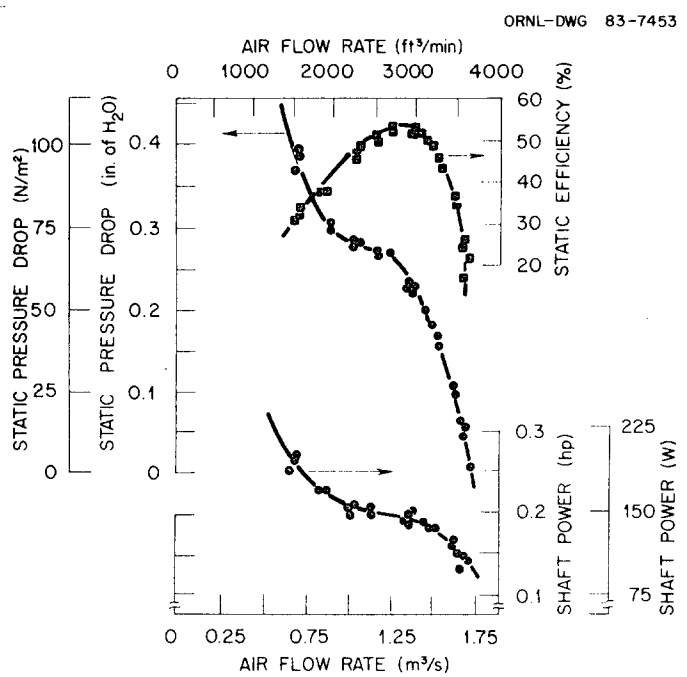

Fig. 8. Graph of high-efficiency fan performance characteristics. SOURCE: T. C. Wright, P. C. Krause, R. S. Lackey and S. E. Veyo, "A High Efficiency Heat Pump Fan" (to be published).

of 825 rpm, and static pressure drops from 0.2860 to 0.2838 in. H_2O (the range for temperatures between 17 and 95°F), fan efficiencies of 48.5 to 48.7% were calculated.

Performance Results

As shown in Table 1, the differences in the two units compared in this study are that the outdoor air flow rate, outdoor heat exchanger tube rows, compressor displacement, and specific speed of the outdoor fan were all lower for the advanced unit, while pressure drop across the outdoor heat exchanger and the fan/fan-motor efficiency of the outdoor fan were higher. The indoor heat exchanger characteristics, outdoor heat exchanger refrigerant circuits, and fraction of total heat exchanger area (indoor and outdoor) remained the same.

Table 1. Comparison of Heat Pump Configurations

	Advanced	Base
Indoor Heat Exchanger		
Tube rows	3	3
Refrigerant circuits	4	4
Fraction of total area	0.461	0.461
Air flow rate (ft^3/min)	959.0	959.0
Outdoor Heat Exchanger		
Tube rows	1	2
Refrigerant circuits	4	4
Fraction of total area	0.539	0.539
Air flow rate (ft^3/min)	2320.0	2476.9
Compressor Displacement (in^3)	2.694	2.752
Outdoor Fan at 47°F		
Pressure drop (in. H_2O)	0.2883	0.1242
Specific speed (rpm)	101,000	196,200
Fan-motor efficiency (%)	67.5	55.0
Fan/fan-motor efficiency (%)	33.0	22.3

The results are presented for steady-state conditions and dynamic conditions in a Nashville climate. The steady-state results reveal an increase in the cooling COP and a decrease in the heating COP for the advanced unit compared with the base unit. The dynamic results show that the performance is better for the advanced unit in both heating and cooling. The reason for the dynamic results showing an improvement in the heating season while the steady-state results showed a decrease in performance is the elimination of frosting and defrosting losses. Thus, the advanced unit will show higher savings in areas where it will operate many hours at or below the frosting point (43°F).

Steady-state Operation

The steady-state performance of the two units for temperatures ranging from -13 to 97°F is shown in Table 2. The results show a decrease in the heating COPs and an increase in the cooling COPs for the advanced unit compared with the base unit.

Examining an energy breakdown of the two units at a heating (17°F) and cooling (82°F) temperature gives an explanation for the difference. Table 3 shows the input energy for the advanced unit was 333 Btu/h higher at 17°F and 79 Btu/h lower at 82°F than the base unit. The reason for this is explained by the compressor power. In cooling, the compressor power decreased substantially, more than

Table 2. Steady-state Performance Parameters for Advanced and Base Systems

Temp. (°F)	Heating COP Adv.	Heating COP Base	Cooling COP Adv.	Cooling COP Base	Capacity (Btu/h) Adv.	Capacity (Btu/h) Base
-13	1.588	1.721	-	-	5,080	4,936
2	2.054	2.165	-	-	11,328	11,129
17	2.520	2.609	-	-	17,575	17,321
32	2.987	3.053	-	-	23,823	23,514
47	3.453	3.496	-	-	30,071	29,707
62	3.920	3.940	-	-	36,318	35,899
67	-	-	4.193	4.123	35,290	35,065
73	-	-	3.940	3.875	34,231	34,000
79	-	-	3.687	3.627	33,173	32,935
85	-	-	3.433	3.379	32,114	31,869
92	-	-	3.138	3.090	30,879	30,627
97	-	-	2.927	2.884	29,997	29,739

Table 3. Energy Balance for Advanced vs Base Unit at Ambient Temperatures of 17°F and 82°F

	17°F (Heating) Advanced	17°F (Heating) Base	82°F (Cooling) Advanced	82°F (Cooling) Base
Heat from condenser to air (Btu/h)	17,069	16,816	38,880	39,054
Heat to evaporator from air (Btu/h)	13,223	12,918	33,254	33,012
Power to indoor fan (Btu/h)	506	506	610	610
Power to outdoor fan (Btu/h)	812	444	805	436
Power to compressor motor (Btu/h)	5,655	5,689	7,755	8,203
Shell heat loss (Btu/h)	848	853	1,164	1,230
Total heat to/from indoor air (Btu/h)	17,575	17,322	32,644	32,402
COP (steady-state)	2.520	2.609	3.560	3.500

offsetting the increase in power to the outdoor fan caused by the fluidized bed. In heating, the compressor power stayed approximately the same. Thus, there was nothing to offset the increase in outdoor fan power.

Table 4 shows that the enthalpy difference of the refrigerant across the compressor decreases 2.7% at 17°F and 4.5% at 82°F for the advanced unit; this would imply a small decrease in input power to the compressor. However, the refrigerant mass flow rate increases 2.1% at 17°F and decreases 1.0% at 82°F for the advanced unit, which accounts for the increased power requirement in heating and decreased power requirement in cooling. Further investigation revealed that changes in the condensing and evaporating temperatures, brought on by the increased effectiveness of the outdoor heat exchanger (i.e., fluidized bed), were causing the changes in the refrigerant mass flow rate.

Dynamic Operation (Nashville Climate)

The results of annual and seasonal performance along with energy consumption are presented for the advanced and base systems. The reason for presenting both performance factors and energy consumption is that performance factors alone, usually the most common method of ranking heat pumps, do not address the area of most interest (i.e., energy savings).

A comparison of the annual performance factor and energy consumption in Table 5 shows an improvement of 8.2% in APF and a decrease of 7.6% in energy consumption for the advanced heat pump. A breakdown of the energy consumption in Table 6 reveals the largest portion of the energy savings (85.9%) was due to elimination of the frosting and defrosting losses while the rest was due to improvements in steady-state operation (12.7%) and decreased use of resistance heat (3.5%).

Table 4. Energy Balance for Compressor at Ambient Temperatures of 17°F and 82°F

	17°F (Heating) Adv.	17°F (Heating) Base	82°F (Cooling) Adv.	82°F (Cooling) Base
Refrigerant mass flow rate (lbm/h)	177.08	173.44	444.01	448.39
Refrigerant enthalpy exiting compressor (Btu/lbm)	133.62	134.23	124.92	125.64
Refrigerant enthalpy entering compressor (Btu/lbm)	106.48	106.35	110.07	110.09
Δ refrigerant enthalpy (Btu/lbm)	27.14	27.88	14.85	15.55
Compressor work (Btu/h)	4,807	4,836	6,591	6,973
Shell heat loss (Btu/h)	848	853	1,164	1,230
Power to compressor motor (Btu/h)	5,655	5,689	7,755	8,203

From Tables 7 and 8, it can be seen that of the 652 kWh saved annually by the advanced system, approximately 606 kWh (86.3%) are saved in the heating season and only 47 kWh (13.7%) are saved in the cooling season. This inequity is due to the large savings realized from elimination of the frosting and defrosting losses which occur in the heating season. For the cooling season, most of the energy savings came from improvements in steady-state operation.

Energy Savings

The annual energy savings for each region (Fig. 9) are the product of the annual energy savings per unit and the number of units to which the savings apply (Table 9). The energy savings on a unit basis were calculated using the steady-state

Table 5. Annual Energy Consumption Breakdown (Nashville Climate)

	Base	Advanced
ANNUAL PERFORMANCE		
Annual performance factor	2.451	2.652
Heating and cooling delivered (MMBtu)	72.2	72.2
Energy consumption (kWh)	8,630	7,978
Hours of operation	3,089	2,934
ENERGY CONSUMPTION BREAKDOWN		
Steady-state operation (kWh)	6,290	6,207
Resistance heat (kWh)	639	616
Cycling losses (kWh)	1,141	1,155
Frosting and defrosting losses (kWh)	560	0

Table 6. Comparison of Annual Energy Consumption for Advanced and Base Systems (Nashville Climate)

Energy Consumption Breakdown	Base (kWh)	Adv. (kWh)	Difference (kWh)	Percent Savings
Steady-state operation	6,290	6,207	83	12.7
Resistance heat	639	616	23	3.5
Cycling losses	1,141	1,155	-14	-2.1
Frosting/ defrosting losses	560	0	560	85.9
Total	8,630	7,978	652	100.0

Table 7. Heating Energy Consumption Breakdown (Nashville Climate)

	Base	Adv.	Difference
SEASONAL PERFORMANCE			
Heating season performance factor	2.192	2.466	—
Heating delivered (MMBtu)	40.8	40.8	0
Energy consumption (kWh)	5,456	4,850	606
Hours of operation	1,970	1,822	148
ENERGY CONSUMPTION BREAKDOWN			
Steady-state operation (kWh)	3,643	3,601	42
Resistance heat (kWh)	639	616	23
Cycling losses (kWh)	614	633	-19
Frosting/defrosting losses (kWh)	560	0	560

performance parameters in Table 2, dynamic loss parameters, and binned load data for representative cities in each region. The steady-state performance parameters are for units optimized for a Nashville climate, which was selected as roughly average for the nation based on its heating and

Table 8. Cooling Energy Consumption Breakdown (Nashville Climate)

	Base	Adv.	Difference
SEASONAL PERFORMANCE			
Cooling season performance factor	2.896	2.939	—
Cooling delivered (MMBtu)	31.4	31.4	0
Energy consumption (kWh)	3,174	3,128	46
Hours of operation	1,119	1,112	7
ENERGY CONSUMPTION BREAKDOWN			
Steady-state operation (kWh)	2,647	2,606	41
Cycling losses (kWh)	527	522	5

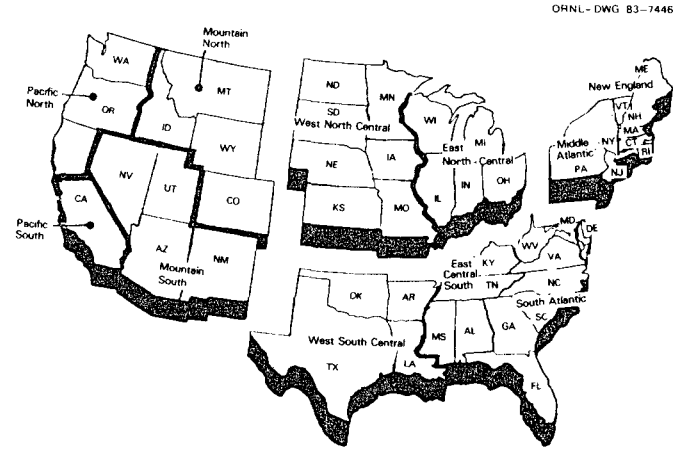

Fig. 9 Regional breakdown of states.

cooling loads.[8] The ideal approach would have been to optimize the system for each region; however, this was impractical because of the amount of time and computer costs required. The number of units in each region to which the savings apply was determined from assumptions of percentages of homes in each equipment category that would consider utilizing the advanced heat pump system and the number of units in each equipment category as determined from housing information from the U.S. Bureau of Census.[5] The national annual energy savings of the advanced unit, based on total market penetration, are shown in Table 9 to be approximately 15.0 x 10^{12} Wh/yr, which is equivalent to 0.15 quads/yr of primary energy. The regions yielding the highest savings are regions 3 and 5. The high savings in region 5 are mainly the result of a large number of units in that region, while the savings in region 3 are due to a combination of a large number of units and high unit savings. The lowest regions, 9 and 11, had both low unit savings and a small number of units.

Results for unit energy consumption by function (heating and cooling) along with performance parameters for the advanced and base systems are given in Tables 10 and 11 respectively. The results show that the largest portion of total

energy savings comes from heating rather than cooling for nine of the regions. This was generally expected since the energy consumption for heating was considerably higher in eight of the regions for the base case. Region 7 was the only region where the higher energy savings occurred in the opposite function to what was expected. An interesting observation is that energy consumption for heating in region 11 was actually higher for the advanced system than for the base system. An examination of the binned load data shows that the unit operated below the frosting point (43°F) approximately 2% of the time.1 As previously discussed, elimination of frosting losses caused the seasonal heating COPs to be higher for the advanced system even though the steady-state COPs were lower. Thus, since there was essentially no improvement from the elimination of the frosting losses, the region 11 unit consumed more energy during the heating season.

It was an interesting exercise to rank the annual energy savings, base energy usage and improvement in the APF to examine the basis for energy savings (i.e., were large savings due to the unit using a large amount of energy initially or did they arise from a substantial improvement in efficiency). The results, shown in Table 12, reveal that the energy savings in regions 1, 4, and 11 were equally attributable to the base energy usage and improved efficiency. For regions 5, 6, 8, and 10, the savings were mainly the result of improvement in the APF, while for regions 2, 3, 7, and 9, the savings were more from large initial energy usage.

Table 9. Regional Energy Savings for Advanced System Relative to Base System

Region	Representative City	Number of Units (10^6)	Annual Unit Energy Savings (kWh/yr)	Annual Region Energy Savings (10^9 Wh/yr)
1	Boston	1.212	615	745
2	Syracuse	3.324	378	1,256
3	Chicago	5.588	649	3,627
4	Kansas City	2.886	642	1,853
5	Atlanta	7.184	476	3,420
6	Knoxville	2.056	514	1,057
7	Fort Worth	4.492	286	1,285
8	Boise	0.654	777	508
9	Phoenix	0.886	165	146
10	Portland	1.422	727	1,034
11	Los Angeles	1.754	29	51
		31.458		14,982[a]

[a] Equivalent to 0.15 quads/yr primary energy.

Table 10. Regional Energy Usage and Savings of Advanced System vs Base System

Region	Heating Energy (kWh/yr)			Cooling Energy (kWh/yr)			Total Energy (kWh/yr)		
	Base	Advanced	Savings	Base	Advanced	Savings	Base	Advanced	Savings
1	8,486	7,931	555	1,284	1,224	60	9,770	9,155	615
2	10,255	9,970	285	1,716	1,623	93	11,971	11,593	378
3	10,472	9,877	595	1,825	1,771	54	12,297	11,648	649
4	7,249	6,659	590	3,240	3,188	52	10,489	9,847	642
5	4,352	3,926	426	3,134	3,084	50	7,486	7,010	476
6	4,878	4,409	469	2,785	2,740	45	7,663	7,149	514
7	3,134	2,933	201	5,364	5,279	85	8,498	8,212	286
8	8,091	7,387	704	2,098	2,025	73	10,189	9,412	777
9	1,783	1,712	71	6,145	6,051	94	7,928	7,763	165
10	5,410	4,703	707	1,154	1,134	20	6,564	5,837	727
11	1,331	1,333	-2	1,890	1,859	31	3,221	3,192	29

Table 11. Seasonal Performance Parameters for Advanced and Base Systems by Region

Region	Base System			Advanced System		
	COPH[a]	COPC[b]	APF[c]	COPH	COPC	APF
1	2.196	2.808	2.277	2.349	2.946	2.429
2	2.157	2.655	2.229	2.218	2.807	2.301
3	2.130	2.718	2.217	2.258	2.802	2.341
4	2.094	2.823	2.319	2.280	2.868	2.470
5	2.265	2.958	2.556	2.512	3.006	2.729
6	2.262	2.892	2.493	2.504	2.940	2.671
7	2.292	2.718	2.562	2.449	2.763	2.651
8	2.187	2.880	2.331	2.396	2.982	2.522
9	2.304	2.598	2.532	2.399	2.639	2.586
10	2.340	3.213	2.493	2.693	3.267	2.805
11	2.562	3.129	2.895	2.559	3.182	2.922

[a] COPH = coefficient of performance, heating.
[b] COPC = coefficient of performance, cooling.
[c] APF = annual performance factor.

Summary and Conclusions

The results reveal that the energy savings by region for the advanced unit range from a low of 29 to a high of 777 kWh/yr with a unit-weighted average of 476 kWh/yr. The percentage savings range from 0.9 to 11.1% with a unit-weighted average of 5.2%. The largest savings, which occurred during the heating season, were mainly due to elimination of the frosting and defrosting losses.

Table 12. Ranking of Annual Energy Savings, Base Energy Usage, and Improvement in APF by Region

Region	Annual Energy Savings (Rank) (kWh/yr)	Base Energy Usage (Rank) (kWh/yr)	Improvement in APF (Rank) (%)
8	777 (1)	10,189 (4)	8.2 (2)
10	729 (2)	6,564 (10)	12.5 (1)
4	649 (3)	12,297 (1)	5.6 (7)
3	642 (4)	10,489 (3)	6.5 (6)
1	615 (5)	9,770 (5)	5.7 (5)
6	514 (6)	7,663 (8)	7.1 (3)
5	476 (7)	7,486 (9)	6.8 (4)
2	378 (8)	11,971 (2)	3.2 (9)
7	286 (9)	8,498 (6)	3.5 (8)
9	165 (10)	7,928 (7)	2.1 (10)
11	29 (11)	3,221 (11)	0.9 (11)

The results show that the fluidized-bed heat exchanger, when substituted for the outdoor heat exchanger in a residential heat pump, could save substantial amounts of energy provided the increase in initial cost is not prohibitive. The energy savings would be highest in areas of the country where the frosting and defrosting losses of a conventional heat pump are high, such as the northeast. Areas where these losses are low, such as the southwest, would experience minimal savings, if any.

Potential problem areas which should be investigated include elutriation (blow-out of bed material) and clogging of the bed in the presence of moisture, such as rain or condensation. In addition, the heat transfer coefficient should be verified in actual testing, along with testing to determine if frosting is eliminated using the selected particles.

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FREE PISTON STIRLING ENGINE HEAT PUMP DEVELOPMENT

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FREE PISTON STIRLING ENGINE HEAT PUMP DEVELOPMENT

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Abstract

The Free-Piston Stirling Engine (FPSE) is an important prime mover option in developing a viable gas-fueled heat pump for residential applications. The Stirling engine, while potentially attractive from a performance, life, and cost standpoint, is on a long-term development track. The reported results show that progress has been made in improving engine performance and reliability. The engine development is being funded by the Gas Research Institute (GRI), with the main technical goal being integration of the most advanced, reliable system components into a prototype FPSE heat pump module for laboratory testing and evaluation. Development of the compressor for the FPSE-driven heat pump is being supported by a coordinated and parallel Department of Energy (DOE)-sponsored program.

This paper highlights the technical goals that were achieved during 1983. Specifically, the external heat system efficiency was increased from 70 to 80% (HHV), and the engine's indicated mechanical efficiency was increased from 24 to 32%. The engine's displacer control motor efficiency was also improved, and a one-third reduction in parasitic power was achieved.

In addition to engine efficiency gains, reliability and durability were also substantially increased. Over 1000 hours of engine operation were achieved during an endurance test, a portion of which involved duty cycling, as well as internal gas bearing operation. Several components with unacceptable failure rates were identified for improvement. Finally, an economic analysis was performed for both heat pumps and cogenerator systems, the results of which (based on component test data) showed that the FPSE Heat-Actuated Heat Pump (HAHP) configuration may have good economic potential for residential space conditioning.

Introduction

The FPSE has advantageous operating characteristics that are generic to most Stirling-engine designs, i.e., high efficiency, quiet and clean operation, and multifuel capability. In addition, the FPSE has the potential for long life with high reliability resulting from its operation with only two moving parts and gas bearing lubrication. Because of these unique features, the FPSE was identified as a candidate prime mover for operation in a Gas-Fueled Heat Pump System.

During 1981, Mechanical Technology Incorporated (MTI) designed and built, under private funding, a fifth-generation FPSE configured to power a linear alternator. The unit was termed the Engineering Model (EM). The original purpose of the EM was to demonstrate the efficiency and reliability potential of the FPSE for system applications. In December, 1982, GRI initiated a program to develop the EM to its full potential as a gas-fueled prime mover for residential heat pumps. In a parallel program, DOE is sponsoring the development of a novel design concept for hermetically coupling the EM with a freon compressor. The object of this program is to demonstrate technical feasibility. Overall, the common goal of the two programs is to establish component readiness leading to system integration and evaluation.

Efficiency Development Goals

The development process is centered on the three major components that comprise the FPSE power module: 1) a natural-gas combustor; 2) a displacer-controlled engine; and, 3) a diaphragm-coupled compressor (these components are schematically depicted in Figure 1). Taken as an integrated unit, the power module was designed to be capable of providing three tons of refrigeration at the 95°F day with condenser and evaporator temperatures of 120°F and 45°F using R-22 refrigerant. At the beginning of the development effort, the measured performance of each component was below its potential. The measured and potential performance at the rating point for each component is listed in Table 1.

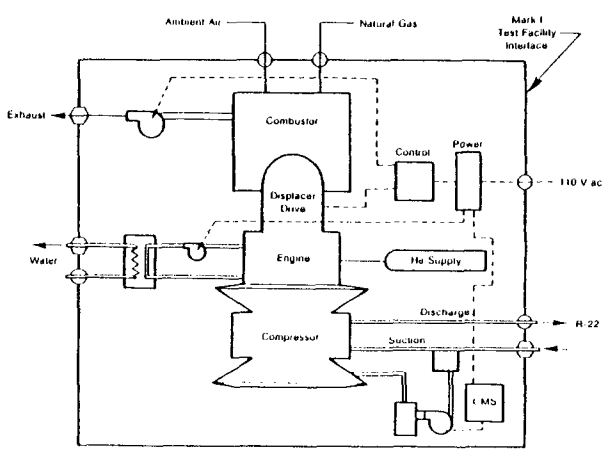


Figure 1 FPSE Heat Pump Compressor Schematic

An analytical effort was undertaken to determine the economic sensitivities of a gas-fueled heat pump to component performance. This study, headed by Lennox Industries, MTI's commercialization partner on the project, was intended to identify the importance of component efficiency. The analysis took into account component cost projection as well as climatic and energy price variation. Table 2 summarizes the overall results of the study, ignoring variations in results due to regional dissimilarities.

Together Tables 1 and 2 point out the importance of component efficiency, and the requirement for improving the power module performance over the baseline performance measurements. With the performance targets in perspective, the development program was started. The following sections summarize the progress achieved.

Table 1
FPSE Heat Pump Compressor Performance

	Baseline Measurement	Design Potential
Engine Power	3.0 kW	3.0 kW
Combustor Efficiency (HHV)	70 %	80 %
Engine Efficiency	31 %	44 %
Compressor Efficiency	54 %	65 %
Compressor Capacity @ 95°F	18,000 BtuH	36,000 BtuH

Table 2
Economic Sensitivity to Component Performance

Parameter	Low Perform-ance	High Perform-ance	Development Project Target
Combustor Efficiency	75%	80%	78%
Engine Efficiency	34%	44%	37%
Compressor Efficiency	50%	60%	55%
Average Annual Energy Cost Savings	$104	$279	$171
Premium Price*	$938	$938	$938
Payback Period*	9 yrs.	3.4 yrs.	5.5 yrs.

*As compared to a high-efficiency (95%) gas furnace and high-performance (EER = 14) air conditioner.

Combustor Development

The original EM combustors had a measured HHV efficiency of below 70%, and required 25" H_2O supply pressure gas fuel. The unit, with a turbulent-diffusion flame combustion process, was known to have significant overboard leakage of combustion gas, which contributed to its relatively low operating efficiency. The overall objectives were to develop a gas-fired external heat system that operates with no leakage, and to identify an approach for operating with natural gas at a typical residential supply pressure of 3" H_2O.

The primary performance indicator of the external heat system is combustor efficiency defined as:

$$\eta_{EHS} = Q_{HH}/E_{in} = \frac{\text{Energy Transferred Across the Heater Head}}{\text{Energy of Fuel and Air at Combustor Inlet}}$$

Prior to the start of the combustor development effort, mass leakage from the combustor was known to be significant. It was speculated that if the mass leakage could be reduced, combustor efficiency would improve to well above 80% Lower Heating Value (LHV). Quantifying the energy lost due to leakage was identified as the first area of investigation in order to obtain good system energy balance, and to verify the potential for achieving improved performance.

The curves in Figure 2 show that if all le ked energy (assumed to leave the combustor by external mass leakage at the heater head exhaust) was channeled through the hot side of the preheater, combustor efficiency would increase from 70% Higher Heating Value (HHV) to 78% HHV at 9 kW (air/fuel (A/F) = 15). Under the GRI Program, advanced combustor with an efficiency goal of 75% HHV was designed and fabricated for reduced leakage. Figure 3 shows the combustor that was developed under this task.

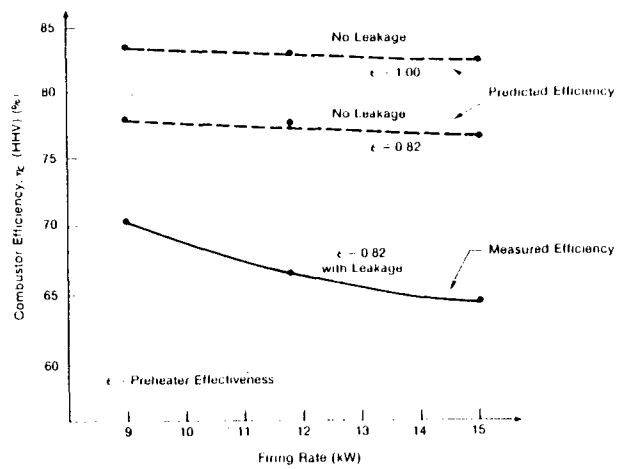


Figure 2 G3 Combustor Efficiency Comparison (A/F = 15)

Figure 3 The G6 Combustor

One of the primary objectives of the new combustor design was to reduce external leakages. To address this objective, special attention was given to the mechanical joints and preheater brazing technique so as to reduce or eliminate known leakage paths identified during characterization testing of the early combustor design. Figure 4 shows that this was accomplished. The leakage obtained from a combustor mass flow balance, i.e., % leakage = $(M_{in} - M_{out})/M_{in}$, was reduced from levels of 15-20% to levels of 6-8% in the new combustor.

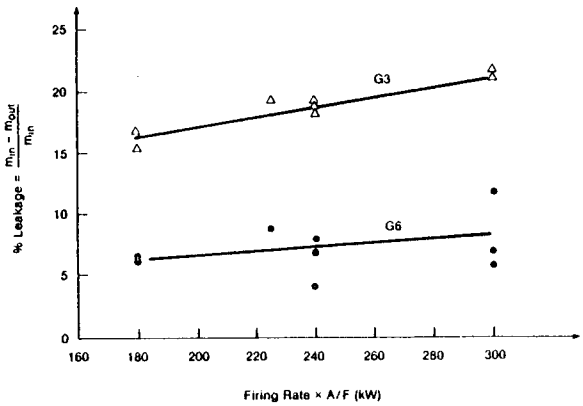


Figure 4 External Leakage Rates - Upgraded (G6) versus Original (G3) Combustor

In addition to leakage control, a significant combustor hardware improvement was made to the new combustor which allowed for operation at A/F ratios as low as 12 (stoichiometric A/F = 9.6). The combustor cup, originally made of alloy steel onto which stainless-steel tubes were welded, was changed to an assembly on the new combustor made entirely of alloy steel, thereby eliminating a serious tube-plugging problem caused by the oxidation of stainless steel.

The combustor was tested at various levels of A/F ratios. The tests indicated a strong inverse relationship between efficiency and A/F ratio (shown in Figure 5). This trend is caused by an increase in the adiabatic flame temperature at lower A/F ratios, thereby increasing the potential for heat transfer into the heater head (since heater head temperature is maintained at a constant 700°C). A peak efficiency of 81% HHV (90% LHV) was obtained at an A/F ratio of 12, which was maintained as the minimum level of A/F ratio due to consideration for temperature limits of the combustor cup material.

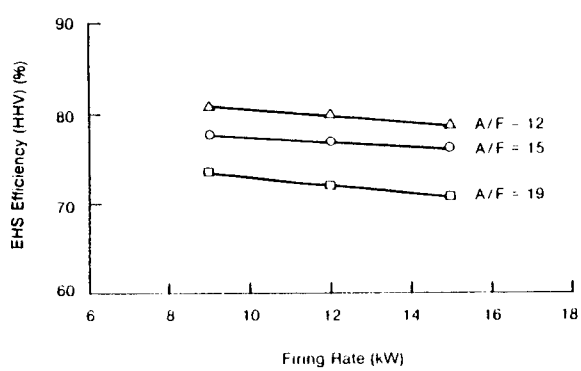


Figure 5 Upgraded Combustor Efficiency for for Air/Fuel Ratios

Both the original MTI combustor and the new combustor operated with natural gas at an available supply pressure of up to 25" H_2O. Although acceptable for laboratory purposes, a method needed to be determined for operating the engine on 3" H_2O - the average natural-gas supply pressure available for residential applications. The question of how to operate the external heat system on the available 3" H_2O natural-gas supply pressure is really a question of what engine auxiliaries are needed (schematically illustrated in Figure 6). The six concepts evaluated during this study are listed with their necessary engine auxiliaries.

Figure 6 summarizes the results of the concept selection study. The conclusions reached as a result of this study can be summarized below:

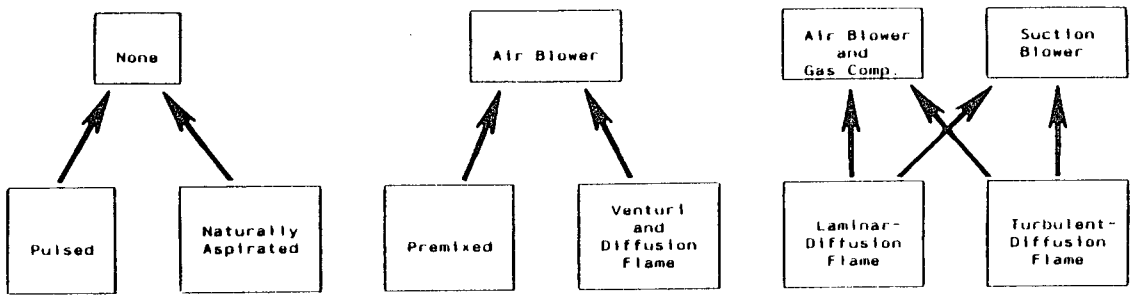
- A naturally aspirated combustion system is not feasible because of the relatively low flow rates generated by this type of system.
- A premixed combustion system is not feasible because of the high preheat air temperatures of the Stirling engine.
- The pulse-combustion system, while feasible, would need a large amount of development work.
- Of the remaining three options - a laminar-diffusion flame burner, a turbulent-diffusion flame burner with a venturi section - it is concluded that:
  - the parasitic losses saved with a laminar-diffusion flame burner would be on the order of 30 watts, and would not justify the extra development requirement; and,
  - the parasitic losses in a suction blower are about the same as the air blower; however, the fuel nozzle design is simplified.

Based on the above conclusions, it was decided that a low-pressure-drop, turbulent-diffusion flame burner of ~10" H_2O total system pressure drop using a suction blower would be developed for the residential FPSE external heat system. This approach has been detailed, and is currently undergoing fabrication for development testing during the early part of 1985.

Engine Development

One of the major technical objectives of this program is to increase the overall efficiency of the Stirling engine through development of the displacer drive system and engine thermodynamic/dynamic components. The approach used was to analyze power module design parameters in order to identify areas for achieving increased conversion efficiency and power output. Based on the analysis, component modifications were made to the 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.

One of the unique features of the EM is the use of a linear motor to control the motion of the displacer piston and, consequently, the engine power output. The initial performance characterization of the EM indicated an unacceptably high displacer motor input power requirement that is particularly acute at the



			ΔP ("H_2O)			
Natural	∿14	∿0.1				
External			8-25	20-35	∿4	8-25
Advantages	Eliminate parasitic power/aux equipment.	No parasitic or auxiliary equipment.	Simplified fuel-injection system.	Uses current combustor design.	Low Parasitic requirements.	Uses current combustor design; low development.
Disadvantages	Significant development required.	Insufficient flow.	Autoignition at desired preheat temp.	Moderate development required.	Increased volume; moderate development.	Hot exhaust fan required.
Decision	Dropped	Dropped	Dropped	Back-up candidate	Dropped	Prime candidate

Figure 6 Low-Pressure Natural-Gas Combustor Concept Alternatives

higher displacer strokes. Analysis suggested that this performance was due to a poor efficiency characteristic of the displacer motor, and/or a loading on the displacer motor, that significantly exceeded the load design point of 400 watts shaft power at full displacer stroke.

To evaluate this high power loss, a series of bench tests were conducted on the motor alone. MTI's Ling hydraulic-shaker test stand was used to characterize the motor, and its mechanical output was directly measured. This test data confirmed that an electrical-conversion motor deficiency did indeed exist. As further evidenced, Figure 7 shows that motor efficiency decreased sharply from its design point as its stroke was increased. Examination of the motor geometry indicated that the magnetic flux linkage path was being decreased to an unacceptably low level at large displacer strokes.

In the original displacer motor, the AC coil geometry and plunger magnet configuration were chosen for ease of fabrication; however, initial testing indicated that this straight-coil slot arrangement produced a flux pattern that decreased sharply with increasing stroke, limiting the generated voltage capacity of the motor. The modified stator/magnet assembly was fabricated to incorporate the features of longer magnets and different AC coil windings. These modifications were predicted to strengthen the flux pattern at high stroke. A comparison of the measured input power requirements for the initial and modified motor is shown in Figure 8. For each of these tests the engine was operated under similar conditions (i.e., stroke ratio, phase angle, temperature ratio and regenerator material). The results verify that the modified motor produced a significant reduction in motor input power.

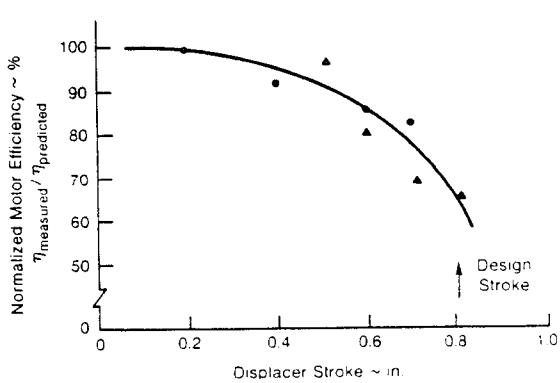


Figure 7 First-Generation Displacer Motor Performance

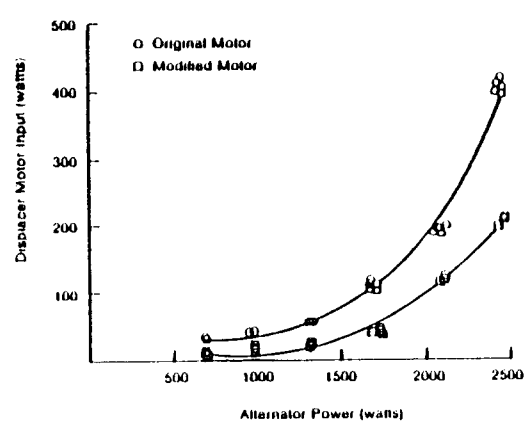


Figure 8 Power Amplification Comparison

At the beginning of this program, the EM had a measured level of efficiency that was below its design point. Thermodynamic analysis showed that the engine's level of efficiency could be improved significantly through thermodynamic component development. A further goal of this task was to establish improved performance through analysis and subsequent hardware modification and test. To this end, a matching analysis was performed. Subsequently, hardware modifications were made and tests conducted involving continuous correlation between test and analysis. Based on the matching analysis, several engine modifications were made to effect improved performance. Specifically, two major areas were addressed: 1) all critical seals were machined to the original design requirements; and, 2) a series of regenerator matrices were constructed to evaluate variation in porosity and wire geometry.

Figure 9 illustrates the effect of decreasing the leakage losses through reduction of the seal clearances to the original EM design specifications (it should be noted that the original EM builds included hardware that was not machined to the specified tolerances. As is evident from Figure 9, the performance improvement became more pronounced at higher power output due to the engine pressure wave (the driving force for seal leakage) becoming greater at larger levels of power output. Overall, ~3.5 points of improved efficiency were measured at the 3000-watt power point.

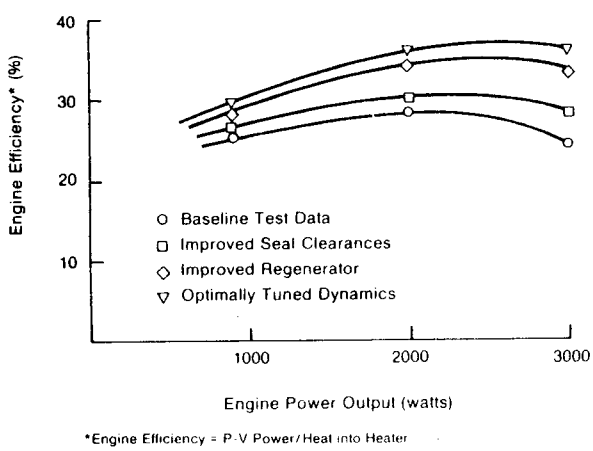


Figure 9 FPSE Efficiency Improvement

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 significant gains could be realized by changing to screen regenerators. 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 9, 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.

To evaluate the reliability potential of the EM, a 1000-hour endurance test was performed with an EM engine and a linear alternator load. A complete chronological history of the endurance test, indicating significant events during the test program, is given in Table 3.

In general, testing was smooth with few interruptions. Only two major unplanned shutdowns occurred during the course of the program - one caused by heater head instrumentation problems, the other by improper assembly of a magnet segment that eventually broke loose.

As can be derived from the dates given in Table 3, actual test hours accumulated on the Endurance Engine, as a percentage of the total hours available for test, increased dramatically throughout the program. This rapid increase in test hours, obtained as a function of time, was a result of the continued improvement in unattended test cell operation, e.g., the last 400 hours of duty-cycle testing were completed in only 432 hours, or 18 days. Further test plans call for a 9000-hour test of the EM to be conducted with internally supplied bearings.

Compressor Development

The compressor designed for integration with the EM consists of a novel configuration that employs diaphragms and a hydraulic transmission to hermetically isolate the engine and refrigerant working fluids. The design is detailed in several recent technical papers[1,2,3*]. The purpose of this paper is to highlight the development progress achieved. To that end, several accomplishments are noteworthy; specifically, advancements were made in compressor efficiency by reducing hydraulic fluid losses, and by reducing the blow-by of freon across the compressor ring seals.

Several transmission fluids with varying viscosities were employed in the compressor testing. The viscosity of fluids tested ranged from 10 centistokes to 300 centistokes, rated at 100°F. It was found that the 10-centistoke fluid resulted in a hydraulic loss reduction of as much as 50%.

The piston ring seals were improved to reduce the blow-by of freon by as much as 42%. Together with the gains achieved in hydraulic fluid losses, the overall compressor efficiency was improved by as much as 40%. Table 4 summarizes the improvements achieved.

Table 4
MTI FPSE Heat Pump Compressor
Development Progress

Parameter	January '84	June '84	Goal
Capacity Btu/hr	16,200	28,200	30,000-36,000
Leakage	33%	19%	5%
Hydraulic Loss	1635 watts	986* watts	906 watts
Efficiency	45%	63%	69%
Oil Management System	Manual	Automatic Open Loop	Automatic Closed Loop

*250-watt debit for counterweight assumed

*Superscript numbers designate references found at the end of the paper.

Table 3
Endurance Test Chronological History (April 8 to October 15, 1983)

Date	Event	Comments
4/8/83	Initiated 100-hr Low-Power Test	
4/15	Completed 100-hr Low-Power Test	Total 100 Hours
4/15-25	Planned Inspection	No Measurable Hardware
4/25-5/20	Limited Full-Stroke Test	Heater Head Instrument Problems
5/20-7/11	Heater Head Refurbishment	Heater Head Thermocouples Completely Replaced
7/13	Initiated 300-hr Full-Stroke Test	Minor Unattended Operation Mode Problems; Corrected During Test
8/3	Completed 300-hr Full-Stroke Test	Total 400 Hours
8/4	Engine Run Without Bearings	
8/5-15	Planned Inspection	No Measurable Hardware Changes
8/17	Initiated 600-hr Duty-Cycle Test	
8/29	Unscheduled Shutdown at Hour 210 of 600-hr Test	Small Magnet Section Separated, Jamming Displacer; Damaged Liner
8/29-9/14	Teardown	Engine Rebuilt with New Motor Liner; Repaired Displacer
9/14	Restarted Duty-Cycle Test	
10/2	Completed 600-hr Duty-Cycle Test	Total 1000 Hours
10/2-9	Continued Duty-Cycle Test an Additional 100 Hours	Total 1100 Hours
10/9-15	Planned Inspection	No Measurable Hardware Changes
10/15/83 - 2/19/84	Internal Bearing Development	Check Valve Optimization
2/13-15	Start/Stop Operation with Internal Bearings	Full Bearing Operation Achieved within 40 cycles
2/16	Planned Inspection	No Measurable Hardware Changes

The remaining compressor development involves implementing a vibration balancer and a hydraulic fluid management system into the compressor operation. Significant progress is being achieved in each of these two areas, as the compressor development work is scheduled to continue through the remainder of 1984.

Direction of Future Development

The work completed to date has been directed toward improving component performance on an individual basis. Future effort will involve evaluating and improving the performance of the entire power-conversion module. The program plan is to incorporate the best of the improvements achieved to date into an integrated gas-fueled Stirling-engine-driven compressor termed the Mark I, which will be a stand-alone, power-conversion device capable of operation with natural gas at typical residential supply pressure, and with automatic controls. The unit is envisioned to be tested with a calorimeter load to determine its operating envelope so that optimized refrigerant heat exchangers can be designed for subsequent integration of the Mark I into a heat pump system. The current schedule is to complete the fabrication of the Mark I, and begin characterization testing during 1985. Pending successful test results, field testing of prototype gas-fueled heat pump systems is scheduled for the 1987-1988 time frame.

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THE BATTELLE DUAL-CYCLE ABSORPTION HEAT PUMP

by W. T. Hanna, W. H. Wilkinson, D. A. Ball, and A. R. Maret

W. T. Hanna

Dr. Hanna's technical specialty is in the thermodynamics of industrial energy conservation. He has directed and participated in several studies of industrial processes and plants where the project goal was the more effective use of fuel supplies. He is currently directing a program aimed at demonstrating the capabilities of new and unique waste heat recovery devices for industrial and government sponsors. These efforts involve technical efforts designed to assist manufacturers in the commercialization of new products based upon Battelle developments. In addition to his background in thermodynamics, he also has a strong background and interest in fluid mechanics and heat transfer. He has participated in and managed research tasks in programs on heat exchanger development for advanced cogeneration systems, and on the analysis of heat transfer for advanced cogeneration systems, and on the analysis of heat transfer in arctic soils. In addition, he has conducted studies of dispersion of dense gases and vapors in the atmosphere.

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Bio not available.

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THE BATTELLE DUAL-CYCLE ABSORPTION HEAT PUMP

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Introduction

Rapidly increasing prices of natural gas provide an important and clear need for more energy-efficient gas-using equipment. The gas-fired heat pump offers the potential of a significant efficiency increase over conventional gas furnaces. Yet, despite intensive development efforts over the past 5 to 10 years, it has not been possible to successfully demonstrate a cost-effective unit for residential applications in the U.S.

Gas-fired heat pumps now under development are basically of two types: engine-driven and absorption. Engine-driven systems are often complex and expensive; significant technical advances appear to be necessary to reduce cost and improve reliability before successful commercialization will be possible. Absorption systems offer the promise of being simpler and more reliable than engine-driven units and on this basis, they represent the best opportunity for early commercialization of a cost-effective, reliable gas-fired heat pump.

Background

The Battelle Dual Cycle Absorption Heat Pump (BAHP) is based upon two patented concepts of a Battelle Senior Scientist, W. H. Wilkinson; one for the heat pump's cycle and the other for a key set of heat exchangers and their configuration in the cycle. In late 1983, the first phase of an overall five-phase development project was begun for the Gas Research Institute, Chicago. The overall project objective is to develop the heat pump as a commercially-available product for the 1988 heating season. In the current Phase I effort, the objective is to develop and test a breadboard system that can provide a cooling COP of 0.94 at 95 F (35 C), and heating COPs of 1.80 at 47 F (8.3 C) and 1.39 at 17 F (-8.3 C) (all COPs include 90 percent burner efficiency).

System Description

Overview

The BAHP system basically links two simple single-effect absorption heat pump cycles. Each has the expected four major heat exchangers of an absorption system: absorber, desorber (or generator), evaporator, and condenser. Thus, the BAHP has a total of eight major heat exchangers; however, four of these heat exchangers (the desorbers and condensers) are contained in a single modular component, the Absorption Power Module (APM). The evaporator and absorber heat exchangers in each cycle can be paired in common components so that all the heat exchangers of the system are contained in three components. These three components, along with the indoor and outdoor coils for hydronic/air heat exchange are shown schematically in Figure 1. This arrangement provides high performance with a modest number of components, actually requiring only one more component than in the simplest possible absorption heat pump.

The BAHP has three distinct operating modes:

- Air Conditioning (A/C)
- Warm Ambient Heat Pumping (WAHP)
- Cold Ambient Heat Pumping (CAHP)

Figure 1 shows the solution and "refrigerant" flow circuits connecting these major components. For simplicity, only the solution and refrigerant flows are shown. The hydronic fluid flow circuits are shown in a later figure.

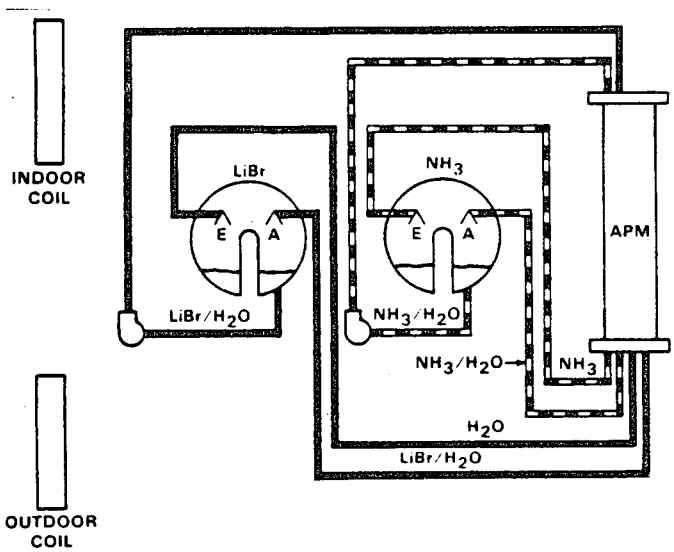

Figure 1. Battelle Absorption Heat Pump Solution Flows

Absorption Power Module

The key to the simplicity of manufactured components for the BAHP is the APM, which stages the heat input to both absorption cycles. It is intended to also allow for low manufacturing cost through the use of high performance heat exchangers and modular design. The APM, shown in a cutaway in Figure 2, provides primary heat input into the BAHP system from the burner. APM also provides for heat transfer from the high temperature cycle to the low temperature cycle, thus staging the heat input to the system in much the same manner as in a conventional, multiple-effect absorption heat pump.

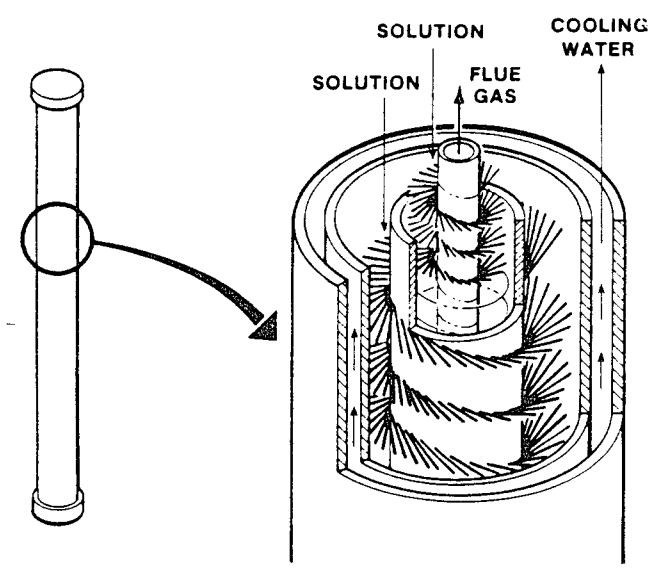

Figure 2. Absorption Power Module

The key to the compact APM design is a heat exchange surface for both desorber sections that allows for very effective heat and mass transfer. The desorber surface used in the APM is particularly well-suited to high heat transfer coefficients and to counterflow heat exchanger, both of which reduce the temperature difference needed for heat transfer. The heat exchanger surfaces are vertical tubes shown with fins on the outer tube surface, both to hold and to heat the falling film of solution. The smooth inner surfaces of these tubes are for primary heat input to LiBr/H_2O desorption (for the inside tube) and for water vapor condensation (on the second tube from the center). The third tube out from the center condenses ammonia on its inside surface and heats the hydronic fluid at its outer surface. The outer-most tube is a simple shell for hydronic fluid containment.

The cutaway diagram of Figure 2 shows the arrangement of concentric tubes, with the solution supply. Small baffles at the bottom of the APM provide separation of each solution from the H_2O or NH_3 condensate. The diagram indicates a tall, smaller diameter module, and our preliminary tests indicate that appropriate sizes for a 10,000 Btu/hr (2.9 kW) module range from a 4-inch (10.2 cm) overall diameter with 5-foot (1.5 m) height to a 5.5-inch (14.0 cm) diameter with 2-foot height. Given these overall dimensions, the pressure-containing chambers have an inner diameter of 1 to 2 inches (2.5 to 5.1 cm) and an outer diameter of 3 to 4 inches (7.6 to 10.2 cm). The finned tubing shown for the two desorber surfaces is a commercial product available in a range of materials and sizes, based upon standard pipe grades; it is one of several commercially available surfaces for these desorbers.

Solution Flow Loops

The solution flow circuits are the same for all modes of operation. Despite changes in operating mode, the solution and refrigerant flows are undirectional and the solution flow rates are steady. Solution concentration and refrigerant flow rates will be varied to meet load changes. Although the BAHP can be operated in an on-or-off mode, the use of modulating burners combined with the ability to heat only a preselected number of the APMs will allow for steady, efficient operation at partial loads, reducing cycling losses.

The operating ranges for both the salt and ammonia subsystems are shown in Figure 3. This figure shows the trapezoidally shaped equilibrium operating envelopes (labeled) for both working fluid pairs in terms of solution temperature and refrigerant saturation temperature and pressure. Each figure is similar to pressure-temperature-concentration diagram for a salt system. However, plotting these figures in terms of solution temperature and corresponding refrigerant saturation temperature permits showing temperatures in both subsystems on one figure. These figures clearly show that the highest temperature heat input is to the LiBr/H_2O desorber, and that the H_2O condenser always operates at a lower temperature to heat the NH_3/H_2O desorber; finally, the NH_3 condenser operates at a still lower temperature. The specific operating temperatures, for the absorbers and evaporators, are

determined by operating mode and ambient temperatures. The ammonia evaporator always operates at a lower temperature than the water evaporator, so that the water temperature is kept as far as possible from 32 F. Similarly, the salt system absorber is operated at a lower temperature than the ammonia system absorber to keep the salt system as far as possible from crystallization, even though both absorbers deliver heat to the same heat sink.

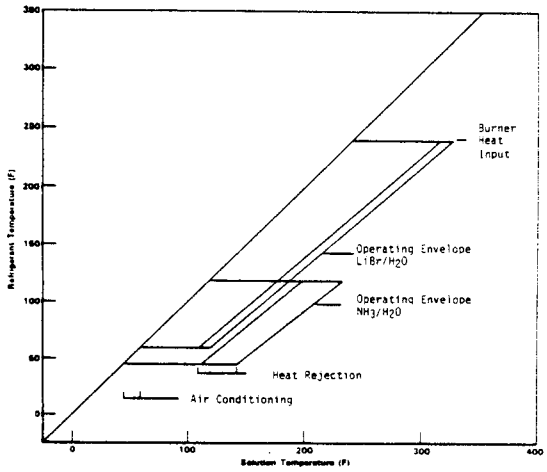

Figure 3. Refrigerant and Solution Temperatures for the BAHP in the Air Conditioning Mode

The salt system effectively buffers the ammonia system from the 65-100 Atm pressures that are usually seen in double effect ammonia systems. The use of ammonia for the low temperature heat input and high temperature output from the ammonia absorber buffers the salt solution from crystallization. Further assisting in these beneficial effects is the staging of the evaporators' heat input, so that the water evaporator provides the initial cooling of the residence's return air and the ammonia evaporator provides the final cooling of this air. This temperature staging also allows the salt system evaporator and absorber to operate at a higher pressure than would be possible without having the ammonia system, further reducing the potential for crystallization in the salt absorber.

Hydronic Flow Loops

The hydronic flow loop schematic diagrams are shown in Figure 4 for the A/C operating mode. Only the hydronic flows are shown in this figure. There are two three-way valves for connecting the indoor and outdoor coils to the BAHP. Also included are six single-acting valves for switching from WAHP to the CAHP mode. In this figure, the lowest temperature fluid circuit is shown by a heavy dashed line. The high temperature circuit always includes both absorbers and the ammonia condenser (in the APM). This circuit also includes the water evaporator in the cold ambient mode. In warm ambients, both evaporators are in the low temperature circuit; in the cold ambient mode, only the ammonia evaporator is in this circuit.

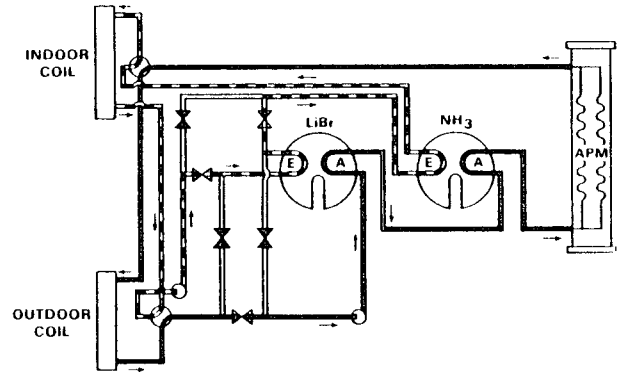

Figure 4. Battelle Absorption Heat Pump Hydronic Circuit Air Conditioning Mode

In the air conditioning mode, the evaporators draw heat from the dwelling via the indoor coil while the absorbers and the APM reject heat to the ambient. The hot hydronic fluid from the APM outlet can also be used for preheating domestic hot water, thereby reducing the heat losses in the A/C mode and increasing system efficiency.

The only changes required to change from A/C to WAHP are to cycle the three-way valves and thus interchange the two coil locations in the BAHP. In the CAHP mode, only the single acting valves are cycled.

Performance Predictions

Baseline System

The BAHP system has been analyzed in detail over a range of outdoor temperatures from 17 F (-8.3 C) to 105 F (41 C) to assess its performance potential.

A computer model was developed to perform mass and energy balances on the BAHP and its two subsystems, the high temperature system ($LiBr/H_2O$) and the low temperature system (NH_3/H_2O), in each of the three operating modes. The model is based upon the following assumptions:

- Some subcooling of the desorber inlet solution is allowed.
- All other state points are assumed to be in vapor-liquid equilibrium.
- An ideal rectifier (NH_3/H_2O cycle) is modeled yielding the minimum energy requirement for 100 percent water removal; the model results correlate well with the performance data for an ARKLA system with a real rectifier.
- An analyzer (NH_3/H_2O cycle) is used to bring the vapor entering the rectifier into equilibrium with the solution entering the analyzer/generator.

The model provides as output the energy flows through the system's heat exchangers (in Btu/lb of "refrigerant" vapor flow), given input of system pressures or temperatures and concentrations. Working fluid properties are taken from standard (usually ASHRAE) references.

The results of using this model to predict COP as a function of ambient temperature are shown in Figure 5. The results are shown for the baseline, denoted by the numeral "1" on the figure. The results for the following four potential improvements are also shown as curves 2 through 5.

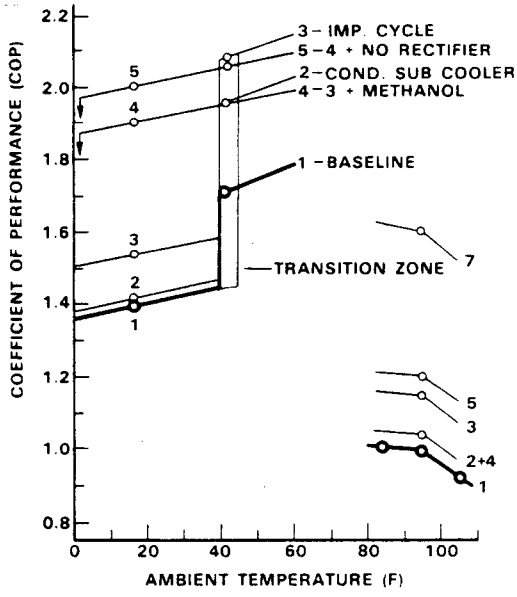

Figure 5. Heat Pump COP Versus Ambient Temperature

System Improvements Possible with the Proposed Fluids

Subcooling of the liquid refrigerants leaving the condensers has been identified as a means of increasing the CAHP performance by about 6 percent and the WAHP and A/C performance by about 17 percent. This merely involves increased recuperative heat transfer and, in advanced evaporator, absorber configurations based on the proprietary concepts, this surface has the potential of being conveniently and compactly included. This improvement is shown by the curve number 2.

The next system improvement involves a more complex low temperature cycle which includes modest amounts of recuperative absorption, desorption, and rectification and which has the potential for:

(1) Increasing the low temperature system's COP by 15-20 percent.
(2) Increasing the recovery factor from the high temperature cycle by about 10 percent by further subcooling.

This involves both some additional cycle complexity and increases the peak cycle temperature by at least 50 F. Advanced proprietary components have the potential for making these additions cost-effective, while the increased temperatures are still within potentially acceptable materials limits. The improvements are significant; a total of 29 percent improvement in CAHP thermodynamic performance and a total of 29 percent improvement in WAHP and A/C thermodynamic performance. These improvements are shown by curve number 3.

Implications of Other Known Fluids

A unique characteristic of this cycle is that neither "fluid" pair is required to tolerate the full thermodynamic extremes of the systems. Consequently, working fluids can be optimized for narrow ranges of operation within the overall cycle which still gives high seasonal performance.

One improvement alternative involves the replacement of the $LiBr/H_2O$ working fluid with fluids whose minimum evaporator temperatures are significantly reduced. The objective of this refrigerant switch is to lower the ambient temperature for the WAHP/CAHP transition. We expect the fluids such as $CH_3OH/LiBr$ to have the potential for extending the WAHP/CAHP transition from about 40 F (4.4 C) to about 15 F (-9.4 C) and that they may be practically cost-effective. This would be true even if WAHP and A/C performance were degraded by a few percent due to the change to the alcohol refrigerant. High temperature stability needs to be confirmed, along with the ability to control alcolate formation in the low pressure portion of the cycle. Changing the 17 F (-8.3 C) performance from a CAHP value of 1.5 to a WAHP value of about 1.9 by this improvement seems very attractive. This effect is shown by curve number 4.

Changing the low temperature pair from H_2O/NH_3 to a nonrectified pair or a ternary set can provide additional benefits by reducing system complexity and improving performance. On the order of 20 percent improvement in the simple cycle performance has been estimated merely by changing the sorbent in an ammonia system from water to salt such as sodium thiocyanate. The results of such fluid changes could lead either to reduced system complexity and moderately improved performance (CAHP @ 1.5, WAHP @ 1.9, and A/C @ 0.9), or to alternative improvements with a modest increase in system complexity and higher performance (CAHP @ 1.6, WAHP @ 2.0, and A/C @ 1.0). These results are shown by curve number 5.

Results of Testing

The first stage of testing has been completed. A breadboard of the $LiBr/H_2O$ system has been assembled, shaken down, and tested. The APM has been operated at an input rating of 10,000 Btu/hr (2.9 kW) and the breadboard has attained its COP goals in the A/C mode. Detailed performance data will be presented at the conference, but a performance summary is given here.

The primary test mode was as an air conditioner where the system COP goal is 0.94 at 95 F, including a 10 percent stack loss. This COP requires that the High Temperature Cycle (the "salt" system) have a COP of 0.66. In actual tests, the measured COP averaged 0.63 and ranged up to 0.69, indicating that COP goals could be achieved. Construction of the Low Temperature Cycle breadboard is now under way with testing planned for October.

KOOL-FIRE - WHAT IT DOES AND HOW IT WORKS

by James B. Sears

James B. Sears

Bio not available.

Paper not available.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

FACTORS AFFECTING SEASONAL PERFORMANCE OF HEAT PUMPS IN CANADA

by U. Gino Di Rezze

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Employed by Canadian General Electric, assignment included National Service Manager for heating and air conditioning products. Presently co-owner and Vice President of a well established (50 year) firm in the mechanical contracting field.

Presented at the 7th Heat Pump Technology Conference
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FACTORS AFFECTING SEASONAL PERFORMANCE OF HEAT PUMPS IN CANADA

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ABSTRACT

Factors influencing the performance of the residential air source heat pump in Canada were studied. Six cities were chosen to represent a cross section of climates, fuel costs, fuel availability and population centres. The same heat pump and model residence were used for each of the cities. A computer model was used to vary pertinent factors to generate comparative data and output in a form useful for analysis.

The energy and economic performances were studied in detail. The air source heat pump performed better in cities with warmer climates, higher alternative fuel costs, and where few alternative fuels were available. The ground source heat pump is introduced and shown as a strong alternative.

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1. INTRODUCTION

The purpose of this study is to acquire data to analyse the performance of the air source heat pump in the Canadian climate. The method used was a model residence and a computer to generate performance data. The study included (a) performance in different climatic conditions, (b) comparisons with alternative heating systems, (c) effects of fuel costs and availability and (d) installation and operating characteristics. Since the heat pump is able to heat and cool, paybacks and return on investments were examined in cases where the cooling features were valued or discounted. The energy performance was studied separately from the economic performance. The energy performance is influenced by technology and climate, both of which are relatively stable. The economic performance is also investigated as a (potential) viable alternative heating system for cold climates.

Since the cooling hours in Canada are typically less than 8% of the heating hours, cooling is studied to a lesser extent.

The local climate has a bearing on the seasonal performance factor, so that the same heat pump will have a different seasonal performance in a different climate, (table 3).

A model residence was chosen which can be described as 2000 square feet, brick veneer frame, 2 stories, with insulation value of R20 on walls and R30 in attic. The home in Toronto has design losses of 50,000 BTUH heating (ΔT of 71°F) and 24,000 BTUH cooling. Theoretically, this same home along with a family having the same living habits is placed in 5 other cities.

Halifax, Montreal, Toronto, Winnipeg, Calgary and Victoria. Under different climatic conditions not only will the home experience different heat losses and heat gains, but the heat pump will also perform differently.

The cooling load varies from 1 ton to 2 tons in four (4) of the cities and there is no need for cooling requirements in the coastal cities of Halifax and Victoria. The cost difference of systems below 2 tons is negligable. Since the cooling requirement is small compared to the heating requirement and to provide consistancy in the study, a 2 ton General Electric/Trane BWR724 heat pump was used in all six (6) cities.

The General Electric/Trane computer and software related to heating and air conditioning was used. The computer contained climatic data for the six (6) Canadian cities, data pertaining to the heat pump performance, and different application configurations. The computer was also used to perform accounting calculation for the payback analysis.

To properly evaluate the air to air heat pump as a viable heating system, the energy performance and the economic performance were studied. The energy performance will change only with improvements in the components, climatic changes and changes in living

habits. However, changes in alternative energy costs compared to hydro is more likely to change yearly which in turn change the economic performance of the heat pump.

2. ENERGY PERFORMANCE

CLIMATE:

The weather data used to study the performance of the heat pump in the six (6) Canadian cities was taken from the General Electric/Trane computer data bank which, in turn, used the Canadian Weather Bureau as a source. Figure 1 shows the yearly frequency distribution of outdoor temperatures for Halifax, Montreal, Toronto, Winnipeg, Calgary and Victoria, based on average temperature over a 10 year period.

Climate for the cities studied can be grouped as zones. One being the east and west coasts (Halilfax and Victoria) and can be generalized as being moderate with few extremes. Both have a daily temperature range of 16°F and have the highest heating hours. The majority of these hours occur above 20°F. The degree hours which are the better measure of the temperature show that Halifax and Victoria are among the warmest in the country, (table 2). The central or prairie cities, Winnipeg and Calgary, have a relatively flat distribution of outdoor temperatures and a higher percentage of colder temperatures (Fig. 1). The Eastern cities, Montreal and Toronto, are the intermediate of the two extremes.

The heat output of the heat pump varies with outside temperature. The decreasing outdoor dry bulb temperature will reduce the outdoor coil evaporator temperatures and pressure. The compressor operates at constant speed therefore, the flow rate is also constant. The decreased refrigerant density at the suction decreases the mass flow rate through the compressor. The capacity of the heat pump to transfer energy depends directly on the mass flow rate.

The energy consumption is closely related to the climate as measured by degree hours. Energy used to heat the model residence in Victoria (Pacific Coast) was one half, of that in Winnipeg (Central Canada) and the degree hours were also nearly half.

The climate also effects the amount of defrosting required over the season. There is a definite relationship between relative humidity, outdoor temperature and frequency of defrost. Note that with high relative humidity, and with outdoor temperature approaching 32°F that the defrost frequency increases. The histogram of temperatures figure shows that Halifax, Victoria and Calgary have the highest number of hours between 17°F and 42°F and Toronto, Montreal and Winnipeg have the lowest in that order. The energy used during defrost represents from 3 to 12 percent of annual energy purchased.[5]

The temperature distribution can have an affect as significant as degree hours on the overall performance of the heat pump. If the distribution curve has a higher percentage of hours above the balance point, where auxiliary heat is not required, a better seasonal performance is expected. For example, although Halifax experiences the third lowest degree hours, it has the second lowest percent hours below thermal balance point. This results in the second lowest energy consumption (Table 3).

Warmer outdoor temperatures are generaly favourable for heat pump systems, apart from a potential negative effect of short cycling during mild weather can have an energy degradation affect ranging from 4% for a properly sized heat pump, to as much as 10% for an oversized system.[5] The overall seasonal performance factor (SPF) of a heat pump which includes capacity, defrost, auxiliary heating and cycle degradation, favours more moderate climate conditions.

INSTALLATION FACTORS:

The energy performance of the heat pump also depends on factors related to installation.

The capacity, or the oversizing and undersizing of the heat pump, affects the thermal balance point temperature and the amount of auxiliary heat used. As the capacity is decreased (undersizing the system), the thermal balance point temperature is increased, thus the consumption of lower efficient auxiliary heaters also increases. The nominal 2 ton system is properly sized for Toronto. When the size was reduced to 1½ tons, the balance point rose from 29°F to 35°F and the use of auxiliary heating also increased by 89%, from 26% to 49% of total energy requirement. The seasonal performance factor dropped by 20%.

As the capacity is increased (oversizing the system), the thermal balance point temperature is reduced, thus, the use of auxiliary heaters compared to overall system also is reduced. When oversized by ½ ton (to 2½ tons) the balance point temperature dropped by 9°F. With a 2½ ton unit, 83% of the heating was done by the heat pump, and the SPF increased by 10%. The thermal balance point temperature for a 3 ton system was reduced to 15°F and did not reduce further with an increase to 3½ tons. The seasonal performance factor reached a peak at 2½ tons and declined with further increase in size. This is due to primarily cycling degradation losses.[10]

"Oversizing a conventional air source heat pump in the traditional sense (i.e., sizing to larger than the cooling load) offers little performance advantage in terms of purchased energy savings."[6] This was the conclusion in a study conducted by the National Research Council in Eastern Canada when one of the two heat pumps was oversized by 90%.

SYSTEM CONFIGURATION:

The energy consumption of an add-on and an all electric were compared by running the two programs with the same variables. The all electric system with its ability to stage the auxiliary heat as needed and working simultaneously with the heat pump, is the more energy efficient of the two. Energy consumption was reduced by as much as 11.3% and 10.9% in Toronto and Montreal respectively. This is expected since the climate required substantial auxiliary heat below the balance point (Table 4). The simultaneous use of the heat pump and the staging comparison showed that Victoria, with little need for auxiliary heating, is the least affected by the configuration, showing less than 1% improvement with the

all electric. Halifax, Calgary and Winnipeg also show reduced energy use with the all electric with 8.7%, 6.5% and 6.4% improvements respectively.[10]

Another energy performance feature is the defrost control method. There are basically two types, the demand defrost and the timed defrost. The demand defrost will initiate defrost only as required and as often as required. When the outdoor temperature drops below 42°F, the timed defrost will initiate defrost (on a pre-set interval) whether or not it is needed. A study conducted by the National Research Council in an Ottawa home where the climate is intermediate for Canada shows that the demand defrost has fewer defrosts yearly. The demand defrost uses up 3% of annual energy consumption while the timed defrost uses up to 12%. Defrosting frequency is a function of outdoor temperature and relative humidity (Table 2) . The closer the temperature is to 32°F and the higher the humidity level, the greater the defrost requirement.[10]

Table 2 has a summary of average winter temperatures, humidity and precipitation. On a timed defrosting system it is assumed that Winnipeg and Calgary will have the most defrost cycles, since the running heating hours are the highest at 4726 and 4636 respectively with Victoria having the fewest since the running hours are less than half at 2290. Toronto, Montreal and Halifax are approximately the same. However, humidity and precipitation will prolong the defrost even on a timed defrost so that Victoria and Halifax have the highest. The demand defrost is even less predictable on a quantitive level since it is not running hours that initiates it, it is a combination of below 42°F sustained temperature, high humidity and precipitation.

THERMOSTAT SETTING:

The model residence was studied when the indoor thermostat was varied from 60°F to 75°F using an all electric heat pump in Toronto. The degree heating hours almost doubled from 121,785 to 221,261 and the energy consumption increased by 95% to 17,785 KWH.

The thermal balance point temperature increased from 20°F to 35°F, when the setting was changed from 60°F to 75°F. This means that at the higher thermostat settings the auxiliary source contributes to a higher percentage of the total, thus, reducing the overall performance. The auxiliary heating requirement jumped from 18% at 60°F to 31% at 75°F.[10] The 15°F increased trippled the auxiliary energy used from 1669 KW to 5527 KW and the SPF was reduced by 50%.[15]

The thermostat set-back setting during sleeping and working hours, does reduce energy consumption.

3. ECONOMIC PERFORMANCE

INSTALLATION COSTS:

Installation costs are best summarized on table 7. However, it is interesting to note that the heat pump can cost $1600 more than an air conditioner.

OPERATING COSTS:

The climate, the fuel costs, and the efficiency as well as the controls of the heat pump influence the operating costs. Although capacity and living habits also affect cost, they were assumed to be the same in all cases.

It was found that in the model residence, heated with an oil furnace in the city of Winnipeg, (Central Canada) is the most expensive to heat per year. With the exception of Halifax, oil heating is the most expensive fuel. Electric heating is $1531 in Halifax (the second most expensive to heat the model residence in Canada), (Table 9).

The least expensive heating in Canada on a yearly basis (excluding the ground source heat pump) is in Calgary at $359 when using a high efficiency gas furnace. Based on degree hours and yearly BTU's of heat used, Calgary is the second coldest city, however, the natural gas price is so inexpensive that a high efficiency furnace still makes it the lowest bill in Canada.

Heat pump heating costs are very attractive in Halifax and Victoria where the weather is moderate and the gas is not available.

Generally the heat pump system is an efficient heating system surpassed only where gas is available, using a high efficient gas furnace. In Calgary where gas is so inexpensive, the degree hours below the balance point are high, any gas furnace will match or exceed the performance of the heat pump.[10]

PAYBACK ANALYSIS:

Payback, return on investment and cumulative cash flow, were calculated based on the possibility that $0 value was given for the cooling portion of a $3200 heat pump or $1600 value was given for the cooling portion of the same heat pump, (Table 6). The coastal cities, Victoria and Halifax, do not require cooling. From the six (6) cities studied, Toronto has the largest demand for cooling at 8% of the heating hours. The shortest payback (4.41 years) justifying full $3200 value of the heat pump ($0 cooling allowance) was in Halifax adding on to an electric furnace and 2.6 years in Montreal and Winnipeg on an oil furnace (Table 6). With the longest payback in Winnipeg when added on to a gas furnace, with return on investment ranging from 0 to 45%.[10]

4. THE GROUND SOURCE HEAT PUMP

The air to air heat pump in Canada is a viable alternative heating system and performance would be significantly better if Canada's climatic conditions were more moderate. The system that is only slightly affected by the climatic conditions is the ground source heat pump. This system will extract the heat from the ground rather than the air. In Canada the temperature below the earth's surface is more constant and predictable than above the surface. In populated areas these temperatures are between 40°-48°F.

The author has installed 40 such systems in the last 10 months in the Toronto area. There are over 300 installed in Southern Ontario that have gone through a number of heating seasons. The Ontario Hydro is now monitoring the performance of some of these systems.

PERFORMANCE:

Recent monitoring by Ontario Hydro shows that the performance of the ground source heat pump can be more than twice that of the air source heat pump with the seasonal efficiency of almost 3.0 Although many systems have been installed in Canada since 1980, few formal studies have been conducted.

The cost of the installations is $3000 more than an all electric heat pump and $3300 more tha high efficient gas furnace with air conditioner.

The major disadvantage of this ground source heat pump is obviously the installation cost and the inconvenience to an existing home with a landscaped lot.

The advantages are numerous. The return on investment can be as high as 20 % when based on future energy increases of 12% per annum over a 10 year period. The return on investment is reasonable at less than 5 years when compared to gas and closer to 3 yearspayback when compared to oil or electric furnace with air conditioner, (Table 5,8)

In all areas of Canada the ground source heat pump is the least expensive of all other heating systems except for the energy rich city of Calgary. Since it has such an inexpensive gas that the mid and high efficiency gas furnace is slightly cheaper in operating cost.

The features of the system are to heat and cool as well as heat domestic hot water and/or swimming pool. The payback and return on investment will even improve, in future since the installation costs are decreasing.

The long life expectancy (20 years) is predictable since the ground source is sheltered from the elements and operated under constant conditions with little temperature fluctuations.

In many retrofit installations, 100 A service panel is adequate since the constant high output and the minor electrical backup (10 KW) will supply needs of the average home. One utility's monitoring shows that the peak demand was reduced to 40% of that used by an electric furnace or all electric air source heat pump. Noise pollution is not a concern when compared to an air conditioner or air source heat pump. The unit has no outdoor moving parts.

5. CONCLUSIONS

Energy Performance of the air source in influenced by climate, installation, system type and lifestyle or thermostat setting.

Areas with few degree hours and fewer hours below the thermal balance point show lower energy consumption for the same model residence. Compared to traditional sizing based on cooling, oversizing slightly is beneficial. Further oversizing to one ton and over, increases cycling losses and shows a trend of "diminishing return" on the SPF. Airflow reduction decreases the SPF but more importantly, poor airflow has a detremental affect on the life of the equipment.

The all electric system is the most efficient of the configurations studied since the auximliary heat can be staged and works simultaneously with the heat pump. Compared to an add-on gas, oil or electric system where more auxiliary heat is used. Thus, a saving of 11.3% in Toronto and 1% in Victoria.

Demand defrost is generally preferred over the timed defrost for the Canadian climate.

Areas with higher average winter temperature and lower degree days benefit most from night and mid-day set back. The model residence in Toronto with a thermostat setting 76°F uses nearly twice the energy of a setting of 60°F. Sizable dramatic results can be shown for other Canadian cities.

Economic Performance of the heat pump is mainly affected by the climate, fuel costs and fuel availability.

Where natural gas is available the high efficiency gas furnace is a slightly better economic performer than the air source heat pump. In Calgary (the prairies) which has the cheapest gas in the country, any gas system is more economical than the air source heat pump.

When compared to electric or oil heating, the air source heat pump is the most economical especially in the moderate climatic regions.

Paybacks were calculated based on the two possibilities. Where the cooling feature of the air source is of no ($0) value the payback on a $3200 heat pump investment was less than 6 years (except in Calgary). When the cooling feature was needed the excess system valued at $1600, the payback was reduced to approximatley 3 years.

The ground source initial capital cost may discourage some homeowners to purchase them even if the payback is less than 5 years. However, the drilling and trenching costs are coming down as volume increases.

The seasonal performance of the ground source heat pump is nearly 2 time better than the air source.

Except for Calgary, the ground source is the cheapest heating system in Canada. Unlike the air source that suffers performance in the colder parts of populated Canada, the ground source heat pump shows better dollar savings in areas where the heating season is the longest.

The long life expectancy, the constant output even in adverse climatic conditions, the reduced dependance on auxiliary, the ability to heat domestic hot water and swimming pools, gives the ground source an optimistic future in the Canadian market.[10]

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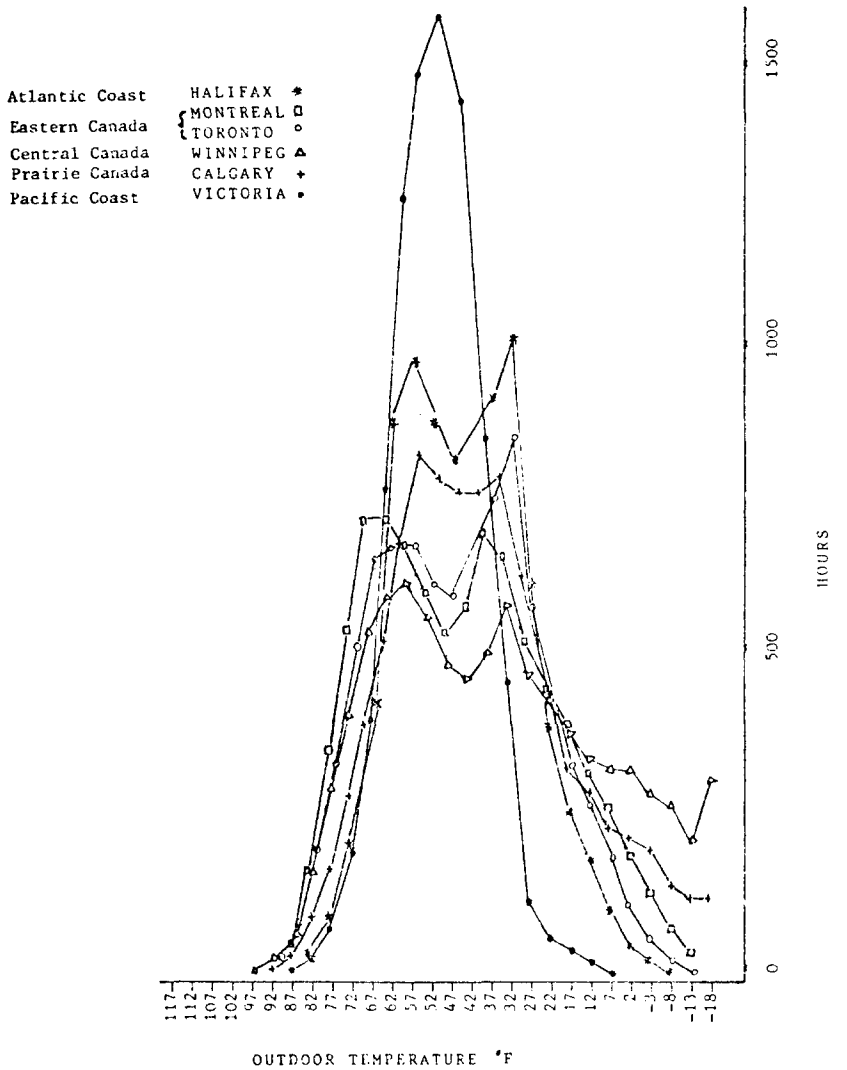
FIG. 1 HISTOGRAM CURVES FOR ALL 6 CITIES



(REF. 10)

TABLE 1 ENERGY COSTS IN CANADA

CITY STUDIED	PROVINCE	OIL $/ LITERS	OIL $/ U.S.GAL.	OIL $/ IMP GAL.	HYDRO $/ KWH	GAS $/ 10 CM	GAS $/ THERM
	NEWFOUNDLAND	.346	1.31	1.57	.0547	N/A	N/A
	P E I	.332	1.26	1.51	.1046	N/A	N/A
HALIFAX	NOVA SCOTIA	.330	1.25	1.50	.0646	N/A	N/A
	NEW BRUNSWICK	.330	1.25	1.50	.0551	2.94	1.038
MONTREAL	QUEBEC	.329	1.25	1.50	.0404	1.95	0.689
TORONTO	ONTARIO	.323	1.22	1.47	.0427	1.81	0.6390
WINNIPEG	MANITOBA	.330	1.25	1.50	.0337	1.58	0.5579
	SASKATCHEWAN	.319	1.21	1.45	.0410	1.16	0.4096
CALGARY	ALBERTA	.270	1.05	1.23	.0449	0.95	0.3350
VICTORIA	BR. COLUMBIA	.324	1.22	1.47	.0473	1.34	0.4732
	NAT'L AVERAGE	.323	1.24	1.49	.0440	1.47	0.5177

SOURCE: VERBAL CONTACT WITH
ENERGY, MINES AND RESOURCES CANADA
OCTOBER 4, 1983.

(REF. 10)

TABLE: 2 WEATHER DATA FOR DEFROST ANALYSIS

CITY	DEGREE HOURS TOTAL HOURS	DEGREE HOURS BELOW 42°F HOURS	DEGREE HOURS BELOW 42°F %	PERCIPITATION INCHES	RELATIVE HUMIDY %	AVERAGE WINTER TEMP(°F)	HEATING HOURS BELOW 42°F
HALIFAX	190177	142170	75	41	79	33.6	4105
MONTREAL	198033	156317	83	24	71	32.7	4087
TORONTO	184663	148857	81	19	75	35.5	4005
WINNIPEG	259548	229573	88	10	74	23.1	4726
CALGARY	234416	191038	82	7	64	28.7	4636
VICTORIA	138650	88776	64	29	82	44.2	2290

SOURCE: ENVIROMENT CANADA AND
COMPUTER RUNS

(REF. 10)

TABLE 3 HEATING DESIGN CONDITIONS FOR VARIOUS CITIES VS SEASONAL PERFORMANCE

CITY	HEAT LOSS (BTUH)	DEGREE HOURS (F)	DESIGN TEMP. OUTDOOR (F)	DESIGN TEMP. ΔT (F)	THERMAL BALANCE POINT(F)	S P F	HEAT PUMP SYSTEM CONSUMPTION HEAT PUMP (KW)	HEAT PUMP SYSTEM CONSUMPTION AUXILIARY (KW)	HEAT PUMP SYSTEM CONSUMPTION TOTAL (KW)	ELECTRIC FURNACE (KW)
HALIFAX	45800	190177	+5	65	25	1.778	11246	2000	13246	23557
MONTREAL	56300	198033	-10	80	30	1.547	10780	6637	17417	26950
TORONTO	50000	184663	-1	71	30	1.671	10630	3728	14358	24005
WINNIPEG	68300	259548	-27	97	25	1.334	10833	12431	23264	31057
CALGARY	65500	234416	-23	93	25	1.512	11804	7390	19194	29024
VICTORIA	33100	138650	+23	47	25	1.912	8199	29	8228	15735

CALCULATIONS BASED ON: HEATING AND COOLING INDOOR DESIGN 70 F, 75 F RESPECTIVELY

(REF. 10)

TABLE: 4 WEATHER RELATED DATA FOR VARIOUS CITIES

CITY	HEAT LOSS MBTUH	HEAT GAIN MBTUH	DEGREE HOURS		DEGREE DAYS	SEASONAL HOURS		TEMPERATURE DAILY F		DESIGN TEMP (F)		BALANCE POINTS (F)			
			TOTAL	BELOW TBP %		HEATING	COOLING	RANGE	AVG. * WINTER	HEATING	COOLING	ECONOMIC			THERMAL
												OIL	GAS	FLE	
HALIFAX	45.8	-	190177	39.1	7924	8034	-	16	33.6	+5	-	+2	-	+2	+27
MONTREAL	56.3	20.0	198033	64.9	8251	6916	392	18	32.7	-10	85	-14	-3	-3	+30
TORONTO	50.0	24.0	184603	50.0	7692	6911	494	22	35.5	-1	87	-7	+2	0	+28
WINNIPEG	68.3	22.0	259548	72.4	10815	7251	381	23	23.1	-27	86	-17	-4	0	+26
CALGARY	65.5	12.0	234416	57.9	9767	7842	147	26	28.7	-23	81	-6	+46	0	+27
VICTORIA	33.1	-	138650	1.68	5777	8060	-	16	44.2	+23	-	-3	-	-3	+24

THE ABOVE DATA WAS COMPILED FROM VARIOUS COMPUTER RUNS. ALL CALCULATIONS ARE BASED ON INDOOR DESIGN OF 70 F FOR HEATING AND 75 F FOR COOLING.

*AVERAGE WINTER TEMPERATURES FROM OCTOBER TO MAY FOR THE PERIOD 1941-1970 READINGS TAKEN AT THE RESPECTIVE AIRPORTS -source ENVIROMENT CANADA

(REF. 10)

TABLE: 5

PAYBACK SUMMARY: GROUND SOURCE VS STANDARD ALTERNATIVE SYSTEMS

	ALTERNATE		GROUND SOURCE		SAVINGS $ 1st YEAR	INITIAL COST DIFF. $	PAYBACK		ANALYSIS CUM. CASH FLOW
	ENERGY	COST $	ENERGY	COST $			YEARS	RETURN ON INV.	
1. GAS FURNACE (STD) + AIR CONDITIONING + DOMESTIC HOT WATER	2031 (THERMS)	1300	10324 (KWH)	444	856	4400	4.2	24.9	10620
2. ELECTRIC FURNACE + AIR CONDITIONING + DOMESTIC HOT WATER	28656 (KWH)	1232	10324 (KWH)	444	788	4600	4.7	21.6	9233
3. OIL FURNACE + AIR CONDITIONING + DOMESTIC HOT WATER	1201 (US GAL)	1465	10324 (KWH)	444	1021	4600	3.8	28.7	13322

* This calculation is done for the Toronto Model Residence

The ground source has cooling and domestic hot water heating features as well as heating. To properly compare to alternative systems initial cost includes cooling and assumption for hot water consumption is $200.

* DATA TAKEN FROM PAYBACK PRINTOUT

(REF. 10)

TABLE 6 PAYBACK SUMMARY FOR ADDON HEAT PUMPS

CITY	PAYBACK IN YEARS WHEN ADDING A HEAT PUMP					
	ASSUME VALUE OF COOLING PORTION OF HEAT PUMP TO BE $ 0			ASSUME VALUE OF COOLING PORTION OF HEAT PUMP TO BE $ 1600		
	OIL	GAS	ELECTRIC	OIL	GAS	ELECTRIC
HALIFAX	5.9	NOT AVAILABLE	4.4	A/C NOT REQUIRED	NOT AVAILABLE	A/C NOT REQUIRED
MONTREAL	4.6	7.4	7.2	2.6	4.5	4.3
TORONTO	4.9	8.9	6.6	2.8	5.5	3.9
WINNIPEG	4.6	9.5	8.8	2.6	5.9	5.4
CALGARY	5.4	NO PAYBACK	6.0	3.1	NO PAYBACK	4.3
VICTORIA	5.4	NOT AVAILABLE	6.5	A/C NOT REQUIRED	NOT AVAILABLE	A/C NOT REQUIRED

1. DATA SUMMARY FOR TABLE 10 EXTRACTED FROM TABLES 7,8, and 9.

(REF. 10)

TABLE: 7 HEATING SYSTEMS' INSTALLATION COSTS AND EFFICIENCIES

SYSTEM NUMBER	HEATING SYSTEM	SEASONAL PERFORMANCE (AVERAGE)	OUTPUT CAPACITY (BTUH)	INSTALLATION COST ($) IN TORONTO
1	GAS FURNACE(STANDARD EFFICIENCY) (chimney liner worth $350 not incl.)	60	84000	1800
2	GAS FURNACE(MEDIUM EFFICIENCY) (chimney liner worth $350 not incl.)	75	71000	2100
3	GAS FURNACE(HIGH EFFICIENCY) (chimney liner not required)	95	76000	2900
4	OIL FURNACE(STANDARD EFFICIENCY) (oil tank worth $400 not incl.)	55	96000	1600
5	ELECTRIC FURNACE(STANDARD EFFICIENCY) (200A service worth $1100 not incl.)	100	51000	1600
6	HEAT PUMP + GAS FURNACE (STANDARD EFF)	160-170	2tons 84000	5000
7	HEAT PUMP + OIL FURNACE (STANDARD EFF) (oil tank worth $400 not incl.)	160-170	2tons 95000	4800
8	HEAT PUMP + ELECTRIC FURNACE (STD EFF) (200A service worth $1100 not included)	170	2tons 51000	4800
9	HEAT PUMP -ONLY (add-on)	170	2tons	3200
10	AIR CONDITIONER + GAS FURNACE (STD EFF) (chimney liner worth $350 not incl.)	60	2tons 84000	3400
11	AIR CONDITIONER + GAS FURNACE (MEDIUM EFF) (chimney liner worth $350 not incl.)	75	24000 7100	3700
12	AIR CONDITIONER + GAS FURNACE (HIGH EFF) (chimney liner not required)	95	2400 7600	4500
13	AIR CONDITIONER + OIL FURNACE (STD EFF) (oil tank worth $400 not included)	55	2400 9600	3200
14	AIR CONDITIONER + ELECTRIC FURNACE(STD EFF) (200A service worth $1100 not incl.)	100	2400 5100	3200
15	AIR CONDITIONER - ONLY (add-on)	-	2400	1600
16	GROUND SOURCE HEAT PUMP	290-300	2½ tons 34000	7800

STANDARD EFFICIENCY GAS FURNACE HAS RATED BTUH: INPUT 105000 ,OUTPUT 84000
MEDIUM EFFICIENCY GAS FURNACE HAS RATED BTUH: INPUT 95000 ,OUTPUT 71000
HIGH EFFICIENCY GAS FURNACE HAS RATED BTUH: INPUT 80000 ,OUTPUT 76000

STANDARD EFFICIENCY OIL FURNACE HAS RATED BTUH: INPUT 120000 ,OUTPUT 96000
ELECTRIC FURNACE HAS RATED BTUH: INPUT/OUTPUT 15KW or 51000BTUH

TABLE: 8 PAYBACK ANALYSIS OF: GROUND SOURCE HEAT PUMP VS ALTERNATIVES

Cooling Size-2 Tons
Furnace Size-15 KW
-50,000 BTUH

SYSTEMS	SYSTEM		COMPARED TO GROUND SOURCE			PAYBACK ANALYSIS		
	ENERGY CONSUMED	FIRST YEAR ENERGY COST $	FIRST YEAR SAVINGS $	FIRST YEAR SAVINGS %	INITIAL COST DIFFERENCES $	YEARS TO PAY BACK DIFFERENCE	RETURN ON INVEST-MENT. %	ACCUMULATED CASH FLOW IN 10 YRS. AFTER PAYING COST DIFFERENCE TAX FREE SAVINGS. $
OIL FURNACE (gallons) +AIR CONDITIONING (KWH) +HOT WATER HEATER (KWH)	500 1,588 5,986	1061	728	69	4450	4.85	20.5	8,328
ELECTRIC FURNACE +AIR CONDITIONING (KWH) +HOT WATER HEATER (KWH)	13,587 1,588 5,986	910	577	63	3350	4.65	21.73	6,770
GAS FURNACE (std) (therms) +AIR CONDITIONING (KWH) +HOT WATER HEATER (therms)	773 1,588 344	824	491	60	4650	6.69	10.81	3,964
GAS FURNACE (Hi-Eff) (therms) +AIR CONDITIONING (KWH) +HOT WATER HEATER (KWH)	487 1,588 344	663	299	47	3050	7.05	9.38	2,201
ELECTRIC FURNACE (KWH) +HEAT PUMP/AIR COND(KWH) +HOT WATER HEATER (KWH)	7,992 1,588 5,986	669	336	50	2950	6.34	12.35	2,950
GROUND SOURCE HEAT PUMP (KWH) +AIR CONDITIONING (KWH) +HOT WATER HEATER (KWH)	4,852 804 2,093	333						

BASED ON JANUARY 1984; FUEL PRICES IN TORONTO

HYDRO @ $ .043 per KWH
gas @ $ .67 per 11 cu. ft. (therms)
oil @ $ 1.47 .per gallon

FURNACE EFFICIENCIES-SEASONAL

oil @ 55%
standard gas @ 60%
high eff. gas @ 95%
air source heat pumps SPF=1.7
ground source heat pump SPF=2.8
SEER=16.0

Based on all fuels increasing at 12% per annum.

(REF. 10)

TABLE: 9 HEATING COST FOR VARIOUS HEATING SYSTEMS

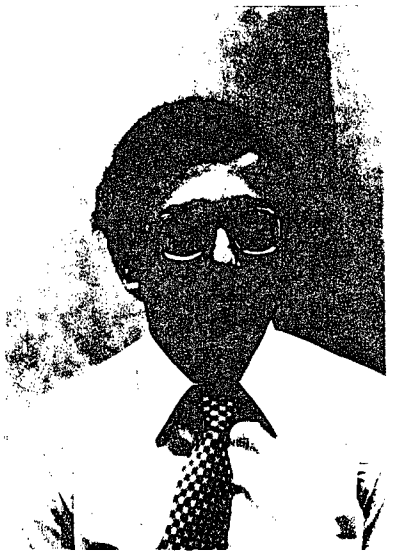
SYSTEM CHOICE	HALIFAX		MONTREAL		TORONTO		WINNIPEG		CALGARY		VICTORIA	
	SYSTEM	COST ($)	SYSTEM	COST ($)	SYSTEM	COST ($)	SYSTEM	COST ($)	SYSTEM	COST ($)	SYSTEM	COST ($)
1	GROUND SOURCE HEAT PUMP	567	GROUND SOURCE HEAT PUMP	399	GROUND SOURCE HEAT PUMP	382	GROUND SOURCE HEAT PUMP	391	GAS- HIGH EFFICIENCY	359	GROUND SOURCE HEAT PUMP	274
2	ALL ELECTRIC HEAT PUMP	862	GAS- HIGH EFFICIENCY	670	GAS- HIGH EFFICIENCY	550	GAS- HIGH EFFICIENCY	618	GAS- MID EFFICIENCY	446	ALL ELECTRIC HEAT PUMP	387
3	OIL + HEAT PUMP ADD ON	880	ALL ELECTRIC HEAT PUMP	697	ALL ELECTRIC HEAT PUMP	618	GAS + HEAT PUMP ADD ON	758	GROUND SOURCE HEAT PUMP	484	ELE + HEAT PUMP ADD ON	391
4	ELE + HEAT PUMP ADD ON	937	GAS + HEAT PUMP ADD ON	748	GAS + HEAT PUMP ADD ON	632	GAS- MID EFFICIENCY	766	GAS + HEAT PUMP ADD ON	547	OIL + HEAT PUMP ADD ON	390
5	OIL FURNACE	1280	ELE + HEAT PUMP ADD ON	773	GAS- MID EFFICIENCY	682	ALL ELECTRIC HEAT PUMP	780	GAS-STANDARD EFF.	557	ELECTRIC FURNACE	740
6	ELECTRIC FURNACE	1531	GAS- MID EFFICIENCY	830	ELE + HEAT PUMP ADD ON	688	ELE + HEAT PUMP ADD ON	830	ALL ELECTRIC HEAT PUMP	858	OIL FURNACE	844
7			OIL + HEAT PUMP ADD ON	910	OIL + HEAT PUMP ADD ON	750	GAS-STANDARD EFF.	958	OIL + HEAT PUMP ADD ON	903		
8			GAS-STANDARD EFF.	1038	GAS-STANDARD EFF.	852	ELECTRIC FURNACE	1056	ELE + HEAT PUMP ADD ON	914		
9			ELECTRIC FURNACE	1078	ELECTRIC FURNACE	1032	OIL + HEAT PUMP ADD ON	1113	ELECTRIC FURNACE	1306		
			OIL FURNACE	1465	OIL FURNACE	1265	OIL FURNACE	1671	OIL FURNACE	1361		

(REF. 10)

1. DATA EXTRACTED FROM COMPUTER RUNS AND TABLES 7, 8, and 9
2. THE HEAT PUMP CAPACITY USED FOR ALL 6 CITIES WAS 24,000 BTUH
3. FURNACE SEASONAL EFFICIENCIES
   (a) STANDARD OIL FURNACE 55%
   (b) ELECTRIC FURNACE 100%
   (c) GAS FURNACE 1. standard 60%
   2. medium 75%
   3. high 93%
   (d) AIR SOURCE HEAT PUMP SPF=1.7
   (e) GROUND SOURCE HEAT PUMP SPF=2.7

RELATIONSHIP BETWEEN THE WATER SIDE, AIR SIDE, AND POWER REQUIREMENTS FOR A WATER-TO-AIR HEAT PUMP

by Ram S. Nagrani, and R. Donald Dexheimer



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R. Donald Dexheimer

BA from North Central College, MBA from Northwestern University, HVAC Application Engineering from Knight Energy Institute. Have been involved in the sale and installation of water source heat pumps since 1980. Have spoken on the subject to national audiences and authored a Water Source Heat Pump Handbook published by NWWA. Have been involved with research projects undertaken by Oklahoma State University, Brookhave National Labs and Illinois Power Company.

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RELATIONSHIP BETWEEN THE WATER SIDE, AIR SIDE, AND POWER REQUIREMENTS FOR A WATER-TO-AIR HEAT PUMP

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A water-to-air heat pump transfers heat and at the same time "amplifies" the temperature from the water side to the air side or vice versa as required. In the process it consumes some electrical power (See Figure 1). When any one of the controllable parameters on the air side or water side is changed, it affects heat pump performance and the power consumed. This paper discusses how changed performance can be determined by changes in easily measured parameters such as temperatures and electric current. Simple charts showing trends are used to provide a better understanding for persons involved in installing and servicing water-to-air heat pumps.

Variable Parameters

Water Side

Entering water temperature, water flow rate, and leaving water temperatures are the variables on the water side. In this paper entering water temperature (TW1) and water flow rate (GPM) are considered as independent variables while the temperature difference between entering and leaving water (ΔTW) is treated as a dependent variable.

Air Side

Air flow rate, entering dry bulb and wet bulb temperatures, and leaving dry bulb and wet bulb temperatures are the air side variables. Because of insufficient test data, and also because of the difficulty in accurately measuring air flow rate (CFM) in the field, it is considered a given constant for purposes of this paper.

In the heating cycle en tering air dry bulb is the only independent variable considered on the air side. Dry bulb temperature difference (ΔTA) is the dependent variable studied.

In the cooling cycle it is necessary to combine dry bulb and wet bulb temperatures into enthalpy. A normal temperature psychrometric chart may be used to determine air enthalpies. Entering air enthalpy (HAI) is the independent variable studied, while the enthalpy difference between entering and leaving air (ΔHA) is the dependent variable.

Electrical

Voltage, frequency, current, power and power factor are the parameters on the electrical side. Inadequate electrical supply could result in lower voltages during heat pump start-up and operation, resulting in compressor burn-out. But for the purposes of this paper it is assumed that adequate electric power is available and the voltage is held constant. The frequency of electrical generation (or Hertz) is also assumed to be constant. The power factor is assumed to be constant in this paper. Although power consumed is of primary concern to the end user, current is the easiest parameter to measure. Since the power factor is considered constant, the power consumed is directly proportional to the current (See Appendix). Consequently, only the current is studied in this paper. See Table 1 for electric power consumption of heat pump.

Test Performance

Simple tests were performed in conformance with ASHRAE Standard 37-78. [1] All controllable independent variables were held constant except the one variable being studied. Test data were obtained and results calculated. Equations used to calculate results were similar to those in ASHRAE Standard 37-78. [1] However, simpler equations are shown in the Appendex. These simpler equations are adequate for field work when applied at normal conditions for air conditioning and space heating.

The results were then plotted and are shown in Figure 2-7. Notice the scales of the dependent variables are different for each plot. This was done to make the height of each point on the plots equal to its heat equivalent. As a result the final chart shows how much each variable changes relative to each other as well as what is the actual value. For instance, the height of the amp curve is equal to the difference in heights of the air side and water side variables. For each variable the slope of the line indicates the relative rate of change.

Heating Cycle

Entering Water Temperature (TW1). This variable has the most effect on heating capacity as indicated by the steep slope of the ΔTA plot in

Figure 2. Notice the ΔTW plot is quite steep as well. This is true not only because it is more difficult for the heat pump to absorb heat at a lower water temperature, but also because there is less heat available in colder water. For example, at 80° F entering water, the heat pump can absorb more than 10° F ΔTW, but at 40° F entering water, this would not be possible since the water would turn into ice before 10° F could be extracted.

Note that even though the power consumed as represented by the amps plot rises with higher entering water temperatures, unit efficiency or COP rises even faster. COP can be calculated by dividing the height of the ΔTA plot by the height of the amps plot at any given entering water temperature.

Water Flow Rate (GPM). The effect of changes in GPM are shown in Figure 3. Heating capacity tends to rise and so does the heat absorbed from the water as the GPM is increased. However, power consumed tends to stay constant after a certain point is reached. Caution: If the GPM is too low an unstable region is experienced causing refrigerant pressures to fall slowly until the heat pump shuts down on low pressure safety switch or water in the evaporator freezes and damges the heat pump.

As is evident from Figure 3, the COP of the heat pump increases as the water flow increases. Though not as effective as raising the entering water temperature, increasing the flow rate (GPM) increases capacity and efficiency of the heat pump. Frequently a high flow rate is used with earth coupled heat pumps because of the low entering water temperatures in late winter.

Notice that there is a direct correlation between the rise in air temperature (ΔTA) and heat pump capacity. This relationship provides the serviceman with another indication of whether a heat pump is operating properly or not.

Note: Instead of ΔTW, the product of ΔTW x GPM is plotted in order to better represent the relationship with the air side and power consumption.

Entering Air Temperature (TA1). Figure 4 shows the effect that entering air temperature has on heat pump performance. Amps increase as entering air temperature increases because the refrigerant pressures increase, which in turn causes the compressor to work harder and use more electric power. See Table 2 for sample refrigerant pressures. ΔTW decreases as entering air temperature increases because the ΔTA is lower. The lower the ΔTA, the less heat of absorption is required which in turn reduces the ΔTW. The ΔTA which represents heating capacity decreases as entering air temperature increases. This relationship is due to the lower temperature difference between the refrigerant coil and the entering air. Therefore, a lower entering air temperature may increase heat pump capacity. However, reducing the entering air temperature too low causes the heat pump to enter an unstable state where liquid refrigerant could enter the compressor and destroy the compressor valve plates.

Cooling Cycle

Entering Water Temperature (TW1). Increasing the entering water temperature reduces the cooling capacity represented by ΔHA plot in Figure 5. The heat of rejection represented by ΔTW decreases as entering water temperature increases because it is harder to transfer heat to warmer water. The reason that amps increase as entering water temperature increases is that the resulting higher refrigerant pressures cause the compressor to work harder and use more electric power. Heat pump EER declines as entering water temperature increases. Nevertheless, reducing the entering water temperature below 55° F does not significantly increase the cooling capacity.

Water Flow Rate (GPM). The effect of changing the water flow rate is shown in Figure 6. Cooling capacity, heat of rejection and power consumed remain essentially constant as GPM is increased. However, when water flow rate is reduced to very low levels the heat pump becomes unstable and head pressure rises until the compressor shuts off on the high pressure safety switch. This should be avoided because if it occurs very often the compressor motor windings will burn out.

Note: Instead of ΔTW, the product of ΔTW x GPM is plotted as in Figure 3 in order to better represent the relationship with the air side and power consumption.

Entering Air Enthalpy (HAI). A sling psychrometer should be used to obtain a wet bulb temperature reading. Then together with a dry bulb temperature reading and a psychrometric chart one can obtain the entering air enthalpy. Figure 7 shows how the entering air enthalpy affects heat pump performance. The power consumed remains essentially constant as represnted by the amps plot. But the ΔTW and ΔHA curves representing cooling capacity and heat of rejection rise as the entering air enthalpy increases. This relationship exists because the entering water temperature to a water source heat pump is relatively constant over a 24 hour period. Compartively, the ΔHA plot for an air conditioner would be relatively flat or decrease slightly because the outdoor air temperature changes considerably over a 24 hour period.

Two important points are illustrated in Figure 7. First, the cooling capacity of the heat pump increases as the entering air enthalpy increases. This is especially beneficial in a hot humid climate such as Houston, Texas. The second point is that with successive cycles of operation the heat pump rides down the ΔHA curve. This causes the heat pump to cycle longer and dehumidify better than an air conditioner.

From Figure 7 one can see that the EER of a water-to-air heat pump increases as the entering air enthalpy increases.

Note: During the tests conducted to obtain these curves moisture was condensing on the air side coil. The dry bulb temperature was held constant and the wet bulb temperature was changed to obtain different entering air enthalphies.

Conclusion

Easily measurable parameters were used to make this paper useful to installers and servicemen. The charts were plotted in a manner to make the relative changes easier to comprehend. Other topics which have not been discussed in this paper are the effects of CFM changes and power factors. Study of over-voltage and under-voltage variations would provide a better understanding of extending compressor life.

In addition to the above external factors, servicemen are concerned about other parameters such as refrigerant charge, refrigerant pressures, and refrigerant temperatures which are internal to the heat pump. (3) These factors depend on individual heat pumps and their designs. Therefore, it is for the manufacturer to provide the data and understanding of its specific heat pump. See Table 2 for sample refrigerant pressures at various conditions from manufacturer.

Taken together, the external and internal measurements of various parameters should help in proper diagnosis of any problem and fine tuning of heat pumps for efficient operation in various situations.

Appendix

Equations used most commonly are listed below.

Water Side

$$Q_W = 500\ (GPM)\ (\Delta TW) \qquad (1)$$

Air Side

$$Q_{A,H} = 1.088\ (CFM)\ (\Delta TA) \qquad (2)$$

$$Q_{A,C} = 4.5\ (CFM)\ (\Delta HA) \qquad (3)$$

Electrical

$$P = (V)\ (I)\ (\phi) \qquad (4)$$

$$QP = 3.412\ (P) \qquad (5)$$

List of Variables

CFM = Air flow rate, cu. ft./min.
ΔHA = Difference between entering and leaving air enthalpies, Btu/lb. of dry air
ΔTW = Difference between entering and leaving air dry bulb temperatures, ° F
GPM = Water flow rate, gal/min.
I = Current, amps
P = Power, watts
QA,C = Air side cooling capacity, Btu/hr.
QA,H = Air side heating capacity, Btu/hr.
QP = Heat equivalent of power consumed, Btu/hr.
QW = Water side heat absorbed or rejected, Btu/hr.
V = Voltage, volts
ϕ = Power factor

References

1. "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment" ANSI/ASHRAE 37-1978

2. "Laboratory Tests of a Residential Unitary Water-Source Heat Pump" by William J. Mulroy and George E. Kelly, NBSIR 81-2287

3. "Water Source Heat Pump Handbook" by R. Donald Dexheimer, Publisher: National Water Well Association, 1984

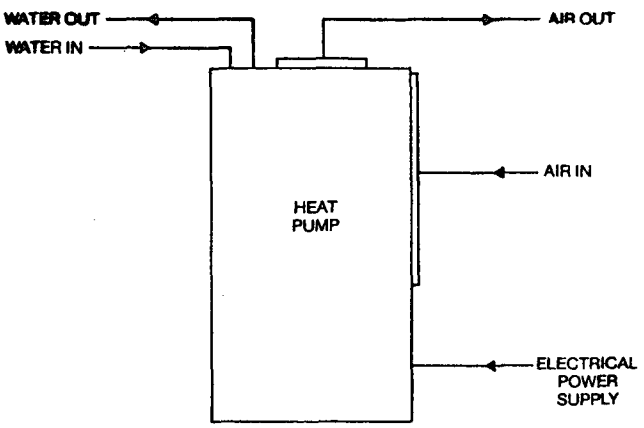

FIG. 1.

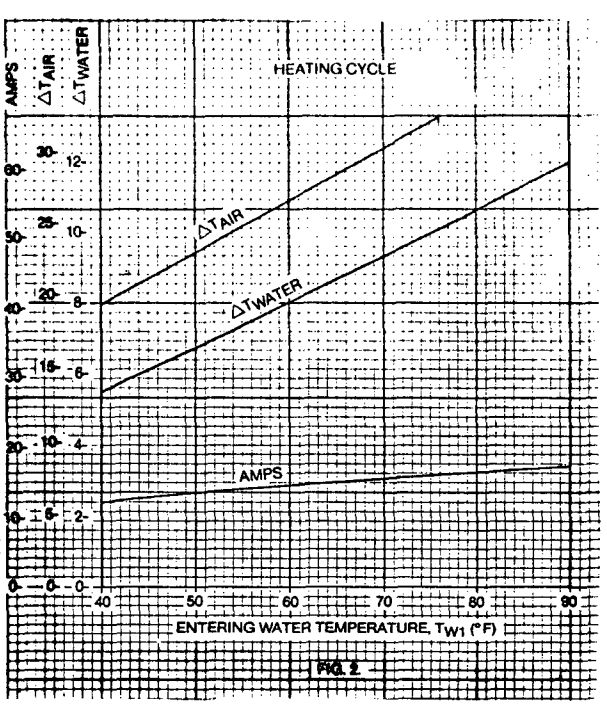

FIG. 2

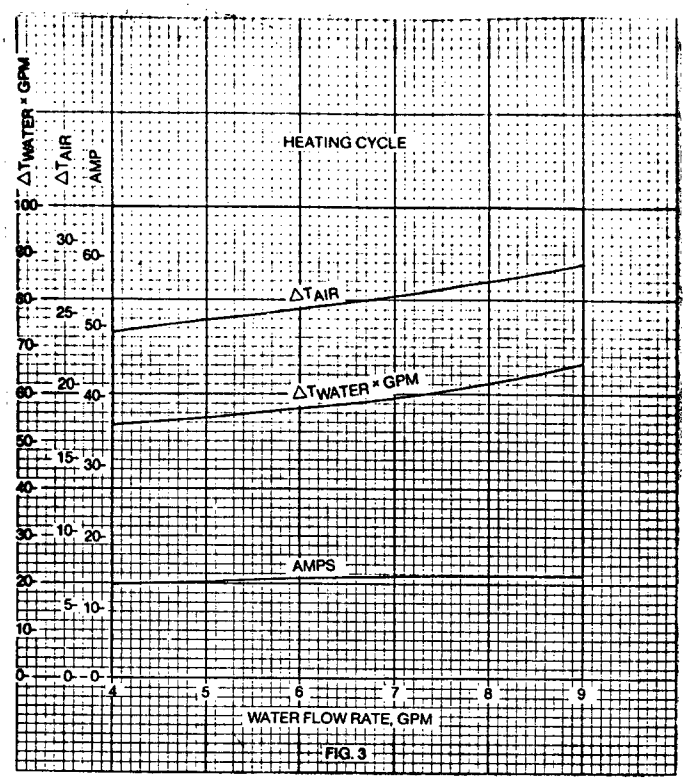

FIG. 3

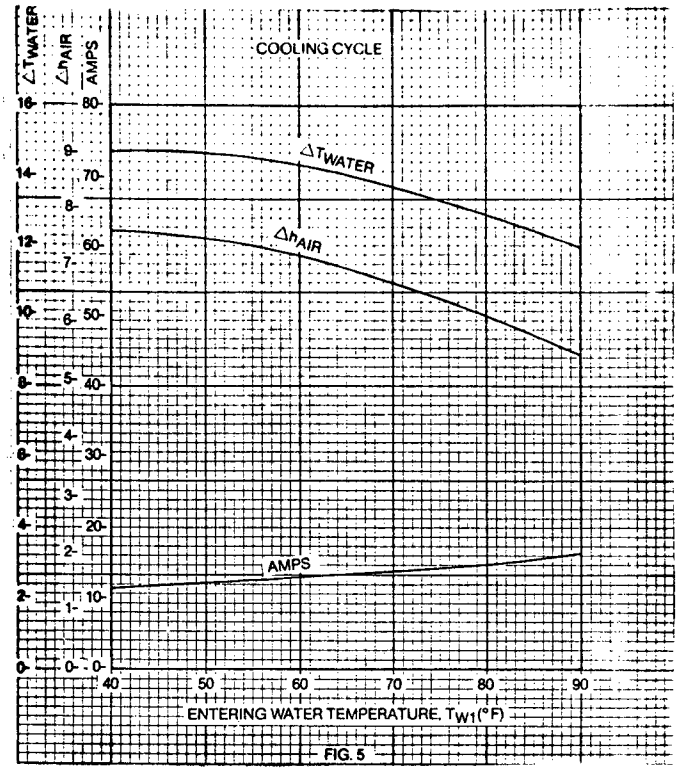

FIG. 5

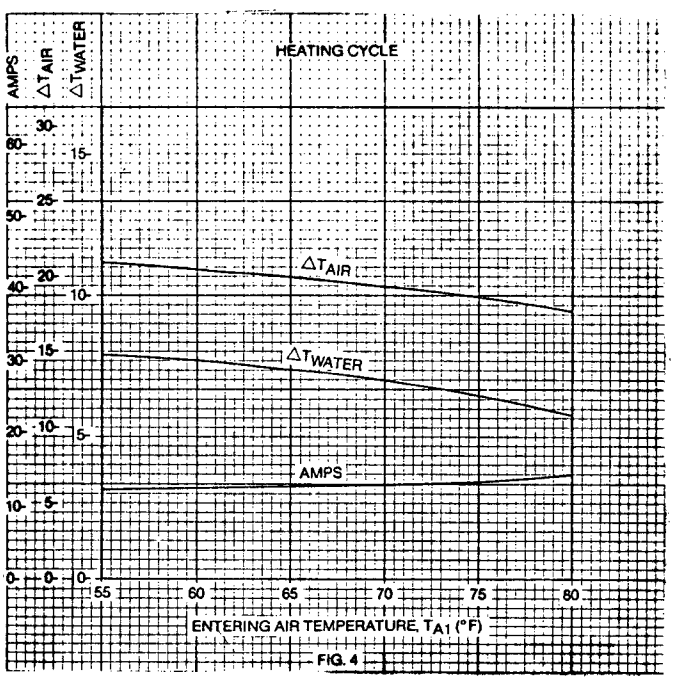

FIG. 4

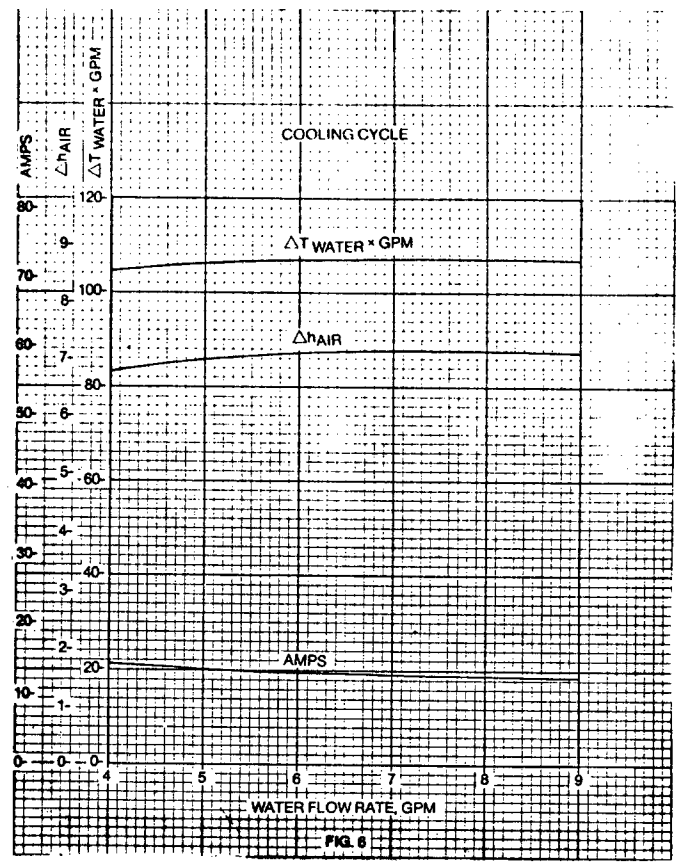

FIG. 6

CAPACITY AND EFFICIENCY RATINGS

MODEL 035

ENTERING WATER TEMP. 6	GPM	WATER PRESSURE DROP (PSI)	NOM. CFM 3	HEATING 1				COOLING 2			
				MBH	WATTS 4	COP	ADJ. 5 COP	MBH	WATTS 4	EER	ADJ. 5 EER
45	5.0	1.03	1435	28.4	2560	3.25	2.87	46.7	2740	17.04	15.14
	6.0	1.42		30.0	2680	3.28	2.83	48.3	2620	18.44	15.87
	7.5	2.15		31.4	2800	3.29	2.75	48.7	2520	19.33	15.86
50	5.0	1.03	1435	31.1	2750	3.31	2.95	45.6	2800	16.29	14.50
	6.0	1.42		32.6	2830	3.38	2.94	46.7	2700	17.30	14.95
	7.5	2.15		34.1	2890	3.57	2.91	47.5	2610	18.20	15.03
55	5.0	1.03	1435	33.7	2860	3.45	3.08	44.6	2880	15.49	13.83
	6.0	1.42		35.2	2930	3.52	3.08	45.6	2780	16.40	14.24
	7.5	2.15		36.7	2990	3.60	3.04	46.4	2700	17.19	14.28
60	5.0	1.03	1435	36.3	2970	3.58	3.21	43.4	2970	14.61	13.09
	6.0	1.42		37.8	3040	3.64	3.20	44.4	2870	15.47	13.48
	7.5	2.15		39.3	3090	3.73	3.16	45.1	2790	16.16	13.50
65	5.0	1.03	1435	38.9	3080	3.70	3.33	42.1	3050	13.80	12.40
	6.0	1.42		40.4	3140	3.77	3.32	43.0	2970	14.48	12.67
	7.5	2.15		41.9	3190	3.85	3.28	43.8	2880	15.21	12.77
70	5.0	1.03	1435	41.5	3200	3.80	3.43	40.7	3140	12.96	11.68
	6.0	1.42		42.9	3250	3.87	3.42	41.7	3060	13.63	11.97
	7.5	2.15		44.6	3290	3.97	3.40	42.4	2970	14.28	12.05

TABLE 1

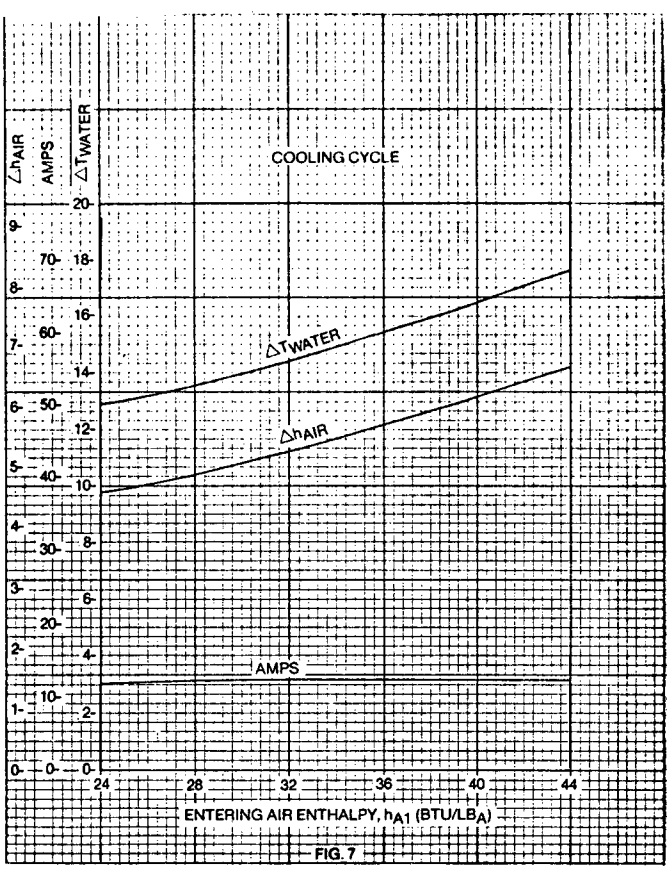

FIG. 7

REFRIGERANT PRESSURES

SOL-A-TERRA II, SIZE 35

EWT ° F.	GPM	HEATING SUCTION PSIG	HEATING DISCHARGE PSIG	COOLING SUCTION PSIG	COOLING DISCHARGE PSIG
45	5	38-43	173-183	71-76	145-155
	6	43-48	187-197	69-74	129-139
	7.5	44-49	190-200	68-73	123-133
50	5	41-46	190-200	71-76	150-160
	6	46-51	196-206	70-75	139-149
	7.5	48-53	198-208	68-73	133-143
55	5	45-50	197-207	71-76	161-171
	6	50-55	202-212	70-75	147-157
	7.5	53-58	204-214	68-73	142-152
60	5	49-54	205-215	71-76	173-183
	6	55-60	211-221	70-75	155-165
	7.5	57-62	215-225	68-73	150-160
65	5	55-60	210-220	72-77	184-194
	6	59-64	217-227	71-76	166-176
	7.5	62-67	218-228	71-76	153-163
70	5	59-64	215-225	72-77	195-205
	6	65-69	225-235	72-77	177-187
	7.5	67-72	227-237	71-76	168-178

Table 2

AN ASSESSMENT OF GROUND-COIL HEAT-PUMP TECHNOLOGY

by Robert D. Fischer, George H. Stickford, and Frank E. Jakob

Robert D. Fischer

Mr. Fischer's work at Battelle-Columbus has included design and performance analysis of air, water, and ground-source heat pumps; reciprocating, rotary vane, and rolling-piston refrigerant compressors; conventional and condensing gas furnaces; oil-fired burners; and induction and exhaust systems for automotive engines. He has also participated in major programs including development of a heat transfer code to predict temperatures during pipeline welding; development and verification of the SP43 model to predict transient performance of warm-air residential heating systems; and updating of the degree-day method for residential space conditioning loads.

He is presently a Principal Research Engineer at Battelle and holds BSME and MSME degrees from University of Minnesota and Bradley University. He is a member of ASHRAE and is a registered Professional Engineer in the State of Ohio.

George H. Stickford

Bio not available.

Frank E. Jakob

Bio not available.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

AN ASSESSMENT OF GROUND-COIL HEAT-PUMP TECHNOLOGY

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Introduction

This paper summarizes results from an analytical comparison of the performance and economics of residential space conditioning with three types of ground-coupled heat-pump systems and air-source systems in ten metropolitan areas of the United States.[1] Additional results from parametric analyses of the effects of costs, soil properties, operating strategy, and alternate system configurations[2] are also summarized.

This work was conducted with major inputs from the Oak Ridge National Laboratory (ORNL) in selecting ten representative cities, characterizing the water table and soil layers in each city, and computing representative residential space conditioning loads and air-source heat-pump performance in each city.

Computer models were prepared for deep-well and multiple shallow-well ground coupling which operate stably with time steps of about one-quarter hour. The model for the horizontal coil is based on the Brookhaven GROCS model[3,4] with revisions to include thermal resistances due to convection within the tube and conduction through the wall of the plastic tubing.

Effects of the water table and different thermal properties of soil layers are explicitly accounted for in the well-type models. Moisture migration and freezing are not modeled, but different thermal properties for the soil above the water table are provided for winter and summer operation. All three computer models have similar subprograms for computing weather data and space conditioning loads, performance of the water-source heat pump, soil properties, convective heat-transfer coefficient at tubing walls with one of four brines, and far-field ground temperatures. The models are well suited for parametric study. For this initial effort, reasonable values for many variables had to be selected based on intuition and experience; however, the paper does provide parametric data for major variables such as depth or length of the ground coil.

Methodology

Ten representative cities were selected for use in this program based on a review of climatic, geologic, and demographic features. Space conditioning loads for an 1800 sq ft single-family building for each month of the heating and cooling seasons were computed for each city using binned weather data.

Building insulation was appropriately selected for each city, and loads reflected occupancy, lighting, and solar influences. These loads were then approximated on a continuous time basis using a simple UA heat loss or gain model with infiltration so that the real effect of the earth thermal capacitance will be realized in long-term performance. Duration of heating or cooling operation in each of the fringe months of April or October was apportioned based on the relative heating or cooling loads in the ORNL bin data. Ambient temperatures were computed with a sinusoidal relationship for a typical day of each month using monthly values of maximum and minimum temperatures from long term weather data for each city.[5]

For the economic analyses, simple payback and life-cycle costs were evaluated from operating costs and marginal equipment costs compared to that for an air-source heat pump. Residential power costs in each city were based on rates for a consumption of 2500 kWh[6]. Table 1 shows estimated costs for the three-type of ground coils.

Table 2 shows characteristic values for density, specific heat, and thermal conductivity that were selected as a function of depth based on appropriate values for representative layered soil and water table for each city. Far-field earth temperatures and surface temperatures were computed using the Kusuda relationship[7] for earth temperature at any depth as a function of time of year, thermal diffusivity of the earth, and three constants, which can be determined from monthly ambient temperatures.

Performance characteristics of water-source heat pumps were based on values for units with good performance available in four capacities from a nominal cooling capacity of 22,000 to 42,000 Btu/hr with 75 F inlet water temperature. Values for cooling and heating capacity and cooling coefficient of performance for each of these units were fitted as a linear function of inlet water temperature for use in the program. A single polynomial for heating coefficient of performance as a function of inlet water temperature was used for all units. Capacity scaling was provided with an input variable multiplier. Any heating capacity not met by the equipment was provided by resistive heating. These relationships and constants provide a realistic basis for a quasi-steady-state performance model in which the effects of variation in inlet water temperature and mismatching are reflected in system performance.

The heat pump was controlled to operate with 3 to 4 on-off cycles per hour, depending on stability requirements. The relative on-off time periods were apportioned to match the load. Performance degradation due to cycling operation was accounted for by increasing power consumption by a degradation factor of 1.15 whenever the capacity of the unit exceeded the space-conditioning load.

Comparative performance data for a space conditioning system with a standard midline, high-quantity-production air-to-air heat pump was computed for each city. The size of heat pump selected for each city was based on lowest annual life-cycle cost. Cycling losses were computed using C_D degration factors of 0.25 for both heating and cooling. Defrosting losses were computed using the Parken and Kelly frosting data.[8]

Descriptive information on the three models for horizontal, deep-well, and multiple shallow-well ground coils is provided in the Appendix.

Performance and Economics

A total of 59 yearly runs were made to provide performance and economic data for each of the three types of ground coil systems in 10 cities. More data are available for some cities from attempts to define optimum coil geometry or to improve cost predictions. Runs were begun in each city at the start of the heating season in October and continued for 8760 hours.

A summary of seasonal performance data is provided in Reference 1 and will not be repeated here. This information included summed electrical power consumption and resistive heating for both ground-coupled and air source heat pumps, deficit (if any) in ground coil cooling capacity, and total "on" time in the heating and cooling seasons. Inlet fluid temperatures at the end of each month were plotted or tabulated for each run to help access the approach of coil length to the optimum based on performance considerations.

Values of electric power consumption were lower with the ground-coil systems for all but one run. For cooling, electric power consumption was lower in 33 of the 59 runs. Of the 26 instances of higher cooling electric power consumption with ground-coil systems, some were due to inadequately sized ground coils, and the balance were in Phoenix and Houston where ground-coil performance was poor due to high cooling load and low water table.

Tables 3 to 5 show that values of seasonal coefficient of performance for heating with the ground-coil systems were substantially higher than that for air-source systems. For cooling operation, the seasonal coefficient of performance for ground-coil systems is strongly dependent on coil sizing. Values consistently lower than that for air-source systems were experienced in Phoenix and Houston. In other cities, values higher than that for air-source systems can be attained with properly sized coils.

Savings in electrical power and resulting costs must be adequate to provide a payback of the additional costs attributable to the ground-coil systems within a reasonable time to be of interest to the average homeowner. Some more affluent homeowners will be more interested in life-cycle cost savings. Tables 6 to 8 show economic values for the three ground-coil systems compared to that for an air-source system. Heating and cooling costs are shown separately so that contributing factors to annual power savings or loss can be readily observed. Substantial annual power savings are evident for all systems for all cities except Phoenix, Houston, and Seattle. Annual power savings with each of the three systems configured for minimum simple payback were fairly equivalent.

Simple payback was lowest with horizontal and deep-well systems and was as low as 8.1 years in Portland, Maine. Simple payback was 10 years or less in four cities with deep-well systems and in two cities with horizontal systems. Simple payback with any system in cities with a high cooling load and deep water table as in Phoenix and Houston was excessive or nonexistent.

Deep-well systems clearly have lower simple payback than multiple shallow-well systems in Portland, ME and Washington, D.C. due to the improved properties of underlying rock layers. Depth of multiple, shallow wells was selected for ease of drilling with a light rig, whereas a heavier-duty rig, used for drilling a deep well, could be used to bore into rock layers.

Life-cycle costs with a 2.0 percent discount rate for all three types of ground-coil systems were lower than that for air-source systems in Portland, ME, Washington, DC, Pittsburgh, Kansas City, and Minneapolis. In addition similar values were lower in Atlanta and Louisville with deep-well systems.

Simple payback and life-cycle costs for systems in Seattle were adversely affected by low-space conditioning loads and very low electrical rates.

Parametric Analysis

Parametric analyses were conducted on trenching and installation costs, thermal properties of the soil, thermal recovery due to extended off time, and alternate system configurations with direct ground-loop cooling and desuperheater water heating during cooling.

Trenching and Installation Costs. Tables 9 to 11 show resulting savings when costs for trenching or drilling and installation are reduced to 75 and 50 percent of baseline costs. This level of cost reduction from prevailing rates, used in the original analyses, may be justified with improved methods or machines expected in the development of a mature technology.

The lowest simple payback was reduced from about 8.2 to 5.1 years for both horizontal and deep-well systems in Pittsburgh, PA. Payback decreased from about 10 years to a value of under 7 years at the 50 percent reduction in cost with horizontal systems in Portland, ME and with deep-well systems in Washington, DC, and Kansas City, MO. Payback for northern cities with deep-well systems was generally under 6 years. Absence of payback is indicated in Tables 9 to 11 with a negative zero.

Savings in life-cycle costs improved substantially with lower trenching and installation costs. The highest value of savings in life-cycle costs was

24.1 percent for a deep-well system located in Pittsburgh, PA. With deep-well systems, the savings was 19 percent or more in four of the nine cities. In a fifth city, the savings exceeded 14 percent.

Best-Case Thermal Properties of the Soil. The effect of best-case soil properties on costs with horizontal-coil systems was explored in three cities. Soil properties at field capacity rather than dry conditions were used for cooling operation. Soil properties at the arithmetric average for unfrozen and frozen saturated soil rather than for unfrozen saturated soil were used for heating operation.

Table 12 shows economics of horizontal systems with best-case soil properties. Simple payback in Portland, ME decreased from 11.7 to 9.3 years (20.5 percent reduction) and in Washington, DC from 12.4 to 10.6 years (14.5 percent reduction). Life-cycle costs were similarly improved with best case soil properties. A savings was indicated in Houston, so additional runs were made to optimize the coil length. These results show an optimum coil length of about 2300 ft in Houston with a simple payback of 38 years.

Thermal Recovery Due to Extended Off Time. The effect of thermal recovery of the earth (due to extended off time) on heat-pump performance was explored by varying the size of the heat pump and extending the range of on-off operation. Horizontal and deep-well models were modified so that the period of one on-off cycle can extend over multiple time-steps depending on the capacity of the heat pump in relation to the load. With higher heat pump capacity, the heat pump will come on for a short time required to meet the load for that period and then will remain off for the balance of the period. In the deep well model, the fluid was assumed to be thermally coupled to the casing wall by conduction when the heat pump is off.

Tables 13 and 14 show information on the thermal recovery runs for horizontal and deep-well systems in Minneapolis, MN. In the first series of runs for both types of coils, heat pump size was varied over a range of about 3 to 5 tons. Heat pump performance was scaled from the largest (3.5 ton) heat pump available and blower and pump power were scaled with size.

Two values for seasonal COP for heating are listed in Table 13. The COP value listed separately is higher because it does not include resistance heating. With resistance heating included, the seasonal COP, in general, rises with increasing heat-pump capacity because resistance heating falls off.

Values for seasonal COP which do not include the effect of resistance heating should indicate the beneficial effect of earth recovery (if any) because the obscuring effect of resistance heating is eliminated. Such values for seasonal COP with a horizontal coil in Table 13 (Runs 1 to 5) indicate marginal improvement in COP due to the beneficial effects of earth recovery during the extended off time with larger sizes. Similar data for the deep-well coil do not indicate any improvement in COP with increasing heat-pump capacity. Table 14 shows that the lowest simple payback and greatest savings in life-cycle costs with respect to air-source systems are obtained at a heat-pump capacity of about 4.5 tons for both the horizontal and vertical systems.

In the second series of runs, the heat-pump size was held constant at a relatively large value to minimize the obscuring effect of resistance heating. Then, the period for one on-off cycle was varied from 0.5 to 1.25 hour for the horizontal-coil system and from 0.67 to 1.67 hour for the vertical-coil system, as shown in Table 13 (Runs 6 to 9 and 14 to 17). No significant trends are evident in this data to support such operation with an extended on-off cycling period.

Heating and Direct Ground-Loop Cooling. For this analyses, the space-conditioning system was assumed to consist of a ground-coupled heat pump for heating and a chiller coil coupled to the ground loop for cooling. The performance model for chiller coil was based on data from the supplier of a specifically sized coil. The location selected for analysis of this system was Minneapolis, MN where cooling and humidity loads are moderate.

Table 15 and 16 show information on the runs with direct ground-loop cooling. Each set of four to six runs consists of two runs with the ground-coupled heat pump used for both heating and cooling; followed by two to four runs with ground-coupled, heat-pump heating and direct ground-loop cooling. A 3.5 ton heat pump was used for all runs.

In Table 15, seasonal COP for cooling operation was significantly improved, as expected, with direct ground-loop cooling. For example, with a horizontal coil, cooling COP increased from 2.58 (Run 2) to 4.61 (Run 4). With a 250-ft deep-well coil, cooling COP increased from 3.38 (Run 14) to 9.28 (Run 16).

Table 16 shows that a savings in payback of about 1.4 years can be realized with direct ground-loop cooling (see Runs 2 and 4); however, the cooling deficit for the entire cooling season was about 20.3 percent with this direct ground-loop cooling configuration. With the cooling deficit reduced to 10 percent by increasing the coil length from 1200 to 2100 ft (see Run 7), the savings in simple payback with direct ground-loop cooling is lost. Simple payback with the deep-well coil was only slightly higher than with the horizontal coil, but the cooling deficit was eliminated. Life cycle costs at the 2 percent discount rate were about 11.2 percent lower with direct ground-loop cooling in comparison to that with an air source unit.

Data for Runs 8 to 11 show that cooling COP for a horizontal-coil system with direct ground-loop cooling can be improved substantially with 6-ft coil depth, but simple payback is higher than at the 4 ft coil depth. Data for Runs 10 and 12 show that lowering the cost for the cooling coil by one half reduced the simple payback by about 9.1 percent (from 11 to 10 years).

Desuperheater Water Heating During Cooling. In this analyses, a desuperheater was used to preheat hot water and to reduce the amount of heat rejected to the ground coil during cooling operation. Cities selected for this study were those with a significant cooling load where coil length is determined primarily by cooling operation. An additional criterion for selection was that the city should also have an adequate heating load to improve the yearly utilization of the heat pump.

The model prepared for the desuperheater water heater was based on the configuration and limited performance data for an available unit. Water is circulated from the bottom of the hot water tank to the desuperheater heat exchanger with a small pump. When there is a hot-water demand, cold water from the supply main is supplied directly to the desuperheater heat exchanger.

A counterflow design of heat exchanger is used with refrigerant vapor from the compressor outlet at condensing pressure on one side and the circulated water on the other side. A heat-exchanger effectiveness of 0.7 was used to characterize a well designed, cost-effective unit. Effectiveness of available units appears to be about 0.33. A value for the minimum thermal capacity rate was determined from performance data for the available units. Compressor outlet temperature was correlated as a linear function of the condensing temperature during cooling operation. Finally, condensing temperature was determined from temperature of the return fluid from the ground coil.

A daily hot water demand of 75 gal/day was divided into an hourly usage schedule with four 2-1/2 hour periods with hot-water demand. Any deficiency in production of this amount, which is equivalent to 63,400 Btu/day, was made up by resistance heating.

Cost of the desuperheater water heater was based on the dealer's cost increment with upward adjustment to cover profit and overhead and the assumption of improved effectiveness.

Table 17 shows the effect of desuperheater water heating during cooling operation on simple payback of costs for a horizontal-coil system in five cities. Energy for heating hot water during the heating season is not included in the analysis.

Comparisons in Table 17 show a substantial reduction in simple payback with desuperheater water heating during cooling operation for all cities. The gain in simple payback would be less if the reference air-source heat pump were also equipped with a desuperheater water heater. Future evaluations shou include such a comparably equipped air source heat pump reference system and should address the effect of integrated water heating of the economics of other ground coil configuration.

Coil length can be substantially reduced while maintaining equivalent cooling performance with desuperheater water heating during cooling operation. As a result, the installed cost (including cost of the desuperheater water heater) is lower in three of the five cities and about equal in the other two cities as shown in Table 17.

Conclusions

This study has shown that simple payback of less than 10 years for deep-well systems appears to be feasible in four cities and in two cities for horizontal systems. A deep-well system appears to be a strong competitor to a horizontal system in Portland, ME, Washington, DC, and Pittsburgh, PA; although, these results are probably influenced considerably by the good thermal properties for rock layers that occur at depths of under 100 ft in these cities. Multiple shallow-well systems do not appear to be a clear choice over deep-well systems in any city. Ground-coil systems do not appear to be appropriate for cities with low electrical rates (like Seattle), or cities with a high cooling load, low heating load, deep water table, and no rock layers (like Phoenix and Houston).

Parametric analyses have shown that economic viability of ground coupled systems can be significantly improved by reducing costs for trenching or drilling and installation of ground coils. A substantial reduction in simple payback of 14 to 20 percent, due solely to using the best-case soil properties, suggests that additional work related to characterization and modeling the effects of time-dependent soil properties should be encouraged. Any beneficial effect on performance due to thermal recovery of the earth by extending the period for one on-off cycle appears to be small. Sizing the ground coil for direct ground-loop cooling operations does not appear to offer sufficient operating cost savings to provide a net economic advantage over conventional air conditioning operation based on our work on such a system in Minneapolis, MN. This work has shown that desuperheater water heating during cooling substantially improves economic viability of horizontal ground-coil systems, primarily because the length of the ground coil can be substantially reduced.

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APPENDIX

Descriptive information on the three models for horizontal, deep-well, and multiple, shallow-well ground coils is provided below.

Horizontal Ground-Coil Model

The finite-difference model for a horizontal serpentine ground coil is based on the GROCS model[3,4]. A description of features of the adapted version of GROCS was presented previously[9] and will not be repeated here.

Deep-Well Ground-Coil Model

A finite-difference deep-well ground-coil model was prepared for this program which has a nodal geometry as shown in Figure A-1. Some features of this model were patterned after the more detailed ORNL deep-well model[10]. As shown in Figure A-1, the model is segmented into six modes in both radial and depth coordinates Saturated properties were used for all nodes below the water table. Thermal capacitance of the pipe and casing material is ignored and the soil nodes were made large enough to permit time steps of one-fourth hour or more without stability problems. Resistance for heat flow from the fluid to the adjacent soil nodes included a laminar convective heat transfer coefficient with entrance-length correction for flow from the bottom to top of the annulus and thermal conductance of the casing material. Temperatures in the surface node and boundary nodes were computed using the Kusuda approach.[7]

Computing time for one month was reduced from about 780 sec with matrix inversion to about 42 sec with an explicit, forward, finite-difference solution. This time compares to about 36 sec for the horizontal-coil model. With explicit solution, all fluid nodes are lumped into one, but thermal resistances at the casing wall are computed as in the more detailed model.

Multiple Shallow-Well Model

Multiple shallow wells are drilled to a convenient depth with relatively light weight well drilling equipment. Then, a coil in a "U" tube configuration is placed in the well bore with supply and return piping in contact with the soil. Since the earth at the coil will experience the mean coil temperature at all elevations, temperature of the fluid in the coil can be characterized by the mean fluid temperature.

A finite-difference model for multiple shallow-well ground coupling was prepared in this program with some features patterned after work underway in Sweden. The wells are clustered in patterns of 4 or 9 well bores to enhance seasonal thermal storage. Coil tubing is placed in the well bores and connected in parallel to distribution manifolds located at the top of the tubes below the frost line. This central core of clustered tubes is surrounded by an outer "store" of earth.

Figures A-2 and A-3 show the nodal geometry for the 4- and 9-pattern coils and Figure A-4 shows the nodal geometry of the outer "store". Symmetry of the nodal pattern and heat flow is utilized to minimize the number of total nodes. Two sets of nodes in the depth coordinate provide explicit modeling of the water table, as in the deep-well model. To minimize computing time, new values for temperatures in the outer "store" are computed at every fourth time step of the central core. Surface temperatures and outer node temperatures in the outer "store" are computed using the Kusuda relationship.

Computing time with the original approach with matrix inversion was reduced from about 340 sec per month to about 50 sec for the 4-pattern coil and about 2800 sec per month to 63 sec for the 9-pattern coil with an explicit, forward, finite difference solution.

TABLE 1. ESTIMATED COSTS FOR GROUND COILS

	Cost/ft, $
Horizontal Coil	
Coil Material	0.50
1-3/8" ID x .12" wall polyethylene tubing	
Trenching	0.75
Laying; backfilling ($25/hr at 100 ft/hr	0.25
Deep-Well Coil	
Coil Material	
1-3/8" ID x .12" wall polyethlene supply tube	0.50
0.12" wall polyethylene casing; inside dia = 2.5"	0.88
5.0"	1.71
7.5"	2.55
Drilling of 3.0 to 8.0" well, except as noted	
Unconsolidated soil	6.00
(3" well)	4.00
Mantle rock	7.00
Bed rock	8.00
Placing coil ($25/hr at 100 ft/hr plus allowance for casing joints)	0.40
Multiple, Shallow-Well Coil	
Coil Material	
1-3/8" ID x .12" wall polyethylene tubing, two tubes per bore)	1.00
Drilling (Special purpose Ditch Witch unit limited to 110 ft deep)	
Unconsolidated soil	4.00
Mantle rock	6.00
Bed rock	8.00
Placing tubing ($25/hr at 100 ft/hr)	0.25

TABLE 2. SOIL-LAYER PROPERTIES ASSUMED FOR EACH CITY

City	Layer Interval, ft.	Soil Description	Soil Moisture	Density, lbm/ft^3	Specific Heat, Btu/lbm-F	Thermal Conductivity Btu/hr-ft-F
1. Atlanta, GA	0-40	Top soil(1)	Dry(2)	94.7	0.26	0.51
			Wet	116.1	0.40	0.76
	40-85	Mantle rock	Wet	153.	0.30	2.4
	85	Bedrock	Wet	164.	0.26	3.3
2. Pittsburgh, PA	0-25	Top soil	Dry	96.7	0.24	0.51
	25-60	Top soil	Wet	118.	0.38	0.78
	60	Bedrock	Wet	164.	0.26	3.3
3. Louisville, KY	0-65	Top soil	Dry	99.4	0.26	0.54
	65-130	Top soil	Wet	119.	0.38	0.78
	130	Bedrock	Wet	164.	0.26	3.3
4. Phoenix, AZ	0-150	Top soil	Dry	93.7	0.23	0.53
	150-2000	Top soil	Wet	118.	0.39	0.82
5. Kansas City, MO	0-13	Top soil	Dry	100.	0.25	0.54
	13-20	Top soil	Wet	119.	0.38	0.78
	20-106	Sand/gravel	Wet	125.	0.40	1.4
	106	Bedrock	Wet	164.	0.26	3.3
6. Houston, TX	0-100	Top soil	Dry	103.	0.27	0.54
			Wet	120.	0.38	0.78
	100	Sand/gravel clay/mix	Wet	110.	0.45	1.3
7. Portland, ME	0-20	Top soil	Dry	99.0	0.22	0.51
	20-40	Top soil	Wet	122.	0.37	0.99
	40-60	Mantle rock	Wet	153.	0.30	2.4
	60	Bedrock	Wet	164.	0.26	3.3
8. Seattle, WA	0-50	Top soil	Dry	96.3	0.21	0.55
			Wet	121.	0.37	1.02
	50-250	Sand/gravel clay	Wet	110.	0.45	1.3
	250	Bed rock	Wet	164.	0.26	3.3
9. Washington, D.C.	0-28	Top soil	Dry	103.	0.22	0.76
			Wet	124.	0.35	1.11
	28-46	Gravel, broken sandstone	Wet	103.	0.36	1.73
	46-76	Mantle rock	Wet	153.	0.30	2.4
	76	Bed rock	Wet	164.	0.26	3.3
10. Minneapolis, MN	0-45	Top soil	Dry	105.	0.23	0.61
	45-120	Top soil	Wet	125.	0.35	0.96
	120-170.	Sand, gravel	Wet	125.	0.40	1.40
	170.	Bedrock	Wet	164.	0.26	3.3

(1) Top soil properties were compiled by ORNL

(2) Properties for winter operation were assumed to be wet values to account for the effect of moisture migration. "Wilting" dry values were used for summer operation.

TABLE 3. SEASONAL COEFFICIENT OF PERFORMANCE WITH HORIZONTAL GROUND COIL

City	Coil Length, ft	Heating COP		Cooling COP	
		Ground-Coil	Air-Source	Ground-Coil	Air-Source
Portland, ME	900	2.04	1.74	2.04	2.30
	1200	2.18		2.41	
	1500	2.25		2.60	
Atlanta, GA	700	2.83	1.89	1.57	2.35
	900	2.90		2.06	
	1100	2.94		2.36	
Washington, D.C.	700	2.61	1.84	1.83	2.32
	1000	2.74		2.33	
	1300	2.79		2.57	
Pittsburgh, PA	1200	2.57	1.79	2.54	2.28
Louisville, KY	900	2.62	1.83	1.81	2.33
	1100	2.70		2.17	
Phoenix, AZ	1600	2.96	1.97	1.52	2.15
	1800	2.95		1.66	
Kansas City, MO	1000	2.55	1.77	2.01	2.28
	1100	2.59		2.17	
Houston, TX	1300	3.04	1.97	1.67	2.31
	1500	3.04		1.89	
Seattle, WA	500	2.40	1.92	1.43	2.42
	700	2.60		2.11	
Minneapolis, MN	900	2.09	1.66	2.15	2.22
	1200	2.17		2.49	

TABLE 4. SEASONAL COEFFICIENT OF PERFORMANCE WITH DEEP-WELL GROUND COIL

City	Coil Length, ft	Heating COP		Cooling COP	
		Ground-Coil	Air-Source	Ground-Coil	Air-Source
Portland, ME	200	2.32	1.74	3.43	2.30
	280	2.56		3.55	
	360	2.66		3.61	
Atlanta, GA	120	2.87	1.89	2.00	2.35
	180	3.02		2.82	
	240	3.06		3.11	
Washington, D.C.	140	2.75	1.84	2.69	2.32
	200(1)	2.83		2.80	
	200	2.92		3.04	
	200(2)	2.96		3.16	
	260	2.98		3.20	
Pittsburgh, PA	240	2.77	1.79	3.43	2.28
	280	2.83		3.49	
Louisville, KY	210	2.80	1.83	2.67	2.33
	250	2.90		2.93	
Phoenix, AZ	400	3.09	1.97	1.51	2.15
	500	3.07		1.86	
Kansas City, MO	210	2.77	1.77	2.98	2.28
	250	2.85		3.15	
Houston, TX	340	3.11	1.97	1.89	2.31
	400	3.11		2.18	
Seattle, WA	200	2.61	1.92	3.38	2.42
Minneapolis, MN	170	1.88	1.66	2.81	2.22
	250	2.28		3.33	

(1) Inside dia of casing changed from 5.0" to 2.5"
(2) Inside dia of casing changed from 5.0" to 7.5"

TABLE 5. SEASONAL COEFFICIENT OF PERFORMANCE WITH MULTIPLE, SHALLOW-WELL GROUND COIL

City	Coil		Heating COP		Cooling COP	
	No. of Coils	Depth, ft	Ground-Coil	Air-Source	Ground-Coil	Air-Source
Portland, ME	9	3.3-80	2.39	1.74	3.49	2.30
Atlanta, GA	9	3.3-40	2.76	1.89	1.49	2.35
	9	3.3-50	2.91		2.22	
	9	3.3-60	2.97		2.65	
Washington, D.C.	9	3.3-46	2.71	1.84	2.60	2.32
	9	3.3-60	2.84		2.93	
Pittsburgh, PA	9	3.3-70	2.58	1.79	3.39	2.28
Louisville, KY	4	3.3-110	2.71	1.83	2.24	2.33
Phoenix, AZ	4	3.3-100	2.99	1.97	1.46	2.15
Kansas City, MO	4	3.3-100	2.74	1.77	2.85	2.28
Houston, TX	9	3.3-80	3.06	1.97	1.42	2.31
	9	3.3-100	3.05		1.52	
Seattle, WA	4	3.3-70	2.59	1.92	3.09	2.42

TABLE 6. ECONOMIC COMPARISON WITH HORIZONTAL GROUND COIL

CITY	COIL LENGTH, FT	TOTAL ELECTRIC POWER COST, $ HEATING GROUND COIL	HEATING AIR SOURCE	COOLING GROUND COIL	COOLING AIR SOURCE	ANNUAL POWER SAVINGS, $	INSTALLED COST OF GROUND COIL, $	SIMPLE PAYBACK, YRS	LIFE-CYCLE COSTS AT 2.0 PERCENT DISCOUNT RATE, $ GROUND COIL	AIR SOURCE
PORTLAND, ME	900	739	867	116	103	115	1350	11.7	15326	15864
	1200	694		98		178	1800	10.1	14751	
	1500	671		91		209	2250	10.8	14703	
ATLANTA, GA	700	170	256	409	274	-49	1050	∞	10526	8661
	900	166		312		51	1350	26.3	9173	
	1100	164		273		93	1650	17.7	8789	
WASHINGTON DC	700	257	364	253	200	54	1050	19.4	9386	9219
	1000	245		198		121	1500	12.4	8743	
	1300	240		180		144	1950	13.6	8818	
PITTSBURGH, PA	1200	467	670	143	159	220	1800	8.2	11766	13566
LOUISVILLE, KY	900	203	291	277	216	27	1350	50.5	9200	8288
	1100	198		231		78	1650	21.2	8663	
PHOENIX AZ	1600	86	130	781	556	-181	2400	∞	16584	11216
	1800	87		716		-116	2700	∞	15825	
KANSAS CITY, KY	1000	273	393	309	273	84	1500	17.9	11023	10896
	1100	269		287		110	1650	15.0	10743	
HOUSTON TX	1300	76	117	710	513	-154	1950	∞	14792	10308
	1500	76		626		-71	2250	∞	13726	
SEATTLE, WA	500	116	145	30	18	17	750	44.4	3145	2672
	700	107		21		35	1050	29.6	3141	
MINNEAPOLIS MN	900	714	781	143	139	63	1350	21.5	15368	15042
	1200	669		123		128	1800	14.1	14748	

Note: Coil length not determined solely by economics. Limitations on ground coil fluid temperatures also a factor. (Refer to Ref. 1, Table 16 for pertinent fluid temperatures).

TABLE 7. ECONOMIC COMPARISON WITH DEEP-WELL GROUND COIL

CITY	COIL DEPTH, FT	TOTAL ELECTRIC POWER COST, $ HEATING GROUND COIL	HEATING AIR SOURCE	COOLING GROUND COIL	COOLING AIR SOURCE	ANNUAL POWER SAVINGS, $	INSTALLED COST OF GROUND COIL, $	SIMPLE PAYBACK, YRS	LIFE-CYCLE COSTS AT 2.0 PERCENT DISCOUNT RATE, $ GROUND COIL	AIR SOURCE
PORTLAND, ME	200	651	867	69	103	250	2022	8.1	13792	15864
	280	591		67		312	2870	9.2	13626	
	360	568		69		337	3719	11.0	14070	
ATLANTA, GA	120	168	256	322	274	40	1148	29.1	9163	8661
	180	159		229		142	1785	12.6	8130	
	240	157		207		165	2421	14.7	8382	
WASHINGTON DC	140	244	364	172	200	148	1363	9.2	8168	9219
	200(1)	237		165		162	1742	10.8	8318	
	200	230		152		182	2000	11.0	8247	
	200(2)	227		146		191	2168	11.4	8269	
	260	225		144		194	2636	13.6	8677	
PITTSBURGH, PA	240	433	670	106	159	290	2426	8.4	11246	13566
	280	424		104		302	2851	9.4	11483	
LOUISVILLE, KY	210	190	291	188	216	129	1968	15.3	8148	8288
	250	184		171		152	2393	15.8	8202	
PHOENIX AZ	400	83	130	786	556	-182	3444	∞	17645	11216
	500	83		641		-38	4305	∞	16152	
KANSAS CITY, KY	210	251	393	209	273	206	2016	9.8	9539	10896
	250	244		197		225	2441	10.8	9659	
HOUSTON TX	340	74	117	625	513	-68	2927	∞	14356	10308
	400	74		544		13	3444	270.4	13544	
SEATTLE, WA	200	107	145	13	18	44	1722	39.3	3677	2672
MINNEAPOLIS MN	170	847	781	109	139	-35	1464	∞	17089	15042
	250	636		92		192	2312	12.0	14211	

(1) Inside diameter of casing changed from 5.0" to 2.5"
(2) Inside diameter of casing changed from 5.0" to 7.5"

Note: Coil depth not determined solely by economics. Limitations on ground coil fluid temperatures also a factor. (Table 17 for pertinent fluid temperatures.) (Refer to Ref. 1.)

TABLE 8. ECONOMIC COMPARISON WITH MULTIPLE SHALLOW-WELL GROUND COIL

CITY	COIL NO OF WELLS	COIL DEPTH, FT	TOTAL ELECTRIC POWER COST,$ HEATING GROUND COIL	HEATING AIR SOURCE	COOLING GROUND COIL	COOLING AIR SOURCE	ANNUAL POWER SAVINGS, $	INSTALLED COST OF GROUND COIL,$	SIMPLE PAYBACK, YRS.	LIFE-CYCLE COSTS AT 2.0 PERCENT DISCOUNT RATE,$ GROUND COIL	LIFE-CYCLE COSTS AT 2.0 PERCENT DISCOUNT RATE,$ AIR SOURCE
PORTLAND, ME	9	3.3- 80	634	867	68	103	269	4704	17.5	16170	15864
ATLANTA, GA	9	3.3- 40	175	256	427	274	-71	1734	∞	11568	8661
	9	3.3- 50	165		290		75	2387	32.0	9829	
	9	3.3- 60	162		243		125	3039	24.3	9659	
WASHINGTON DC	9	3.3- 46	248	364	178	200	138	2018	14.6	8976	9219
	9	3.3- 60	236		158		170	2932	17.2	9371	
PITTSBURGH, PA	9	3.3- 70	465	670	107	159	258	3512	13.6	12863	13566
LOUISVILLE, KY	4	3.3- 110	197	291	223	216	87	2241	25.9	9113	8288
PHOENIX AZ	4	3.3- 100	86	130	816	556	-215	4569	∞	19304	11216
KANSAS CITY, KY	4	3.3- 100	254	393	218	273	194	2031	10.5	9751	10896
HOUSTON TX	9	3.3- 80	75	117	833	513	-277	3624	∞	18472	10308
	9	3.3- 100	76		778		-223	4569	∞	18532	
SEATTLE, WA	4	3.3- 70	108	145	14	18	42	1401	33.6	3390	2672

Note: Coil length/depth not determined solely by economics. Limitations on ground coil fluid temperatures also a factor. (Table 18 for pertinent fluid temperatures.) (Refer to Ref. 1.)

TABLE 9. EFFECT OF TRENCHING AND INSTALLATION COSTS ON ECONOMICS WITH HORIZONTAL COIL

CITY	COIL LENGTH, FT	SIMPLE PAYBACK, YRS TRENCHING AND INSTALLATION COST, PERCENT OF BASELINE VALUE			SAVINGS IN LIFE-CYCLE COST, PERCENT* TRENCHING AND INSTALLATION COST, PERCENT OF BASELINE VALUE		
		100	75	50	100	75	50
Portland, ME	900	11.7	9.7	7.8	3.4	4.8	6.2
	1200	10.1	8.4	6.7	7.0	8.9	10.8
	1500	10.8	9.0	7.2	7.3	9.7	12.0
Atlanta, GA	700	-.0	-.0	-.0	-21.5	-19.5	-17.5
	900	26.3	21.9	17.6	-5.9	-3.3	-.7
	1100	17.7	14.8	11.8	-1.5	1.7	4.9
Washington, DC	700	19.4	16.2	13.0	-1.8	.1	2.0
	1000	12.4	10.3	8.3	5.2	7.9	10.6
	1300	13.6	11.3	9.0	4.3	7.9	11.4
Pittsburgh, PA	1200	8.2	6.8	5.5	13.3	15.5	17.7
Louisville, KY	900	50.5	42.1	33.7	-11.0	-8.3	-5.6
	1100	21.2	17.6	14.1	-4.5	-1.2	2.1
Phoenix, AZ	1600	-.0	-.0	-.0	-47.9	-44.3	-40.7
	1800	-.0	-.0	-.0	-41.1	-37.1	-33.1
Kansas City, MO	1000	17.9	14.9	11.9	-1.2	1.1	3.4
	1100	15.0	12.5	10.0	1.4	3.9	6.5
Houston, TX	1300	-.0	-.0	-.0	-43.5	-40.3	-37.2
	1500	-.0	-.0	-.0	-33.2	-29.5	-25.9
Seattle, WA	500	44.4	37.0	29.6	-17.7	-13.0	-8.3
	700	29.6	24.6	19.7	-17.6	-11.0	-4.5

* Savings with respect to costs with an air-source heat-pump system; 2% discount rate.

TABLE 10. EFFECT OF DRILLING AND INSTALLATION COSTS ON ECONOMICS WITH DEEP-WELL COIL

CITY	COIL DEPTH, FT	SIMPLE PAYBACK, YRS DRILLING AND INSTALLATION COST, PERCENT OF BASELINE VALUE			SAVINGS IN LIFE-CYCLE COST, PERCENT* DRILLING AND INSTALLATION COST, PERCENT OF BASELINE VALUE		
		100	75	50	100	75	50
Portland, ME	200	8.1	6.5	4.9	13.1	15.6	18.0
	280	9.2	7.4	5.6	14.1	17.7	21.2
	360	11.0	8.9	6.7	11.3	15.9	20.5
Atlanta, GA	120	29.1	23.5	17.9	-5.8	-3.2	-.7
	180	12.6	10.2	7.7	6.1	10.1	14.1
	240	14.7	11.8	8.9	3.2	8.7	14.1
Washington, DC	140	9.2	7.5	5.7	11.4	14.3	17.1
	200	10.8	8.5	6.2	9.8	13.8	17.7
	200	11.0	8.9	6.7	10.5	14.8	19.0
	200	11.4	9.3	7.3	10.3	14.5	18.8
	260	13.6	10.9	8.3	5.9	11.5	17.1
Pittsburgh, PA	240	8.4	6.7	5.1	17.1	20.6	24.1
	280	9.4	7.6	5.7	15.4	19.5	23.6
Louisville, KY	210	15.3	12.3	9.4	1.7	6.2	10.8
	250	15.8	12.8	9.7	1.0	6.6	12.1
Phoenix, AZ	400	-.0	-.0	-.0	-57.3	-51.6	-45.9
	500	-.0	-.0	-.0	-44.0	-36.9	-29.7
Kansas City, MO	210	9.8	7.9	6.0	12.5	16.0	19.6
	250	10.8	8.7	6.7	11.4	15.7	20.0
Houston, TX	340	-.0	-.0	-.0	-39.3	-34.0	-28.7
	400	270.4	220.1	169.9	-31.4	-25.2	-19.0
Seattle, WA	200	39.3	32.0	24.7	-37.6	-25.6	-13.7

* Savings with respect to costs with an air-source heat-pump system; 2% rate.

TABLE 11. EFFECT OF DRILLING AND INSTALLATION COSTS ON ECONOMICS WITH MULTIPLE SHALLOW-WELL COIL

CITY	NO. OF WELLS	COIL DEPTH, FT	SIMPLE PAYBACK, YRS DRILLING AND INSTALLATION COST, PERCENT OF BASELINE VALUE			SAVINGS IN LIFE-CYCLE COST, PERCENT* DRILLING AND INSTALLATION COST, PERCENT OF BASELINE VALUE		
			100	75	50	100	75	50
Portland, ME	9	3.3-80	17.5	13.8	10.0	-1.9	4.4	10.7
Atlanta, GA	9	3.3-40	-.0	-.0	-.0	-33.6	-29.5	-25.5
		-50	32.0	24.5	18.8	-13.5	-7.8	-2.1
		-60	24.3	19.3	14.2	-11.5	-4.2	3.1
Washington, DC	9	3.3-46	14.6	11.6	8.7	2.6	7.1	11.5
		-60	17.2	13.7	10.1	-1.6	4.9	11.5
Pittsburgh, PA	9	3.3-70	13.6	10.8	8.0	5.2	10.6	15.9
Louisville, KY	4	3.3-110	25.9	20.6	15.4	-10.2	-4.5	1.0
Phoenix, AZ	9	3.3-100	-.0	-.0	-.0	-72.1	-67.2	-58.1
Kansas City, MO	4	3.3-100	10.5	8.3	6.2	10.5	14.3	18.1
Houston, TX	9	3.3-80	-.0	-.0	-.0	-79.2	-72.1	-65.0
		-100	-.0	-.0	-.0	-79.8	-70.8	-61.8
Seattle, WA	4	3.3-70	33.6	26.8	20.0	-26.9	-16.3	-5.7

* Savings with respect to costs with an air-source heat pump system; 2% discount.

TABLE 12. ECONOMICS OF HORIZONTAL SYSTEMS WITH BEST-CASE SOIL PROPERTIES

CITY	COIL LENGTH, FT	TOTAL ELECTRIC POWER COST, $ HEATING GROUND COIL	HEATING AIR SOURCE	COOLING GROUND COIL	COOLING AIR SOURCE	ANNUAL POWER SAVINGS, $	INSTALLED COST OF GROUND COIL, $	SIMPLE PAYBACK, YRS	LIFE-CYCLE COSTS AT 2.0 PERCENT DISCOUNT RATE, $ GROUND COIL	AIR SOURCE
PORTLAND, ME	900	730	867	95	103	145	1350	9.3	14840	15864
WASHINGTON DC	1000	237	364	185	200	142	1500	10.6	8401	9219
HOUSTON TX	1500	73	117	546	513	12	2250	195.5	12370	10308
	1700	73		516		41	2550	61.8	12184	
	2000	73		487		70	3000	42.8	12162	
	2300	73		468		89	3450	38.8	12304	
	2500	73		459		98	3750	38.2	12452	

TABLE 13. INFORMATION ON THERMAL RECOVERY RUNS

Run No.	Description	Heat Pump Size, Tons	On-Off Period, Hr.	Seasonal COP* Htg.	Htg**/Clg
		Horizontal Coil (1200-ft-long x 4-ft-deep coil)			
1	Effect of HP Size	3.0	1.	2.52	2.04/2.60
2	"	3.5	1.	2.34	1.99/2.53
3	"	4.0	1.	2.26	2.05/2.43
4	"	4.5	1.	2.35	2.27/2.32
5	"	5.0	1.	2.31	2.28/2.22
6	Effect of on-off Periods	5.0	0.5	2.13	2.08/2.24
7	"	"	0.75	2.07	2.01/2.22
8	"	"	1.00	2.15	2.12/2.21
9	"	"	1.25	2.12	2.08/2.20
		Single, Deep-well Coil (250-ft-deep coil)			
10	Effect of HP Size	2.8	1.67	2.49	1.90/3.25
11	"	3.5	"	2.37	2.04/3.29
12	"	4.2	"	2.28	2.12/3.33
13	"	4.9	"	2.18	2.11/3.38
14	"	5.6	"	2.13	2.11/3.40
15	Effect of on-off Period	"	1.33	2.14	2.12/3.43
16	"	"	1.00	2.15	2.12/3.47
17	"	"	0.67	2.16	2.14/3.53

* Includes energy for pump
** Includes energy for electric-resistance heating

TABLE 14. ECONOMIC DATA WITH THERMAL RECOVERY

CITY	Run No.	TOTAL ELECTRIC POWER COST, $ HEATING GROUND COIL	HEATING AIR SOURCE	COOLING GROUND COIL	COOLING AIR SOURCE	ANNUAL POWER SAVINGS, $	INSTALLED COST OF GROUND COIL, $	SIMPLE PAYBACK, YRS	LIFE-CYCLE COSTS AT 2.0 PERCENT DISCOUNT RATE, $ GROUND COIL	AIR SOURCE
					Horizontal Coil (1200 ft long)					
MINNEAPOLIS MN	1	635	781	118	139	167	1800	10.8	14110	15042
	2	649		122		149	1800	12.1	14405	
	3	630		126		163	1800	11.0	14170	
	4	570		133		218	1800	8.3	13279	
	5	568		138		214	1800	8.4	13336	
	6	622	781	137	139	161	1800	11.2	14209	15042
	7	644		138		138	1800	13.1	14588	
	8	610		138		172	1800	10.5	14028	
	9	622		139		158	1800	11.4	14251	
					Single Deep-Well Coil (250 ft deep)					
	10	682		94		144	2312	16.0	14998	
	11	637		92		191	2312	12.1	14237	
	12	614		93		213	2312	10.9	13879	
	13	619		90		211	2312	10.9	13899	
	14	621		91		208	2312	11.1	13952	
	15	615		89		215	2312	10.7	13831	
	16	611		88		221	2312	10.5	13744	
	17	604		87		229	2312	10.1	13613	

TABLE 15. INFORMATION ON RUNS WITH DIRECT GROUND-LOOP COOLING

Run No.	Description	Coil Length or Depth, ft.	Coil Depth, ft.	Seasonal COP* Htg.	Seasonal COP* Htg**/Clg
		Horizontal Coil			
1	GC Htg/Clg	900	4	2.23	1.88/2.20
2	GC Htg/Clg	1200	4	2.37	2.03/2.58
3	GC Htg, HX Clg	900	4	2.23	1.88/3.85
4	GC Htg, HX Clg	1200	4	2.37	2.03/4.61
5	GC Htg, HX Clg	1500	4	2.46	2.14/5.17
6	GC Htg, HX Clg	1800	4	2.53	2.22/5.65
7	GC Htg, HX Clg	2100	4	2.55	2.26/6.08
8	GC Htg/Clg	900	6	2.27	1.93/2.18
9	GC Htg/Clg	1200	6	2.43	2.11/2.61
10	GC Htg/HX Clg	900	6	2.27	1.93/4.75
11	GC Htg/HX Clg	1200	6	2.43	2.11/5.83
12	Same as Run 10 except HX cost is 1/2 that for Run 10				
		Single Deep-Well Coil			
13	GC Htg/Clg	170	--	1.85	1.55/2.85
14	GC Htg/Clg	250	--	2.42	2.08/3.38
15	GC Htg/HX Clg	170	--	1.85	1.55/5.15
16	GC Htg/HX Clg	250	--	2.42	2.08/9.28
17	GC Htg/HX Clg	330	--	2.64	2.36/12.32

* Includes energy for pump.

** Includes energy for electric-resistance heating.

TABLE 16. ECONOMICS OF GROUND-COUPLED SYSTEMS WITH DIRECT GROUND-LOOP COOLING

Run No.	Coil Length, ft	Total Electric Power Cost, $ Heating Ground Coil	Heating Air Source	Cooling Ground Coil	Cooling Air Source	Annual Power Savings, $	Installed Cost of Ground Coil, $	Simple Payback, Yrs	Life-Cycle Costs At 2.0 Percent Discount Rate, $ Ground Coil	Air Source
				Horizontal Coil						
1	900	690	781	139	139	92	1,350	14.7	14,893	15,042
2	1,200	638		119		153	1,800	11.0	14,177	
3	900	689		58		174	1,740	10.0	13,943	
4	1,200	638		58		229	2,190	9.6	13,489	
5	1,500	605		50		265	2,640	10.0	13,355	
6	1,800	584		48		289	3,090	10.7	13,415	
7	2,100	573		45		301	3,540	11.8	13,659	
8	900	672	781	141	139	108	1,755	16.3	15,038	15,042
9	1,200	613		117		189	2,340	12.4	14,292	
10	900	672		53		195	2,145	11.0	13,996	
11	1,200	613		47		259	2,730	10.5	13,535	
12	900	672		53		195	1,950	10.0	13,801	
				Single Deep-Well Coil						
13	170	835	781	108	139	-22	1,464	∞	16,874	15,042
14	250	623		91		206	2,312	11.2	13,987	
15	170	835		48		37	1,854	50.4	16,295	
16	250	623		33		264	2,702	10.3	13,434	
17	330	549		25		346	3,551	10.3	12,929	

TABLE 17. EFFECT OF DESUPERHEATER WATER HEATING WITH HORIZONTAL COIL

City	Baseline (Table 6) Lowest Simple Payback, yrs.	Baseline Coil Length, ft.	Baseline Installed Cost	With Desuperheater Water Heating Lowest Simple Payback, yrs.	With Desuperheater Water Heating Coil Length ft.	With Desuperheater Water Heating Installed Cost**
Atlanta	17.7	1100	$1650	6.8	700	$1300
Washington, D.C.	12.4	1000	1500	7.0	700	1350
Pittsburgh	8.2	1200	1800	6.7	900	1600
Louisville	21.2	1100	1650	9.0	900	1650
Kansas City	15.0	1100	1650	7.3	850	1575

* With desuperheater water heater

** Includes cost of desuperheater water heater

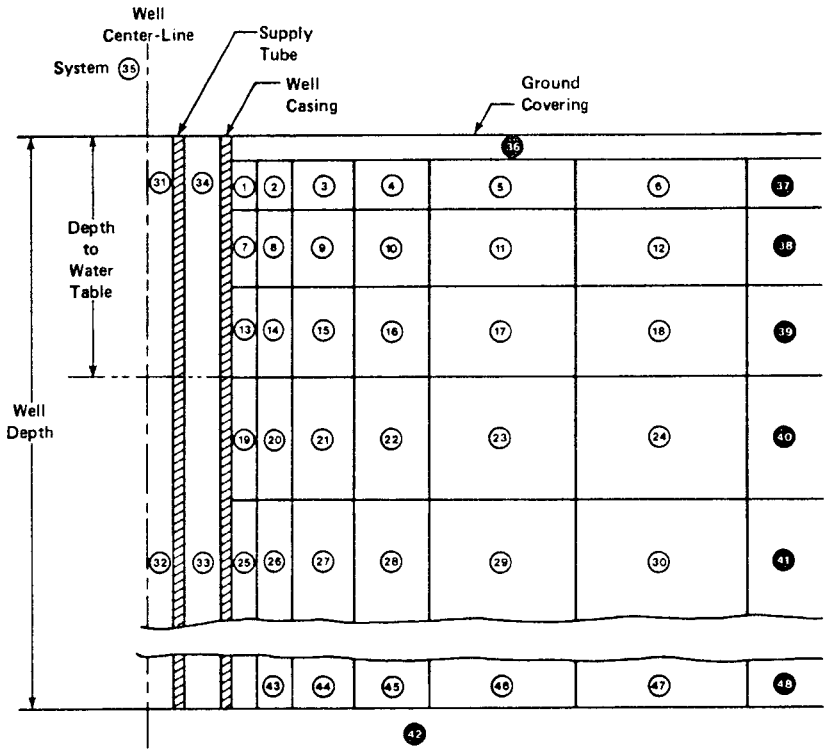

FIGURE A-1. NODAL GEOMETRY FOR DEEP-WELL GROUND COIL MODEL

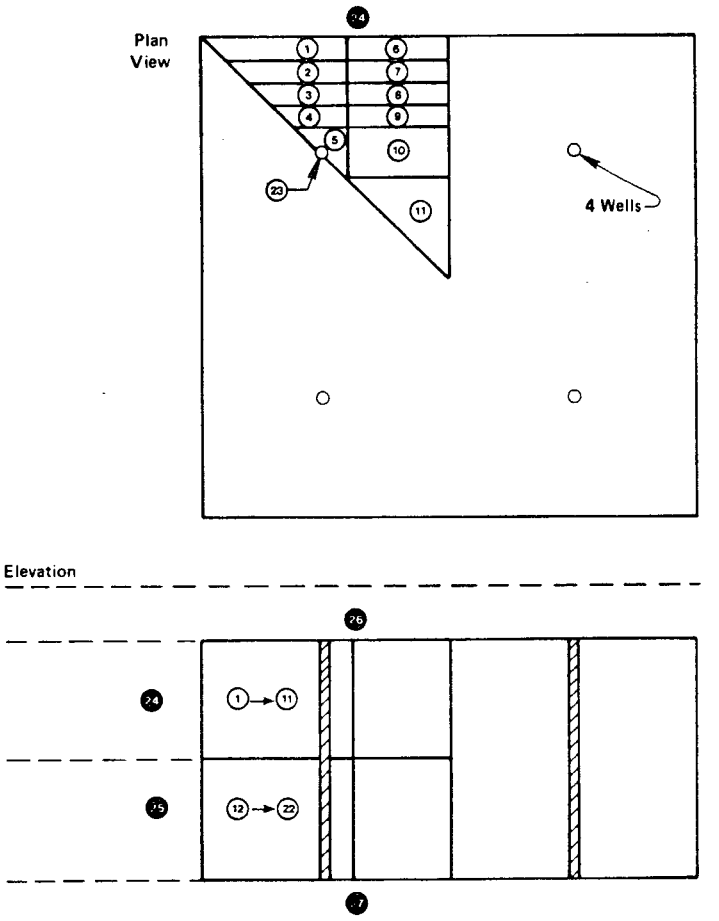

FIGURE A-2. NODAL GEOMETRY FOR MULTIPLE SHALLOW-WELL MODEL WITH FOUR WELLS

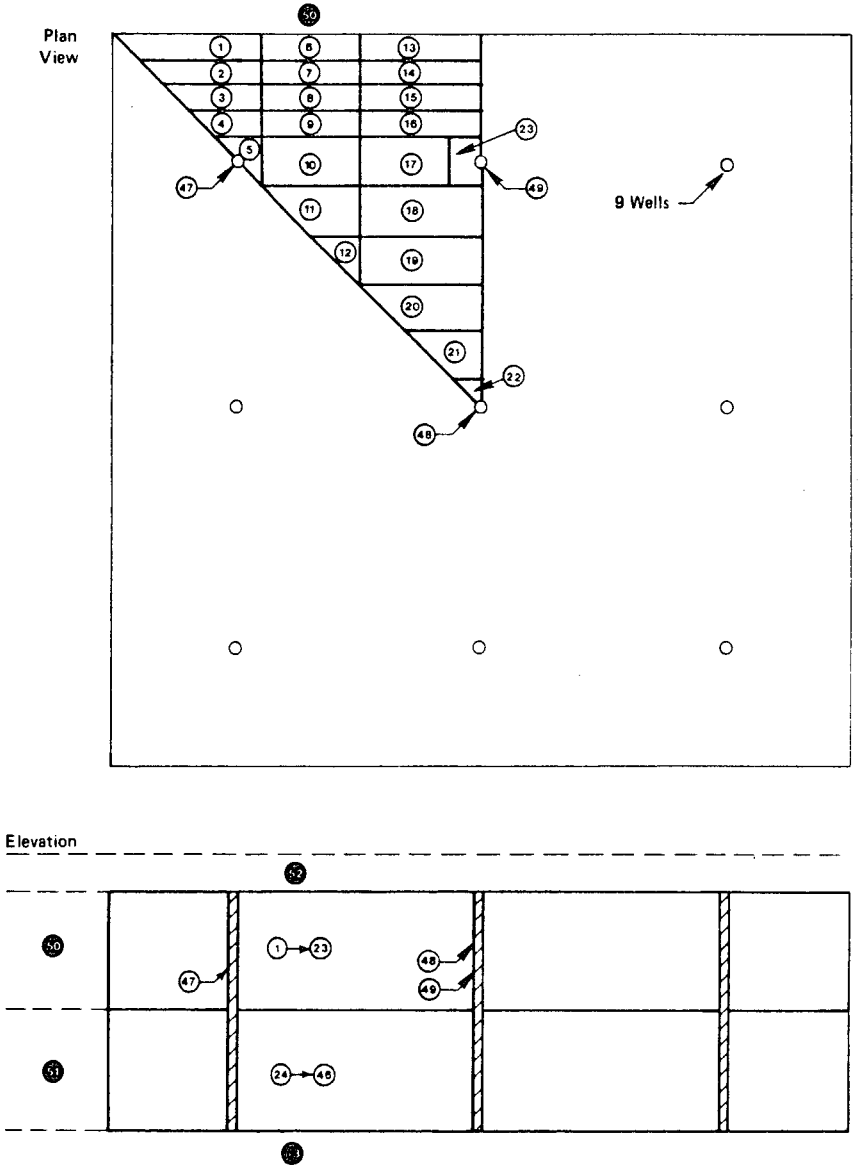

FIGURE A-3. NODAL GEOMETRY FOR MULTIPLE, SHALLOW-WELL MODEL WITH NINE WELLS

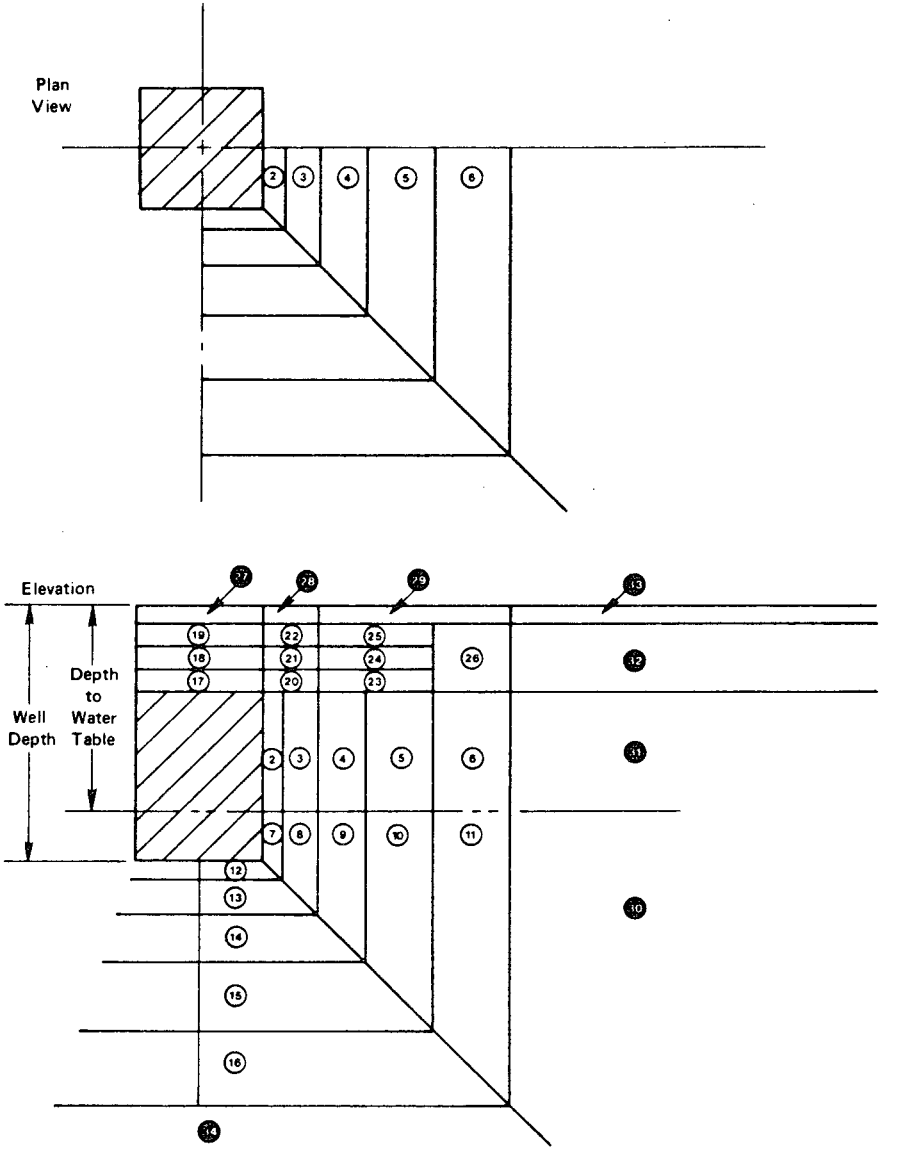

FIGURE A-4. NODAL GEOMETRY FOR OUTER "STORE" OF MULTIPLE, SHALLOW-WELL MODEL

OPTIMIZATION OF A GROUND COUPLED HEAT PUMP

by V. D. Baxter, and M. A. Catan

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OPTIMIZATION OF A GROUND COUPLED HEAT PUMP

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Summary

This is the first comprehensive report on a project to optimize a ground coupled heat pump (GCHP) system for a northern climate. The project received its impetus from the well-circulated question of how well such a system could perform with a heat pump which was designed specifically for ground coupled heat pump systems operating in a northern climate. The project originally sought to optimize the Annual Performance Factor (annual performance factor, or APF, is the ratio of heating and cooling load of a house to the electrical energy consumed in meeting that load) of a heat pump system with a given ground coupling device. This objective grew in several stages to the present objective: to minimize the life cycle cost for the entire system. In order to achieve this objective a number of modeling tools were developed which will likely be of interest in their own right.

For the present we are only concerned with systems with horizontal ground coils in a given house[1] in the Pittsburgh area. The types of components used in the heat pump are essentially conventional, only their relative dimensions may be considered unconventional. Subsequent work will consider other climatic regions and ground coupling devices and unconventional heat pump components and, possibly, other 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 a specified configuration and operating conditions. A predetermined 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 of parameters for which the highest coefficient of performance (COP) is predicted by the heat pump model. The parameters have fixed ceilings to limit the expected cost of the optimized heat pump. In the present study, the set of parameters to be varied include some pertaining to the ground coil as well as those 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, and a parameterized cost model.

The quest for this automatically derived configuration for a ground coupled heat pump system has been marked by several excursions. First, the heat pump steady-state performance was optimized for one heating source temperature and one cooling source temperature. Second, the sensitivity of steady-state COP to various heat pump parameters was explored. Third, a number of advanced component options were examined in a cursory way to see if it was worthwhile to develop the modeling capability needed to include them in a full scale optimization. Fourth, a matrix of APF values for various combinations of ground coil length, heat pump size, and heat pump COP's was generated. At the time of this writing we have not proceeded with a full scale life cycle cost minimization using the optimizer but the preceeding work has given an approximate picture of the expected results.

The modeling programs developed for this project can be operated independently and are installable on some microcomputers. The heat pump model is the largest. It was modified to simulate water source heat pumps via the installation of a set of experimentally validated water-refrigerant heat exchanger subroutines. The cost model was developed by Friedrich Climate-Master Corporation under a subcontract to Brookhaven National Laboratory (BNL). From the same set of inputs used by the heat pump model it calculates the first cost of the heat pump to the consumer. The model contains no overt cost information. The ground coil model, called FTECM (Fast Transient Earth Coil Model) accounts for cycling operation of the ground coil and uses a finite element approach to model three dimensional underground heat flow. It is a speeded-up stand-alone version of the newly refined GROCS/TRNSYS program[4] which has been compared to exact solutions to conduction problems and to experimental data from the BNL Test House. A subroutine has been added which generates a 3-D grid for any serpentine horizontal earth coil given the desired length, depth, pipe spacing, and number of parallel passes making this the only input, aside from thermal loads, required to simulate a horizontal ground coil.

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Method

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 resulting optimized heat pump design iteratively, to generate a new bin model. Since these optimizations represent a substantial computer-time burden an alternative 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 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 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 exchanger is a very cost and performance sensitive component of a water source heat pump[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 performance: 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 have 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 preposed 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. The laboratory heat pump, or Subcomponent Test Stand (STS; see Figure 1) is a water-to-water heat pump with a

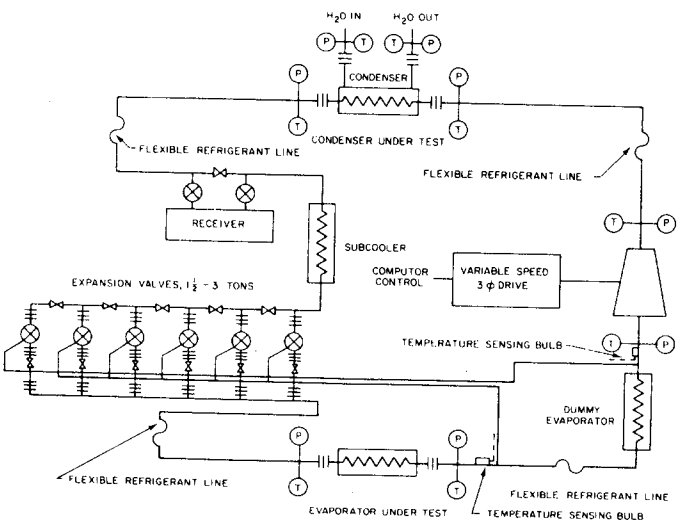

Fig.1. Subcomponent test stand configured for evaporator and condenser testing.

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.

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 the experimental data is shown graphically in Figure 2. For each set of pressure drop and heat transfer correlations to be tried, an optimizer routine would find the free constants which minimized the error between the experimental data and the subroutines' predictions for all the test conditions. In this way the pressure drop correlations and the heat transfer correlations were fitted simultaneously.

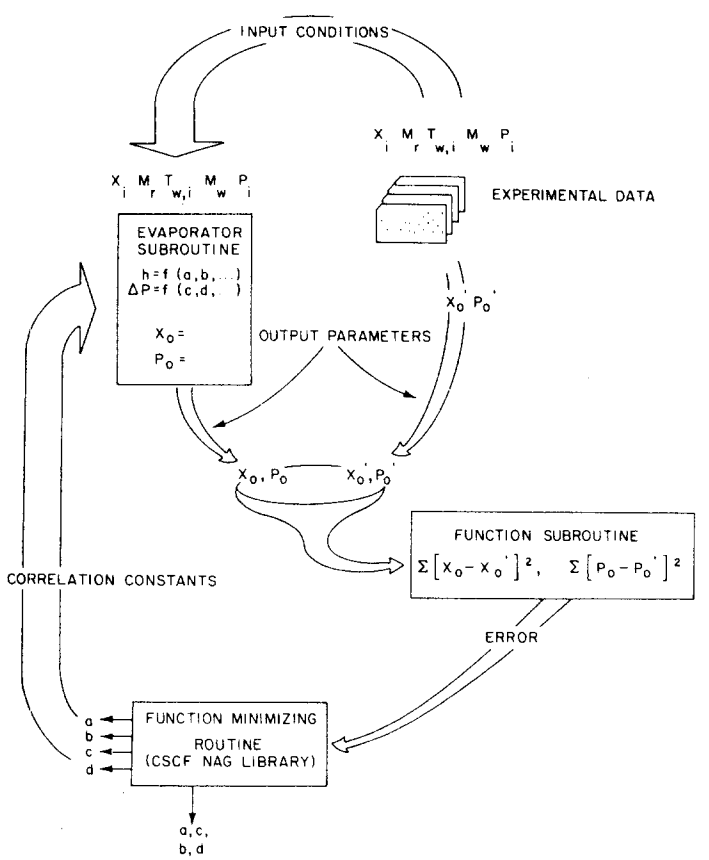

Fig. 2. Procedure for fitting evaporator model to experimental data.

In the evaporating model, separate correlations are used for the three heat transfer regions: wet-wall forced convection evaporation, dry-wall forced convection evaporation, and single phase forced convection. In the condensing model three heat transfer regions are also assumed: incipient condensation with superheated gas, forced convection two phase condensation with saturated vapor temperatures, and subcooled single phase forced convection. In both models, pressure drop is calculated with the assumption of non-separated flow.

Much more detail on the heat exchanger subroutines can be found in reference 5.

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 desireable. The approach used by the model GROCS is a good basis for the ground coil simulation routine required in the 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 device 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.

FTECM uses a modified version of GROCS (which does the same thing as the original only faster) to model the underground heat flow everywhere except in the immediate vicinity of the pipe where a routine called TYPE40 takes over. The simulation of an isolated buried pipe would normally require small timesteps and small earth subdivisions and thus would be costly to run. TYPE40 finds the entering source temperature of the ground coil via a resistance-capacitance R-C) model of pipe fluid-earth interaction which has been shown to give accurate results through comparisons with analytical solutions to simple problems[5].

In the R-C network model the pipe wall is considered to be a resistance in parallel with a capacitance. The interaction between the pipe wall and the surrounding earth is modeled as a pure resistance. This network links the ground coil to the GROCS program. The governing differential equation for the pipe-fluid network is:

$$Q = -C\dot{\theta} - 1/R\theta \qquad (1)$$

where Q is the heat rate during heat withdrawal, C is the pipe fluid capacitance, θ is the temperature difference across the pipe wall, and R is the pipe wall resistance.

The solution of this equation leads to the time-average source temperature during heat extraction

$$\theta_I = 1/t_1 \int_0^{t_1} \theta dt$$
$$= -\dot{Q}_o R + \frac{\dot{Q}_o R^2 C(1-e^{-(t_2-t_1)/RC})(1-e^{-t_1/RC})}{t_1(1-e^{-t_2/RC})} \quad . \qquad (2)$$

The ratio of θ_I to θ_c

$$\theta_I/\theta_c = \frac{1 - \frac{(1-e^{-w})\ (1-e^{-\emptyset})}{w[1-e^{-(w+\emptyset)}]}}{[w/(w+\emptyset)]} \qquad (3)$$

which is the ratio between the ΔT's for cycling and steady heat withdrawal, is factored into the pipe wall resistance and the steady state load used to compute the temperature drop between the adjacent earth block and the pipe fluid.

An energy balance is set up at each point along the pipe leading to the outlet temperature from a pipe of length L, initial fluid temperature, and adjacent earth block temperature T_1 giving

$$T_{E,i} = T_o e^{-(1/R\dot{m}Cp)} + T_{1i}[1-e^{-(1/R\dot{m}Cp)}] \qquad (4)$$

where T_o is the fluid temperature entering the ground coil, T_{1i} is the block temperature adjacent to the ith pipe segment, and $\dot{m}$ is the flowrate. A succession of blocks and pipe lengths is normally used so the inlet temperature for each succeeding pipe length is set equal to the outlet temperature of the preceding pipe length. This leads ultimately to an equation for the ground coil return temperature for the entire succession of pipe lengths. The derivation of these formulas and the justification for the assumptions made in the process can be found in reference 4.

The FTECM program has been run using load data from an actual ground coupled heat pump installation at BNL to determine the accuracy of the model's predictions. The error given by equation (5) was found to be slightly more than two percent for the heating season and approximately six percent for the cooling season.

$$\frac{\sum_i |COP_{c,i} - COP_{m,i}|\ Qi}{\sum_j Q_j} \qquad (5)$$

where the subscripts i and j refer 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 the air coil face area. All of the major component costs, which account for eighty percent 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 coil fin pitch
- air coil tube wall thickness
- air coil fin thickness
- air flowrate
- external static pressure
- water coil length
- water coil shell outer diameter
- unit aspect ratio
- labor cost
- overhead rate
- cost of aluminum
- cost of copper

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.

Results

Steady State Optimization

As mentioned before, the fully automated optimization of a ground coupled heat pump system using life cycle cost as the output to be minimized requires a great deal of computer time, making program bugs quite painful. To acquire experience with the optimizer routine, and to learn something about the design tradeoffs to be dealt with, a set of steady-state optimizations was performed. The cost of the heat pump was constrained to a fixed value while the COP was maximized by the optimizer for a fixed source temperature and compressor dispalcement. The optimizer tries to build the best heat pump it can for the money. Since the displacement is fixed the optimizer can't improve the COP by reducing the capacity of the heat pump. It must weigh various tradeoffs in system features to find the optimum balance by actually trying various combinations out on the heat pump model. For the steady-state optimization runs which have been performed, the COP's, and the values of the major parameters which were varied, are given in Table 1. The base case units which represent the configuration of an actual purchasable water source heat pump were run on the heat pump model and the resulting predicted COP are shown in Table 1. The heat pump was optimized for the same first cost and compressor displacement as the base case unit, and subsequently for 1300.00, 1400.00, and 1500.00 dollar first costs.

The COP improvements which are "bought" with the incremental cost increase are justified if the energy conserved during the life cycle is less costly than the improvement. A rough calculation of heating mode SPF using a 60 MMBtu load, steady-state performance at 30°F suggested that performance improvements costing more than 150 dollars are not justified for a 3.5 year simple payback.

Several trends are evident in the way the optimizer improved the COP of the base case heat pump. For the heating mode, where no cost was added to the base case, the water coil was made smaller and the air coil bigger. The aspect ratio of the air coil was not changed dramatically. For the cooling mode, where no cost was added to the base case the, the face area of the air coil was increased and the depth decreased. After the initial change in the balance of heat transfer surface in the water coil to that in the air coil, the size of both heat exchangers increased steadily as the cost constraint rises. The depth of the air coil hits the upper limit of 4.0 tube rows in the heating mode case. The water coil's preferred configuration for cooling is larger and thinner than that for heating at each cost step except for the 1500 dollar one where the water coil becomes smaller and fatter.

The optimum combination of parameters for the heating mode often differs from that for the cooling mode. Whether this truly leads to a "conflict of interests" can be determined by studying the sensitivity of the COP to changes in the various parameters. This is a way of making up for the "blindness" of the automated optimization approach.

Steady State Sensitivity Studies

The optimizer finds the best configuration without giving any feeling for the sensitivity of the COP to the various parameters. From a knowledge of this one may find ways to combine some of the heat pump parameters into lump parameters and/or eliminate them entirely, making the optimizations much cheaper to run. Since the steady-state optimization considers only one mode of operation it cannot make tradeoffs between changes which improve heating performance and changes which improve cooling performance. A knowledge of the sensitivity of COP to the various parameters in the vicinity of the optimum configuration can suggest ways of compromise which do not affect significantly the performance in either mode.

Table 1. Steady State Optimization Runs

	Cost (Dollars)	S.S. COP	S.S. Capacity (Btu/hr)	Air Coil Face Area (ft^2)	Air Coil Tube Rows	Water Coil Length (inches)	Water Coil Shell Dia. (inches)
HEATING MODE:	1216.00*	2.89	26600	3.33	3.0	168	1.39
	1216.00	3.00	26000	3.53	3.3	144	1.36
	1300.00	3.23	26100	4.23	4.0	152	1.44
	1400.00	3.41	26000	5.04	4.0	190	1.54
	1500.00	3.49	26500	6.73	4.0	192	1.55
COOLING MODE:	1216.00*	2.82	33200	3.33	3.0	168	1.39
	1216.	3.30	34900	3.86	2.9	172	1.20
	1300.	3.42	36800	4.61	3.7	217	1.26
	1400.	3.62	37100	5.19	3.7	227	1.47
	1500.	3.63	37900	6.78	4.0	207	1.52

* Unoptimized; off-the-shelf; base case configuration

To obtain this sensitivity information contour plots of the COP, capacity, and cost versus two independent optimization parameters were made. Two parameters were chosen at a time and the parameters varied about the optimum values determined by the optimizer. The contour plots show lines of constant COP, capacity, or cost in the range of values of the two independent parameters. Superimposed on these contour plots is a curve of constant cost showing the cost constraint used by the optimizer. By following this constant cost curve one can find a cost optimum for the two variables plotted. If the cost curve and the COP curves are roughly parallel it means that the COP needn't be compromised to trade one parameter for another at constant cost. An example of this type of situation is described below.

Figures 3 and 4 are plots of COP versus air coil face area and number of tube rows for heating and cooling modes respectively. In this case the heating COP is almost completely insensitive to trading face area for depth at constant as evidenced by the fact that the COP curves are parallel to the cost curves. On the other hand, the cooling COP is markedly affected by the same tradeoff. This means that the cooling mode can be ameliorated without hurting the heating performance by choosing the cooling model optimum for the two sets of variables.

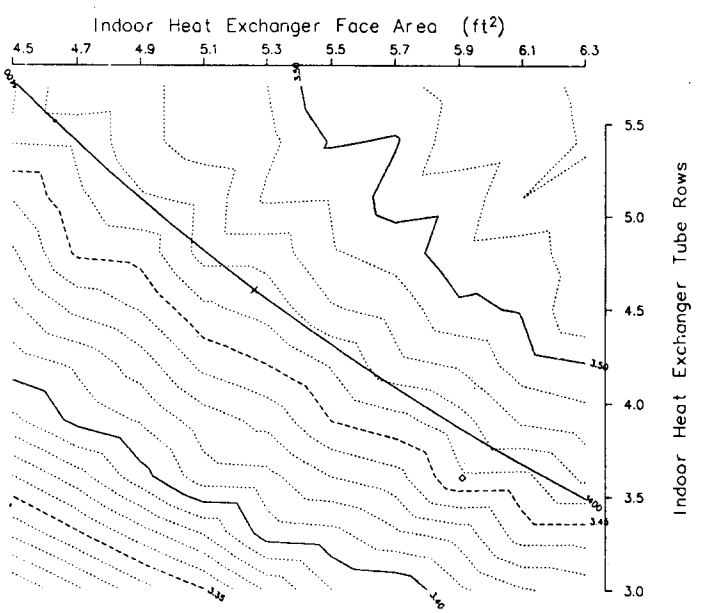

Fig. 3. Contour plot of COP vs air coil face area and number of tube rows for heating mode.

A plot of COP versus air coil face area and water coil length where diameter of the water coil is proportional to the length was made. What we wanted to see here is how the COP of the heat pump changes as heat trasfer surface in the water coil is traded for heat transfer surface in the air coil along a line of constant cost. The highest COP obtainable for the ranges of the two variables, but within the cost constraint, is the point at which the cost curve is tangent to the curves of constant COP. In this case, the optimum for the heating mode and the optimum for the cooling mode are very close. This suggests that, for these two

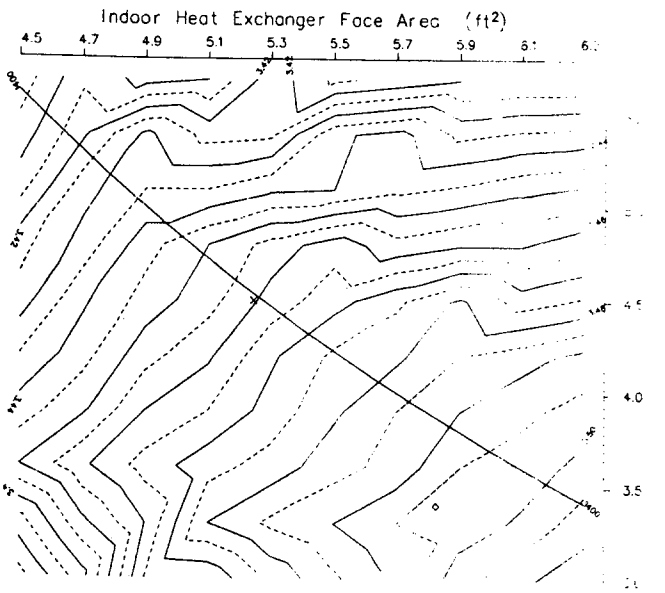

Fig. 4. Contour plot of COP vs air coil face area and number of tube rows for cooling mode

parameters, no compromise in cooling performance need be made for the sake of heating performance and vice versa.

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, in the manner described above, 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 1216, 1300, 1400, and 1500 dollar optimized base cases respectively.

Table 2. System Performance and Economics for Heat Pump Optimized with a Cost Constraint of 1216 Dollars*

Capacity Scale Factor	Heat Pump Cost (dollars)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Net Present Value (dollars)	Simple Pay-back 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 Dollars*

Capacity Scale Factor	Heat Pump Cost (dollars)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Net Present Value (dollars)	Simple Pay-back 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 Dollars*

Capacity Scale Factor	Heat Pump Cost (dollars)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Net Present Value (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	8453	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 Dollars*

Capacity Scale Factor	Heat Pump Cost (dollars)	Ground Coil Length (feet)	Energy Use (kWh) Heating I^2R	Heating Total	Cooling	Performance Factors Heating	Cooling	Annual	Net Present Value (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

$C_{d,H} = (0.3022 - 0.00235\ Q_{EWT})$

Cost Constraint	EWT	COP_H	Capacity
1216	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

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

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 two columns in each table give, what we refer to as, the simple payback (SPB) relative to an alternative system. This is the difference in the capital cost of the system minus that of the alternative 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 net present value (NPV) 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 NPV 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 NPV, 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 analysis is deficient in several respects. First the ideal system chosen has struck two independent parameter boundaries; the smallest ground coil and the highest efficiency heat pump. Second and third, the number of independent variables and their resolution are limited. The last phase of this project, the life cycle cost system optimization, will substantially overcome these deficiencies. The number of independent variables which will be optimized will be much greater and the step size used by the optimizer limited only by the resolution of the models.

Another point to note is that soil moisture freezing is not taken into account in these simulations. The effect of taking freezing into account should be to boost the performance of the systems with small ground coils more than that of systems with large ground coils. We intend to reiterate the above procedure using the refined FTECM program, which will include freezing. In addition the system optimization will probably use this new model.

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. The changes in relative capacity are somewhere in-between these two in their cost/performance effect.

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.

Because of the need for fast run times in the modeling tools created for this project, all will run on small computers. The heat pump model, which is the largest, requires no more memory than most modern microcomputers possess. It is hoped that these models will be of value to people involved in ground coupled system design.

References

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WATER SOURCE HEAT PUMP CLOSED LOOP PERFORMANCE OF IOWA SUGGESTED EQUIPMENT SIZING ADJUSTMENTS DUE TO WINTER AND SUMMER LOOP TEMPERATURES

by Alvin C. Bierbaum

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Presented at the 7th Heat Pump Technology Conference at Tulsa, Oklahoma, October 15-16, 1984

WATER SOURCE HEAT PUMP CLOSED LOOP PERFORMANCE OF IOWA
SUGGESTED EQUIPMENT SIZING ADJUSTMENTS DUE TO WINTER AND SUMMER LOOP TEMPERATURES

by

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Background

I became interested in marketing water source heat pumps in 1981 after concluding a two year study comparing various types of heating systems. The table below lists our findings for 120 houses of similar size and construction.

Heating System	Avg. KW Demand	Avg. KWHR Used	Avg. Size (sq.ft.)	Avg. Age (years)	Avg. Window Area (sq.ft.)	% with Heated Base-ments
Electric Furnace	16.84	17502	1811	15.5	196	100%
Air-Air HP	20.87	16287	1914	18.0	234	100%
Radiant	12.37	14678	1628	30.8	193	50%
GWHP	4.26	6257	2640	3.5	344	100%
Add-on HP	3.03	4250	1847	35.0	234	100%
Dual-Fired Electric Furnace	19.01	27989	1785	32.6	214	100%

The above results pretty well convinced us that air-to-air heat pumps and electric furnaces are a poor load for us and a poor deal for our consumers. The groundwater heat pump results, on the other hand, convinced us that this is the heating-cooling system of the future.

If we put dollars and cents to the above information, the Iowa RECs serve around 150,000 rural consumers in Iowa. We presently have around 15% of our members using electric heating, causing a winter peak situation for our wholesale power supplier. If we were to actively market air-to-air heat pumps and were to induce 15% more of our members to convert to air-to-air heat pumps, they would create a 475 MW new peak demand on the coldest day of the year. At $1,500/KW, we would have to build a 3/4-billion dollar power plant that would not generate enough revenue to pay for the interest on the investment because there would be little or no spring, summer and fall utilization of the facility. In reviewing these sobering statistics, we decided that promoting water source heat pumps was in our, as well as our consumers, best interest.

1982-83 Marketing Experiences

The Iowa RECs started actively marketing water source heat pumps the summer of 1982 at the Iowa State Fair by having a working model using a barrel of water. The response really surprised us, with over 1,000 people showing a lot of interest. Unfortunately, this left us with a myriad of questions we could not answer.

Our first flyer basically listed the economic advantages of water source heat pumps and nothing else. Realizing that our rural area was the most logical place to start promoting and installing water source heat pumps (because of better competing fuel advantages, such as oil and propane), and having the open spaces for a loop system or an existing well, we took it upon ourselves to develop a handout that appeared to resolve most of the technical questions that were coming up. This has become my best seller (a 15-page question and answer booklet on water source heat pumps). We have given out over 3,000 copies and intend to continue to make it available to our members at no charge.

By the time the Iowa State Fair rolled around in 1983, we were much better prepared to deal with the many technical questions that were being raised. The year 1983 was a year of confusion for most of us. There was a lot of conflicting information coming from contractors, well drillers, and manufacturers concerning loop size and length, parallel vs. series loops, etc. Most of our member service people recognized also that there needed to be an easier way to perform heat loss/heat gain calculations for equipment sizing.

These problems all came to a head for me late last summer when I installed a water source heat pump in my own home to replace a failed air-to-air heat pump. I acted as my own contractor and discovered quite rapidly that I had a lot to learn about plumbing, pumps, well drilling, etc. After considerable struggling, I accomplished the task.

I made some mistakes, and as a result, decided that what the industry needed was an installation handbook on the subject. Since there was not a complete handbook available, I spent last winter in my spare time writing one. It has been well received, with copies being sold all over the United States and Canada. We went public with the handbook last May when we used it as a textbook for a water source heat pump workshop that was attended by over 150 REC personnel, dealers and contractors in Iowa.

Closed Loop Performance Questionnaire Results

One of the main topics of discussion at our workshop pertained to earth coupled design considerations.

We presented projections of loop water temperatures for various sized load, loop sizes and lengths from computer runs made at Oklahoma State University and then compared them to the results obtained from a closed loop questionnaire that we had sent out earlier. Note copy of questionnaire below:

Geo-graphic Loca-tion	Heat Pump Size ton	Total Loop Length ft.	Pipe Size ins.	Loop Depth ft.	Pipe Type	Jan. Loop Temp. (F.)	No. of Systems
			HORIZONTAL LOOP				
SW Iowa	3	1,500	1-1/2	6	PE	37	5
W Iowa	3	2,500	1-1/4	6&4*	PB	28-30	10
S Minn	4	2,280	1-1/2	10	PE	37	5
SW Iowa	3	1,700	1-1/2	7	PE	42	24
S Iowa	3	2,200	1	6&4*	PE	30	1
N Iowa	2-1/2	1,100	2	6&4*	PB	36	1
			VERTICAL LOOP				
N Cen Iowa	3-1/2	1,200	3/4	4	PB	31	2
SW Iowa	3	750	1-1/2	3	PB	29-35	50

*Two pipes in the same trench

General Comment: The vertical loops appear to have faster heat recovery than the horizontal loops.

It is my opinion that the results of the questionnaire correlate quite well with the projections made by Oklahoma State University on loop performance. By and large, our recommendations for loop lengths in Iowa consist of 500 ft. of 1-1/2 in. pipe at 6 ft. depth for horizontal loops and 150 ft. of wetted borehole per ton or 300 ft. of 1-1/2 in. pipe for vertical loops. Most of the systems mentioned in the questionnaire went in under these guidelines. You will note that in most cases, our loop temperatures dropped to the middle to low thirties and high twenties where two pipes were put in the same trench at 6 ft. and 4 ft.

Water Source Heat Pump Performance at Varying Water Temperatures

If you look at the low temperature performance of most water source heat pumps on the market, you will note that there are not very many units that are rated for low temperature water. Note listing below as taken from my handbook.

S. Comparison of GWHP Capacity at Various Water Temperatures (36,000 BTU Unit)
Air entering indoor coil at 70 F. D.B. at 1200 cfm.
Water Flow at 5-7 GPM

Water Temp.	Command-Aire SWPR 350 BTU/Hr	Freidrich* (805-036) BTU/HR	Bard (HWP 36) BTU/Hr	Mammoth SOLA-Terra II (035) BTU/Hr	Comfort-Aire (WPH 34) BTU/Hr
25F	25200		22000		
30F			25300		
35F	29850		27900		
40F			30400		
45F	34450	32100	32740	31400	27900
50F		34500	35200	34200	29900
55F	39050	36900	37300	36700	32000
60F		39050	39678	39300	34000
65F	43700	41675	41922	41900	36400
70F		44300	44370	44600	38700

NOTE: The heating capacity figures shown are taken from literature supplied by the manufacturers listed as there are no A.R.I. (Independent Lab) low temperature ratings available.

*Both units have lower temperature capabilities. As of 4/1/84 no additional low temperature data has been available from the manufacturer.

T. Comparison of GWHP Cooling Capacity at Various Water Temperatures (36000 BTU Unit)
Air entering indoor coil at 70 F. W.B. at 1200 cfm.
Water Flow at 5-7 GPM

Water Temp.	Command-Aire SWPR 350 BTU/Hr	Freidrich* 805-036 BTU/HR	Bard HWP 36 BTU/Hr	Mammoth SOLA-035 BTU/Hr	Comfort-Aire WPH 34 BTU/Hr
55F	41900	40348	37875	44600	38900
60F		39248	37168	43400	38800
65F	40800	38435	36461	42100	38700
70F		37623	35754	40700	
75F	39700		34946		38600
80F			34239		
85F	38600		33532		37500
90F			32724		
95F	37500		31815		33200

U. Comparison of GWHP Capacity at Various Water Temperatures (44000 BTU Unit)
Air entering indoor coil at 70 F. D.B. at 1400 cfm.
Water flow 10-15 GPM

Water Temp.	EconAir (400)	Tetco (HECWZ-042C)	Weather-King (YPH-44)	Temp-Master (W-2.50A)	Can-Therm* (DUO520)
25F	34200				40000
30F	37200				43000
35F	39700				46000
40F	42000	36400	43000		50000
45F	44400	36000	46400	42970	53000
50F	45900	38000	48800	44865	57000
55F	47850	41000	51900	47945	
60F	49400	43400		51730	
65F				52135	
70F				53609	

*50000 BTU Unit

Having seen the air-to-air heat pump industry flounder back in the sixties due to poor applications and sizing, it became apparent to me that we need to inform our members and consumers about the selection

of the right brand of water source heat pump for earth coupled applications, or make sure that additional loop is installed so that water temperatures will not drop below the level at which the water source heat pump can operate. I have already been involved in a number of complaints of poor performance where this has happened. Fortunately, the contractor and equipment supplier were gracious enough to replace the water source heat pump with another brand that was designed for lower water temperatures to solve the problem.

I have even seen some complaints on the performance of water source heat pumps using 52 degree well water. A unit designed for the south (70 degree incoming water) was inadvertently installed in northern Iowa. At the water temperature listed, the unit was performing at 60% of design, thus requiring an excess amount of resistance heating during colder weather. Fortunately, I am seeing more and more equipment manufacturers being concerned about this problem and they are supplying their dealers with micro-computers and other technical support to insure that enough loop is put in the ground to insure proper operation of their particular brand of water source heat pump.

Water Source Heat Pump Loop Sizing Practices in Iowa

If someone were to call me today (middle summer, 1984), I would still be recommending so many feet per ton, etc., and warn them that the recommendations given would result in 30 degree water and that they should take care in selecting a brand of water source heat pump that will perform well at these water temperatures. I hope that in the very near future there will be more units on the market that will perform better with 30 degree water. I believe that the software program for closed loop performance developed by Oklahoma State University for micro-computers will take a lot of the guesswork out of loop sizing.

Once I get a handle on soil densities and types in Iowa and I better understand how to use the software program, I intend to purchase the system. I intend to encourage each REC in the state of Iowa to also purchase the micro-computer and software programs mentioned so they can help their local dealers.

One of the questions that keeps bothering me is the economic trade off between loop length and water source heat pump performance. Should we try to install enough loop to keep water temperatures in the forties for higher BTU output per KWHr input or can we do as well in terms of investment payback by reducing our investment in loop piping? I believe this can be quantified, but there will be a different answer for each brand of equipment.

Having not attempted to quantify the above(I believe that every manufacturer should provide information to us as we are at their mercy in terms of supplying accurate unit performance at lower water temperatures), but I have a hunch that we ought to supply enough loop piping to keep loop water temperatures above 35 degrees F. during the coldest month of the year. The loop length recommendations mentioned earlier appear to fulfill these requirements.

Sizing of Water Source Heat Pumps-Earth Coupled vs. Open System

One of the surprises I ran into last year was trying to perform an accurate heat loss-heat gain calculation for my residence. When I built my house back in 1975, the local utility suggested that I needed 30 Kw of strip heaters if I installed an electric furnace and that I should go with at least a 4 ton A-A heat pump. When I went through a Manual J calculation, I came up with 27,000 btu cooling and 42,000 BTU heating, which is about 1/2 of the capacity that we originally installed.

In doing a number of other sizing formats, I have come to the conclusion that Manual J is too conservative for older homes having average to poor windows. In performing a Manual J calculation of a number of other older homes and making the calculations to correlate annual energy usage with design conditions, I have come to the conclusion that an additional .35 air change per hour should be added for better correlation.

In looking at a typical heat loss/heat gain calculation in Iowa, we find that the typical residence in Iowa has twice the heating load than cooling.This obviously poses a sizing dilemma.

In the handbook that I have published, I suggest that we can safely oversize air conditioning capacity by 175% or more. In my own experience, the A-A HP in my own home was oversized by nearly 200% and I never once sensed that I was not getting proper dehumidification from short cycling. Other than going with two smaller units using both for heating and one for cooling, it behooves us to try to size as close to our heating requirements as possible.

In writing my handbook, I tabulated the heating-cooling performance of a number of units for water temperatures starting at 25 degrees up to 100 degrees F. One of the things I noticed was that those units designed for superior low water temperature performance also had poorer high temperature performance.

It then occurred to me that this could work to our benefit for earth coupled systems, because when loop temperatures warm up, the water source heat pump has to run longer due to decreasing BTU output from the warmer water.

To demonstrate this fact, I went through a sizing example in my handbook for a house in NE Iowa requiring 63,000 BTUs for heating and 29,000 BTUs for cooling. In selecting a water source heat pump, we have three choices of units as shown by figures provided by a well known manufacturer:

Nominal BTU Rating	BTU Heating @ 50 Degrees F.	BTU Cooling @ 50 Degrees F.
35,000	39,150	43,000
41,000	44,600	49,400
51,000	48,200	56,100

If we use the 175% oversizing criterion and multiply it times our air conditioning design heat gain for the house, we find that a unit rated at 50,750 BTUs for cooling would give us adequate dehumidification. In looking at the above chart, it appears the 41,000 BTU nominally rated unit would be the logical

choice. Heating capacity is high enough at 44,600 BTUs, so that around 14% of the total heating BTUs required would need to be supplied by strip heaters and 49,000 cooling BTUs is less than our 175% criterion.

The story changes dramatically when we look at an earth coupled system where we expect the loop water to drop to 35 degrees in the winter and 75 degrees in the summer, as shown below:

Nominal BTU Rating	BTU Heating @ 30 Degrees F.	BTU Cooling @ 70 Degrees F.
35,000	31,700	41,300
41,000	36,200	47,000
51,000	39,200	53,100

As you can see, heating performance dropped off considerably while cooling performance decreased less significantly. In this instance, I believe, we should select the next larger unit. The penalty for dropping heating capacity to 36,200 BTUs is too great (27% of the heating requirement would have to be supplied by strip heaters) when compared to the small penalty incurred by oversizing air conditioning a couple more percent.

In the above example, design conditions called for 15 degree gain in the summer to maintain a 73 degree inside temperature. The homeowner in this case may have to lower his thermostat a couple more degrees to achieve proper dehumidification. By setting his thermostat down to 68 degrees, for instance, for a couple of days, the design cooling load increases to approximately 32,000 BTUs per hour. 175% of this figure is 56,000 BTUs, which is around 3,000 BTUs higher than the larger unit's cooling output at 75 degree water.

Conclusion

In conclusion, I believe we need to carefully consider the above factors when sizing a water source heat pump. As we get more experience, more information will become available about what we can expect from loop temperature performance. I believe my recommendations may be counter to what many manufacturers are saying (it is getter to undersize than oversize). I do not disagree, except that we should not undersize any more than 70% of peak hour design conditions.

COOLING PERFORMANCE IMPROVEMENT OF A HORIZONTAL GROUND COUPLED HEAT PUMP

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

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Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

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COOLING PERFORMANCE IMPROVEMENT OF A HORIZONTAL GROUND COUPLED HEAT PUMP

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Abstract

Performance of the horizontal-coil ground-coupled heat pump system in TECH House I at the Tennessee Energy Conservation in Housing Facility is reported for the summers of 1983 and 1984. The overall coefficients of performance (COP) were 1.80 (1983) and 2.41 (1984) while the overall cooling seasonal performance factor (SPF) was 1.11 (1983) and 1.43 (1984) with the system located within the conditioned space. If the system had been outside the conditioned space, an SPF of 1.31 (1983) and 1.76 (1984) would have been realized.

In both years, the systems were identical with the exception that prior to the 1984 cooling season, the ground coil was excavated and a new one of indentical length was installed using sand as a backfill material. Although the 1984 SPF was 29% higher than that measured in 1983, the improvement cannot be totally attributed to the backfill material. During the summmer of 1983, 30% more cooling degree days were incurred than in 1984. A decrease in runtime (as a result of lower ambient temperatures) would also increase the performance factor.

Introduction

The concept of the ground-coupled heat pump has attracted much attention in recent years, because the ground temperature is normally more favorable than the air temperature to obtain high performance. Success has been achieved with ground-coupled systems in the heating mode. Over two seasons of operation, the system described herein has provided a seasonal performance factor of 2.6 while requiring no electrical backup heating.[1] In a 1982 survey of ground source heat pump investigations, Ball 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.[2] However, he reports only two values for cooling, 1.8 and 2.7. Cooling season performance data appear to be scarce relative to heating data.

The University of Tennessee Energy, Environment, and Resources Center (UT-EERC) has been contracted by the Oak Ridge National Laboratory (ORNL) for the procurement, installation, maintenance, operation and instrumentation of a horizontal ground coil heat pump (GCHP) system. Tech House I at the Tennessee Energy Conservation in Housing (TECH) Facility incorporates the horizontal ground coil heat pump system. This facility is located on the University of Tennessee Institute of Agriculture's Experiment Station in Knoxville, Tennessee.

ORNL contracted with Battelle, Inc. to design the GCHP system for Tech House I.[3] That design (based on the cooling load) called for a ground coil pipe length of 700 feet utilizing 1-1/4" IPS polybutylene pipe. This design was based on an assumed cooling load of 1.41 tons. The actual load, however, has been measured to be 2.0 tons. Using a "rule of thumb" of 400 feet per nominal ton of heat pump cooling capacity and a correction factor of 1.25 for 1.25 inch pipe, the pipe length should be at least 1000 feet in length.[4] This estimation represents a 48% increase over the actual installed length of 675 feet.

The GCHP system was operated during both the 1983 and 1984 cooling seasons. Between the 1983 and 1984 cooling seasons, the polybutylene ground coil was excavated and a new high density polyethylene ground coil was installed using sand as a backfill material. The former coil had been backfilled with indigenous clay. The 1984 cooling season operated with the same pipe length as during 1983 in order to quantify the effects of sand backfill versus indigenous clay backfill.

System Description

1983 Ground Coil

Although the design called for a ground coil length of 700 feet, the actual installed length is 675 feet due to rock that was encountered during the installation. The pipe is polybutylene nominal 1.25 inch IPS with an inside diameter of 1.380 inches and wall thickness of 0.120 inches. The ground coil is buried four feet deep with no run of pipe closer than six feet to any other section of the ground coil. The brine in the ground coil is a solution of water, 20% (by weight) of methanol, and corrosion inhibitors.

A hole was dug in the ground coil field near the mid-length on August 9, 1983, in order to inspect the ground coil and surrounding soil. As the hole was dug, air gaps were observed at the interface between the soil in the trench in which

the coil was buried designated as disturbed soil, and the surrounding undisturbed soil. The gaps at the interface occurred intermittently from the surface to the depth of the coil, approximately four feet from the surface. Several other air spaces were encountered in the disturbed soil as the hole was dug. When the pipe was uncovered, an air gap approximately two inches long and 0.25 inches wide was discovered parallel to and in contact with the coil wall. This type of air gap occurred where the coil approached the interface between disturbed and undisturbed soil and was not evident when the coil was in the middle of the disturbed soil.

On October 2, 1983, a second hole was dug at about 20 percent of the total coil length from the inlet to the ground. At this location the interface between the disturbed and undisturbed soil was not as distinct as the interface of the hole dug in August. Air gaps were not encountered as the hole was dug. The soil seemed well packed and moist to the touch in the vicinity of the coil.

Based on the 1983 cooling season performance data, it was decided that the ground coil be reinstalled using sand as a backfill material to decrease the thermal contact resistance between the ground coil and the soil.[5]

1984 Ground Coil

Prior to the 1984 cooling season, the ground coil was reinstalled using nominal 1.25 inch high density polyethylene pipe. The material was changed in order to use butt fusion welded joints rather than barbed fittings. In lieu of using indigenous clay backfill around the pipe, fine river sand was placed approximately three inches all around the pipe. Indigenous clay was then backfilled and carefully tamped. Although additional pipe length was installed as well as other features such as a soaker pipe and a brine to domestic water heat exchanger, these features were not used. The system that operated during the 1984 cooling season was identical to the system that operated during the 1983 cooling season with the exception of the pipe and backfill material.

Heat Pump

Considering pressure drop, capacity, and efficiency, one of the heat pumps suggested by Battelle was a TETCO unit. This heat pump is a water to water, heating only heat pump. Space cooling is accomplished by re-directing the water-methanol brine with manually controlled three way valves.

Since cooling performance data for the heat pump were unavailable from the manufacturer, tests were undertaken at the TECH Facility to evaluate the performance of the Tetco unit. Figure 1 shows measured performance data compared to two other units[6] (water to air heat pumps).

Building

The GCHP system is used to provide space conditioning for TECH House I (formerly called the UT Solar House). TECH House I is an occupied 1800 square foot residential building. It has a design heating load of 30,000 Btu/hr and a design cooling load of 2.0 tons.

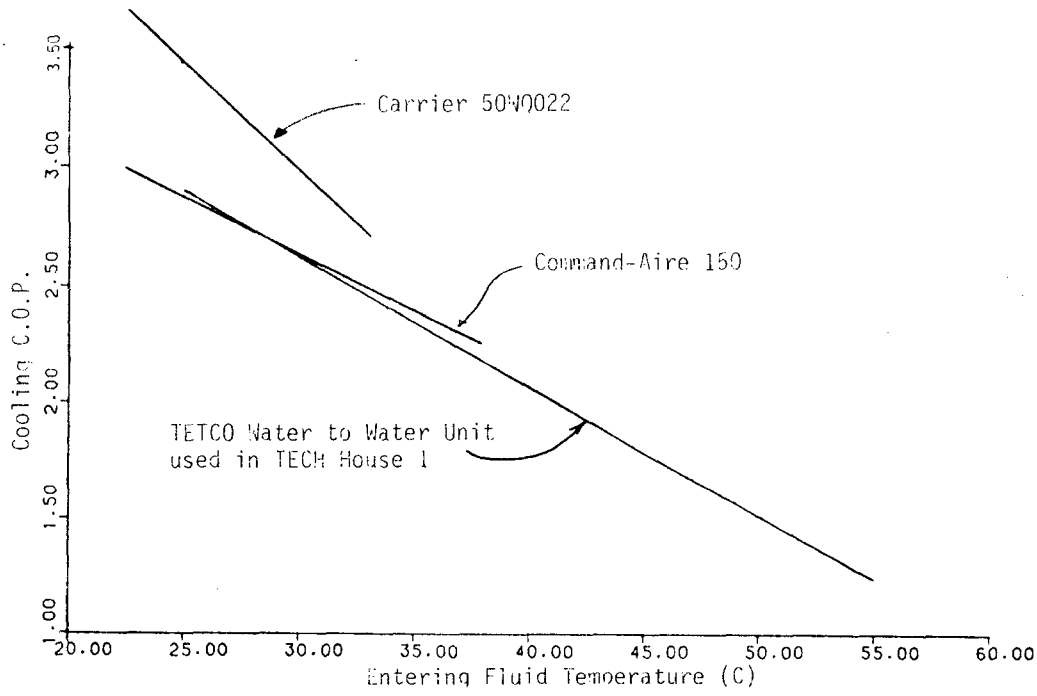

Figure 1. Cooling Coefficient of Performance Curves

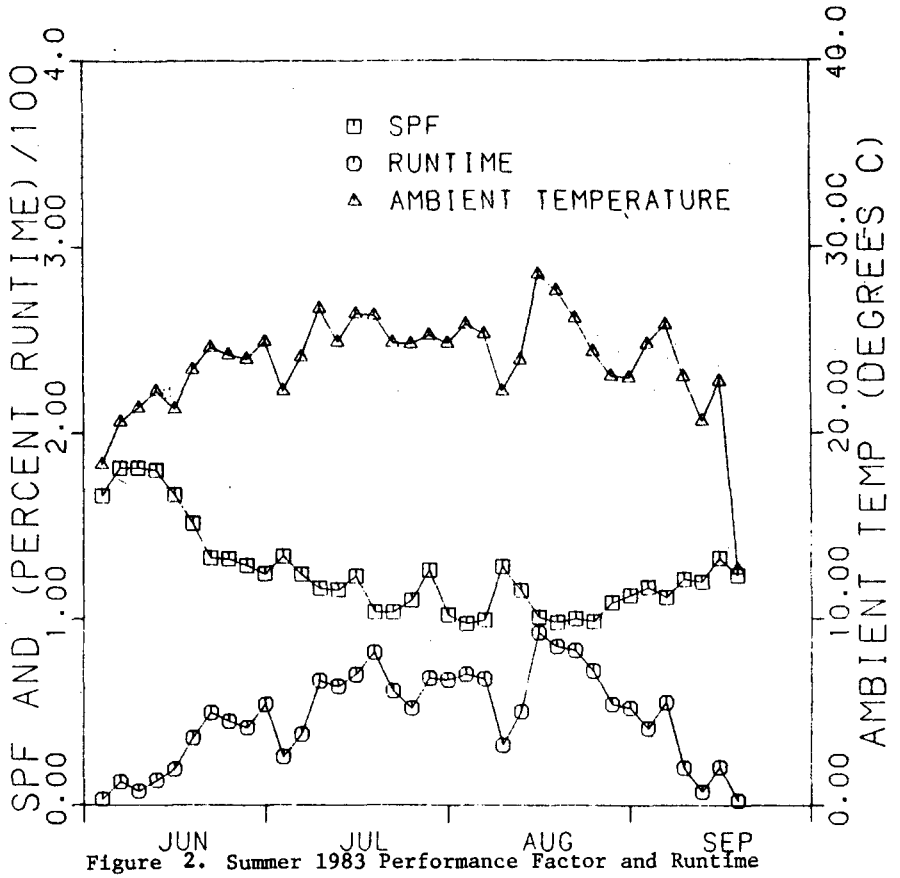

Figure 2. Summer 1983 Performance Factor and Runtime

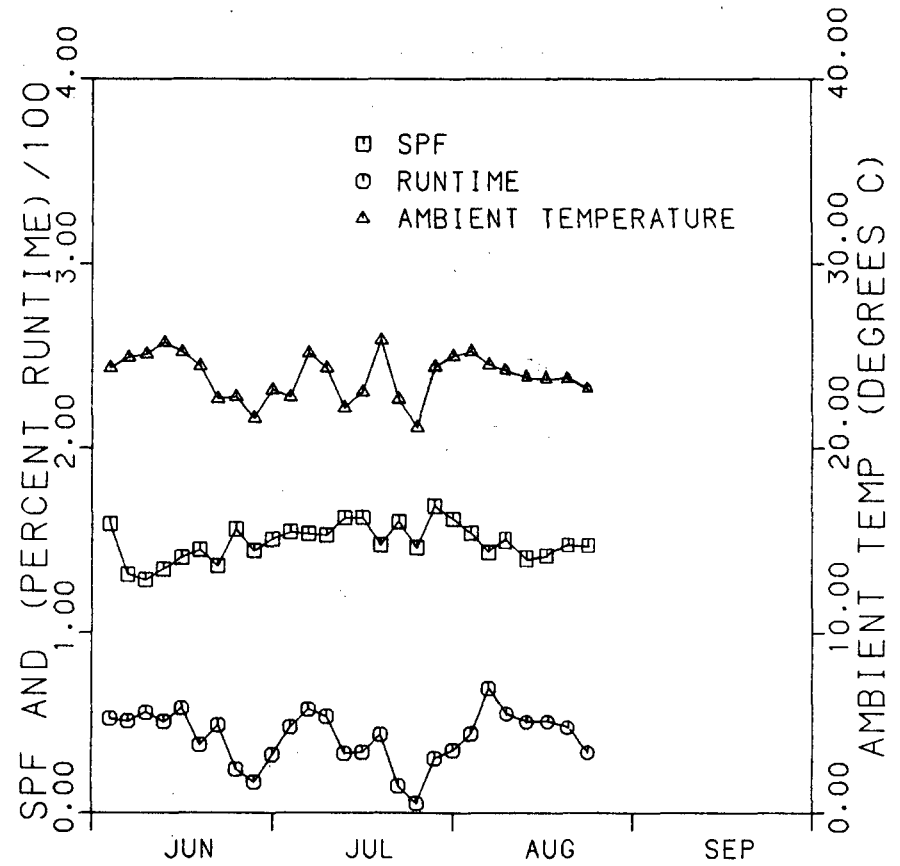

Figure 3. Summer 1984 Performance Factor and Runtime

Instrumentation

Soil and brine temperatures are measured with platinum resistance temperature devices (RTDs) using standard resistance thermometry. Conditioned space dry and wet bulb temperature measurements are made with solid state temperature transducers. Twenty three different soil and brine temperature measurements were made hourly throughout the 1983 cooling season. Additional probes were installed during the ground coil reinstallation and fifty four different soil and brine temperature measurements were made hourly throughout the 1984 cooling season.

Ceramic soil moisture sensing devices were utilized to measure the moisture content of the soil. These devices were calibrated using actual soil samples from the site. The electrical resistivity of these devices changes with moisture content of the surrounding soil.

Four watt-hour meters are used to measure the power consumption of the heat pump compressor, supply air blower, brine circulation pumps, and total house power. In each of these meters the revolutions of the eddy disk are counted by optical sensors. The output of each optical sensor is digitized and then sent to a summing circuit or counter.

A flow meter and two RTDs form the basis of each measured heat flow. The heat into/out of the ground, and the water-to-air coil heat flow are measured with these devices. The temperature difference measured by the two RTDs is electronically converted to a frequency. That frequency is then electronically multiplied by the frequency output of the flow meter. The product of the flow rate and the temperature difference, which is directly proportional to the heat flow, is sent to a summing circuit in the DAS.

The latent cooling load is measured by recording the total mass of condensate removed from the air by the water-to-air coil. The condensate flows by gravity through a pipe to a tipping bucket rain gauge. Each tip of the bucket sends a 10 volt digital signal to the data acquisition system (DAS). Knowing the mass of water required to tip the bucket, one can then determine the mass of the condensate.

The digital heat flows and electric power signals from the different modes of system operation are accumulated by counters. Each hour on the hour the counters are summed (and then zeroed) and the analog signals are scanned. These data are recorded on magnetic tape and printed on paper. The magnetic tape contains, then, the hourly heat flows, energy consumptions and specified temperatures in the system as well as complete on site meteorological conditions.

Thermal Performance Data

Data were recorded hourly throughout the 1983 cooling season. At the time of this writing, data for the 1984 cooling season are presented for the period through August 27, 1984.

Meteorological Conditions

Presented in Table 1 is a comparison of cooling degree days and rainfall for both cooling seasons.

Table 1. Summer 1983 and Summer 1984 Meteorological Conditions.

Month	Inches of Rain	Cooling Degree Days (base 65°F)
Jun 1983	3.26	235
Jul 1983	3.18	425
Aug 1983	3.89	462
Sep 1983	0.95	217
Jun 1984	2.00	289
Jul 1984	7.04	255
Aug 1-27 1984	.98	282

Seasonal Performance Factors

The seasonal performance factor (SPF) is defined as the net cooling (or heating) effect of the total system divided by the total electric power required to deliver that effect. 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 methods of calculating the performance are presented. SPF_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. The cooling SPF_i is then calculated as:

$$SPF_i = \frac{WTAC - BLWR - PUMPS - CCL}{POWER}$$

where:

- WTAC is the water-to-air coil heat flow.
- BLWR is the electric power required by the blower.
- PUMPS is the electric power required by the circulation pumps.
- CCL is the compressor can loss calculated as the sum of the WTAC and compressor electric power consumption less the ground coil heat flow.
- POWER is the total electric power required by the system.

SPF_o is the performance factor for the case in which the heat pump package containing compressor and pumps are located outside the conditioned space (i.e., a garage). The cooling SPF_o is then calculated as:

$$SPF_o = \frac{WTAC - BLWR}{POWER}$$

Table 2 presents the monthly and seasonal heat pump performance data for both the 1983 and 1984 cooling seasons. Clearly, locating the heat pump package outside the conditioned space is more beneficial than locating it inside the space during summer months. The converse is true, however, during the winter months.[1]

Table 2. Heat Pump Performance Data

1983 Cooling Season	Electric Power (kWh) Blower	Pumps	Comp	Total	Heat Flows (kWh) WTAC	GRND	SPF_i	SPF_o
Jun	67	45	426	538	968	1274	1.37	1.67
Jul	187	118	1197	1502	2227	3218	1.14	1.36
Aug	242	154	1631	2027	2701	4130	1.04	1.21
Sep	77	50	538	665	891	1373	1.06	1.22
Jun-Sep	573	367	3792	4732	6787	9995	1.11	1.31

WTAC: Water-to-Air Coil Heat Flow
GRND: Ground Coil Heat Flow

1984 Cooling Season	Electric Power (kWh) Blower	Pumps	Comp	Total	Heat Flows (kWh) WTAC	GRND	SPF_i	SPF_o
Jun	126	83	727	936	1788	2213	1.36	1.77
Jul	117	75	714	906	1717	2242	1.47	1.77
Aug 1-27	142	90	874	1106	2077	2714	1.45	1.75
Jun-Aug 27	385	248	2315	2948	5582	7169	1.43	1.76

Effect of Meterological Conditions

Figures 2 and 3 show the relationship between runtime and the performance factor during the 1983 and 1984 cooling seasons, respectively. During 1983, there exists a clear and direct correlation between runtime and the performance factor. As the runtime increases, the performance factor decreases. This result was perhaps unexpected because the heat pump would be operating more in the transient mode rather than steady state as the runtime decreased. The COP of a heat pump itself is lower in transient operation than in steady state. The positive effects of cycling upon heat transfer to the soil, however, evidently outweigh the negative effects of transient operation of the heat pump itself.

Since TECH House I is a skin load dominant building, there exists a direct relationship (as depicted in Figures 2 and 3) between ambient temperature and runtime. Ambient temperature, and consequently runtime, decrease during a period of rain.

The runtime and performance factor correlation is also evidenced during June and August of 1984. The correlation is not as clear during July, 1984. This lack of a correlation during July, 1984 may be a result of the unusually high rainfall experienced during that time period. Figures 5 and 6 depict the performance factor and rainfall during the 1983 and 1984 cooling seasons, respectively. Figure 4 shows the soil moisture measured at a point six inches horizontally from the ground coil midpoint during both cooling seasons. Although the quantity of rain during each season was comparable, the periods in which it occured were not. During 1984, virtually all of the rain occured from mid June through the latter part of July with the other time periods experiencing little rain (Figure 6). A similar trend in 1984 soil moisture is also seen with the soil moisture being higher from mid June through early August. During 1983, the amount of rain was evenly spread out until mid August when it dropped off sharply (Figure 5). The soil moisture in 1983 gradually decreased throughout the summer. By a comparison of Figures 4,5 and 6, it is observed that for both the 1983 and 1984 cooling seasons, the trend of both soil moisture and SPF is the same. This indicates that soil moisture, which significantly affects thermal conductivity, plays an important role in system performance.

Comparison of COP and SPF

Figure 7 shows the 1984 measured COP and SPF_i. The SPF_i is essentially the net cooling effect divided by the total system power whereas COP is the gross cooling effect divided by only the compressor power. This comparison is made in order to point out the large difference between the two numbers. When comparing the performance of ground coupled heat pumps with air to air heat pumps, one must use the SPF of each system because the SPF considers the total interaction of the heat pump with the building whereas the COP does not.

Comparison of 1983 and 1984 Performance

The SPF_i measured during the 1983 Cooling Season was 1.11. For the period through August 27 during the 1984 Cooling Season, the SPF_i was measured to be 1.43. The performance factors measured during each cooling season are plotted on Figure 8. The 1984 performance factor represents a 29% improvement over the performance factor measured during 1983. This improvement, however, cannot be totally attributed to the change in backfill procedures. During comparable periods, 30% more cooling degree days were incurred in the summer of 1983 than in the summer of 1984. As was pointed out earlier, a decrease in ambient temperatures (and consequently runtime) will increase the performance factor.

Summary and Conclusions

The cooling seasonal performance factor of a horizontal ground coupled heat pump was measured to be 1.11 and 1.43 during 1983 and 1984 cooling seasons, respectively. In both seasons, the systems were identical with the exception prior to the 1984 cooling season, the ground coil was excavated and a new one of identical length was installed using sand as a backfill material. The intent of the use of sand backfill material was to improve the contact between the earth and the ground coil. Although the 1984 SPF_i was 29% higher than that measured in 1983, the improvement cannot be totally attributed to the backfill material. During the summer of 1984, 30% more

cooling degree days were incurred than in 1984. A decrease in runtime (as a result of lower ambient temperatures) will tend to improve the performance factors.

It is recognized that the length of the ground coil is undersized and a water-to-air heat pump should be used in lieu of the existing water-to-water heat pump. With these changes and additional efforts to optimize the horizontal ground-coil heat pump system, it will be determined if the cooling seasonal performance factor can be competitive with a conventional air-to-air heat pump.

Future Work

Preliminary plans are currently being made for a joint project combining the expertise of the ORNL, the UT-EERC, and the private sector. A new horizontal ground coupled heat pump will be designed and installed by the industry at the TECH Facility. This system will be instrumented and evaluated by the ORNL and the UT-EERC. Seasonal performance evaluation should begin in the summer of 1985.

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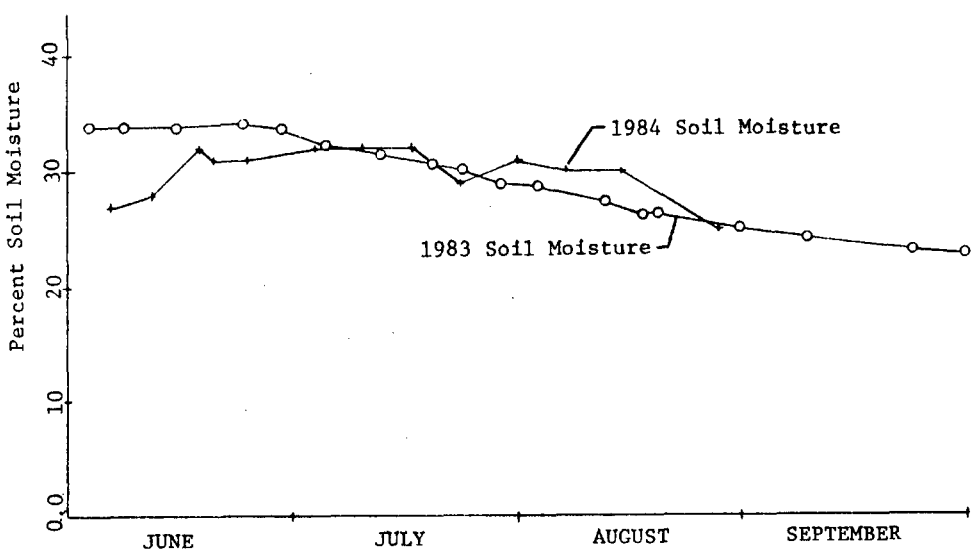

Figure 4. Soil Moisture Measurements

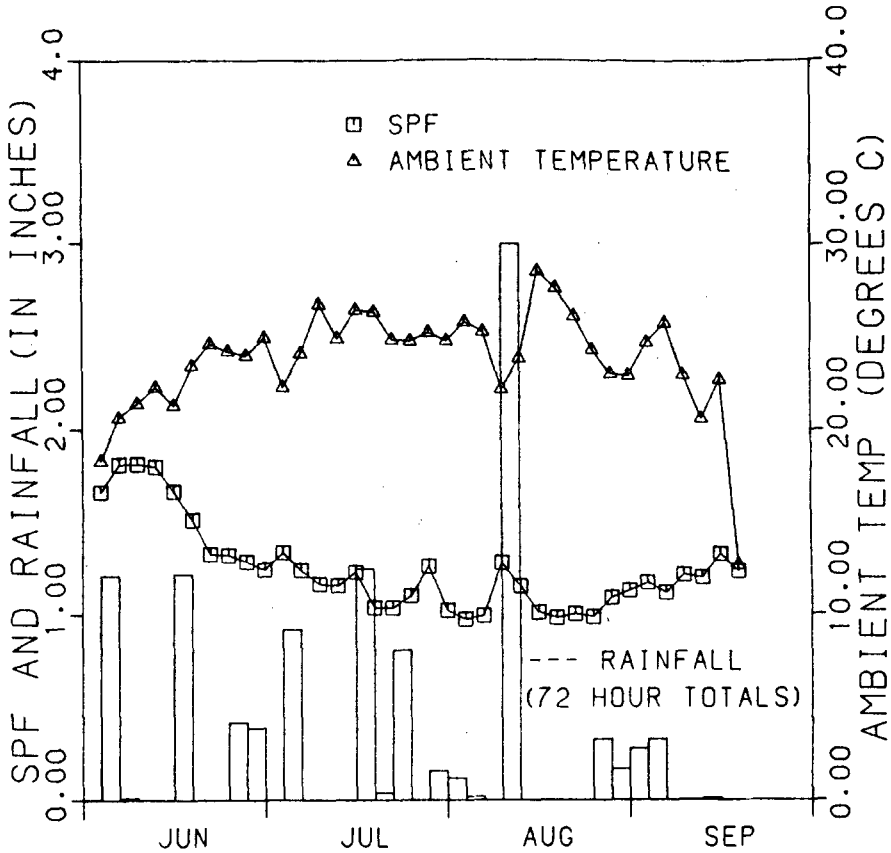

Figure 5. Summer 1983 Performance Factor and Rainfall

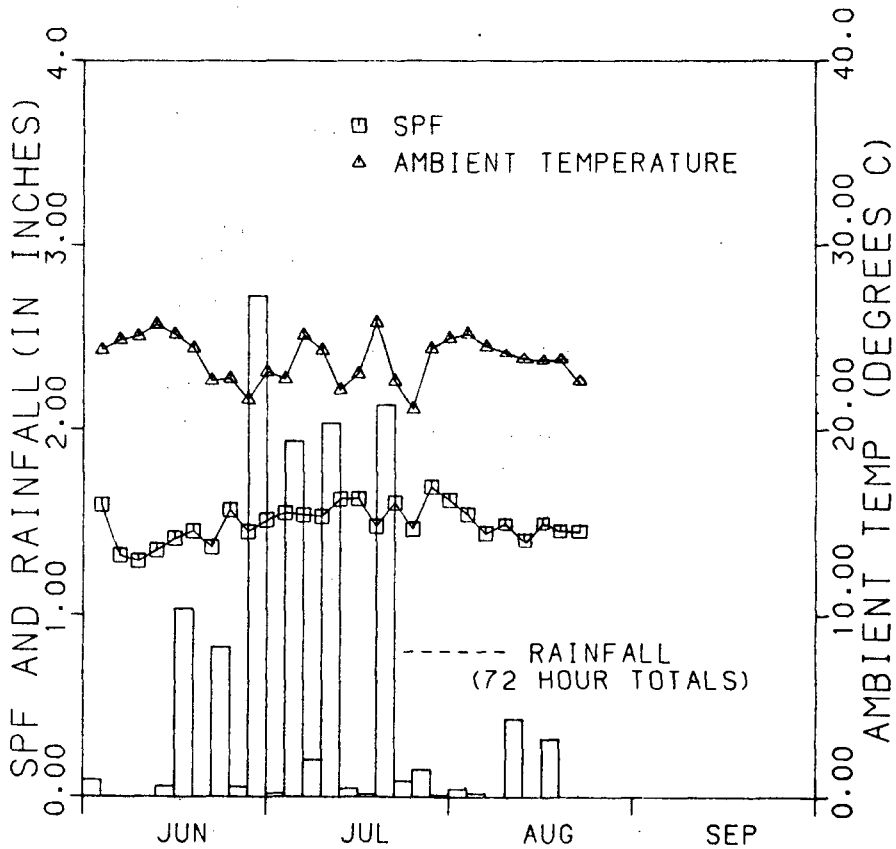

Figure 6. Summer 1984 Performance Factor and Rainfall

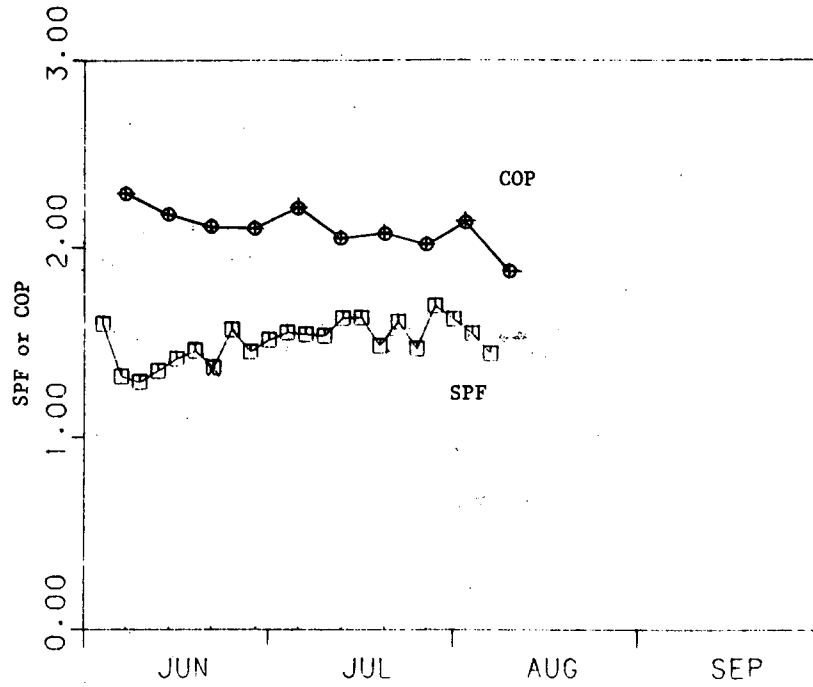

Figure 7. Comparison of 1984 SPF and COP

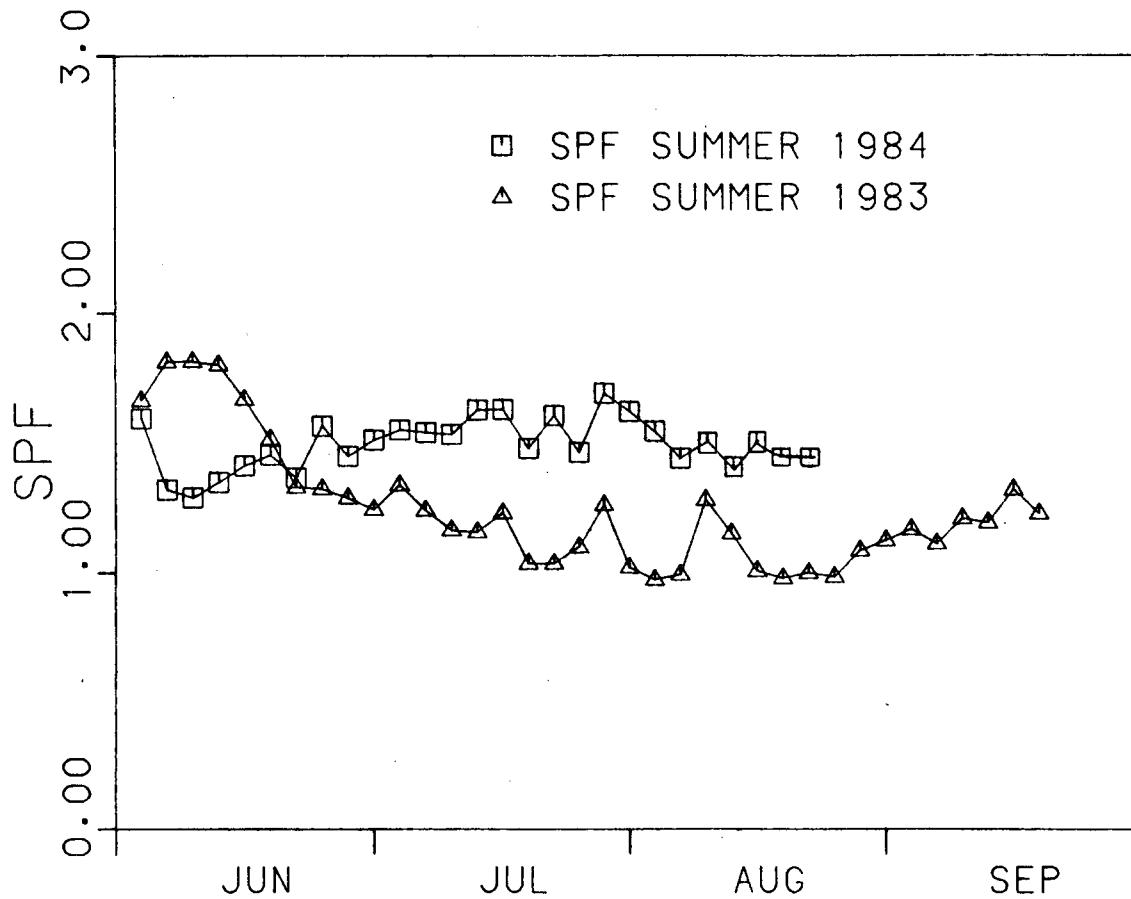

Figure 8. Comparison of 1983 & 1984 Performance Factors

EARTH-COUPLED HEAT PUMPS FOR RESIDENTIAL HEAT/COOL AND HOT WATER

by Harry Braud

Harry J. Braud

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EARTH-COUPLED HEAT PUMPS FOR RESIDENTIAL HEAT/COOL AND HOT WATER

by

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Vertical closed-loop heat exchangers were used for earth heat exchange for heat pumps. Residential energy saving with the earth-coupled system as compared to air-source air-conditioning and natural gas heat was 5839 kwh (21 per cent) and natural gas for central heat was discontinued. Average gas consumption was 16.6 mcf/year before. The system used two water-to-air heat pumps for space heat/cool and a water-to-water heat pump for hot water needs in excess of the de-superheater output of one of the space conditioning units.

Introduction

A closed-loop earth-coupled heat pump system has the energy-saving advantage inherent in water-source heat pumps and there is no water supply or disposal problem. The system draws on the stable temperature of the earth for a heat source in winter and uses the relatively cool temperature of the earth for a heat sink in summer. Research in Oklahoma, Bose et al.[1,2], Partin[3], in Louisiana, Oliver and Braud[5], Braud[6] and in New York Metz[7] has led to the design of vertical-bore and horizontal-grid plastic pipe heat exchangers for heat pumps. Different pipe configurations have been evaluated, and several designs and plastic pipe materials are now being used by contractors who are in the business of installing water closed-loop earth-source heat pump systems in Louisiana.

The Project

In 1981, the author received a U.S. Dept. of Energy Grant to install and monitor a closed-loop earth-source heat pump system in his residence. The goal was to monitor the energy consumption of the components of the system to compare to the conventional heating and cooling equipment that was used in the residence prior to the change. The new system employed two zoned water-to-air heat pumps for space heat/cool and one water-to-water heat pump for direct hot water production, Figure 1. One of the space heat/cool units had a de-superheater attachment that generated hot water whenever the unit ran. Earth-source heat exchange was achieved with vertical-bore plastic pipe heat exchangers of two designs, a concentric pipe and a U-bend, Figures 3 and 4. Sizing of the heat exchangers was based on research findings at Louisiana State University where several earth-coupled heat exchanger configurations were being tested, see Braud[6].

The residence was previously equipped with an air-source air-conditioner and a gas furnace. With the new earth-coupled heat pump system there was no need for gas service to the residence, so it was discontinued and the meter deposit was returned by the utility company. Some temperature, flow meter, and electric energy instrumentation was already in place in the residence for a project on air-source heat pump water heaters. Over the two year period 1980-81, air-source heat pump water heaters were tested in switch back comparison to electric resistance water heating, see Braud and Klimkowski[4].

Monthly utility billing records for both gas and electricity had been kept for the four years prior to the DOE project with the closed-loop water-source heat pumps. These records served as a base to which the energy consumption with the new closed-loop heat pump system was compared.

To define energy use within the system, electric watt-hour meters were put on each of the heat pumps and the closed-loop circulator pump. A meter was also installed on the domestic water well pump and the electric clothes dryer. Other electric uses in the house were found by subtracting the submetered items from the total residential metering by the utility company. See Figure 2 for specifications of the equipment.

House Description

The residence was built in 1970 with 4 bedrooms, 2 1/2 baths, single gable roof, brick veneer construction and 195 m^2 (2100 ft^2) of living area. Family size was two adults and four children ages 6 to 12. The wall insulation was 9 cm (3 1/2 inches of fiberglass insulation). Ceiling insulation was originally 20 cm (8 inches) of blown fiberglass which was increased by an additional 9 cm (3 1/2 inches) of batt insulation in 1978. Glass area was only 7 per cent of floor area. The original heating system consisted of central gas furnace unit with split air-source air-conditioner (EER = 6.5). The residence used an electric clothes dryer. The original gas water heater was discontinued in use in 1979 when an air-to-water heat pump water heater was put on test. In the period October 1979 to March 1982 the heat pump water heater was operated in week-by-week switch-back with electric resistance water heating to determine the performance factor for the heat pump unit. The annual performance factor was found to be 2.1 (see Braud and Klimkowski[4].) Residential electric use in that period was thus indicative of the mixed mode of water heating - both heat pump assist and electric resistance heating. Gas use after October 1979 was for space heat only.

Heat Exchanger Specifications

U-Bend

Side-by-side U-bend 122 m (400 feet) deep, 3.2 cm (1 1/4 inch) SDR11, Phillips "Driscopipe" polyethylene pipe in continuous lengths with shop-fabricate bronze 180° close elbow fitting at the bottom, Figure 3. Bore hole size, 10.2 cm (4 inch) nominal. Three fifty-pound steel weights 7.6 cm (3-inch) diameter scrap shaft stock were attached to U-fitting with short cables to give dead weight and were dropped in bore hole first to allow insertion of pipes in bore hole mud. Water table depth was, 4 m (15 feet). The pipes were filled with tap water prior to insertion into the bore hole. With the 68 kg (150 pounds) of dead weight, the pipes slipped easily down the bore hole without seizure.

Concentric Pipe Heat Exchanger

Using a pipe-in-a-pipe configuration, a 91.4 m (300 feet) exchanger was inserted into a 10.2 cm (4" nominal) bore hole. The pipe casing was 6.4 cm (2 1/2 nominal) diameter SDR 21 PVC. A check-type back-wash valve was installed at the bottom of the casing to allow earth fluid to enter casing to allow earth fluid to enter casing during insertion into the bore hole, Figure 2. Back washing caused seating of check valve. The inner pipe was 3.2 cm (1 1/4 inch) diameter schedule 40 PVC pipe. Because the concentric pipe has more water flow friction loss than a U-bend, the length was made shorter for this heat exchanger.

Service Lines

Service lines to both heat exchangers were 2.3 cm (1 1/4 inch) diameter schedule 80 PVC pipe. Service lines were buried approximately 38 cm (15 inches) deep. All service lines were put in the same trench without concern about cross-over heat flow due to the thick wall and low thermal conductivity of PVC plastic pipe.

Energy Consumption

Annual Use

Total electric energy consumption for the residence for the 1982-83 test year on heat pumps and the three year period 1979-81 when the residence had air-source air conditioning and natural gas furnace for heat is given in Table 1. Electric use was 28,684 kwh in 1979, 28,810 kwh in 80, and 26,979 in 81. The average annual use for the three years was 28,154 kwh. On the new closed-loop earth-source heat pump system installed in 1982, the annual use dropped to 22,315 kwh even though the electric-powered heat pumps supplied all the space heating energy in the absence of gas service to the residence. The reduction with the new system amounted to 28,154 kwh - 22,315 kwh = 5,839 kwh worth $379.53 at the current electric rate of $0.065/kwh. This is for electricity only.

The average annual gas consumption for space heat in the three previous years (79-81) was 470 m^3 (16.6 mcf), Table 2. With the new heat pump system, gas supply to the residence was disconnected so that use in 82-83 winter was zero. The savings in gas was 470 m^3 x $0.265/$m^3$ = $124.55.

Table 1

Residential Electric Energy Consumption

	Air-conditioning and natural gas heat			Heat pumps
	--------KWH---------			--KWH--
	(1979)	(1980)	(1981)	(1982)
April	1631	1858	2108	1755
May	2160	3018	2405	1943
June	2501	2977	2773	1924
July	3106	3470	2932	2440
August	3021	3227	2976	2192
September	3374	2939	2345	1902
October	2617	2169	2065	1541
November	2124	1836	1811	1572
December	1953	1952	2025	1912
	(1980)	(1981)	(1982)	(1983)
January	2272	2040	1920	1713
February	1895	1779	1685	1805
March	2030	1545	2014	1616
Total	28684	28810	26969	22315
Average (79-81)	-------28154-------			22315
Difference: (79-81 vs. 82)		28154	-	22315 = 5839 KWH (21%)

Savings, @ $.065/KWH = 5839 KWH x .065 = $379.53

Table 2

Natural Gas Consumption for Space Heating Braud Residence

1979-80		1980-81		1981-82		1982-83	
m^3	COST($)	m^3	COST($)	m^3	COST($)	m^3	COST($)
399	$83.31	518	$118.95	484	$128.12	0	0
0.203$/$m^3$		0.230 $/$m^3$		0.264$/$m^3$		Zero	
Average=--------470 m^3----------						Zero	
79-82 =--------0.265 $/$m^3$--------							

Total dollar savings for the residence was $379.53 for electricity plus $124.55 for gas = $504.08 per year, Table 3.

The above values for the two systems can only be fairly compared in light of the weather conditions in each of the heating and cooling seasons. Table 5 gives heating and cooling degree-days for the test years. The 79-81 winters were colder than the 82-83

winter by 38 degree-days. All winters were colder than normal in the heating season. Summers were warmer than normal in the cooling season. Summer 1982 was warmer than the 79-81 summers by 334 cooling degree-days.

Table 3

Total Annual Savings

Electric	- 5839 KWH	- $379.53	- 21 Per cent
Gas	- 470 m^3	- $124.74	- 100 Per cent
Total		$504.27 per year	

Annual Savings on All-Electric Equivalent Basis:

To replace 470 m^3 gas with electric resistance heat would require 3,319 KWH. Gas efficiency @ 65 percent and 100 percent for electric. 28,154 KWH + 3,319 KWH = 31,473 KWHall-electric basis. 31,473 KWH - 22,315 KWH = 9158 KWH savings and 9,158/31,473 = 0.29 or 29 percent annual savings worth $595.27.

Contractors estimate the cost of a new closed-loop heat pump system to be $500 to $600 per ton of cooling capacity more than a good air-source heat pump system. This cost estimate includes hot water generators which are factory-installed on some water-source heat pumps. In the test residence the nominal size of the original system was 4 tons cooling. The cost for the earth heat exchangers was $2025 which was the added cost for earth-coupling over air-source heat pumps.

Monthly Energy Use

In Louisiana with heavy air-conditioning loads, energy demand peaks in the summer months. The peak energy use in the residence was measured in July 1980 when 3470 kwh were used. Peak month in 1979 was September with 3374 kwh and August 1981 with 2976 kwh (Table 1). The closed-loop heat pump system reduced peak loads significantly. Peak use month in 1982 was July with 2440 kwh.

Direct monthly comparisons of electric use in the winter months are very interesting because gas heat was used in the 1979-81 period and the new heat pump system used electricity only. Note in Table 1 that the new heat pump with its space heating capability used less energy than the residential electric use in previous years with gas heat.

Monthly electric energy use in the residence is shown graphically in Figure 3. The curve of average use per month shows the usual summer peak for air-conditioning and another in winter. As a base for comparison to the new heat pump system for heating, the gas use in 1979-81 was converted to electric resistance energy equivalent for the same months to give the adjusted curve. Gas system efficiency was taken as 65 percent. The adjusted curve has the winter heating peak. The lower curve is the energy consumption with the closed-loop heat pump system. It is flatter than the other curves and much below the adjusted curve. The more-level energy use curve is attractive to electric utility companies.

Seasonal Energy Use

Sub-metering of several of the appliances and the heat pumps gave a clear account of where the energy was being used in the residence. Weekly meter readings (Sunday morning) were used to calculate fractions of total use. For display of true energy use by the heat pumps for space heat/cool and hot water, the energy consumed by the circulator pump was charged to the heat pumps in proportion to the run time of each of the heat pumps.

A breakdown of energy used by seasons is shown in Table 4 where residence total is broken down to clothes dryer, space heat/cool, water heating, and other i.e. lights, cooking and other uses not sub-metered. In summer, air-conditioning consumed 35 percent of the total use. Water heating with the water-to-water heat pump took only 5 percent. It should be noted that one of the space heat/cool heat pumps had a hot water generator which took waste heat from the compressor to make "free" hot water. Operation of the clothes dryer took only 11 percent.

In fall, heat/cool took only 18 percent of total energy used; water heating increased to 8 percent, and the clothes dryer increased to 17 percent.

In winter, space heating consumed 13 percent of the total, hot water took 23 percent and the electric clothes dryer consumed 15 percent. Note that water heating, even with the very efficient water-to-water heat pump, took more energy than space heating.

Lighting and other appliances use remained near 50 percent of total household use year-round. The clothes dryer consumed a much larger fraction than expected, 14 percent over the year.

Energy for hot water generation was measured with a watt-hour meter on the water-to-water heat pump. Whenever the day-living zone space heat/cool heat pump ran, the de-superheater also produced hot water which went into the storage tank, giving two sources of energy transfer from the earth-source water loop. This "free" energy input was not measured separately. Total energy use for water heating was 2777 kwh for the two modes of energy transfer from the loop to the hot water tank.

In a previous study of air-to-water heat pump water heaters (Braud and Klimkowski[4]), the annual energy use with resistance water heating was found to be 6584 kwh. An annual performance factor can be defined as the ratio of energy consumption to produce hot water with resistance electric energy to the consumption with the water-source heat pump system. Assuming that hot water consumption is equal, the performance factor PF is:

$$PF = \frac{\text{energy for hot water production with electric resistance energy}}{\text{energy for hot water production with heat pump system}}$$

$$PF = \frac{6584 \text{ kwh}}{2777 \text{ kwh}} = 2.37$$

In a previous study Braud and Klimkowski[4] of heat pump water heaters (air-to-water type) vs. electric resistance hot water heating, the performance factor was found to be 2.1.

Table 4
Electric Energy Consumption
By Season

Month	House Total KWH	Clothes Dryer KWH	Space Heat/Cool KWH	Water Heat KWH	Light, Other KWH
----------SUMMER----------					
(Weeks 28 to 40)					
July Sept.	6662	751	2306	362	3243
Percent	100	11	35	5	49
KWH/WK	513	58	177	28	249
----------FALL----------					
(Weeks 41 to 46)					
Sept October	2072	344	379	169	1180
Percent	100	17	18	8	57
KWH/WK	345	57	63	28	197
----------EARLY WINTER----------					
(Weeks 47 to 52)					
November December	2371	351	212	451	1357
Percent	100	15	9	19	57
KWH/WK	395	59	35	75	226
----------MID WINTER----------					
(Weeks 1 to 8)					
January February	3340	487	446	763	1644
Percent	100	15	13	23	49
KWH/WK	418	61	56	95	206
----------LATE WINTER----------					
(Weeks 9 to 14)					
February March	2279	377	161	492	1249
Percent	100	17	7	21	55
KWH/WK	380	63	27	82	208
----------SPRING----------					
(Weeks 15 to 22)					
April May	3237	528	528	416	1765
Percent	100	11	31	5	53
KWH/WK	405	66	66	52	221
----------SUMMER----------					
(Weeks 23 to 27)					
June July	2273	257	693	124	1199
Percent	100	11	31	5	53
KWH/WK	455	51	139	25	240
Total (Annual)	22234	3095	4725	2777	11637
Percent	100	14	21	13	52

A performance factor of 2.37 amounts to a 58 per cent reduction in energy use for hot water. The waste heat generation of the space heat/cool heat pump along with the water-to-water heat pump gave significant energy savings. The kwh reduction of 6584 - 2777 = 3807 kwh is worth $247 @ $.065/kwh.

Table 5
Heating and Cooling Degree Days
Baton Rouge, LA

SEASON	HEATING	COOLING
1979		2379
79-80	1898	
1980		2691
80-81	1952	
1981		2706
81-82	1580	
AVERAGE	1810	2592
1982		2926
82-83	1772	
DIFFERENCE	-38	+334
NORMAL	1670	2585
DEPARTURE	+140 (79-81) +102 (82-83)	+7 (79-81) +241 (82)

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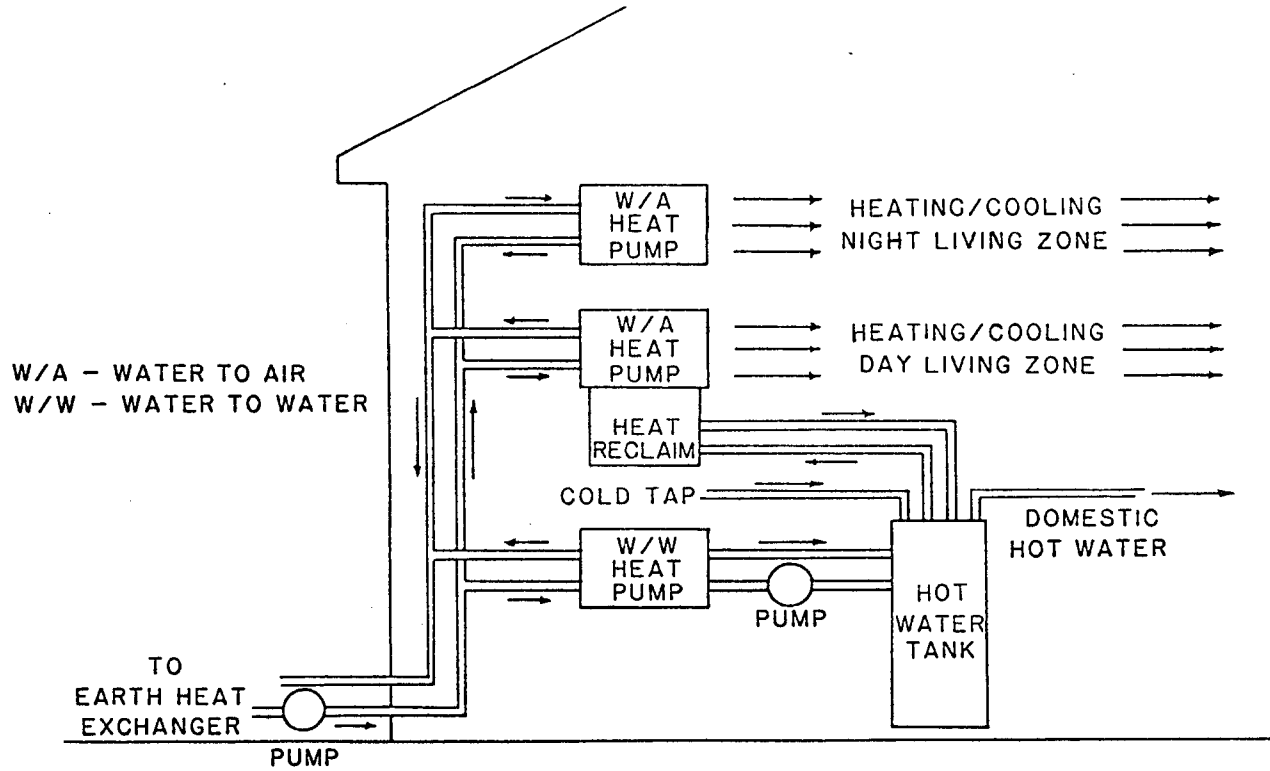
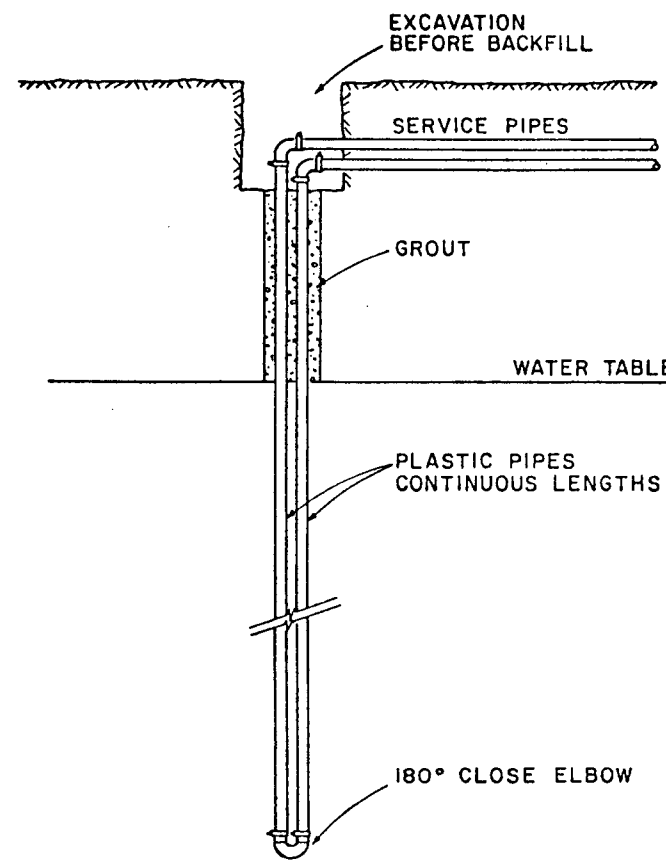


Figure 1. Closed-loop water flow in parallel for three water-source heat pumps.

Electric Sub-Metering and Equipment Specifications

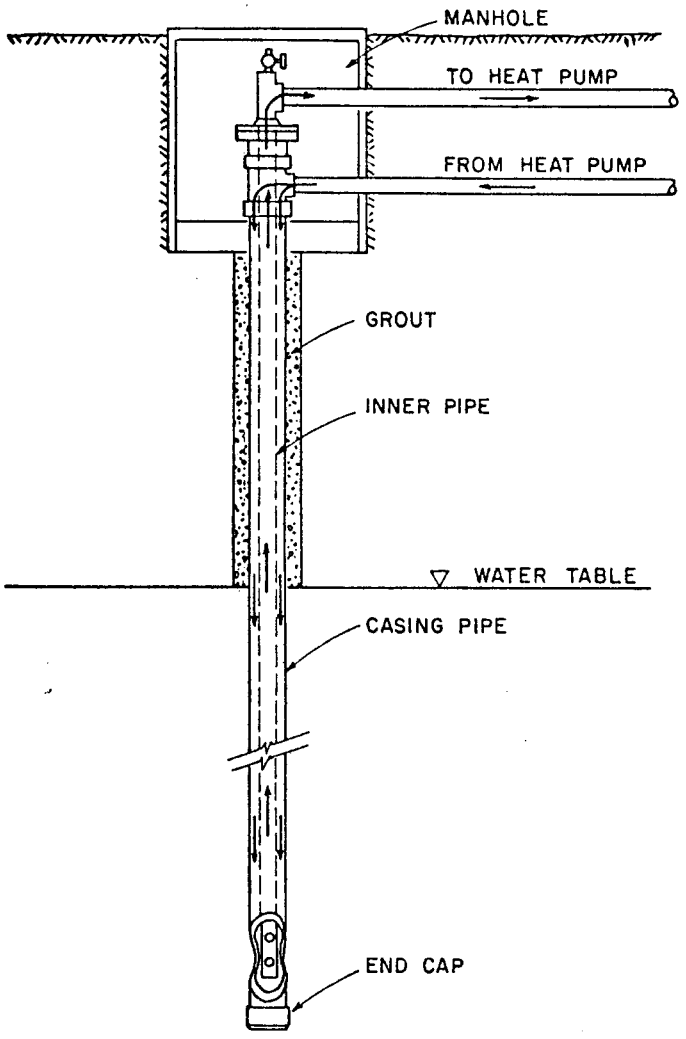
Unit	Specifications
Water Heater Heat Pump	Fedders Model SOCFO20, 5.6 kw, COP=5.0 @ 49°C (120°F) delivery, 21°C (70°F) entering water
Space Heat/Cool Heat Pump Day Living Zone	Florida Heat Pump Model HE 27 EER=11.5 @ 29.4°C (85°F) entering water, COP=3.8 @ 18.3°C (65°F) entering water
Space Heat Cool Heat Pump Night Living Zone	Friedrich Model 804024 GE. EER=10.4 @ 29.4°C (85°F) entering water. COP=3.9 @ 21°C (70°F) entering water
Water Circulating Pump for Heat Pumps	0.186 kw (¼ HP) Flint and Walling Centrifugal Vari jet
Well Water Pump for Domestic Supply	0.747 kw (1 HP) Ruth Berry shallow well centrifugal with jet
Electric Clothes Dryer,	Whirlpool Model LE7000XKWO 5.2 kw
Lighting and Appliances (Except as Listed Above) (By Subtraction)	

Figure 2



U-BEND DESIGN
(Typical Detail)

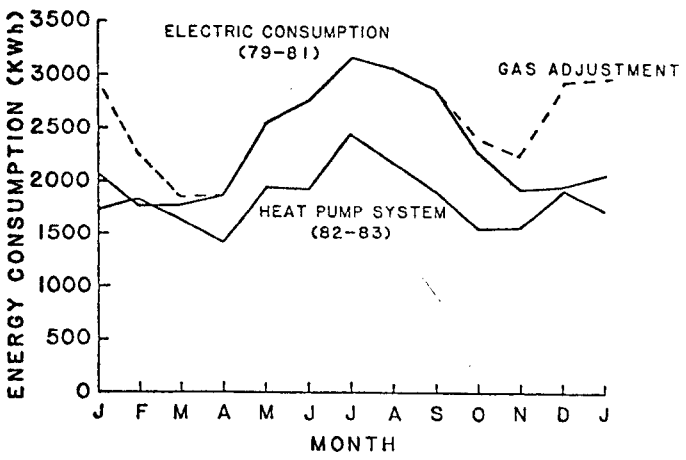
Figure 3.



CONCENTRIC PIPE
(Typical Detail)

Figure 4.

Figure 5. Energy consumption for residence. Electric consumption (79-81) line is three year average. Dotted line is electric heat energy equivalent of gas added to actual electric consumption for 79-81. Heat pump curve is total use for water-source closed-loop system.



A METHOD FOR PERFORMANCE TESTING OF EARTH COUPLINGS

by James R. Partin

James R. Partin

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A METHOD FOR PERFORMANCE TESTING OF EARTH COUPLINGS

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Summary

The interest in earth coupling a water source heat pump is growing very rapidly. A variety of materials are being used with a still larger number of piping configurations to construct the buried closed loop exchangers. As a result it is difficult to compare different earth exchangers.

The author proposes definitions of quantities such as far field earth temperature, earth coupling temperature and overall heat conductivity per unit length of earth coupling. Also proposed are methods and philosophies for measuring unknown variables and selecting other arbitrary quantities. Factors that affect long term heat transfer performance such as cycling and total run time are discussed.

Using the proposed methods, actual test data are presented and are reduced to a form producing performance numbers that can be compared directly to data from other locations, configurations, pipe lengths, heat pumps and soils.

Introduction

Literature available in the field is sparce but a short bibliography of both classical and contemporary sources is included. Most recent results refer to earth coupled heat pump system coefficients of performance. While the systems are very efficient, little effort has been expended to develop a systematic procedure for performance testing of earth couplings.

We desire to performance test an earth coupling in such a manner that data from one configuration and location can be compared to those from another. Since we have no simple means to measure quantities such as soil density, moisture content and conductivity directly, a test procedure is proposed whose results include the effects of these variables, but does not explicitly measure any soil property.

As a practical matter we define an overall U_o at a given point in time to be

$$U_o = \frac{Q}{L\ \Delta T}\ .$$

U_o is the amount of heat that can be rejected to (extracted from) the earth coupling per unit of characteristic length L and temperature difference ΔT with far field temperature.

Here we inject some philosophy into the definitions of Q, L and ΔT. The heat rate Q may vary somewhat with time when real operating systems are used to load the earth coupling. It is preferable to hold Q constant through a long period or through a short cycle. Figures included later show U_o values plotted for both sustained run and cycles while Q is held constant. If the heat rate has varied during the duration of a measurement sequence it is preferable to use the latest or final rate Q in calculating U_o. (A plot of U_o is similar to a trajectory problem where recent guidance and thrust are more relevant than events in the past.)

The characteristic length of the earth coupling takes on slightly different definitions for horizontal and vertical earth coupling configurations. For U-bend geothermal wells we choose to measure L as the depth of the backfilled bore. The most common wells utilize two pipes per bore hole however some experimentation has involved four pipes. For horizontal (trenched) earth couplings we measure L as the length of the backfilled trench. It is typical to have two pipes per trench, however, as with wells more than two are sometimes placed in the same trench and spaced apart.

The temperature difference ΔT between the earth coupling and far field depends on a philosophy of defining these quantities. We choose to define the earth coupling temperature T_e as the average of water-in and water-out temperatures. This will prove handy working with heat pumps since the earth coupling average temperature is the same as the heat pump average water temperature. Far field temperature T_{ff} is that temperature we obtain when water is circulated in the earth coupling but before the actual application of a significant heat load. The temperature differences are then

$$\Delta T_c = T_e - T_{ff} \qquad \text{(cooling)}$$

$$\Delta T_h = T_{ff} - T_e \qquad \text{(heating).}$$

Since the earth coupling is to be tested by rejecting heat to the earth, we use for earth ΔT when calculating U_o:

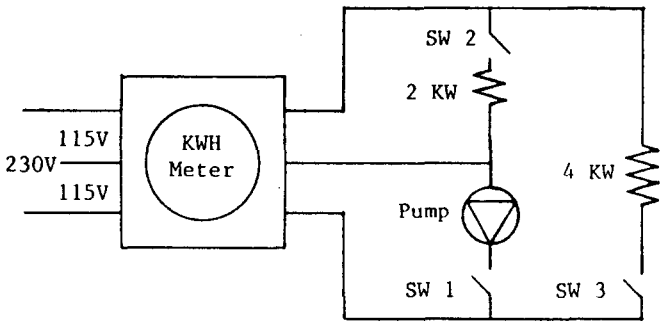
$$\Delta T = \Delta T_c = T_e - T_{ff}$$

Having defined the components of U_o we can now calculate it at a given time by measuring Q, L and ΔT.

Measurement Apparatus

The objective of the test is to calculate U_o accurately by measuring heat rejected, characteristic length and earth coupling temperature difference with

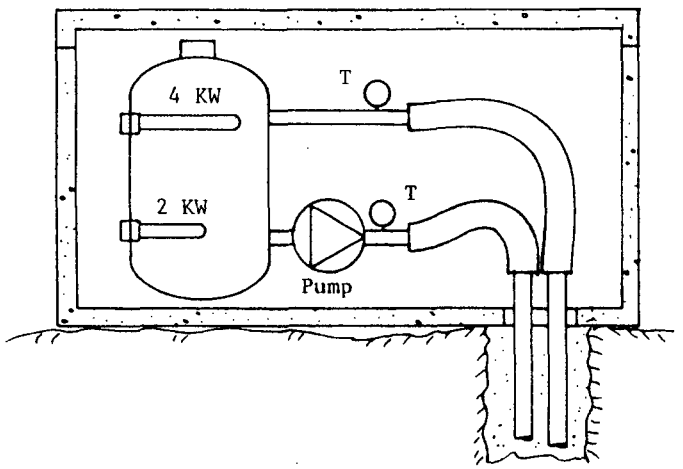
the earth. To accomplish this we heat water using electric heater elements and circulate it through the earth coupling, measuring KWH (BTU/H) with a watt hour meter, temperature in and out, and time. Figures 1 and 2 show the electrical and water piping schematics for the test.



Electrical Schematic

Figure 1

Insulated Case



WATER PIPING

Figure 2

A list of equipment includes:

- Low wattage circulator pump
- Water reservoir
- Electric heating elements
- Electronic probe thermometer accurate to 0.1 degree F
- KWH meter accurate to 0.01%
- Insulated container
- Miscellaneous switches, valves, hookup hardware
- Digital time piece

Water flow rate can be set indirectly by throttling the flow to obtain approximately 10°F difference between entering and exiting water to the reservoir.

The volume of the reservoir should be small compared to the earth coupling.

The insulated container minimizes the effects of wind and air temperature on the test. A foam lined ice chest is suitable for this container.

Testing the Earth Coupling

Pretest Data

Physical Data. Information about the earth coupling pipe and the manner in which it is buried must be recorded. These data will prove useful in making comparisons to other systems. Pipe information should include:

- Nominal size and pressure rating
- Material specifications
- OD and ID
- Length
- Number pipes per bore/trench
- Spacing of pipes

The excavation (bore hole or trench) information should include:

- Location
- Diameter/width of bore/trench
- Depth of bore/trench
- Backfill material
- Depth of bore/trench in saturated moisture conditions
- Soil Description

Characteristic Length. Simply measure the total bore depth for wells or the trench length for horizontal systems. It may be practical to drop a weight on a measuring tape down a U-bend well if the depth is in doubt. Record this length L for later use in calculating U_o.

Far Field Temperature. Fill the earth coupling with water and allow it to come to equilibrium. One might allow at least 24 hours for this if the plastic pipe had not been filled prior to the test. Ordinarily the pipe will have been filled at the time of burial and the water inside would be at far field temperature.

Turn on the circulator pump and observe the water temperature returning from the earth. Some judgement will be required in selecting T_{ff} since the earth temperature varies with depth below the surface. Record T_{ff} for subsequent use.

Taking Run Data

The time at which the heat load is applied to the earth coupling would be the initial entry. One would simply record the date, time, remarks, supply and return temperatures and KWH. Test time intervals of one hour for short term and 12 or 24 hours for long term tests seem most appropriate to yield good results.

Calculating Overall Conductivity

A simple procedure that is adaptable to a computer is used to calculate U_o at each time data is taken after initial startup. The procedure for each of the calculated values follows:

Column 2:

$$\Delta time = \text{Current Time} - \text{Previous Time}$$

$$= ①_R - ①_{R-1}$$

Column 5:

$$\text{Avg Temp } T_e = \frac{\text{Supply Temp} + \text{Return Temp}}{2}$$

$$= \frac{③ + ④}{2}$$

Column 6:

$$\Delta T = T_{ff} - \text{Avg Temp}$$

$$= T_{ff} - ⑤$$

Column 8:

$$\Delta KWH = \text{Current KWH} - \text{Previous KWH}$$

$$= ⑦_R - ⑦_{R-1}$$

Column 9:

$$BTU/H = \frac{3412\ \Delta KWH}{\Delta time}$$

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Conclusions

Selecting Design U_o

Actual test data from a 1" IPS Polybutylene U-bend well system are given in figure 3 and plots of the first three days' test are given in figure 4. Changes in heat loading and some effects of cycling are illustrated in figure 5.

We conclude that this geothermal well under short term loading has a U_o in the range of

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$$U_o = 2.0 = \frac{Q}{L\ \Delta T} \quad \text{(design value)}$$

EARTH COUPLING TEST DATA

EXCAVATION:

Location: A. G. Service - Stillwater, OK

Diameter/width of bore/trench: 6"

Depth of bore/trench: 75

Backfill material: drilling tailings + mud

Depth to saturated moisture cond.: 2 ft

Soil description: Renfro clay + sandrock

PIPE:

Nominal size & pressure rating: 1" IPS 160 psi

OD and ID: 1.231, 1.049

Material specs.: 2110 Polybutylene

Length: 73 ft wetted borehole

No. pipes per bore/trench: 2

Spacing of pipes: together in U-bend

Characteristic Length, L = 73'

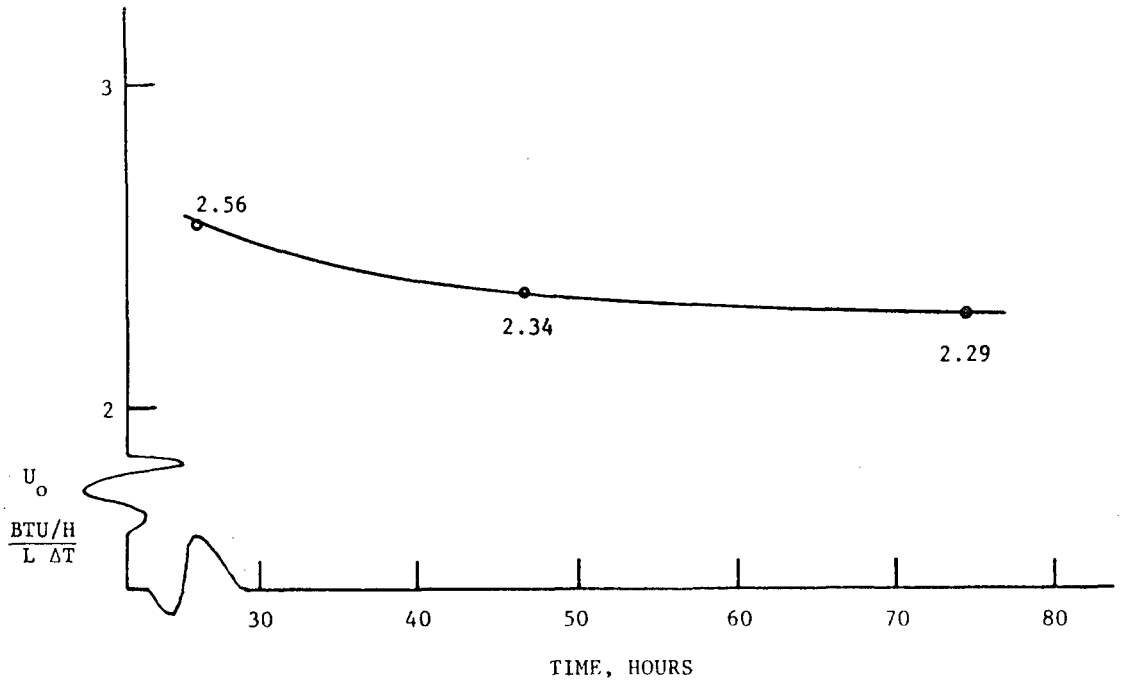
Far Field Temperature, T_{ff} = 62.4°F

4 KW Heater used.

◯ - Column Number
R - Row Number
✓ - Original Recorded Data

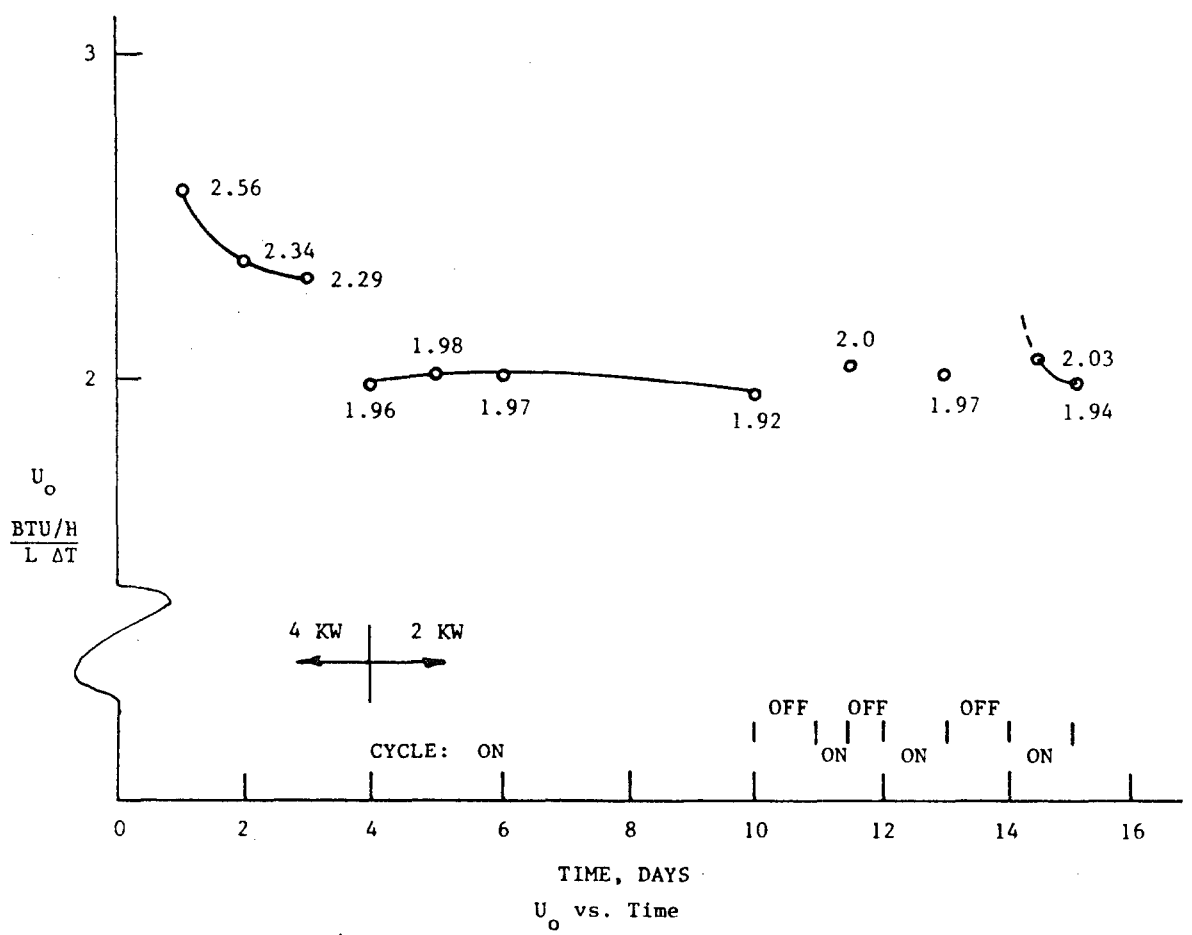
Row No.	Remarks	1984 Date	① Time	② Δ Time	③ Supply Temp	④ Return Temp	⑤ Avg Temp	⑥ Earth ΔT	⑦ KWH	⑧ ΔKWH	⑨ BTU/H	⑩ U_o
	✓	✓	✓	$①_R - ①_{R-1}$	✓	✓	$\frac{③+④}{2}$	$T_{ff} - ⑤$	✓	$⑦_R - ⑦_{R-1}$	$\frac{3412\ ⑧}{②}$	$\frac{⑨}{L ⑥}$
01	Startup	6/1	8:30A	—	—	—	—	—	1644.0	—	—	—
02	24 Hr read	6/2	10:30A	26.0	140.4	128.8	134.6	72.2	1746.8	102.8	13491	2.56
03	48 Hr read	6/3	8:30A	22.0	148.3	135.9	142.1	79.7	1834.5	87.7	13601	2.34
04	72 Hr read	6/4	8:30A	24.0	149.8	138.1	143.95	81.55	1930.2	95.7	13605	2.29

Fig. 3



U_o vs. Time

Fig. 4



U_o vs. Time

Fig. 5

Matching Length to Real System

If one solves this equation for L,

$$L = \frac{Q}{2.0\ \Delta T},$$

we have a method for sizing the earth coupling as a function of Q (rejected or extracted heat) and cooling or heating ΔT. To go further in the design process one needs heat pump data:

Heating Mode:

Q_{ext} - heat extracted from earth at temperature T_{eh}

T_{eh} - Minimum design water temperature (average in and out)

Cooling Mode:

Q_{rej} - heat rejected to earth at temp T_{ec}

T_{ec} - maximum design water temperature (average in and out)

Knowing the heating and cooling machine data one can calculate a heating and cooling earth coupling length:

$$L_h = \frac{Q_{ext}}{U_o\ \Delta T_h}$$

$$L_c = \frac{Q_{rej}}{U_o\ \Delta T_c}$$

where

$$\Delta T_h = T_{ff} - T_{eh}$$

$$\Delta T_c = T_{ec} - T_{ff}\ .$$

Having calculated both the heating and cooling earth coupling length, it would be sound to choose the longer of the two for design purposes.

Experience has shown that the cooling length will be longer in southern U.S. climates while the heating length dominates in northern climates.

The example given here has led to the selection of a design $U_o = 2.0$ This is for a relatively small and thick walled polybutylene U-bend in a wet bore hole. Larger $1\frac{1}{2}$ inch polybutylene U-bends have yielded U_o values near 3.0. In general the larger the pipe, the greater the U_o value for a given location, pipe type, and configuration.

Design U_o From Short Term Test

As more short and long term data are gathered it would be very desirable to be able to select the design value of U_o based on a fairly short term test, i.e., 12 hours or less. It is difficult to perform long term tests and sustain constant conditions. In the case of horizontal trenched earth coils the far field temperature T_{ff} is not constant but changes slowly throughout the year in a predictable sinusoid-like wave whose temperature depends on depth, time of year, soil conditions and so on. For these horizontal systems a short term test may have more meaning than long term, provided a rationale or philosophy for derating short term performance data is developed. For instance, in this example, the selected design $U_o = 2.0$ is about 77% of the 24 hour value. More testing using this or some other organized procedure should ultimately yield a valid and complete method for short term prediction of the design U_o.

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OPTIMIZATION OF VERTICAL GROUND-COUPLED HEAT PUMPS

by Steve Kavanaugh

Steve Kavanaugh

Steve Kavanaugh is a graduate student in the Mechanical and Aerospace Engineering Department at Oklahoma State Unviersity. Prior to his enrollment at OSU he was an instructor of Mechanical Engineering at Lamar University in Beaumont, Texas. Previous research has included comparison of open and closed loop water source heat pump systems and energy efficient house design in humid climates. His experience has included two years as a utilities engineer in the petrochemical industry.

Presented at the 7th Heat Pump Technology Conference
at Tulsa, Oklahoma, October 15-16, 1984

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Steve Kavanaugh

School of Mechanical and Aerospace Engineering
Oklahoma State University
Stillwater, Oklahoma 74078

Introduction

This paper reports on the current status of a project at Oklahoma State University investigating methods of optimizing vertical ground-coupled heat pumps. In many applications vertical ground couplings may be preferable to horizontal systems. Vertical couplings can be installed in locations of limited ground area. Secondly, ground temperatures at depths greater than 25 feet have negligible annual temperature variation, while this temperature variation for horizontal couplings at a depth of four feet is typically 10° to 20°F above and below the average yearly air temperature.

Vertical ground coupling installations require 100 to 175 feet of bore for a heat pump with a nominal rating of one ton. This length varies with soil conditions and type of ground coupling. Installation costs vary from $3.00/ft. to above $10.00/ft., again depending on soil conditions and the type of ground coupling used. At $5.00/ft. this would raise the first cost approximately $750.00 per nominal ton above a heat pump of standard efficiency. A variety of ground coupling designs are used and general trends concerning optimum arrangements are known. However, direct comparisons cannot be accurately made between systems because of the large number of variables that cannot be controlled or accurately measured. These variables include ground thermal properties, ground moisture content, heat pump operation characteristics and installation procedures.

This project proposes to develop a model that can be used for both design and system simulation and is currently about 75% complete. The approach to development is as follows:

1. Comparative experimental study of six ground couplings operating in parallel.
2. Experimental study of convective heat transfer coefficients in vertical ground couplings.
3. Application of experimental results and heat pump performance to ground-coupling system simulation using finite difference equations (F.D.E.'s).
4. Application of experimental data, F.D.E. results and heating/cooling load calculations into a simplified design and simulation method.

Comparative Ground-Coupling Study

The experimental system comparing six typical vertical ground couplings is shown in Figure 1.

The heat pump used in the experiment has a nominal capacity of 48,000 Btuh. Manufacturer's data state that this unit is capable of operating with entering water temperatures as low as 45°F. Water exits the heat pump into a common header and branches off into the six 100 foot ground couplings. Water returns to the heat pump through individual pipes. The water from each loop passes through a valve and flowmeter before mixing into a common header at the pump suction. Two small circulating pumps operating in series discharge into the heat pump. Pump power input ranges from 300 watts at 7.5 GPM to 350 watts at 12.0 GPM.

The operation of the heat pump is controlled by a timer rather than a thermostat. This timer is capable of cycling the unit in 15 minute increments and is programmed on a 24 hour basis. During the winter portion of testing, the unit was operated 75% at night and 50% during the day, unless other tests were performed. This level of operation was maintained for 2 1/2 months for an equivalent run fraction of approximately 65% from mid-January until April 4.

Peak heat transfer rates of earth coupled heat pumps are typically lower in the heating mode than in the cooling mode of operation. The amount of heat absorbed by the coupling device is the unit capacity less the input of the compressor and pumps. However, it cannot be assumed that earth couplings sized to the cooling load are sufficiently sized for heating operation even in areas of moderate heating requirements. The heating mode of operation is critical because of the possibility of freezing in the evaporator and because of the sharp drop in capacity of heat pumps not specifically designed for operation with low water temperatures.

The necessity of antifreeze solutions in vertical ground couplings in Oklahoma is debatable. It is necessary to examine actual heat absorption rates and local ground conditions before making a decision. However, it appears that in installations with a climate and ground properties similar to Stillwater, antifreeze is unnecessary provided that ground couplings similar to those used in this test are installed with 150 to 175 feet of hole per nominal ton capacity and water flow rates at least 3 GPM per ton. This assumes that the

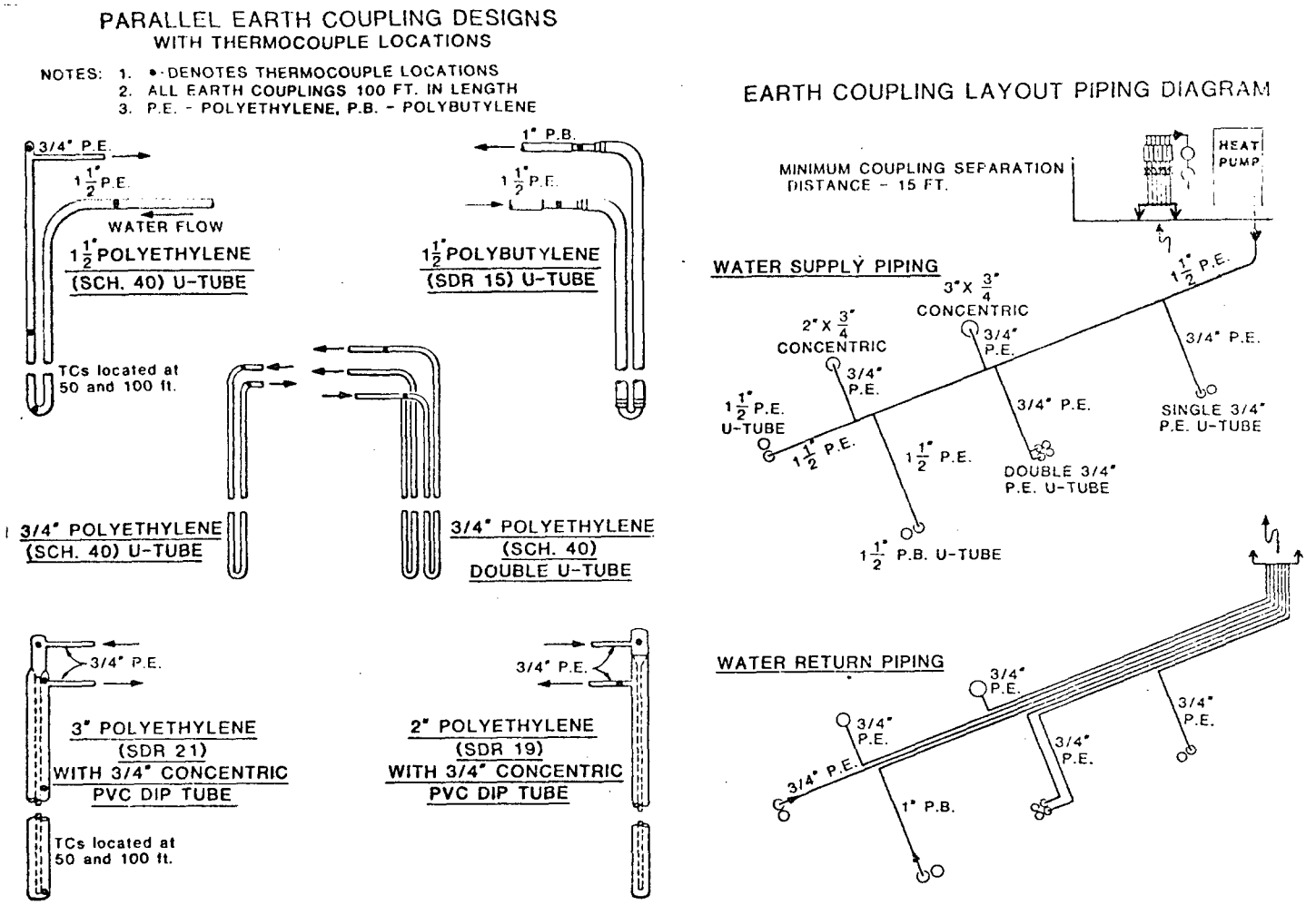

FIGURE 1 - COMPARATIVE GROUND-COUPLING EXPERIMENT

system will be sized to the load and the monthly run fraction will not exceed 75%. No antifreeze solution is being used in this experiment. The unit shuts down when the water leaving the heat exchanger drops below 36°F.

Caution must be used in applying the temperatures recorded in these tests directly to installations. Figure 2 appears to indicate the performance of 5 of these loops is almost identical. However, since the loops are returned to a common pipe, the ones having a larger heat transfer capacity compensate for the ones of lower capacity.

Examination of Figure 2 shows the results of a continuous run performed March 17-18, after the system had been operated at over 60% run time for two months. This test would be equivalent to the load experienced by a properly sized unit operating in Stillwater with a 13°F outdoor temperature for 16 hours. The test was preceded by an eight hour 75% run, which included a 15 minute off period immediately before the unit was turned on. The performance results are typical. The double 3/4 inch polyethylene U-tube normally returns water 0.2 to 0.6°F warmer than the 1 1/2 inch U-tubes and the 3 inch concentric coupling. The 2 inch concentric arrangement matches the performance of the 1 1/2 inch U-tubes for continuous runs of 4 hours or less,

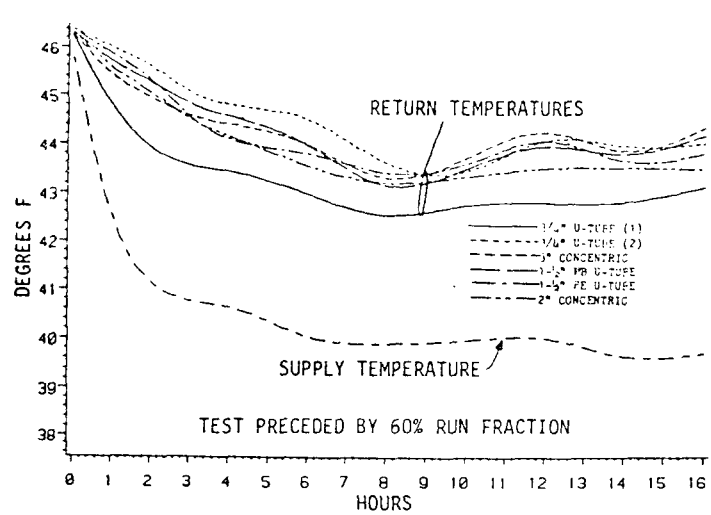

FIGURE 2 - CONTINUOUS RUN AT 2.0 GPM PER LOOP

but temperatures are 0.2 to 0.4°F lower for most other runs. The single 3/4 inch polyethylene coupling typically returns water 1.0 to 1.2°F cooler than the double 3/4 inch. This translates into

Column 5:

$$\text{Avg Temp } T_e = \frac{\text{Supply Temp} + \text{Return Temp}}{2}$$

$$= \frac{③ + ④}{2}$$

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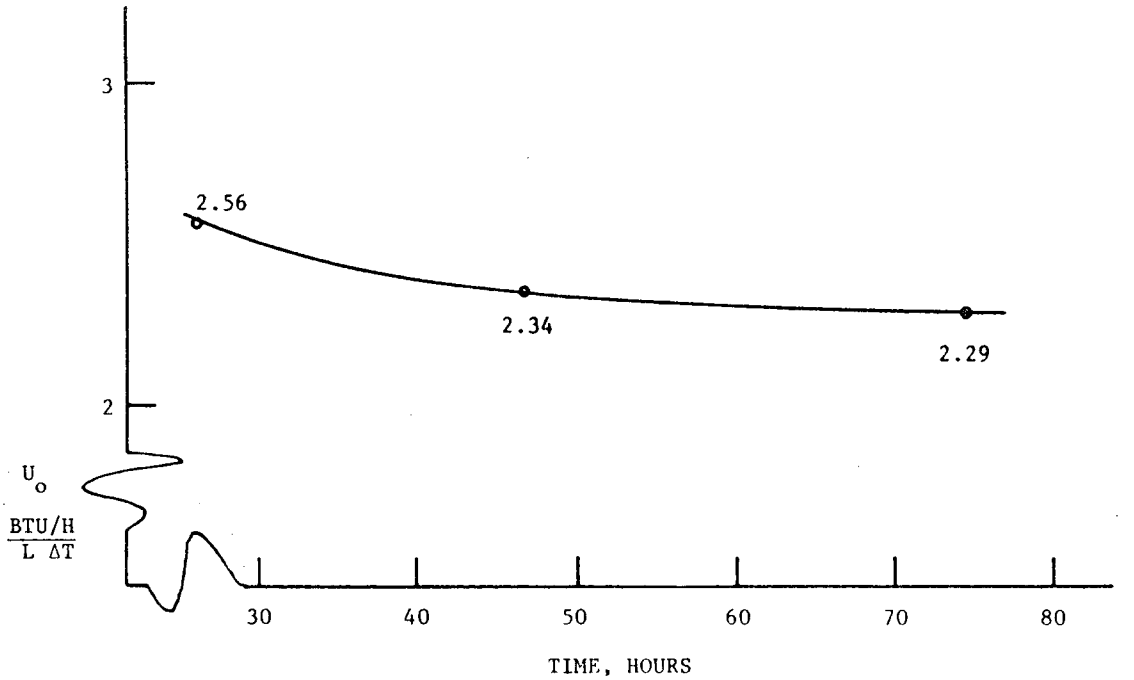
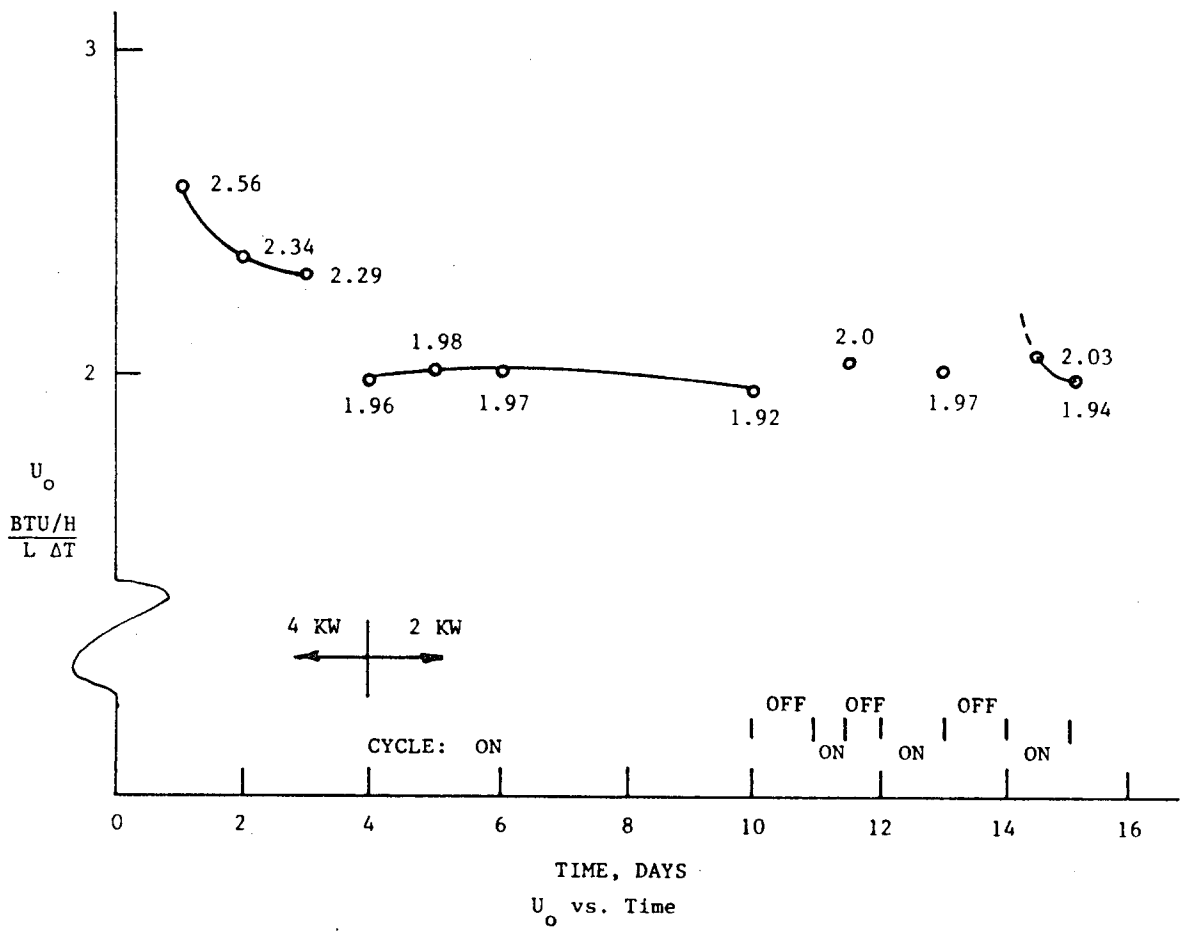
Fig. 3



U_o vs. Time

Fig. 4



U_o vs. Time

Fig. 5

Simplified Model

The performance of standard air-coupled heat pump systems are primarily dependent upon outdoor air conditions. Capacity and efficiency can be calculated by knowing the outdoor temperature. Similar performance calculations of ground-coupled heat pump systems are dependent on a great many more variables. These include soil temperature, soil moisture, soil density, pipe thermal characteristics, pipe size, pipe length, water flow rate and amount of time the system has operated before the performance calculation is made. This last variable requires that some type of building load simulation be performed in conjunction with a simulation of the ground coupling in order to calculate the water temperature entering the heat pump. Since peak conditions usually occur two to four months from the beginning of a heating or cooling season, the simulation should be performed over this entire period. Exact models therefore require a large amount of computation.

Presently the alternative to this type of model would be estimating return water temperature by a rule of thumb. However, rules of thumb can vary significantly in ground coupled heat pumps with soil conditions and load patterns.

An alternative method is proposed by Bose, Parker and McQuiston[6]. The simplified model of this project incorporates the modified Kelvin Line Source Theory utilized in (6) in connection with a transfer function load calculation program developed by Harris[7]. The method will be verified and supplemented by the results of the experimental system and F.D.E. computer programs previously mentioned. This portion of the project is currently in the developmental stage.

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SMALL SCALE TESTING METHODS TO PREDICT EARTH COIL PERFORMANCE

by Rick J. Couvillion

Rick J. Couvillion

Dr. Couvillion, Age 31, received his BS degree from the University of Arkansas in 1975 and then was a project engineer for ALCOA in Bauxite, Arkansas, until he entered graduate school in 1977 at Georgia Tech, where he received his PhD in 1981. His thesis was an analytical and experimental investigation of drying near buried heat sources. Dr. Couvillion has been on the Mechanical Engineering Faculty at the University of Arkansas since November 1981, and has taught courses in Thermodynamics, Fluid Mechanics, Heat Transfer, Thermal Systems Design and Numerical Methods. Research interests include moist porous media heat and mass transfer, biomedical fluid mechanics, and heat transfer problems in poultry processing.

Presented at the 7th Heat Pump Technology Conference
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SMALL SCALE TESTING METHODS TO PREDICT EARTH COIL PERFORMANCE

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Summary

Current earth coil design methods are based on line source theory. Unfortunately, these methods cannot account for a number of factors - including moisture movement and drying, shrinkage, surface affects, etc. - which can have a dramatic effect on earth coil performance. Eventually computer models of earth coils which include these factors should be developed. Until they are, small scale experimental methods offer a way to quickly and inexpensively determine how well an earth coil will perform in a particular soil environment. Small scale tests can also provide valuable information to aid in computer model development which cannot be obtained from field data.

The theoretical basis for small scale testing is presented in this paper along with proven and potential applications. A procedure for developing a computer model with the aid of small scale tests is also presented.

Introduction

A Swiss patent issued in 1912 was the initial recognition of the benefits of earth coupled heat pumps. Later, the thermodynamic advantage was dramatically demonstrated in the 1940s by using brine circulated through buried metallic coils as a heat source. Soil corrosion problems made earth coils impractical and forced the development of air source heat pumps in use today. The corrosion problems have been overcome recently by the use of plastic piping, and research on earth coupled heat pumps is again being pursued. Full scale experimentation carried out at Oklahoma State University[1], Oak Ridge National Lab[3], Louisiana State University[2], and Brookhaven National Lab[4] indicates that both horizontal and vertical coils will work for both heating and cooling when installed in a suitable soil environment.

Prediction of how well an earth coil will perform in a particular soil environment is often very difficult because there are a number of factors which can have a dramatic effect on coil performance. These include

1. moisture movement and drying
2. surface effects such as ambient temperature variaion, rainfall, evaporation, solar flux, etc.
3. shrinkage of soils with a high clay fraction
4. water table location
5. variation in soil properties along the length of the coil
6. rewetting of the soil around the coil through fissures in the soil.

The effects of items 1 and 3 above are often observed in field tests[3] and are especially harmful during the summer cooling season. In the summer the earth coil rejects heat to the soil and causes soil mosture to move away from the coil, leaving drier soil adjacent to the coil. For most soils the thermal conductivity drops sharply with moisture content, and thus the dried soil acts as a layer of insulation, causing the coil temperature to rise for a given heat rejection rate. If the heat pump is not used for heating during the winter, the soil temperature at the coil surface typically returns to the far field temperature, but the soil rewets only slightly. Thus, when the heat pump is reactivatated the following summer, the soil around the coil is drier further, resulting in a further decrease in the performance of the coil. If the heat pump is used for heating in the winter, heat is extracted from the soil and flows to the coil; consequently, moisture is drawn back toward the coil. Whether or not the initial moisture content is recovered depends on how long the heat pump is operated in heating mode. The problems caused by drying are much worse for some soils with a high clay content. These soils often shrink and harden when dried, leaving an air gap around the earth coil. The effect on hear transfer is devastating and often permanent.

The effects of all of the factors listed above have been observed in field tests, but there are no present methods, analytical or experimental, which can predict their occurrence. Most analytical models currently used for design are based on line heat source theory and account for few, if any, of these factors. Thus, a significant safety factor is usually added to current designs in the form of additional earth coil, increasing the cost of the systems and making them less competitive with conventional systems. Even with this large safety factor the design can fail dramatically if drying and shrinkage occur.

Methods to inexpensively, quickly, and accurately predict the performance of an earth coil in a particular soil environment should be developed. These methods will allow the cheapest possible working system to be offered to a potential customer and thus help develop markets for earth

coupled heat pumps. The methods will also determine when a soil environment is simply not suitable for an earth coil, preventing the dramatic failures and unhappy customers which can seriously damage the potential market for earth coupled heat pumps. Eventually, a computer model should be developed which can consider all of the factors listed earlier and accurately predict coil performance; the inputs to this model should require that only a few simple lab or field tests be performed on a small sample of soil. Small scale testing should play a key role in the development and confirmation of this model. Until such a model is developed, small-scale experimental methods can potentially be used to quickly and inexpensively determine if a soil is suitable for use as a heat source or sink.

Theoretical Basis

The theoretical basis for small scale testing which can account for the effects of moisture movement and drying are presented in this section. The basis for small scale tests which include the other factors listed earlier can also be derived but are not presented here.

The two governing equations for low intensity heat and mass transfer in moist soil are the conservation of mass

$$\partial\theta/\partial t = \nabla \cdot (D_\theta \nabla_\theta + D_T \nabla T) \qquad (1a)$$

and conservation of energy[5]

$$C\partial T/\partial t = \nabla \cdot [\rho_2 h_{21} D_{\theta v} \nabla\theta + (k + \rho_2 h_{21} D_{Tv}) \nabla T] \qquad (1b)$$

where

C = volumetric heat capacity
ρ_2 = density of water
h_{21} = enthalpy of vaporization
θ = volumetric moisture content
k = thermal conductivity
D_θ = isothermal water diffusivity
$D_{\theta v}$ = isothermal vapor diffusivity
D_T = thermal water diffusivity
D_{Tv} = thermal vapor diffusivity

Define dimensionless variables

$S = \theta/\varepsilon$
$T^* = (T - T_i)\, k_0/q'$
$R = r/r_0$
$Fo = \alpha_0 t/r_0$

where

k_0 = dry conductivity
α_0 = dry diffusivity
r_0 = coil tubing radius
q' = heat transfer rate per length of coil
ε = porosity
T_i = initial temperature
$\nabla^* = r_0 \nabla$
C_0 = dry volumetric heat capacity

Using these dimensionless variables in the governing equations yields

$$\frac{\partial S}{\partial Fo} = \nabla^* \cdot \left[\frac{D_\theta \nabla^* S}{\alpha_0} + \frac{D_T q' \nabla^* T^*}{k_0 \alpha_0 \varepsilon} \right] \qquad (2a)$$

$$\frac{C}{C_0} \frac{\partial T^*}{\partial Fo} = \nabla^* \cdot \left[\frac{\varepsilon \rho_2 h_{21} D_{\theta v} \nabla^* S}{q'} + \frac{(k + \rho_2 h_{21} D_{Tv}) \nabla^* T^*}{k_0} \right] \qquad (2b)$$

For an impermeable, constant-flux coil the boundary conditions are written as

$$\frac{D_\theta \nabla^* S}{\alpha_0} + \frac{D_T q' \nabla^* T^*}{k_0 \alpha_0 \varepsilon} = 0 \quad \text{at } R = 1 \qquad (3a)$$

$$\frac{\varepsilon \rho_2 h_{21} D_{\theta v} \nabla^* S}{q'} + \frac{(k + \rho_2 h_{21} D_{Tv}) \nabla^* T^*}{k_0} = \frac{-1}{2\pi} \quad \text{at } R = 1 \qquad (3b)$$

and the initial conditions are

$$S(\infty, Fo) = S_i \qquad (4a)$$

$$T^*(\infty, Fo) = 0 \qquad (4b)$$

The dependent variables in Eqs. (2a - 4b) are S and T*. The independent variables are Fo and R. Since the tube radius r_0 does not appear anywhere in these equations, the resulting solutions are independent of r_0 if the coefficients in Eqs. (2a - 4b) are not dependent on r_0. The terms ε, ρ_2, h_{21}, k, k_0, and α_0 are properties and thus are independent of r_0. The heat transfer rate per unit length q' is also independent of the source radius. The diffusivities $D_{\theta v}$, D_{Tv}, D_T, and D_θ are functions of θ and T, or in terms of S and T*

$$f(\theta, T) = f(S\varepsilon, T_i + q'T^*/k_0)$$

Thus, solutions for S and T* as functions of Fo and R are independent of the source radius r_0. However, for a given soil the solutions are functions of the initial conditions θ_i, T_i and the boundary condition q'.

Now consider a heat source/sink which varies with time. If the source strength varies sinusoidally

$$q' = q_0' \sin wt$$

where q_0' is the peak strength and w is one cycle per year. The boundary condition becomes

$$\frac{\varepsilon \rho_2 h_{21} D_{\theta v} \nabla^* S}{q_0'} + \frac{(k + \rho_2 h_{21} D_{Tv}) \nabla^* T^*}{k_0} = \frac{-1}{2\pi} \sin(w^* Fo) \text{ at } R=1$$

where

$$w^* = w r_0^2/\alpha_0$$

$$T^* = (T - T_i) k_0/q_0'$$

Now the solutions are functions of Fo, R, and the parameters q_0', w^*, T_i, and θ_i which appear in the boundary and initial conditions. Again, the value of r_0 does not appear in the equations, and the solutions are independent of r_0.

As an application of these results, consider the problem of simulating a full size 1 1/2 in. diameter coil in the lab with a 1/8 in. diameter tube. The soil used in the lab is the same as in the actual installation, and the initial moisture content, θ_i, and initial temperature, T_i, are also the same. The results derived above show that if the lab coil is operated at the same values of q_0' and w^* as the full size coil, then the values of S

and T* are the same for both coils at equal values of Fo and R. Thus, using the subscript "f" for the full size coil and "s" for the small scale coil,

$$Fo_f = Fo_s \text{ or } \frac{\alpha_0 t_f}{r_{0f}} = \frac{\alpha_0 t_s}{r_{0s}} \text{ or } t_s = \left(\frac{r_{0s}}{r_{0f}}\right)^2 t_f = \frac{t_f}{144}$$

and

$$R_f = R_s \text{ or } \frac{r_f}{r_{0f}} = \frac{r_s}{r_{0s}} \text{ or } r_s = \left(\frac{r_{0s}}{r_{0f}}\right) r_f = \frac{r_f}{12}$$

These results show that a one-year cycle, t_f = 365 days, for the 1 1/2 in. oil can be simulated in a small scale test in t_s = 365/144 = 2.53 days. The dimensions of the soil sample are 1/12 of the full scale test volume. For example, if the full scale test volume is 50 ft. x 50 ft. x 12 ft. (deep), the soil sample used in the lab is 4.2 ft. x 4.2 ft. x 1 ft. (deep). If the total length of the full scale coil is 200 ft., the small scale coil is 200/12 = 16.7 ft.

Applications

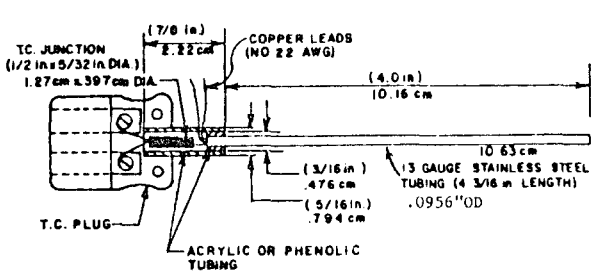
One proven application and one potential application of small scale testing are presented in this section.

Thermal Probe

Thermal probes similar to that shown in Figure 1 have been developed and are currently being used to help determine the capacity of underground electric cables.[6] These probes can be used either in the field or in the lab to determine

(a) the effective thermal conductivity of the soil at its initial moisture content,
(b) the tendency of the soil adjacent to the cable to dry and cause the cable to overheat,
(c) the thermal conductivity of the dry soil.

This test can be completed in 15 minutes or less and requires a soil sample of 0.1 cubic feet or less.



THE HEATER WIRE RUNS DOWN AND BACK THE LENGTH OF THE PROBE. THE THERMOCOUPLE JUNCTION IS LOCATED 5.3 cm FROM THE TIP OF THE PROBE.

Figure 1 - Thermal Probe

The probe consists of a slender tube containing a heating element and a thermocouple which is filled with a high conductivity epoxy. The heat output per unit length of the probe can be controlled by regulating the voltage applied to the heater wire. The probe is inserted into the soil, and the heater is energized at time t = 0. Typical responses are shown in Figure 2 for two different initial moisture contents and heating rates. The T-lnt response for a given moisture content is a straight line until drying begins and the response begins to deviate from this straight line, the drying and deviation beginning sooner for the higher heating rates. The slope of this initial straight line portion is an indication of the effective thermal conductivity of the soil at the initial moisture content. The effective thermal conductivity can be calculated from the familiar line source equation[7]

$$k = q'/4\pi(dT/d\ln t) \qquad (5)$$

As one expects, the slope is greater, indicating lower conductivity, at the lower initial moisture content. After the soil at the surface dries completely and a drying front moves away from the probe, the slope of the T-lnt response again becomes constant and much more steep. This second slope corresponds to the slope that would be predicted by the line source equation for dry soil. Note that all of the curves in the figure become parallel as the surface soil dries in each case. In every case the probe eventually responds as if it is surrounded by dry soil.

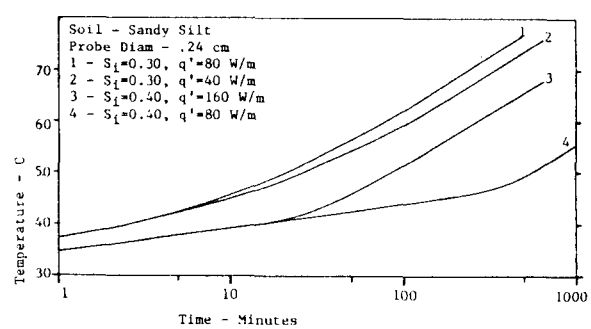


Figure 2 - Thermal Probe Responses

The results of the probe test described above can be used to determine when or if significant drying will occur around an underground electric cable. Let t_p be the time in the probe test at which the T-lnt response begins to deviate from the initial straight line portion. Let t_c, the quantity desired, be the time at which this deviation occurs for a full size electric cable in the same soil at the same initial moisture content with a current such that q' is the same as in the probe test. The results from the previous section show that

$$t_c = t_p \left(\frac{d_{cable}}{d_{probe}}\right)^2$$

The time at which the slope of the cable temperature response reaches the dry value can be calculated in a similar manner from a probe test. A combination of this time and t_c is used to define when significant drying has occurred.[6]

A thermal probe can be used to simulate an earth coil during the cooling season and determine if significant drying will occur. A probe which also has controllable refrigeration capacity would have to be developed in order to simulate the annual

cycle undergone by a reversible heat pump. The effect of both rewet during the heating season and drying during the cooling season could be measured with this probe and used to quantify the performance of the coil in terms of heat flux and heating time/cooling time ratio.

Laboratory Simulator

The earth coil in a typical horizontal installation consists of 150-400 ft. of 1 to 2 in. pipe. The pipe can be replaced by a 1/8 in. or less diameter tube in a lab model. Results given earlier show that all other length dimensions can be reduced by a factor of 12 for a 1 1/2 in. pipe, and a 50 ft. x 50 ft. x 12 ft. (deep) section of soil used by an actual earth coil can be replaced in the lab by a soil sample 4.2 ft. x 4.2 ft. x 1 ft. (deep). The earlier results also illustrate the greatest benefit of small scale testing -- substantial time scale reduction. The annual cycle undergone by a 1 1/2 in. diameter earth coil can be simulated in only 2 1/2 days with a 1/8 in. laboratory scale earth coil.

An elaborate "box" is being built at the University of Arkansas to contain the soil sample, earth coil, and instrumentation. The arrangement is shown in Figure 3. The soil will be prepared at the desired moisture content and packed at the desired density into the box with the earth coil. The operation of the heat pump will be simulated by controlling the flow rate and inlet temperature of the antifreeze mixture pumped through the coil. The annual cycle of the environmental temperature will be simulated by controlling the temperature above the soil. A water table location can be simulated as shown. If needed, a solar flux variation can be added by placing radiant heaters in the box. Perhaps even rainfall simulation could be added. Variation in soil properties in either horizontal or vertical directions can be accounted for by careful packing of the soil. The antifreeze exit temperature will be recorded as a function of time along with the moisture content and temperature at a number of locations in the soil. The apparatus will be controlled by a departmental microcomputer which will also record the required data.

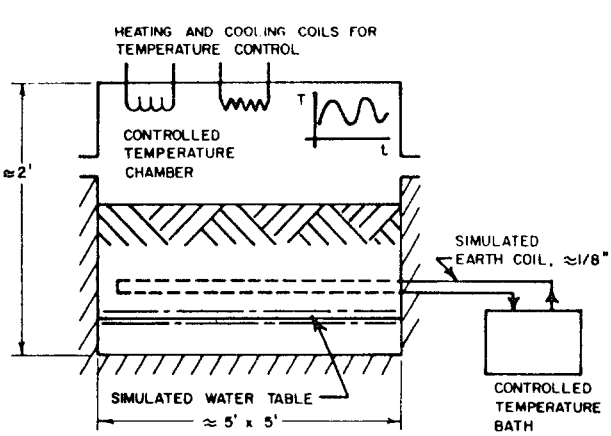


Figure 3 - Earth Coil Simulator

This apparatus will simulate the annual cycle undergone by a full size earth coil in 2 1/2 days. The data taken will allow the thermal resistance of the soil to be determined at any time during the first or subsequent annual cycles of a full size coil. The effects of drying, rewet, and shrinking can be observed and quantified in a short time; it would take a year or more to make the same observations in a field installation.

The simulator will greatly aid the development of an accurate computer model, the eventual solution to the problem of earth coil design. Any computer model should eventually match field data and include all of the factors listed earlier which can have a significant effect on coil performance. The best way to build any model which includes a number of effects is to begin with a fairly simple model, add the effect of one factor, refine the expanded model by comparing it to available data, add the effect of the next factor, and so on until all of the factors have been included and their individual effects on the model verified by data. It is very difficult to use field test data in this type of process because field data includes the effect of all factors, and one cannot determine the relative effect of each factor on the data.

The factors listed earlier can be eliminated or controlled by using full scale laboratory tests to verify proposed models. However, the facilities required to do these tests are very large and expensive, and it takes a long time to perform a test. A full size earth coil, i.e. 1 1/2 in. or so tubing, must be operated for one year in order to measure the effect of drying and rewet which occurs in an annual cycle. The coil must be operated for two or more years if the cumulative effect of dryout and rewet are to be measured.

The factors listed earlier can be either eliminated or simulated in a small scale laboratory setup. In addition, the time required to do a test is substantially less for the small scale lab setup than for a full scale test. The accuracy of a proposed computer model describing coil performance can be determined in only a few days as opposed to the year or more required to obtain field test data or full scale lab data.

The use of a lab setup similar to that shown in Figure 3 will allow a computer model which eventually includes most of the factors listed earlier to be built up in stages. For example, an accurate computer model for a coil in a semi-infinite soil could be expanded to include the effects of constant surface and water table temperatures. The lab setup could be arranged to produce these conditions, and the data obtained would be used to refine the model. The model and lab setup could then be modified to also include ambient temperature variation, and the model would again be refined. The model developed at the end of this process could then be compared to field test data, and further refinements could be made. The procedure described would be a much better approach than attempting to match field test data by including all of the factors listed above in a first model. This same process could be carried out using full-scale lab data, but, as mentioned previously, the time required to set up the test and get the data would be much, much longer.

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A REPORT ON THE ASHRAE PROJECT TO DEVELOP A DESIGN DATA MANUAL FOR GROUND-COUPLED HEAT PUMPS

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Presented at the 7th Heat Pump Technology Conference at Tulsa, Oklahoma, October 15-16, 1984

A REPORT ON THE ASHRAE PROJECT TO DEVELOP A DESIGN DATA MANUAL FOR GROUND-COUPLED HEAT PUMPS

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Summary

A research project at Oklahoma State University, sponsored by ASHRAE, is nearing completion. A draft of the final report has been submitted to the overseeing committee and a revised report will be completed in December, 1984. This paper discusses the objectives of that study and summarizes some of the contents of the final report.

Background

ASHRAE Research Project RP-366 "Development and Preparation of a Design Data Manual for Closed-Loop, Ground-Coupled Heat Pumps," began July 1, 1983 under the oversight of ASHRAE's TC 9.4 and TC 6.9 Technical Committees. Originally intended to be a 12 month study, the project was extended without cost to cover an 18 month period ending December 31, 1984. The objectives of the research study were to:

(1) Collect and organize data from available literature and active projects pertinent to design of the ground coupling device and its use in heating/cooling systems.

(2) Publish a design and installation manual.

(3) Identify additional data needed for the design of these systems.

At the recommendation of the monitoring committee the design data manual technical emphasis was directed to the practicing design engineer. It was to provide data and material for use by the design engineer and not to develop a specific design procedure. By the bringing together the information available in the literature and among those active in the field it was to be a report on the "state of the art" in the ground-coupled industry. It was felt that several "gaps" in information might be filled in, or at least identified and defined more clearly. The study was to be international in scope, covering all climates and every type of soil and building.

A rough draft of the design data manual has been submitted to the ASHRAE monitoring committee. It will likely undergo some changes as a result of this review, and a final version will be submitted in December.

The techniques being used in the design and installation of ground-coupled heat pump systems are changing rapidly. None of the design procedures that have been developed are completely satisfactory and none have been widely adopted. Optimization procedures are very complex and not entirely reliable. This ASHRAE Design Data Manual does provide a lot of useful information and does furnish some design methodologies that are practical and useable. Because of the rapid developments, however, it is likely that any such manual will not be finalized for some time.

Review of the Literature

One of the objectives of the project was to ". . . organize data from available literature . . ." and this was a major emphasis in the early months of the project. A wide variety of literature was surveyed, much of it related to soil properties, soil thermal behavior, and moisture migration in soils. A surprising amount of literature was uncovered, going back several decades, and involving workers in several countries. A bibliography for the Design Data Manual was assembled and is given as a part of this paper. We depended upon previous surveys for part of our efforts, and were especially grateful for the efforts of Ball, et al. (1982).

Contents of the Design Data Manual

The ASHRAE Design Data Manual in its present form consists of three major sections, plus the appendices and bibliography. The first part is an introduction and overview. In this section the ground-coupling concept is introduced and explained, and the earth's temperature variations are described. With this background the thermal behavior of ground-coupling systems are presented. The effect of a horizontal ground-coupling system on the local earth temperature variation is shown in Figure 1. The effect is to create a maximum temperature T_{MAX} in the soil in late summer or early fall due to the cooling operation of the heat pump. Likewise a minimum temperature T_{MIN} is created in the local soil in late winter due to the heating operation. A major objective in heat pump design and operation is to keep one or both of these temperatures within some acceptable bounds. A similar concern exists for vertical ground coupling devices, but they operate more or less free of the seasonal variation in soil temperature that affect the horizontal systems.

The effect of cycling on system performance has been appreciated by most of those with experience in

operating ground coupled systems. This cycling effect and the long term effects of heating or cooling the ground are more easily understood if the Kelvin line source theory, described by Ingersoll and Plass (1948), is used as a study tool.

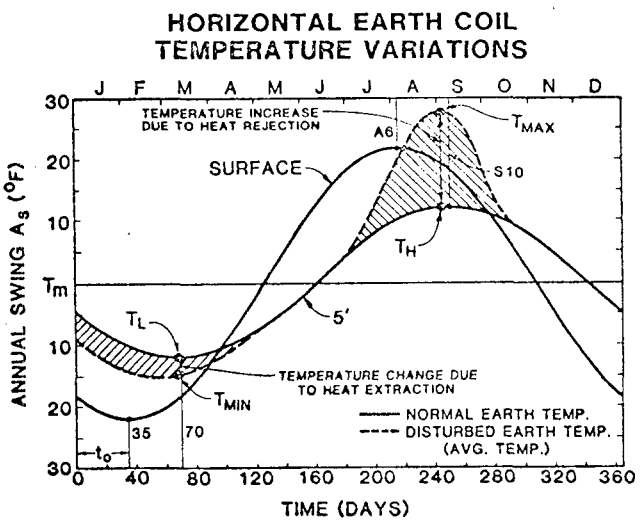


Figure 1. Horizontal Earth Coil Temperature Variations

Figures 2 and 3 show how the theory can be used to determine the effect of different cycling patterns of a heat pump. The effect of recovery of ground temperatures around the pipe when the heat pump is off are shown to be significant, as is the history of prior operation. The dashed lines in Figures 2 and 3 show how averaging of energy inputs may be used to show the long term effects of cyclic operation on ground temperature buildup or decay.

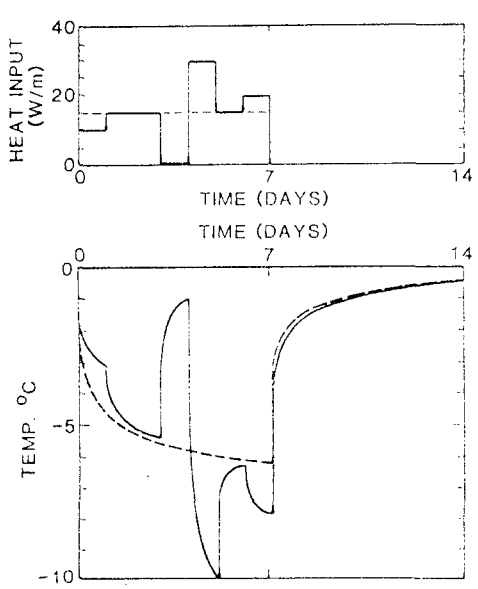


Figure 2. Extraction rate and pipe temperature in a comparison of a constant pulse during a week with variable daily pulses.

This same theory has been used by other researchers to show the inference effects as well as the long term effects of operation of multiple vertical ground coupling systems, Claesson (1983). Figure 4 shows these effects for a 3 x 3 grid of vertical exchangers of various depths and is typical of the kind of information that can be developed with line source theory.

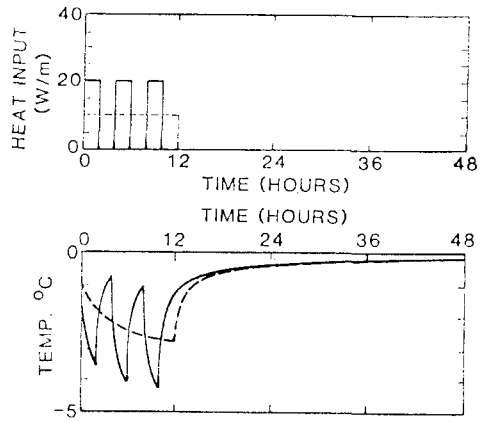


Figure 3. Extraction rate and pipe temperature for two-hour pulses.

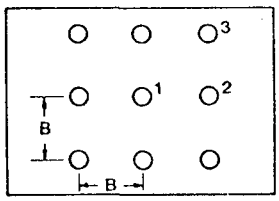
In the second major section of the manual preliminary design procedures are discussed. Significant site, climate and economic factors are given. The importance of a good load calculation is emphasized and a particularly good summary of ASHRAE heating and cooling design load calculations methods is included.

Soil thermal characteristics are a primary factor in any ground coupling design method. The preliminary design procedure section contains a section on this topic and this is reinforced by an appendix section on soil properties. For the first time, soils information related to ground-coupling of heat pumps has been brought together and discussed from this viewpoint. The importance of soil moisture and soil thermal stability are major topics. Figure 5 shows the very strong effect of low soil moisture on the thermal resistivity of several types of soil as well as the large variation of thermal resistivity between soils. Soils are classified in ways that are useful to the ground-coupling designer and practical soil property determination methods are described in the manual.

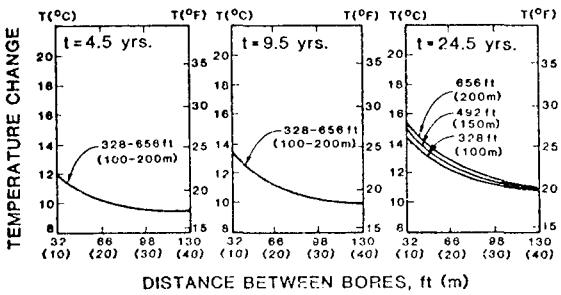
The material on ground-coupling system selection, augmented by an appendix listing ground heat exchanger configurations, should help the reader appreciate the many systems that have been studied, and hopefully to pick out the best system for his own needs.

The third major section covers ground-coupling system design methods. The widely used Kelvin line source theory is applied to develop a field resistance concept for various arrays of pipes buried in the earth. These are useful for comparing different arrangements of trenching and depth of burial, and different pipe diameters. The method gives a very accurate assessment of the interference effects between buried pipes. A design procedure which permits system operation within some specified extremes is given for illustrative purposes. This procedure is the best known for making comparative studies of alternative ground-coupling arrangements. Mathematical background is given in an appendix.

NUMBER OF BORE HOLES: 9
k_s=2.02 Btu/hr·ft·°F (3.5 W/mk) (DENSE ROCK)
PIPE RESISTANCE: R_p=0
INSULATION DEPTH: 16.4 ft (5 m)
CONFIGURATION: 3 X 3



BORE DEPTH	HEAT OUTPUT (ENTIRE FIELD)
328 ft (100 m)	509·10^6 Btu/yr (149.3 MWh/yr)
492 ft (150 m)	765·10^6 Btu/yr (224.3 MWh/yr)
656 ft (200 m)	1020·10^6 Btu/yr (299.2 MWh/yr)



	HEAT OUTPUT (% OF AVERAGE)							
	5 YEARS				25 YEARS			
SPACING (ft/m)	32/10	66/20	98/30	130/40	32/10	66/20	98/30	130/40
BORE NUMBER 1	86	94	98	100	83	87	91	94
BORE NUMBER 2	97	99	100	100	96	97	98	99
BORE NUMBER 3	106	103	101	100	108	106	104	103

HEAT OUTPUT FOR SELECTED WELLS
RELATIVE TO THE AVG. OUTPUT PER WELL

Figure 4. Long term effect of six vertical ground exchangers

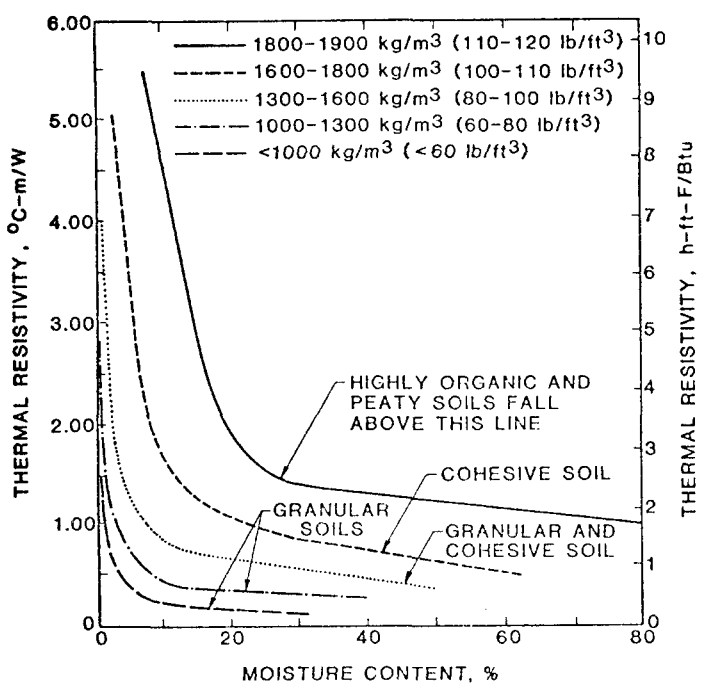


Figure 5. The effect of moisture content and dry density on the thermal resistance of soils Salomone (1979) .

The third section also discusses optimization, simulation and numerical methods. These are the tools that would be needed to assure that a system design was the best of all available alternatives. A major part of the literature survey not related to soils applies to this section. No computer programs are furnished but a summary of the many methods that have been developed is presented.

The appendix, which as already mentioned, augments material in several sections of the report, also contains information on plastic pipe selection and pressure drop considerations, on circulating fluids, and an equipment room requirements.

Conclusions

As has been mentioned, the field of ground-coupling heat pump systems is still rapidly developing. Although the material brought together as a result of this ASHRAE study will be very useful to those who must commit themselves to designs, one can expect some significant changes in methodologies as mores is learned. The manual may need some revising on a regular basis at least over the near term.

Acknowledgments

The authors are particularly grateful to ASHRAE for their support of the study RP-366, and for the members of the monitoring committee for their patience and cooperation. A special thanks is also due Dr. L. A. Salomone for his consultation on soils, and for the friends and colleagues and others who cooperated and furnished information.

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