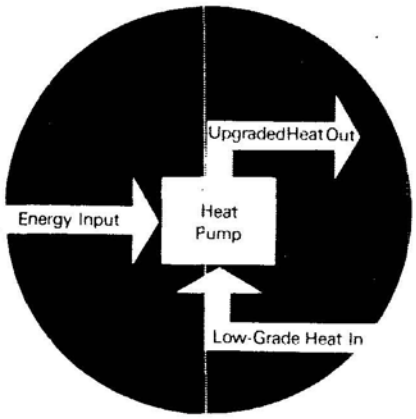
CONF-810672

# Proceedings of the DOE Heat Pump Contractors' Program Integration Meeting




## June 2-4, 1981

## Holiday Inn Tysons Corner
## Mc Lean, Virginia

Sponsored by
**U.S. Department of Energy**
Assistant Secretary, Conservation
and Renewable Energy

Published March 1982

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# Foreword

Heat Pump Technology Contractors' Program Integration Meeting
2-4 June 1981

This meeting was sponsored by the Department of Energy under the Office of the Assistant Secretary for Conservation and Renewable Energy. The purpose of this meeting was to provide an appropriate forum for the exchange of technical and programmatic information and for the stimulation of mutual awareness among participants in heat pump technology programs and projects. Through the active participation of various project organizations and program management offices, this meeting served to promote and facilitate coordination and cooperation among heat pump technology programs, including those within the Department of Energy and those in the private sector.

The heat pump, a device that moves heat from a region of low temperature to a region of higher temperature, is highly adaptable to a variety of energy sources and end-use energy needs of interest to various research and development programs within the Department of Energy. It is not, therefore, surprising that many groups within the Department of Energy sponsor heat pump technology projects. It was the intent of this meeting to bring together these various groups to facilitate information exchange, stimulate coordination, and thereby improve effectiveness in research and development efforts. Participation in this forum included the following offices within the Department of Energy:

- Advanced Conservation Technologies,
- Buildings and Community Systems,
- Industrial Programs,
- Transportation Programs,
- Solar Applications for Buildings, and
- Coal Utilization.

Under the auspices of the above program offices, the Department of Energy is sponsoring heat pump technology projects in diverse areas such as:

- Industrial process heat recovery,
- Solar energy utilization,
- Building space heating and cooling,
- District heating and cooling, and
- Heat engines for heat pump drives.

In addition, groups outside of the Department of Energy (i.e., the Electric Power Research Institute and the Gas Research Institute) are also involved in heat pump research and development and demonstration activities. This meeting also brought together participants from these private-sector organizations.

This proceedings document contains the technical papers presented at the meeting. Overview papers are included, summarizing activities and programs at the national laboratory level. Other technical papers are grouped according to major activity or type of heat pump: absorption heat pumps, electric-driven systems, heat engine systems, heat sources, and advanced heat pump technology.

It is the intent of the Department of Energy in publishing these proceedings to provide researchers and program managers a source of detailed information on current activities in the heat pump technology field in the United States. It should be noted, however, that future DOE-sponsored work identified in several of the papers may be subject to budgetary restrictions and possible redirection.

John D. Ryan
Meeting Chairman

Maxine Savitz
Deputy Assistant Secretary
Conservation and Renewable Energy
Department of Energy

# Conference Ad Hoc Planning Committee

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**Gas Research Institute**

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# Overview

# Overview

HEAT PUMP R&D AT OAK RIDGE NATIONAL LABORATORY*

OAK RIDGE NATIONAL LABORATORY
CONTRACT NUMBER W-7405-eng-26
F. A. CRESWICK AND R. D. ELLISON
ENERGY DIVISION

## OBJECTIVE

The objective of this program is to stimulate or accelerate the development of more efficient, energy-conserving heat pumps for heating and cooling buildings with conventional energy sources.

## BACKGROUND

This program is funded out of Technology and Consumer Products Branch, Division of Buildings and Community Systems.

Heat pump research at ORNL began in 1976 with a program aimed at exploring options for efficiency improvement and providing technical support to the DOE/BCS Program Manager. In FY 1978, responsibility for the program was transferred to ORNL. Projects directly related to the introduction of new heat pumps are carried out by subcontractors, most of which are industrial organizations. These subcontractors also perform some technology development and exploratory tasks. Research tasks at ORNL are aimed at providing the technological basis for long-range development of advanced heat pump systems, analytical tools to promote such development, and, when necessary, to provide technical support to subcontractors. Principal technology areas of concern are:

- heat-actuated system development,
- advanced electric system development,
- ground-source system characterization, and
- heat pump technology development.

Efficiency improvement incentives are illustrated by Figure 1, which displays evolutionary primary-fuel seasonal performance factor (PFSPF) objectives for present and advanced types of equipment. Electric resistance heating systems have a PFSPF of about 0.3--equal to the efficiency of the electric power generation and distribution system. Present top-of-the-line electric heat pumps can deliver an SPF of 2.0 in central regions of the United States; this is equivalent to a PFSPF of 0.6. Our objective for the next generation of advanced electric heat pumps is a 25% seasonal efficiency improvement, or a PFSPF of 0.8.

For furnaces, PFSPF and seasonal efficiency are equivalent if auxiliary electric power requirements are excluded. A properly sized furnace of the type marketed in the past decade can deliver 0.6, while advanced condensing furnaces soon to be marketed will deliver a seasonal efficiency of 0.9 or slightly better. Our PFSPF objectives for heat-actuated heat pumps are 1.1 for single-stage absorption systems and 1.4 for more advanced systems. The goals are approximately equivalent to 25 and 50% improvements, respectively, over the efficiency of the best furnaces expected to be marketed in the future.

*Research sponsored by the Department of Energy, Office of Buildings and Community Systems under contract W-7405-eng-26 with the Union Carbide Corporation.

ORNL-DWG 80-12660

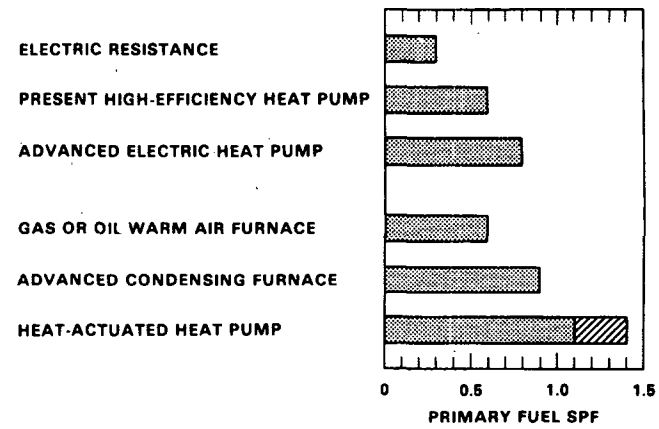


Figure 1. Fuel utilization of alternative heating systems.

## SUMMARY

Four of the advanced heat pump system development projects underway have reached a stage of development where the technical risks are low and prospects for initial commercialization within the next few years appear bright. These include two gas-fired residential absorption systems, an advanced electric residential system, and a light commercial gas-fired, gas-turbine-driven system. Three other system development projects are in progress that presently involve high technical risks and longer development periods: two of these are residential-sized, free-piston Stirling-engine-driven gas-fired heat pumps, and the third is an internal-combustion-engine free-piston compressor.

Field studies in progress or in preparation include full-house experiments with one air-source heat pump, two well-water-source units, and a ground-coupled system. There are also several field experiments being conducted with a variety of heat exchangers for ground coupling.

Laboratory studies on heat pumps have been conducted to compile a data base on air-to-air systems and component performance, and to investigate the effects of frosting on system performance. A test of a water-source heat pump is underway, and preparations are being made to conduct cycling studies of air-source units.

Analytical accomplishments include formulation and validation of a heat pump computer model, development of a technique for heat pump design optimization, an evaluation of design methods for ground-coupled heat exchangers, and an investigation of alternative heat pump cycles.

## TECHNICAL ACCOMPLISHMENTS

The status of most of the individual projects that comprise this program is reported on by other papers in this conference. In the following paragraphs, an overview of these projects will be presented to show how they fit together in subprograms. The material presented shows the program as planned in January 1981. Subsequent uncertainty in funding and the expectation of revisions in program philosophy and objectives have necessitated substantial alterations to the program--these changes are still in progress at the time of this writing. In modifying the program, our near-term objective has been to reduce the financial commitments while keeping our options flexible for future program implementation.

- Heat-actuated heat pumps. Figure 2 lists the four residential-application system development subcontracts in progress. The General Electric and Allied projects are cosponsored by the Gas Research Institute. The two free-piston Stirling projects reflect our belief that this configuration is the superior choice for heat-actuated heat pumps in the long term from the standpoint of efficiency, cost, maintenance, noise, and emissions. However, the technical risks are still very high. The two absorption systems have comparatively modest efficiency, but the technical risks are low and these systems show good promise for near-term commercialization.

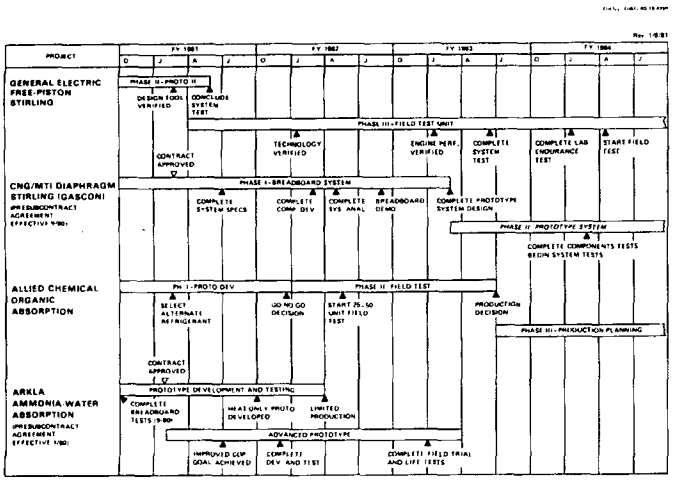


Figure 2. Heat-actuated heat pump development projects.

Figure 3 lists two light commercial systems under development. The AiResearch gas-turbine system development is cosponsored by the Gas Research Institute. Its technical risks are viewed by us as low, and the prospects for commercialization appear strong. The Honeywell project is concerned with application of the Braun linear engine to heat pump use. The existing engine is in a high state of development; the principal technical hurdle is development of a satisfactory refrigerant seal for the compressor section.

- Advanced electric systems. Figure 4 lists two system development subcontracts; only the first is currently active. The Westinghouse project can be viewed as accelerating the next big step in the evolution of heat pump design, i.e., moving a 1990 system

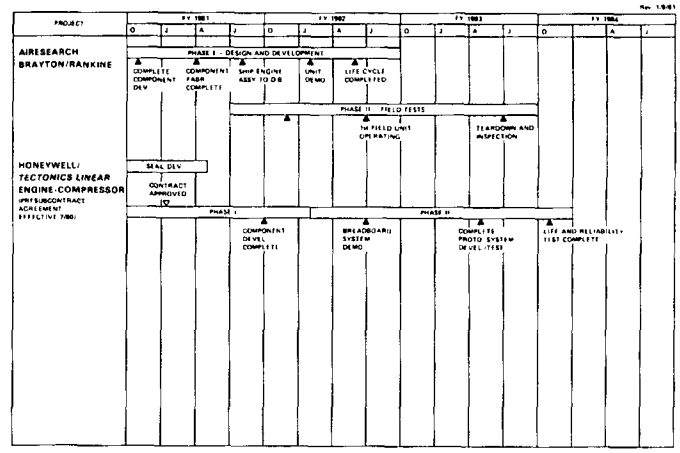


Figure 3. Heat-actuated heat pump development projects (continued).

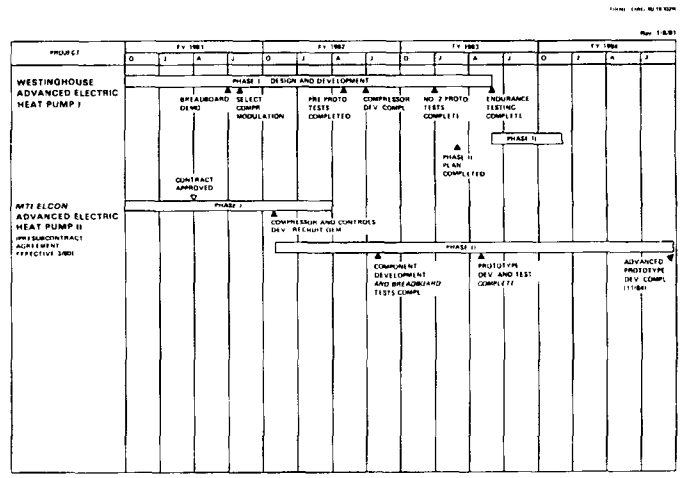


Figure 4. Advanced electric heat pump development projects.

ahead to 1985. Although there was a presubcontract agreement in effect on the MTI Elcon compressor, this procurement has been cancelled for the present. We still view this motor-actuated, variable-stroke unit as a significant advancement of compressor technology.

- Well-water-source heat pumps. Equipment for well-water-source heat pump installations is commercially available; accordingly, we see no need for federally-funded equipment development in this category. Results of modeling work from the National Water Well Association study (see Figure 5) lead us to expect that, over most of the United States and particularly in northern regions, well-water-source heat pumps can operate substantially more efficiently than the best available air-source heat pumps. Our concern is that there is no experimental documentation of their thermal performance and economics, and that environmental restrictions may severely limit the use of this technology. The market size study addresses this latter concern, and perhaps could be better described as a geographic applicability study. Of the two field

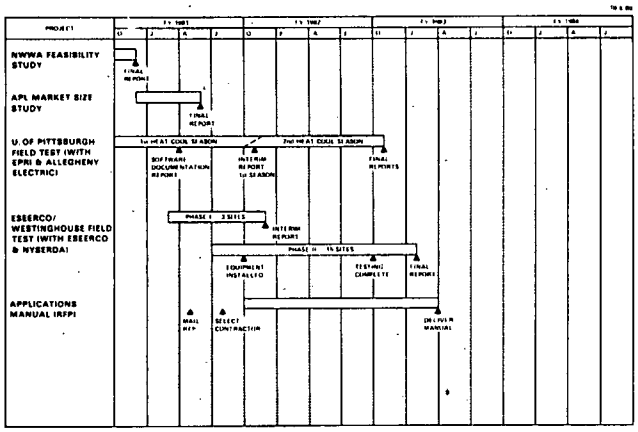


Figure 5. Well-water-source heat pump projects.

tests, the one cosponsored by the Electric Power Research Institute and Allegheny Electric Power is under way; the one to be cosponsored by the Empire State Electric Energy Research Corporation and the New York State ERDA is still being negotiated. The planned subcontract for an applications manual is in question at this time.

- Ground-coupled heat pumps. As with well-water-source heat pumps, satisfactory equipment for ground-coupled systems is available. We view the primary need in this technical area as being the formulation and dissemination of reliable design methods for the ground-coupling heat exchanger. To this end, we engaged Battelle (see Figure 6) to survey and evaluate this technology and supported Brookhaven National Laboratory in the continuation of their ground-coupling studies which had been started under their solar-assisted heat pump program. E-Tech was funded to continue some experiments with direct-expansion ground-coupled evaporators. It appears likely that a number of additional ground-coupled field experiments will be appropriate.

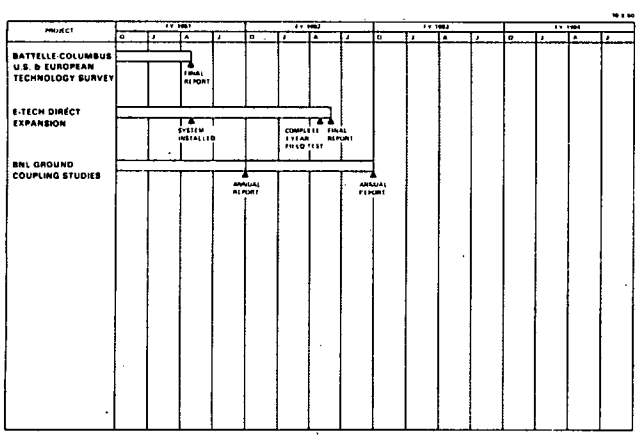


Figure 6. Ground-coupled heat pump projects.

- Heat pump technology projects. Other subcontracts, planned or in progress, are listed in Figure 7. The National Bureau of Standards has developed testing and rating procedures for heat-actuated heat pumps and is presently examining the part-load efficiency of absorption systems. A. D. Little examined prospects for using conventional internal combustion engines as heat pump drivers and concluded that, in commercial applications, the technology is adequate and the economics are attractive. A demonstration project would appear to be the next logical step, if funding and future program objectives permit.

TRW looked at opportunities for R&D to improve the efficiency of large, engineered, built-up HVAC systems, and concluded that economic and institutional factors are presently inhibiting the use of efficient equipment. A number of good proposals were received in response to an RFP on an improved frost control system; however, the procurement was cancelled due to uncertainties in future program objectives. The planned marketing overview subcontract listed in Figure 7 will probably not be implemented.

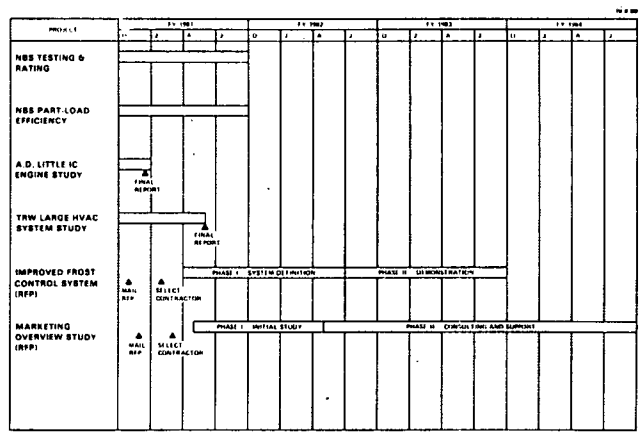


Figure 7. Heat pump technology and program analysis projects.

- In-house research. Table 1 summarizes the in-house activities at ORNL. Details on this work are described in other sections of these proceedings. The in-house work has provided an excellent technology base with which to assess existing heat pump technology, to deduce R&D needs and evaluate potential performance gains and technical and economic risks of project opportunities, and to make contributions to the advancement of heat pump technology.

## FUTURE ACTIVITIES

It is our intent to continue to implement the program described herein insofar as future funding and program philosophy permits. We hope to continue to support system development projects with the objective of reducing technical risks and demonstrating proof of principle. We believe that our present plans for field studies, laboratory, and analytical work are certain to be appropriate for the future.

Table 1. ORNL In-House Heat Pump Research

- Laboratory Performance Evaluation
  - Air source
  - Water source
  - Steady-state
  - Frosting
  - Cycling

  Field Performance Characterization
  - Air source
  - Ground coupled
  - Well-water source

  Heat Pump Computer Model

  Design Optimization Techniques

  Alternative Cycle Evaluation
  - Air cycle
  - Stirling

## POST-CONTRACT ACTIVITIES

Not applicable.

## PUBLICATIONS/REPORTS/REFERENCES

Publications resulting from this contract are given in the accompanying detailed project descriptions.

TECHNICAL SUPPORT: SOLAR ASSISTED HEAT PUMP SYSTEMS*
BROOKHAVEN NATIONAL LABORATORY
CONTRACT NUMBER DE-AC02-76CH00016
JOHN W. ANDREWS
PERIOD OF CONTRACT: BEGIN 10/1/80 END 9/30/81

## OBJECTIVE

The purpose of the project is to provide technical and program planning assistance to the Active Solar Heating and Cooling Division of the U.S. Department of Energy (DOE) in connection with the development of energy conserving and cost effective solar assisted heat pump systems. In order to provide a system which meets the technical and economic constraints, the following elements are required:

High efficiency heat pump hardware with performance characteristics tuned to the solar application;

Low cost storage and an available alternate heat source/sink to supplement the solar energy source;

Low cost means for collecting the solar energy in a form best suited to the system configuration.

A reservoir of technical expertise is required to enable program goals in each of these areas to be defined and attained. These activities fall into three general categories, which in the order listed progress to increasingly active roles in the overall program. These categories are contract monitoring, technical support, and program planning.

Contract monitoring consists of activities which provide information to the DOE on the current status and rate of progress of in-place contracts. Site visits, design reviews, and reviews of progress reports are examples of activities which fall into this category.

Technical guidance consists of activities which results in initiation of new work and changes in the direction of ongoing work. It therefore has wider impact than contract monitoring. Reviews of unsolicited proposals, writing and reviews of Requests for Proposals (RFP's), and the preparation of statements of work and changes in statements of work are examples of activities which fall into this category.

Program planning consists of activities which result in the definition of overall program goals and the means to attain them. It is the most far-reaching type of technical assistance activity. Preparation of research and development plans, recommendations concerning which system configurations should be given priority, and recommendation of the need for and timeliness of RFP's are examples of activities which fall into this category.

To enable these activities to proceed on a sound basis, it has been necessary to undertake selected systems analysis tasks, the outcomes of which have significantly affected the direction of the program.

## BACKGROUND†

Program Genesis (1976 - 1977). The Solar Assisted Heat Pump (SAHP) program in the United States began in 1976 at the time of the formation of the United States Energy Research and Development Administration, forerunner of the present DOE. The fundamental idea of the SAHP system [1] is to collect solar energy at temperatures too low for direct use in space heating, and to use this heat as a thermal source to the evaporator of a heat pump, which processes the heat to the load at a higher temperature. The intent of the concept is to improve the efficiency of both the solar collector and the heat pump by operating in the 5 to 40 C temperature range, which is low for a solar collector (relative to collectors used for direct space heating) but high for a heat pump (relative to air-source heat pumps). The efficiency of a solar collector increases as its operating temperature is lowered, while for a heat pump the potential exists for increasing efficiency by raising the source temperature. The SAHP concept required the redesign of the heat pump to take advantage of these source temperatures, since currently available water-source heat pumps did not achieve the continuously rising coefficient of performance (COP) with rising source temperature that one should expect from the Second Law of Thermodynamics. The concept also envisioned the deployment of special collectors, low in cost, made possible by the restriction of their operating temperature range to values less than 40 C. In 1977 an RFP was issued, inviting proposals to design, fabricate, and test the special heat pumps needed for the SAHP concept. As a result of this RFP, three hardware development contracts were awarded.

Computer Simulation Studies (1977 - 1979). As the heat pump development program began, a parallel program of system simulation using computers also got underway. Out of this work arose some differences in viewpoint which it is necessary to understand if the current status of the program is to be appreciated. These issues arose largely from a consideration of the relaitve merits of the two major solar heat pump system configurations under study at the time, series and parallel (Figure 1). In the series configuration, the

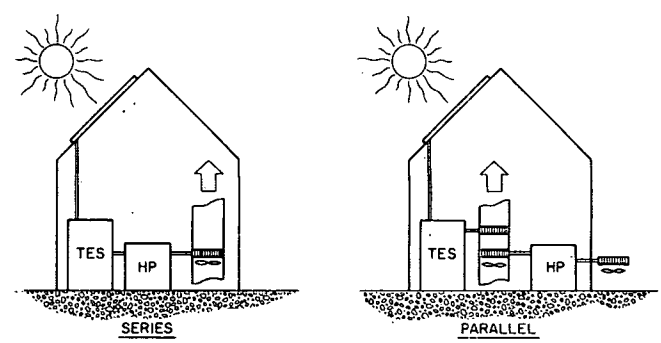


Figure 1. Series and Parallel Solar/Heat Pump Systems

solar collectors provide thermal energy to storage; this heat is transferred to the evaporator of the heat pump and processed to the load. In the parallel configuration, a stand-alone solar heating system of conventional design is used, with an air-source heat pump as the auxiliary or backup heat source for the building, independent of the solar system. The series configuration was the one which the SAHP program was

*Work performed under the auspices of the Active Solar Heating and Cooling Division, U.S. Department of Energy, Contract No. DE-AC02-76CH00016.

†Material in this section is excerpted from my paper, "Solar Energy and Heat Pumps: Can This Marriage Be Saved?", to be presented at the 16th Intersociety Energy Conversion Engineering Conference, Atlanta, GA, August 1981.

established to develop; however, some system analysts, notably those at the University of Wisconsin, [2] reached the conclusion that the parallel configuration was superior. The major difficulty with the series configuration was that in the coldest part of the winter, the storage often became depleted of solar-derived heat, requiring the introduction of auxiliary heat into the system. This condition has become known as thermal starvation. Most but not all of these studies [3,4] assumed that electric resistance was the only auxiliary source of heat available. With this assumption, a quandary arose. If small collector areas were used, the system was depleted of stored heat often, which required the use of large amounts of electric resistance auxiliary heat. This resulted in an inefficient system which did not provide significant savings of purchased energy. On the other hand, if large collector areas were used to insure that the system was well supplied with heat, the cost was driven up to an unacceptable level.

The second major issue concerned the likelihood that the low temperature collectors which were useful in the series configuration (but not in the parallel one) could really be manufactured and installed at the low prices envisaged by proponents of the series system. This is still a major issue in the U.S. solar community. Another possibility, supported by recent work at Brookhaven National Laboratory, [5] is that low-cost materials and methods of manufacture may be extended to medium-temperature collectors suitable for direct heating without going through a heat pump.

In addition to all of this, some analysts took a "curse on both your houses" position, maintaining that neither the series nor the parallel configuration was an attractive option. Some took this position after having concluded that both systems are inherently too expensive.[6] Others took this position on the basis of utility impact.[7] Here the assumption was that the utility must stand ready to meet the design load of the building with electric resistance backup heat at times when solar storage is depleted and the ambient temperature is low. The cost of this standby capacity - coupled with low electric energy sales to the solar residence because of its high efficiency under most conditions - results in a high unit price for electricity to the solar residence if it is not to be subsidized by other users.

Ground Coupling. It was with a view toward overcoming the above obstacles that work was initiated on the use of the ground as a heat source/sink or storage element for the heat pump. Such use of the ground has been called "ground coupling". With the ground available as an alternate heat source to the heat pump, thermal starvation need no longer be a problem. Since the ground maintains a predictable temperature with only long-term variations, the heat pump could be sized to meet the design load of the building from the ground source without recourse to electric resistance backup. This would improve the electric utility's load factor, potentially making it better, not worse, than that of an air-to-air heat pump or a straight resistance heating installation. Initially we perceived ground coupling as an adjunct to a solar system, but it soon became apparent that ground coupling could be used as the sole source and sink to a heat pump. The question then became, not "How can ground coupling help a solar system?" but "Can solar energy be used to advantage with ground coupling?" Because of the importance of ground coupling in the SAHP program, let us review some of the early and more recent work on this subject.

Early Work on Ground Coupling. A fair amount of research was done on ground-coupled heat pumps in the late 1940's and early 1950's. Although, as we shall see, this work indicated that ground-coupled heat pumps could operate satisfactorily, it is nevertheless evident that ground coupling did not enter the mainstream of American heating, ventilating and air-conditioning (HVAC) engineering practice. Among HVAC engineers this work took on a legendary aspect, with the conventional wisdom being that the experiments did not perform satisfactorily. Let us delve into some of the reported literature to see what happened.

To begin with, a quote from one of these papers is illuminating:

> "The heat source or sink often presents a major problem in the use of heat pumps. Where neither air nor water can be utilized, a grid of tubing buried in the earth has often been found satisfactory. On the other hand such grids have often partially or completely failed because of improper size or design. This is not surprising since the ground grid presents a new field of heat exchange design and little is known of the factors involved.
>
> Several experimental studies of this problem have already been made or are now in progress. Perhaps no one study can result in information which can be generally used, since each will be influenced by the climate, locality, type of soil, or apparatus used. However, an accumulation of such information will indicate trends and produce averages sufficiently accurate for most design purposes. Workers in this field of research should be continually aware of the need to present their results in such a manner as to be adopted by designers in any locality."[8]

Virtually all of the modern approaches to ground coupling had been thought of in this earlier period. One source [9] lists some of these:

1. A simple well from which water is pumped, heat is extracted from the water (or rejected to the water) and the water is discarded.

2. A closed system in which a U-tube is sunk into a vertical well and a heat transfer fluid pumped through the tube to pick up heat from the earth.

3. A modified U-tube system in which the entire well casing is utilized as part of the enclosed system for heat-transfer fluid.

4. A horizontal grid of buried pipe through which a heat transfer fluid is circulated.

5. A horizontal grid of buried pipe which forms part of the refrigerant circuit of the heat pump itself (direct-expansion).

6. A large buried storage tank constructed from large-diameter drain culvert pipe.

7. A coil of pipe surrounding the footings of the house foundation.

8. A series of shallow vertical holes, 15 to 20 feet deep, in which heat exchange coils are inserted.

Both theoretical and experimental work was performed. In order for theoretical work to be done at all, the actual situation of the heat exchange device in the ground, as well as the geometry of the device itself, had to be greatly simplified. This was, of course, before the advent of large digital computers. Calculations were done for a number of simplified heat

exchanger geometries, chief among these being the infinite plane, the sphere, and the long straight pipe. Calculations performed on the infinite plane and the sphere gave results which were much more pessimistic than those performed on the pipe. Generally, the calculated heat transfer rate between a plane or sphere and the ground decreased rapidly with time, approaching zero or near zero asymptotically, whereas the heat transfer rate between the pipe and the ground approached a positive steady-state value.

One early paper which calculated only the flat plane solution concluded that over a five month period the heat transfer rate achievable with ground coupling would drop by a factor of sixty.[10] A later study performed a more comprehensive analysis of the plane, the sphere, and the pipe.[11] It concluded that the pipe was superior, providing a significant steady-state rate of heat transfer. The result was not strongly dependent on pipe diameter, a 10 cm (4 in.) pipe providing only 30 to 40% more heat transfer per linear foot than a 2.5 cm (1 in.) pipe.

Experimental results were reported from a variety of locations and pipe configurations.[12-14] Reference 13 reported an installation of 146 m (480 ft) of 2.2 cm (7/8 in.) O.D. pipe to serve a home with a design loss of 51 MJ/hr (48,000 Btu/hr) under extreme conditions of 21 C (70 F) inside a -21 C (-5 F) outside. Actual heat loss from the house was less than this. This system delivered 43.8 GJ (41.5 million Btu) or 12,170 kWh of heat to the house, which was located in Kentucky, consuming 3980 kWh of electricity to operate the compressor, fan, and pump. The seasonal performance factor, or ratio of heat delivered to electricity consumed, both measured in the same units,[13a] was 12,170/3980 or 3.05.

In view of results such as these, the question still must be asked,"Why did ground coupling not catch on?" While a definitive answer to this question would be a good objective for a study in the sociology of engineering practice, it is possible to point out some contributing factors:

1. Copper pipe was expensive [13b] and energy costs were low. Now we have plastic pipe and energy costs are higher.

2. Generally accepted design procedures did not exist, although various methods were used by the innovators of this period to size their systems.

3. Whereas an air-source heat pump could be warranted in its entirety, there were no warranties on ground coils.

4. City lot sizes in the 1940's and early fifties were generally much smaller than suburban lot sizes are today.

One author [8a] expressed the opinion that lot sizes were too small to support ground coils, although his own design procedure led to a field size of only 1800 $ft^2$, or less than 1/20 of an acre. Perhaps our larger lot sizes, which resulted from the postwar exodus into the suburbs and is the source of increased energy consumption to heat larger detached houses and transport us over longer distances, can at least be taken advantage of now to support the higher efficiencies possible with ground-coupled heat pumps.

Recent Ground Coupling Work. A number of research projects on ground coupling have been funded by DOE [15] and monitored by Brookhaven National Laboratory. The contractors involved are Oklahoma State University; E-Tech Incorporated (Southeastern U.S.); FAFCO Incorporated (California); and Kaman Sciences Corporation (Colorado, Ohio, New Mexico).

The Oklahoma State project comprises a horizontal ground coil experiment and a vertical sealed well. The horizontal coil uses ∿ 10 cm (4 in.) O.D. PVC pipe. The vertical well has a ∿ 13 cm (5 in.) outer casing, with a concentric inner pipe for the return flow. Measurements are taken of the heat transfer rates, soil density, soil moisture content, and temperature patterns in the soil immediately surrounding and affected by the installed ground-coupling devices.

The E-Tech project retrofitted six ground coils in existing solar assisted heat pump houses in the southeastern U.S. All but one of the coils were constructed of ∿ 4 cm (1.5 in.) O.D. polyethylene pipe buried at ∿ 1.2 m (4 ft) depth, with the length of pipe ranging from 270 m (885 ft) to 450 m (1480 ft). Heat withdrawal rates were measured over varying portions of a heating season, and cooling season heat rejection rates were measured.

The FAFCO project is intended to compare the heat transfer capability of a horizontal pipe field to that of a set of buried flat-plate heat exchangers. Two 60 ft sections of buried plastic plate are used alternately with 300 ft of ∿ 4 cm (1.5 in.) pipe. Responses of the two systems to nearly identical inputs will be compared.

The Kaman Sciences work comprises three projects in which solar energy is deposited, through much of the year, into buried, uninsulated water tanks, and heat is removed via a heat pump as needed during the heating season. One project is the Phoenix House in Colorado, which utilizes 72 $m^2$ (780 $ft^2$) of collectors and a 26.5 $m^3$ (7000 gallon) storage tank in conjunction with a 5-ton water source heat pump. The second project, in Ohio, utilizes a 2-½ ton heat pump with duplicate arrays of metal and of plastic collectors for comparison testing. The third project, in New Mexico, utilizes a 7-½ ton heat pump with two ground coupled storage units: a 3-compartment tank and a buried flat-plate plastic heat exchanger.

In-house ground coupling research at Brookhaven National Laboratory is described in detail in a companion paper and will therefore not be reviewed here.

## SUMMARY

Only the highest level activities - program planning and associated system analysis - are described below. Contract monitoring and technical guidance, while important, take place at too fine a level of detail to warrant elaboration here. Past program planning accomplishments fall logically into three chronological phases:

1. Program genesis, through the preparation of the Solar Assisted Heat Pump RFP in 1977 and selection of contractors.

2. Setting in place of critical subelements - heat pump, ground coupling, collector, system analysis - in 1978-79.

3. Development of cost/performance goals, and preparation and implementation of a Program Plan, in 1980-81.

System analysis tasks of major importance undertaken under this contract include:

1. Initial analysis of the series SAHP system.

2. Analysis of the impact of varying collector characteristics on system performance in series and parallel SAHP's.

3. Development of an analytical optimization procedure for SAHP systems.

4. Analysis of four options for combining solar energy with ground coupling.

TECHNICAL ACCOMPLISHMENTS

Program Planning Accomplishments.

• Program Genesis. The solar assisted heat pump was conceived in 1976 and put in place in 1977 through the issuance of an RFP, "Solar Assisted Heat Pump Projects for the Heating and Cooling of Buildings." Proposals were reviewed in June of 1977 and contracts were awarded to Singer Corporation (September 1977); Northrup Incorporated (February 1978); Lennox Industries Incorporated (February 1978); and General Electric Company (May 1978).

• Critical Program Subelements. In addition to the need for an advanced heat pump, it was recognized early on that low-cost collectors and a storage element which could avoid thermal starvation were necessary elements in a complete system. Projects in these areas were set in motion. In the collector area, work on lightweight modules employing thin films was begun. In the ground coupling field, work on a number of different heat exchanger geometries in varying geographic regions was initiated.

• Program Element Plan. In 1980 BNL was asked by DOE to prepare a plan which would lay out in some detail the directions the solar assisted heat pump program should take. This plan described two generic types of solar assisted heat pump systems: those which use the ground as a heat source/sink but not as a storage element for actively collected solar energy; and those which do use the ground as an active storage element. In the course of developing this plan, a set of cost/performance goals was develoepd on the basis of payback, cash flow, and life-cycle costing. These goals and their justification have been published separately.[16] Following the submission of this plan, it has been necessary to undertake revisions in order to emphasize those aspects which are currently perceived as appropriate activities to be undertaken under DOE auspices.

System Analysis Accomplishments.

• Initial Analysis. This analysis [1] set forth the basic ideas of the solar assisted heat pump concept, and identified the heat pump and the collector as requiring special design for application in these systems. The potential of the vapor compression cycle to deliver high COP's at solar source temperatures was laid out in theory. A computer simulation and economic analysis of SAHP systems completed this work.

• Collector Impact Studies. The main objective of these studies [17] was to determine the extent to which collectors made to operate efficiently only at relatively low temperatures could be employed usefully in solar assisted heat pump systems. The analysis generated a set of curves giving required collector area as a function of the horizontal intercept $\tau\alpha/U_L$ of the collector efficiency curve. The conditions under which improving the heat pump COP improved system performance were also studied. This latter effort was motivated by a report in the literature which concluded that improving the heat pump had no impact on system performance. The key here proved to be the provision of a backup mode, such as ground coupling, in which an alternate source of low grade heat was processed through the heat pump, as opposed to the use of electric resistance backup. The following conclusions were drawn: 1) Collectors with low values of the horizontal intercept $\tau\alpha/U_L$ may be used in solar assisted heat pump systems if the use of electric resistance backup can be avoided. 2) Improvements in heat pump performance benefit system performance if both the solar derived heat and the backup heat are processed through the heat pump. 3) Collector cost reduction is of paramount importance if active solar heating is to achieve widespread utilization.

• Analytical Optimization Procedure. In the course of the systems analysis work described under BACKGROUND, above, SAHP systems were studied extensively. Most of those studies simulated, on an hour-by-hour basis, several plausible system configurations involving solar collectors and heat pumps. They also assumed that electric resistance was the auxiliary heat source. Although much was learned from those studies, it was difficult to evaluate the sensitivity of their results to all of the assumptions that went into them. The need was seen for an approach [18] which would be simple enough so that all the assumptions could be states concisely. It was also recognized that the average source temperature to the heat pump could be controlled by setting the minimum temperature at which the heat pump switches away from the solar source and turns to auxiliary. With electric resistance backup one always wants to set this cutoff temperature as low as the hardware will handle, but with a more efficient auxiliary heat source such as ground coupling this is not necessarily the case, but is rather the subject of an optimization problem. An analytic procedure was developed which could carry out this optimization by means of simple algebraic equations, rather than with hour-by-hour simulations. The procedure was calibrated against f-Chart so that confidence could be placed in its results. The primary merit of the procedure is that it made transparent some relationships between variables in these systems which heretofore had been obscure. For example, it was seen that as the COP of the auxiliary increases, the optimum solar storage temperature also increases so that the solar-source heat pump may maintain a significant advantage in COP over the auxiliary. Once pointed out, this relationship may appear obvious, but it had been totally overlooked in previous simulations.

• Analysis of Four Options. Building on the work described in the preceeding paragraph, an analysis [19] was performed for four system options in three cities - Madison, New York, and Atlanta - giving a range of heating-season climates from mild to severe. The system options were (Figure 2):

1. Actively collected solar energy and heat removed from the ground are both used as sources of thermal energy to the heat pump (series/direct heating).

2. Actively collected solar energy is delivered directly to the building load, with the ground coupled heat pump as backup (direct heating only).

3. Actively collected solar energy preheats the return air stream from the building, and the heat pump raises the air temperature further (if necessary) to the value required for comfort (high-side boosting).[4]

4. Solar energy collected via direct-gain passive design is used to reduce the building load required to be met by the ground coupled heat pump.

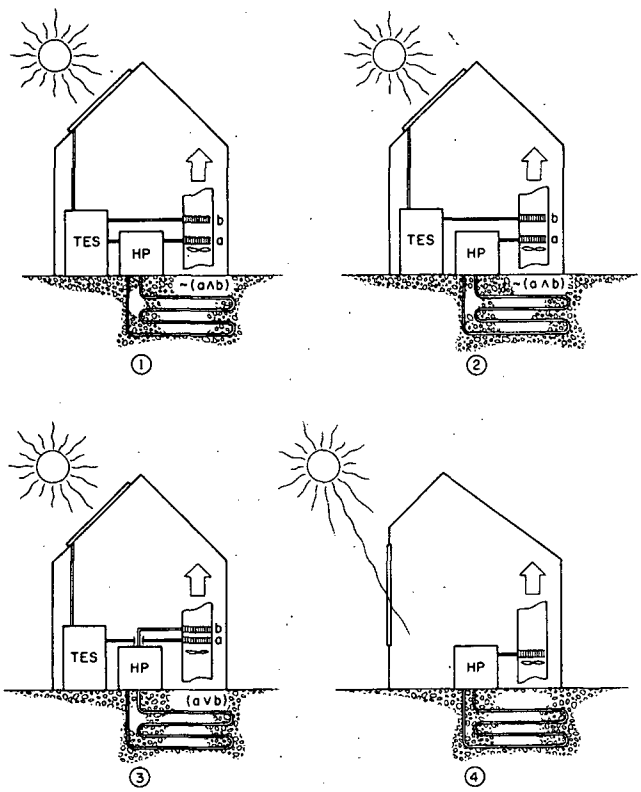


Figure 2. Ground Coupled Solar Heat Pump Options

In each of the three active solar options the heat pump produces hot water to the extent that solar energy is inadequate. In the passive option all of the hot water is produced via the heat pump.

In the three modes which involve active solar subsystems the solar energy is stored in an insulated tank, rather than in the ground, and the ground is used solely as a heat source and sink rather than as a storage element. The ground coupling heat exchanger configuration chosen - a horizontal plane serpentine coil of buried plastic pipe - is particularly suited for use as a source or sink but is not capable of long-term energy storage. Therefore it should be carefully noted that the results of this study apply only to such systems and not to systems in which in-ground storage is attempted.

Two types of collector were employed in the model, a "high -performance collector" with characteristics approximately equal to those of a good single-glazed selective-surface collector, and a "heat pump collector" with an overall heat loss factor twice as great. The heat pump collector faced the dilemma that if it operated at relatively low tempe-atures (< 10 C) the system COP did not greatly exceed that of ground coupling alone, while if the collection temperature was forced upward, the amount of solar energy collected by this "lossy" collector declined markedly. With the high-performance collector, the optimization procedure always selected the direct heating mode over the series mode. The passive collector (with night insulation) provided about the same benefit to the system as the high-performance collector. Remembering that this study explored ground coupled solar heat pump systems in which the ground is used as a heat source (and sink for cooling) but not as a storage element, the most promising systems appear to be ground coupled heat pumps without additional solar input; ground coupled heat pumps with modest passive augmentation; and possibly ground coupled heat pumps with active solar augmentation, if high-performance collectors can be sold and installed at much lower cost than is generally the case now. A severe cost restriction on the solar components arose from the good projected performance of the ground-source heat pump and from the economic criterion used. Three areas of research and development whose pursuit is consistent with the results of this study were identified. They are:

1. Development of ground coupled heat pump technology and design tools.

2. Development of low-cost high-performance collectors.

3. Continued investigation of the merits of ground coupled solar heat pump configurations in which the ground is used as a storage element and not as source/sink only.

## FUTURE ACTIVITIES

The task immediately at hand in the area of program planning is to obtain review and approval from DOE for the latest update of the Program Element Plan for solar assisted heat pump systems. At this writing, the organization of the program, both within the Office of Solar Applications for Buildings and also between that office and the Office of Buildings and Community Systems, remains to be resolved. This resolution will require decisions at the appropriate levels within DOE; it is expected that these matters will be resolved in the near future. The plan envisions approximately one more year of activities designed to permit a decision to be made on each of several candidate systems. Some systems will then be selected for field test and concurrent preparation of design guidelines. It is anticipated that these activities will require approximately two years. The output of the effort will be a set of field test reports and a system design handbook.

Two areas of system analysis remain to be completed. One is the analysis of solar ground coupled heat pump systems which use the ground as a storage element. Such systems can be divided into systems which aim for transseasonal carryover of thermal energy stored at temperatures above the far-field temperature, and systems which use actively collected solar energy to assist the natural recharge of the ground thermal reservoir. This analysis will complement the completed study of systems which use the ground as heat source and sink but not as storage element. The other systems analysis task is the detailed evaluation of the impact of the cooling mode on system design. The assessment of special collector/rejectors with variable heat loss factors is one element of this analysis. The other is the evaluation of alternative in-ground storage devices for their impact on cooling performance.

## REFERENCES

1. J. W. Andrews, E. A. Kush, and P. D. Metz, A Solar Assisted Heat Pump System for Year-Round Space Conditioning, Proc. 3rd Workshop Use of Solar Energy for the Cooling of Buildings, DOE Conf. 780249, February 1978, p. 248.

2. T. L. Freeman, J. W. Mitchell, and T. E. Audit, Performance of Combined Solar Heat Pump Systems, Sol. Energy 22, 1979, p. 125-135.

3. W. Kahan and R. C. Estes, Optimization and Comparison Strategies for Solar Energy Systems, Am. Soc. Mech. Eng. Pub. 79-WA/SOL-26, 1979.

4. W. F. Bessler and B. C. Hwang, Solar Assisted Heat Pumps for Residential Use, ASHRAE Journal, September 1980, p. 59.

5. J. W. Andrews and W. G. Wilhelm, Thin-Film Flat-Plate Solar Collectors for Low-Cost Manufacturing and Installation, BNL 51124, March 1980.

6. P. J. Hughes and J. H. Morehouse, Comparison of Solar Heat Pump Systems to Conventional Methods for Residential Heating, Cooling, and Water Heating, Science Applications, Inc., McLean, VA, Report 80-9060WA, 1979.

7. J. G. Asbury and R. O. Mueller, Solar Energy and Electric Utilities: Should They be Interfaced?, Science, Vol. 195, No. 4277, February 1977, p. 445.

8. G. S. Smith, Factors Useful in Ground Grid Design for Heat Pumps, Transactions of Am. Soc. of Heating and Ventilating Engineers (ASHVE), Vol. 57, 1951, p. 189; a. p. 205.

9. E. N. Kemler, Methods of Earth Heat Recovery for the Heat Pump, Heating and Ventilating, September, 1947, pp. 69-72.

10. B. F. Rober, C. F. Booster, and F. W. Hutchinson, The Heat Pump - How to Analyze Earth as a Heat Source, Heating, Piping and Air Conditioning, March, 1948, p. 83.

11. L. R. Ingersoll, F. T. Adler, H. J. Plass, and A. C. Ingersoll, Theory of Earth Heat Exchangers for the Heat Pump, Transactions of the Am. Soc. of Heating and Ventilating Engineers (ASHVE), Vol. 57, 1951, p. 167; a. p. 173.

12. G. S. Smith, Intermittent Ground Grids for Heat Pumps, Transactions of Am. Soc. of Heating and Ventilating Engineers (ASHVE), Vol. 62, 1956, pp. 473-488.

13. M. Baker, Design and Performance of a Residential Earth Heat Pump, Transactions of Am. Soc. of Heating and Ventilating Engineers (ASHVE), Vol. 59, 1953, pp. 371-394; a. p. 383; b. p. 384

14. D. M. Vestal, Jr. and B. J. Fluker, Earth as a Heat Source and Sink for Heat Pumps, Transactions of Am. Soc. of Heating and Ventilating Engineers (ASHVE), Vol. 63, 1957, pp. 41-68.

15. These contracts are described fully in Proc. Ann. DOE Active Solar Heating and Cooling Contractors' Rev. Mtg., March 26-28, 1980, Conf. 800340, Section 4.

16. J. W. Andrews, Cost Performance Goals for Solar and Ground-Coupled Heat Pump Systems, BNL 51259, September 1980.

17. J. W. Andrews, Heat Pump Impact Upon Solar Collector Design and Cost, Proc. Int. Solar Energy Soc. Congr., Atlanta, GA, May 1979, p. 792.

18. J. W. Andrews, Optimization of Solar Assisted Heat Pump Systems Via a Simple Analytic Approach, Proc. Second Ann. DOE Systems Simulation and Economic Analysis Conf., San Diego, CA, January 1980, Solar Energy Research Inst., Editor, SERI/TP-351-431, pp. 361-6, U.S. DOE, Washington, D.C. 1980.

19. J. W. Andrews, Ground Coupled Solar Heat Pumps: Analysis of Four Options, Proc. ASME Solar Energy Div. Conf., Reno, NV, April 1981.

# OVERVIEW - DOE/NASA AUTOMOTIVE GAS TURBINE AND STIRLING PROJECTS

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SUMMARY

This report presents a brief overview of the automotive gas turbine and automotive Stirling engine technology projects being carried out by NASA Lewis Research Center for the Department of Energy's Automotive Technology Development Division. This report: (1) discusses the projects as they were formulated and being carried out in accordance with PL 95-238 "Auto Propulsion Research and Development Act of 1978;" (2) presents substantive technology accomplishments; and (3) briefly addresses future path options of the program.

The Projects. A brief review of the project history since its inception by EPA in 1970 is presented in Fig. 1. Figure 2 shows the anticipated phasing of government and industry efforts leading from experimental engine technology by government/industry teams to eventual production by industry. The project goal is the same for both the gas turbine and Stirling engines and is presented in Fig. 3. The 30 percent fuel economy improvement is for vehicles of equivalent weight and performance. Accomplishments planned to be completed by the end of the development projects, in order to meet the project goals, are shown in Fig. 4. The project activities are divided into three major areas as shown in Fig. 5, with the major contractors, contract start dates, and contract values given in Fig. 6. (MTI/USS/AMG is a contractor team composed of Mechanical Technology Inc., United Stirling of Sweden, and AM General. DDA is Detroit Diesel Allison Division of General Motors.) The first three contracts listed are engine development efforts; the fourth "Ceramic Applications In Turbine Engines" involves incorporating ceramic components into an existing heavy duty truck gas turbine to obtain early field experience with ceramic components. The three engine development contracts follow the same development logic with similar schedules as shown in Fig. 7, with a Reference Engine design effort to guide the overall effort, component technology development, and two generations of engines (Mod I and Mod II) to be tested both on dynamometers and in vehicles.

Technology Accomplishments-Gas Turbine. Key technology requirements for the successful development of the automotive gas turbine engine are listed in Fig. 8. Figure 9 is a cutaway view of the single shaft gas turbine design under development by Garrett/Ford. The DDA design is a two-shaft machine, and both utilize radial flow turbomachinery. Status and significant accomplishments of the automotive gas turbine project are presented in Figs. 10 to 12.

For the automotive gas turbine, ceramics are the key to high performance. This is illustrated in Fig. 13. Figure 14 presents a cutaway view of the Detroit Diesel Allison heavy duty turbine engine which is being modified with the ceramic components identified in the figure.

This effort is being carried out in three steps of increasing turbine inlet temperature as shown in the schedule in Fig. 15. Maximum temperature requirements and candidate materials for key ceramic components of the automobile engines under development are given in Fig. 16. Two sample ceramic rotors fabricated by Carborundum for the DDA automobile engine gasifier turbine are shown in Fig. 17. These rotors are visually and dimensionally acceptable but do not yet meet the strength requirements. Figure 18 illustrates one of the critical problems with the use of ceramics components. That is, the very small critical flaw size as compared to that for metal parts. This requires the development of new nondestructive evaluation techniques to be able to detect defective parts. As shown in Fig. 19 the AGT/CATE program is now the primary support of the U.S. structural ceramics effort. The areas of ceramic technology advancements still required are shown in Fig. 20.

Technology Accomplishments-Stirling

Figure 21 lists some observations regarding Stirling engine technology status at project inception. This was in sharp contrast to the broad gas turbine technology base existing in this country. The United Stirling P-40 engine, Fig. 22, an experimental four cylinder, double acting engine, has served as the baseline engine for the automotive Stirling development effort. Key technology requirements of the automotive Stirling project are listed in Fig. 23. Figure 24 shows the participants in the major Stirling engine development contract and their primary areas of responsibility. Early in the program a P-40 engine was installed in an Opel automobile, Fig. 25. While this resulted in an underpower, overweight vehicle, it was done to provide early engineering experience with engine-vehicle integration. Status and significant accomplishments of the automotive Stirling project to date are shown in Figs. 26 and 27. Figure 28 is a picture of the partially assembled Mod I Stirling engine now on test at United Stirling. Some very early Mod I engine test data at part power operating conditions is shown in Fig. 29. United Stirling anticipates little difficulty in achieving the design efficiencies with only minor corrections. Figure 30 illustrates a technique, developed in the supporting research and technology (SRT) effort at LeRC, for controlling the permeability of hydrogen through the high temperature metal heater tube walls. Current efforts are underway to determine how small a percentage of CO or $CO_2$ dopant can be utilized while still holding permeability losses within acceptable limits. Figure 31 is a picture of the Ground Power Unit (GPU), a single cylinder, displacer type engine originally built by General Motors, being used in the SRT effort at LeRC to evaluate new component technologies. Figure 32 shows a jet shell heat transfer device currently being tested in the GPU to assess its potential for augmenting heat transfer to the engine heater tubes.

The Future Path. Figure 33 addresses near term activities aimed at phasing the gas turbine and Stirling efforts down to the levels required by the administration's current budget plans. At the same time, an effort is being made to retain some flexibility of options should legislative action result in somewhat higher budget levels. While planning is still in the early stages, it is anticipated that the focus of technology effort starting in FY 82, Fig. 34, will shift from engine development to component technology, with emphasis on the longer range, high pay off technologies.

- EPA REQUEST TO LeRC FOR EMISSION TECHNOLOGY HELP IN 1970.
- ERDA-NASA IAA IN 1976 FOR HEAT ENGINE TECHNOLOGY PROJECT MANAGEMENT - CHANGED TO DOE-NASA IN 1977
- TRANSPORTATION PROPULSION DIVISION WITH GAS TURBINE & STIRLING PROJECT OFFICES FORMED AT LeRC IN 1977.
- TITLE III, PL 95-238 IN FEB 1978 MANDATED DOE TO DEVELOP GAS TURBINE & STIRLING EXPERIMENTAL ENGINE TECHNOLOGIES WITHIN 5 YEARS - AND AMENDED SPACE ACT FOR NASA PARTICIPATION.
- MAJOR TECHNOLOGY DEVELOPMENT CONTRACTS SIGNED IN 1978-1979.

Figure 1. - Some history.

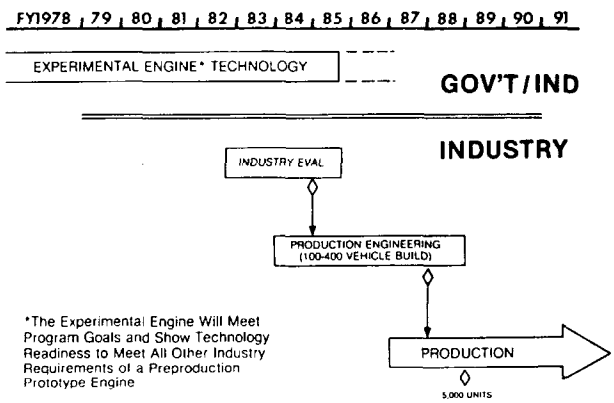


Figure 2. - Automotive heat engine technology.

OVERALL GOAL:

PROVIDE THE TECHNOLOGY BASE WITHIN THE AUTO INDUSTRY TO SUPPORT PRODUCTION DEVELOPMENT OF ALTERNATE AUTOMOTIVE PROPULSION SYSTEMS THAT:

- SHOW AT LEAST 30 % FUEL ECONOMY GAINS OVER SPARK IGNITION ENGINE
- MEET MOST STRINGENT EMISSIONS STANDARDS INCLUDING .4 GM/MI $NO_X$
- CAN USE A VARIETY OF FUELS INCLUDING NONPETROLEUM BASE
- CAN BE SOLD COMPETITIVELY

Figure 3. - Heat engine program gas turbine and stirling.

- EXPERIMENTAL GAS TURBINE AND STIRLING PROPULSION SYSTEMS IN VEHICLES MEETING FUEL ECONOMY EMISSIONS AND MULTIFUEL CAPABILITY OBJECTIVES
- TEST STAND DURABILITY DATA FOR KEY COMPONENTS AND EARLY VERSIONS OF EXPERIMENTAL SYSTEMS
- PROJECTIONS BY INDUSTRY OF MASS PRODUCTION AND OWNERSHIP COSTS
- DEFINITION OF KEY LOW-COST MANUFACTURING APPROACHES

Figure 4. - Accomplishments by end of development projects.

| PRIME CONTRACTS | SUPPORTING ACTIVITIES | EARLY FIELD EXPERIENCE |
|---|---|---|
| • GAS TURBINE | • CERAMIC APPLICATIONS IN GAS TURBINES | • BUS<br>- GREYHOUND<br>- BALTIMORE TRANSIT |
| • STIRLING | • SR&T CONTRACTS & GRANTS | • TRUCK |
| | • LeRC IN-HOUSE SRT<br>- ANALYSIS<br>- COMPONENT TECHNOLOGY<br>- ENGINE EVALUATION | |

Figure 5. - Automotive propulsion systems.

- ADVANCED STIRLING ENGINE DEVELOPMENT

| | | |
|---|---|---|
| MTI/USS/AMG | 3/78 | $90M |

- ADVANCED GAS TURBINE DEVELOPMENT

| | | |
|---|---|---|
| GARRETT/FORD | 10/79 | $57M |
| DDA/PONTIAC | 10/79 | $65M |

- CERAMIC APPLICATIONS IN TURBINE ENGINES (COMPLETION MID 1984)

| | | |
|---|---|---|
| DDA | 1/78 | $43M |

Figure 6. - Major heat engine contracts (completion by May 1985).

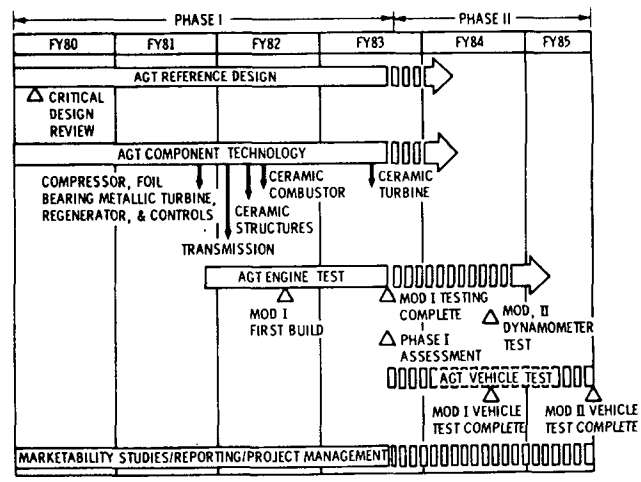


Figure 7. - AGT project plan.

- HIGH STRENGTH, CHEAP STRUCTURAL CERAMIC COMPONENTS
- LOW EMISSION, WIDE OPERATING RANGE COMBUSTORS
- SMALL, HIGH EFFICIENCY TURBOMACHINERY
- HIGH EFFECTIVENESS, LOW LEAKAGE REGENERATOR ASSEMBLIES
- HOT END BEARINGS
- ENGINE INSULATION
- VEHICLE LOW SPEED ACCELERATION
- HIGH STRENGTH MATERIAL FOR COMPRESSOR IMPELLERS

Figure 8. - Automotive gas turbine project key technology required.

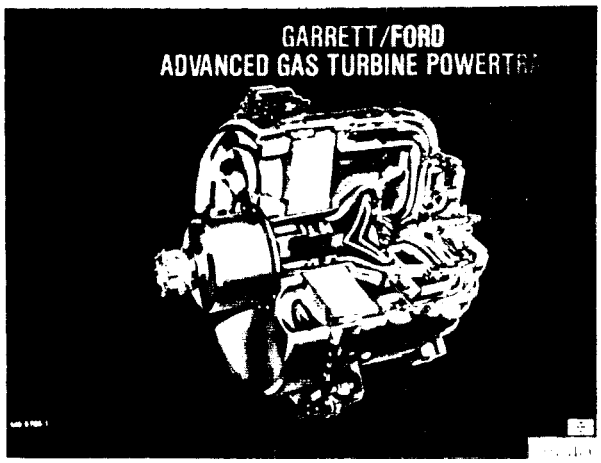


Figure 9.

- GARRETT/FORD PRELIMINARY DESIGN SHOWS ADVANCED POWERTRAIN SYSTEM POTENTIALLY CAPABLE OF MEETING PROJECT OBJECTIVES.
- FIRST MOD I BUILD IN AUGUST
- DDA/PONTIAC DESIGN REVIEWS IN JUNE
- RESULTS OF RECENT TURN-AROUND ON WEIGHT/COST/HEAT LOSS ISSUE WERE JUST EXCELLENT - COST POTENTIAL NOW FITS ACCEPTABLY IN GM'S 1985-90 MARKET PROJECTIONS.
- COMPRESSOR TESTS ARE ABOVE PART-POWER EFFICIENCY TARGET FOR MOD 2!

Figure 10. - Automotive gas turbine status and significant accomplishments.

- CERAMICS (THE KEY TO ULTIMATE SUCCESS)
  - 6657 MILES SUCCESSFULLY RUN OVER-THE-ROAD WITH AN ENGINE WITH CERAMIC TURBINE VANES & REGENERATOR CORES
  - VISUALLY GOOD, DIMENSIONALLY CORRECT, ONE-PIECE CERAMIC ROTORS FABRICATED
  - OVER 700 HOURS OF ENGINE TESTING OF INTEGRATED (36 PIECE) TURBINE NOZZLE COMPLETED (AT 1900° F T.I.T.)
  - 30,000 CYCLES (489 HOURS) TO 2200° F COMPLETED ON ONE PIECE SILICON NITRIDE TURBINE STATOR
  - OVER 5000 HOURS OF SUCCESSFUL OPERATION ON 4 CERAMIC REGENERATOR CORES AT 1800° F IN AN ENGINE

Figure 11. - Automotive gas turbine status and significant accomplishments.

- KEY TECHNOLOGIES
  - ALL COMPRESSOR & TURBINE DATA MEETING INITIAL GOALS
  - ULTRA-LEAN COMBUSTOR CONCEPT SHOWS PROMISE FOR SIMPLER, LOWER COST HARDWARE
  - CATALYTIC COMBUSTOR MATERIAL SHOWS GOOD PERFORMANCE AND DURABILITY FOR 250 HOURS AT 2600° F

Figure 12. - Automotive gas turbine status and significant accomplishments.

| ENGINE TECHNOLOGY LEVEL | PROJECTED FUEL ECONOMY, MPG* |
|---|---|
| BASELINE TECHNOLOGY (CHRYSLER UPGRADED ENGINE) | 18.4 |
| AGT METAL ENGINE (MOD I) | 26.0 |
| AGT CERAMIC ENGINE (MOD 2) | 42.5 |

* DIESEL FUEL, 3000 LB VEHICLE

Figure 13. - The payoff of ceramics technology for an automotive gas turbine engine.

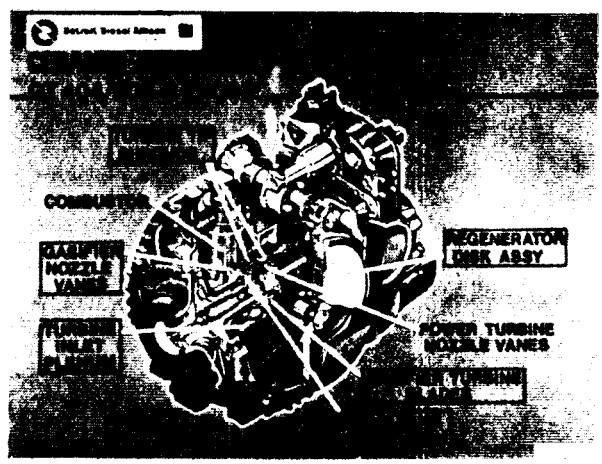


Figure 14.

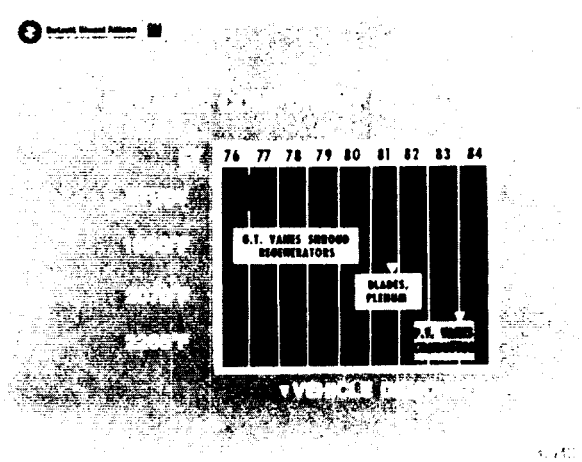


Figure 15.

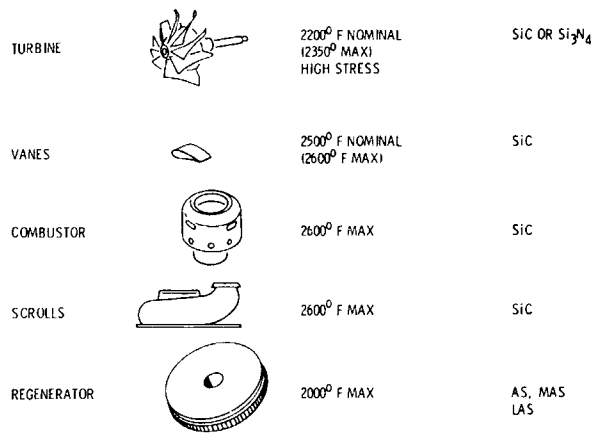


Figure 16. - AGT key ceramic parts.

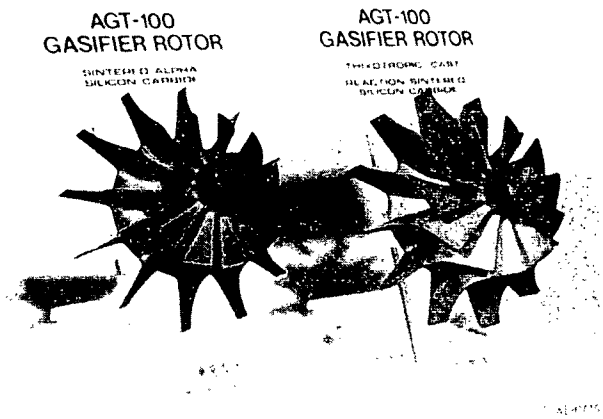


Figure 17.

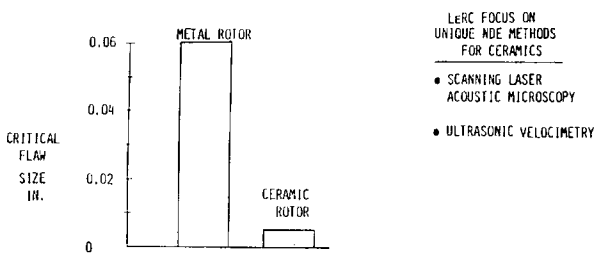


Figure 18. - Nondestructive evaluation: A key need for brittle materials.

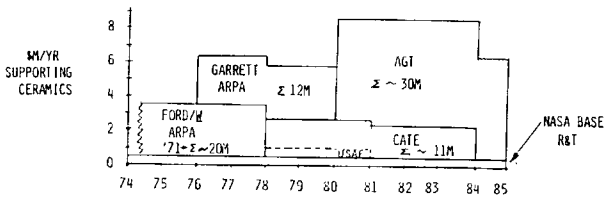


Figure 19. - AGT/CATE now major supporters of U. S. structural ceramics technology.

- MATERIAL PROPERTY IMPROVEMENTS
  - STRENGTH/REPRODUCIBILITY
  - ENVIRONMENTAL RESISTANCE
- PART QUALITY REPRODUCIBILITY
  - UNIFORM PROPERTIES
  - DIMENSIONAL CONTROL
- RELIABILITY/NDE

Figure 20. - Considerable ceramic technology advances still required.

- AN ESTABLISHED TECHNOLOGY BASE FOR STIRLING SIMILAR TO OTTO, DIESEL, RANKINE, OR GAS TURBINE DOES NOT EXIST.
- STIRLING TECHNOLOGY IS NOT PRESENTLY MATURE ENOUGH TO COMPETE COMMERCIALLY IN MAJOR ENERGY ARENAS LIKE INDUSTRIAL OR AUTOMOTIVE APPLICATIONS WITHOUT SIGNIFICANT COMPONENT AND SYSTEM DEVELOPMENT.
- THERE IS A LIMITED NUMBER OF U.S. INDUSTRY PARTICIPANTS FOR ANY SUCH DEVELOPMENT PROGRAM BECAUSE OF LIMITED ACCESS TO: (A) DESIGN COMPUTER CODES; (B) ENGINE HARDWARE; AND (C) ENGINE OPERATING DATA.
- THERE IS NOT YET A RELIABLE, PROVEN ENGINE FOR TEST OR FOR DEVELOPMENT TOWARD A NEW APPLICATION, AND RELATIVELY LITTLE ENGINE OPERATING EXPERIENCE.
- MOST TECHNOLOGY EXISTS IN EUROPE.
- STATIONARY INDUSTRIAL AND SOLAR APPLICATIONS MUST EVOLVE FROM MAJOR AUTO PROGRAMS.

BUT THE ENGINE HAS GREAT POTENTIAL

Figure 21. - Observations regarding stirling technology status.

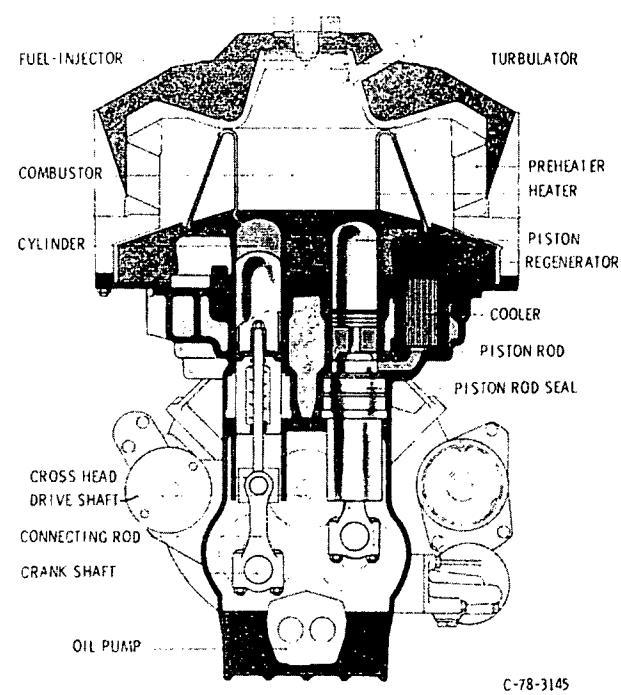


Figure 22.

- LOW COST MATERIALS FOR HOT COMPONENTS
- REDUCED HYDROGEN PERMEABILITY
- LONG LIFE, LOW LEAKAGE, LOW FRICTION SEALS
- IMPROVED COMBUSTOR EFFICIENCY
- IMPROVED AUXILIARY EFFICIENCIES
- REDUCED DRIVEAWAY TIME
- ENGINE WEIGHT REDUCTION

Figure 23. - Automotive stirling engine project key technology required.

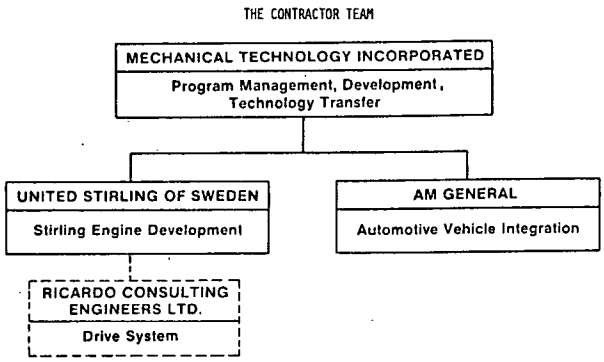


Figure 24. - Automotive stirling engine development program.

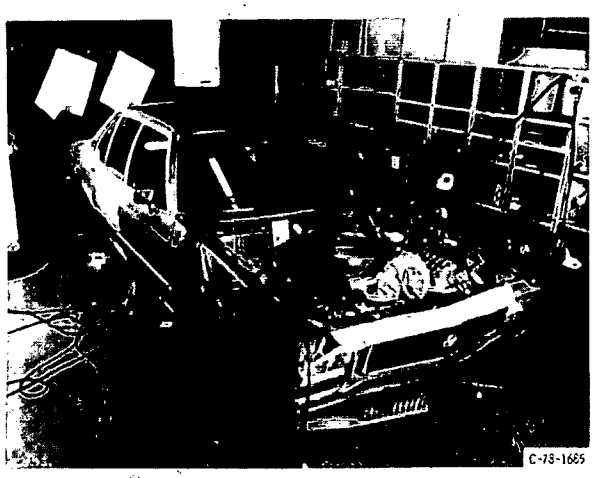


Figure 25.

- FIRST ENGINE DESIGNED FOR AUTOMOBILE (MOD-I) UNDER TEST - 6 MORE TO FOLLOW
  - 147 HRS. AS MOTORING UNIT
  - 36 HRS. TESTING AT FULL TEMP - (BUT LOW PRESSURE)
  - PERFORMANCE WITHIN 2 POINTS OF PREDICTED WITH $H_E$, BETTER WITH $H_Z$
  - NO MAJOR PROBLEMS
  - MOD-I GOAL OF 20 % BETTER FUEL ECONOMY SHOULD BE ACHIEVED.
- ACHIEVED 0.3 GM/M $NO_x$ WITH MOD-I COMBUSTOR RIG.(GOAL IS $<$ 0.4 GM/M $NO_x$)
- STIRLING TECHNOLOGY TRANSFERRED TO U.S.
  - MTI, AMG, & LEWIS PERSONNEL ALL CAPABLE OF DIAGNOSING & CORRECTING ENGINE PROBLEMS
  - U.S. NON-PROPRIETARY PERFORMANCE & OPTIMIZATION CODES NOW EXIST
  - MOD-I ENGINE BEING MANUFACTURED IN U.S.

Figure 26. - Automotive stirling status and significant program accomplishments to date.

- PROJECTED FUEL ECONOMY FOR THE REFERENCE ENGINE IN A 1984 "X" BODY CAR IS 42 MPG (GASOLINE - 56 % IMPROVEMENT OVER 1984 S.I. ENGINE IN SAME VEHICLE.
- TECHNICAL CHALLENGES IN STIRLING APPEAR CAPABLE OF SOLUTION
  - LEWIS INVENTION OF DOPING REDUCES $H_2$ PERMEATION TO ACCEPTABLE LEVELS
  - PUMPING LENINGRADER ROD SEAL VERY GOOD
    -- 3500 HOURS ON SEAL ASSEMBLY IN ENGINE
    -- OVER 70,000 HRS. TOTAL EXPERIENCE ON SEAL
  - MATERIALS BEING DEVELOPED THAT SHOULD HAVE SUFFICIENT HOT STRENGTH WHILE STILL LOW IN COST
  - LeRC 4-CYLINDER DYNAMIC CONTROLS MODEL IS ONLY SUCH CODE IN EXISTENCE.

Figure 27. - Automotive stirling status and significant program accomplishments to date.

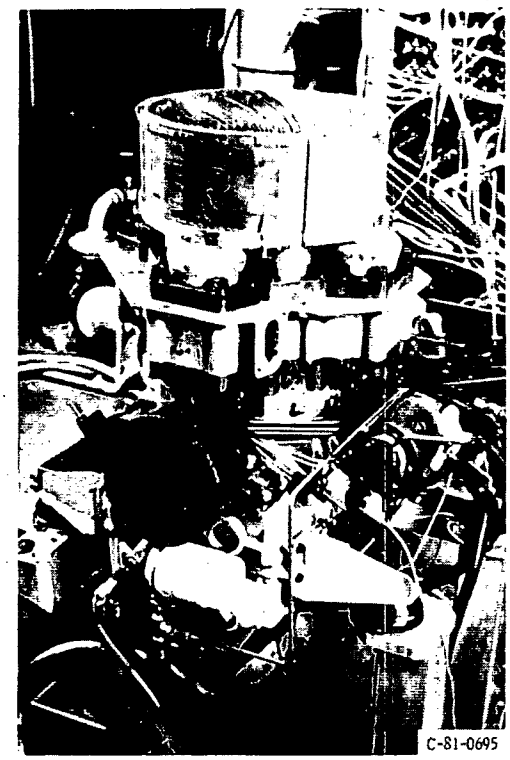


Figure 28.

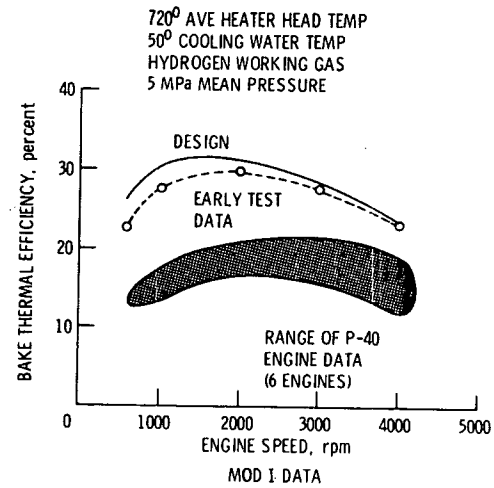


Figure 29. - (Low power operation.)

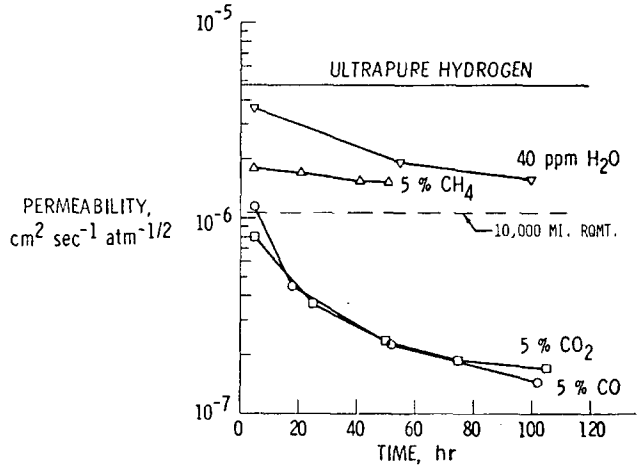


Figure 30. - Effect of dopants on hydrogen permeability through inconel 800H.

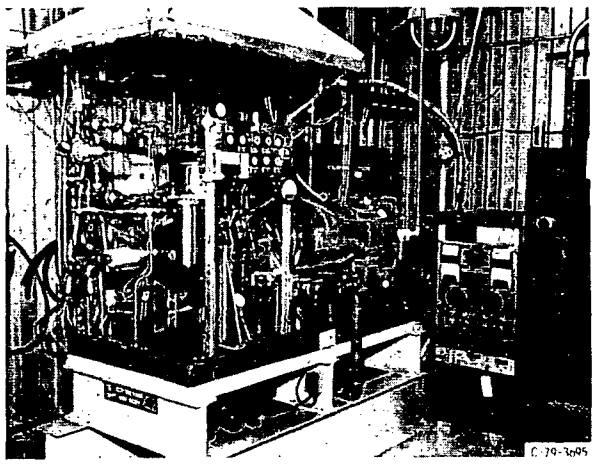

Figure 31.

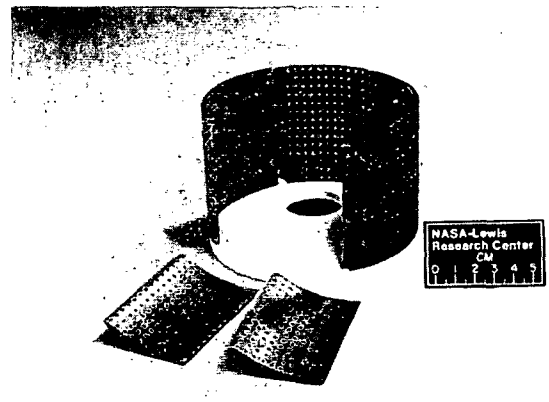


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Figure 32.

- INITIATE PHASE DOWN OF TECHNOLOGY DEVELOPMENT ACTIVITIES IN FY 81 AS RATIONALLY AS POSSIBLE
- MAINTAIN OPTION FOR CONGRESSIONAL REINSTATEMENT
- INITIATE DEFINITION OF FY 82 REDUCED LEVEL TECHNOLOGY PROGRAMS
- GO-NO-GO DECISION POINT WILL OCCUR THIS SUMMER
- IF CURRENT MAJOR CONTRACTS ARE TO BE TERMINATED, ISSUE TECHNOLOGY RFP'S IN OCTOBER

Figure 33. - The future path.

- GAS TURBINE (~ $3M)
  - CERAMICS FOCUS
  - SMALL SCALE AERODYNAMICS
  - USE MOD-I TEST BED
- STIRLING (~ $3M)
  - CERAMIC HEATER HEAD
  - ADVANCED CONCEPTS - VARIABLE STROKE
  - SEALS, REGENERATORS

Figure 34. - The technology path (still in early planning stage).

# OVERVIEW OF ECUT HEAT PUMP RELATED PROGRAM

Robert E. Holtz and James G. Daley
Components Technology Division
Argonne National Laboratory

## OVERVIEW OF ECUT

The mission of the Energy Conversion and Utilization Technology (ECUT) Program is to benefit all energy-use sectors by supporting research to improve efficiency and fuel switching capability in the conversion and utilization of energy. The two ECUT objectives are to develop a technology base for energy conversion and utilization systems that incorporate major improvements in efficiency or are capable of using alternate fuels; also, investigate advanced concepts in energy conversion and utilization which offer high potential benefit to energy end-users.

ECUT consists of two program elements: Energy Conversion Technology and Energy Utilization Technology. These program elements are divided into the following projects:

- Energy Conversion Technology
  - — Open-Cycle Power Systems
  - — Closed-Cycle Power Systems
  - — Direct Heating and Conversion
- Energy Utilization Technology
  - — Materials Development
  - — Physical Processes
  - — Chemical Processes

Within the ECUT Program, the heat pump related activities are being conducted in the Closed-Cycle Power Systems Project.

## CLOSED-CYCLE POWER SYSTEMS PROJECT

The objectives of the Closed-Cycle Power Systems Project are: (1) to resolve technological barriers to the use of closed cycle devices such as Stirling engines, closed cycle Brayton engines and Rankine engines and (2) to contribute to the improved efficiency, performance and fuel flexibility of these systems for environmentally acceptable and cost effective operation in applications covering all sectors of the economy. Development of closed cycle engines operating on waste heat is included; such as Rankine bottoming engines.

A major project area is the development of closed cycle engines to drive heat pump compressors. These engine driven heat pumps have a large market potential in both the residential/commercial and industrial sectors of the economy. Their use would substantially improve overall energy utilization as is shown in Table 1 with comparison against conventional alternatives.

Engine driven heat pumps using internal combustion engines are presently in use in Europe; the energy balance on such a system is shown in Figure 1. It may be seen from Figure 1 that recovery of engine reject heat is responsible for the high performance of these directly fueled systems.[1] While engine efficiency is related to overall efficiency, very good fuel utilization is possible with engines of relatively low efficiency due to recovery of heat from exhaust gases and cooling water. The effect of engine efficiency on system performance is shown in Figure 2.

## HEAT PUMP EFFORTS WITHIN ECUT

Engine driven heat pumps are not presently in the United States and are not part of the product line of any domestic manufacturer. The greatest present need of this technology is that reliable, quiet, cost effective systems be designed. These application needs would best be met by integrating existing heat engines with existing heat pump compressors and developing controls and packaging from existing hardware. Once these systems are developed and operating the barriers to their acceptance can be addressed.

ECUT's role is seen to be in the development of advanced heat engines and associated technology to enhance the performance of these systems by providing higher efficiency, greater reliability, fuel flexibility, better environmental acceptability, and lower cost. A near term engine driven heat pump system would require internal combustion engines, either reciprocating or turbine. Advanced heat engines would employ closed cycle technology and external combustion. Free-piston Stirling engines have good potential for residential heat pumps operating either on gaseous or liquid fuels. ECUT is participating in research leading to an improved fundamental understanding of these engines; both through improved analysis methods and experimentally determining the magnitude of important energy loss mechanisms that occur during operation. Some loss mechanisms to be investigated are:

Compression Space Hysteresis Loss — The working space hysteresis loss is due to irreversible thermal transfers which occur in boundary layers on the various working space surfaces during cyclic pressure variation of the gaseous working fluid. The loss is similar to the hysteresis loss of a simple gas spring and analysis of this loss is based on a simple gas spring model. The validity of this assumption is in question because of the high degree of turbulence in the engine working spaces due to gas mixing and also to the possible effect of the large surface area exposed during operation.

Regenerator Matrix Loss — Regenerator losses are the result of numerous, complex and often interacting phenomena including: matrix-to-gas friction and heat transfer, axial and radial gas and matrix thermal conduction, cycle gas pressure variation during the regenerator process, real gas effects (significant only at high-gas pressures), regenerator bypass leakage, and regenerator gas flow maldistribution radially, axially, and circumferentially.

Displacer Seal Loss — The seal between the displacer piston skirt and the cylinder wall is accomplished either by a close clearance seal or an piston ring. The effect of clearance gap in the case of close clearance seals will be evaluated. Flow through this seal results in parasitic pumping losses and a potential thermal energy transfer from the hot expansion space working gas directly to the cooler.

[1] Source: Increasing Interest in the Gas Engine Heat Pump, Diesel and Gas Turbine Worldwide, October 1980

Table 1. Comparison of Heating Options

| Heating Mode | Overall Energy Utilization Efficiency (BTU of Heat/BTU of Primary Energy) |
|---|---|
| • Electric Heating | 30% |
| • Conventional Boiler | 70% |
| • Unit Gas Heating | 75% |
| • Electrically Driven Heat Pump | 80% |
| • Engine Driven Heat Pump | 140% |

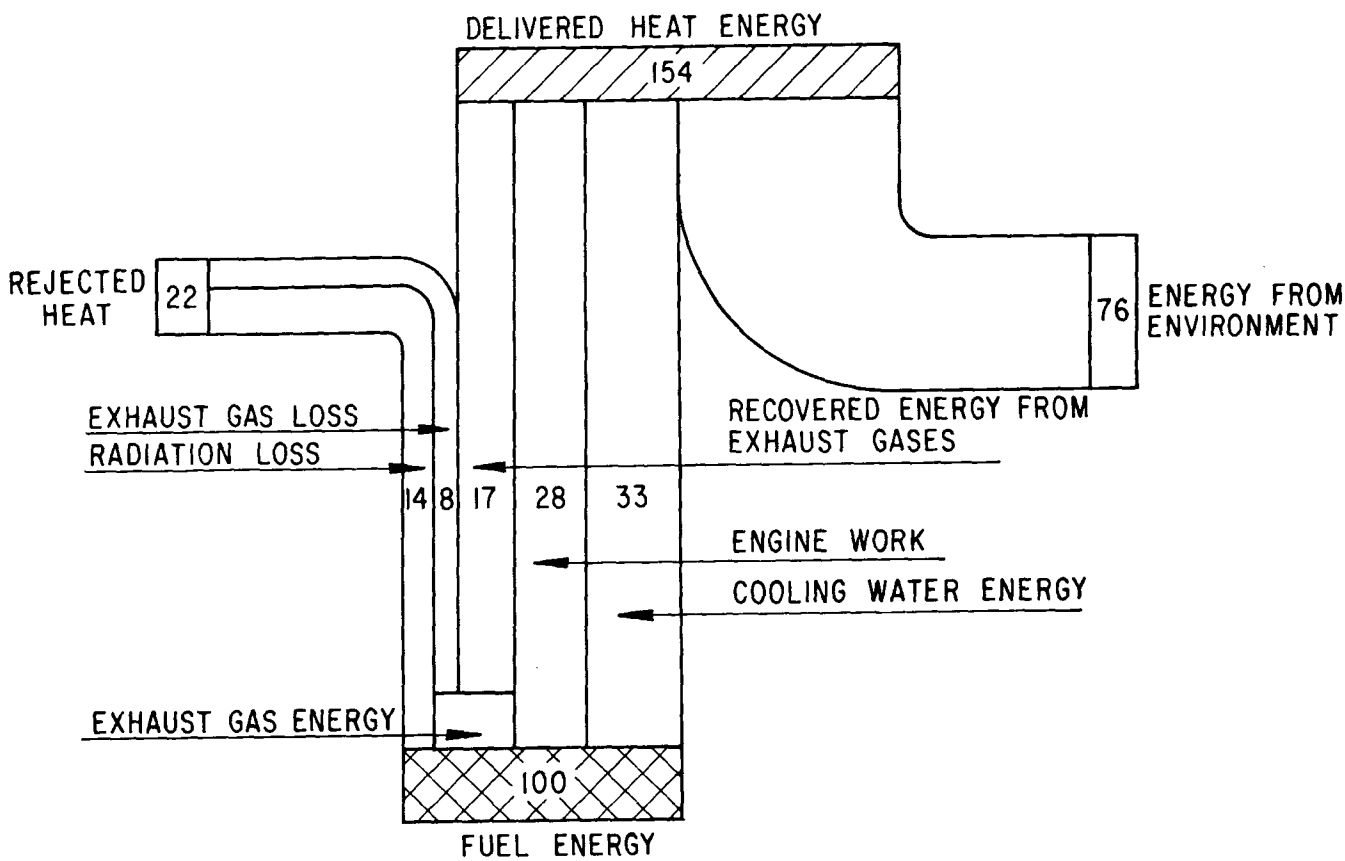



Figure 1. Energy Balance on Engine Driven Heat Pump System

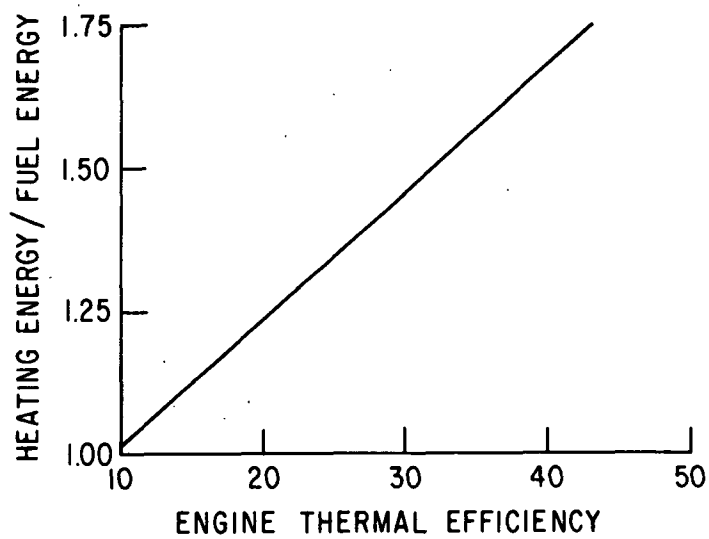



Figure 2. Effect of Engine Efficiency on Engine Driven Heat Pump System Performance

Displacer Appendix Gas Loss — The appendix gap is the thin working gas annulus between the displacer dome and the engine cylinder. Losses are associated with heat and mass transport in this gap region as the displacer reciprocates.

Kinematic Stirling engines have excellent potential for residential/commercial and industrial heat pumps, and ECUT is presently developing the technology needed for successful application of these engines. Kinematic Stirling engines would be used in applications requiring shaft power above 8 kW, while free-piston Stirlings would be useful in power ranges below 10 kW.

Rankine cycle engines are seen as having very good potential for use with heat pumps in the residential/commercial and industrial sectors. Initial ECUT effort in this area involves the effect of working fluid on performance; both in the engine and heat pump. Further effort is planned on defining and meeting the technology needs of heat pumps driven with Rankine cycle engines.

# THE CHEMICAL HEAT PUMP PROGRAM
# AN OVERVIEW

BROOKHAVEN NATIONAL LABORATORY
ALESSIO MEZZINA

## SUMMARY

A brief overview of the Chemical Heat Pump Program is presented as a forerunner to more detailed project and systems descriptions which will follow. Program background, rationale, technology description, and research and development needs are addressed. Chemical heat pumps comprise reversible reactions which can be driven by low-grade heat. Thermal energy is absorbed in one direction and liberated in the reverse direction; thus, serving as a basis for system designs applicable to space conditioning or process heat management and offering the capability for high-density energy storage as an integral part of the system.

## INTRODUCTION AND BACKGROUND

Brookhaven National Laboratory (BNL) provides technical management and oversight support to the Chemical Heat Pump Program on behalf of the Division of Physical and Chemical Energy Storage Systems within the DOE Office of Advanced Conservation Technologies (currently the Office of Energy Systems Research). In this capacity BNL has steered the program on a course consistent with policy guidance and budgetary constraints as identified by the DOE which called for emphasis to be placed on exploiting the near-term potential of chemical heat pump technology.

The DOE/STOR Chemical Heat Pump/Chemical Energy Storage Program began in the CY 75-76 period with funding of five unsolicited proposals to develop thermal energy storage concepts based on reversible thermochemical reactions that can also function as thermally-driven heat pumps. These initial procurements were for the development of concepts based upon:

Calcium Chloride/Methanol, to EIC Corp.;
Magnesium Chloride/Water, to Chemical Energy Specialists, Inc.;
Sulfuric Acid/Water, to Rocket Research Corp.;
Ammoniated Salts, to Martin Marietta Corp.; and
Paired Metal Hydrides, to Argonne National Laboratory.

The first four listed above were under the STOR thermal energy storage budget and collectively became known as the chemical heat pump storage program. Primary management and procurement authority in this general area was assigned to SLL in FY 1977, then to BNL in the last quarter of FY 1979. The last listed also was a chemical heat pump project, but was funded under the STOR hydrogen budget and management retained at headquarters. In FY 1980 BNL was authorized to sponsor further development of hydride heat pumps as part of what is now called Chemical/Hydrogen Energy Systems Program.

The program had been largely limited in scope to solar space conditioning applications. Many of the potentially suitable chemical reactions are kinetically well matched to the diurnal solar cycle and can be driven by unconcentrated solar collectors. These natural connections to a solar application, in which energy storage is an essential element, helped justify STOR sponsorship of the program. When it became increasingly evident that other applications of chemical heat pumps, with and without a need for storage, could have greater near-term potential, the BNL program shifted toward the exploration and development of chemical heat pump and chemical energy storage technologies for reject heat management, electrical load leveling, and combustion heat amplification.

Consistent with the near-term thrust of the program, BNL technical management exercised value judgments in selecting systems for advanced development which offered highest probability of success and which were furthest along the hardware development stages. Projects selected included:

- Sulfuric Acid/Water CHP, at Rocket Research Corporation
- Calcium Chloride/Methanol CHP, at EIC
- Paired Metal Hydrides CHP, at Southern California Gas/Solar Turbines International following a competitive procurement action

## WHY CHEMICAL HEAT PUMPS?

The incentives which drive chemical heat pump development derive from advantages of substituting low-grade thermal input for high-grade energy to achieve goals of energy efficient space conditioning as well as industrial process heat management. The advantages relate to:

- The capability of using solar collector or residual process heat to drive the reactions with minimal dependence on auxiliary power;
- the dual function (heating and cooling) of a single unit;
- the ability to provide high-density energy storage as an integral part of the system at low incremental cost;
- the ability to upgrade residual process heat to higher and more useful temperature regimes;
- the ability to use engine reject heat for providing fuel efficient on-board refrigeration for commercial vehicles.

## TECHNOLOGY DESCRIPTION

All chemical heat pumps are based upon one or two reversible chemical reactions. Physically, these two reactions take place between a vapor (or gas) working fluid and either a liquid or a solid absorbent. The phases present are not as important an issue as are heat and mass transfer and possibly energy storage density as these relate to overall economics. A chemical heat pump can displace some or all of the fossil fuel used by competing space conditioning systems. Economic analyses performed by Arthur D. Little and TRW Energy Systems (McLean, Virginia), have shown that with appropriate development CHPs can be cost competitive for non-solar applications. At the present time, collector costs strongly dominate solar-fired CHP

system costs. Eventually, collector technological development and energy cost escalation will render such systems economically attractive. A CHP can both heat and cool using solar collectors as well as offer high energy density storage with some systems. This greatly increases the role of a solar collector for space conditioning and as a result the collector value is greatly increased.

RESEARCH AND DEVELOPMENT NEEDS

Primary problems pacing cost-effective chemical heat pump development include corrosion, the introduction of noncondensibles (which inhibit vapor transport) as well as freezing of water in water-based systems. One contractor has identified an inexpensive coating for carbon steel which is impervious to attack from $H_2SO_4$. Other coatings have been identified by proposers for $NH_4NO_3$ and $NH_4Cl$ systems. Careful design and cost analysis can overcome these sort of problems. Systems with inherently long cycle times (e.g., $CaCl_2$-$CH_3OH$) due to mass and heat transfer limitations are most applicable to diurnal cycles and as such will need to be developed at a rate commensurate with economic solar collector systems.

Research and development needs which have been identified consistent with the attainment of chemical heat pump cost and performance goals include:

- The selection and characterization of "ideal" reversible reactions which: minimize corrosion and toxicity problems; offer high stability and heat transfer capability over a wide temperature range; operate at pressures approaching one atmosphere.

- The selection of advanced "designed-to-cost" heat exchangers.

The related issues and assessments which must be addressed in the course of R&D activities must consider:

- Legal/institutional factors when introducing noxious working fluids and/or absorbers into the residential/commercial market (safety issues);

- environmental impact associated with wide distribution of toxic/flammable/noxious materials;

- possible impact on utility load factors subsequent to wide application of non-electrical heating and cooling systems;

- Cost/performance evaluations comparisons of current and evolving chemical heat pump technology (TRW);

- Marketing studies of CHP cooling systems (Booz-Allen & Hamilton supporting Southern California Gas Company/Solar Turbines International);

- performance characteristics of emerging solar-driven CHP technologies (EIC, funded by ANL/DOE Solar).

CURRENT STATUS OF CHP DEVELOPMENT ACTIVITIES

The status of projects and other support functions currently funded at BNL are summarized:

- Rocket Research Corporation (RRC) has completed design and fabrication of the $H_2SO_4/H_2O$ Verification Test Unit. RRC is proceeding with the testing of a 150,000 Btu/hr, 1,000,000 Btu storage system in both the HVAC and industrial heat pump mode. Cost and performance characterization will permit tentative plans for engineering evaluation testing at a paper and pulp industrial plant in FY 82.

- EIC Corporation has completed 100 cycle tests of its calcium chloride/methanol chemical heat pump engineering test model. Cost and performance characterization of the system suggests possible shortcomings with regard to its commercial viability because of cost effectiveness uncertainties. Further development has been deferred pending outcome of cost/performance analyses and comparisons with competing systems identified as current and emerging technologies.

- Southern California Gas and Solar Turbines International are pursuing the development of paired metal hydrides chemical heat pumps. The project is in the component selection, testing, and characterization phase setting the stage for proof-of-concept of the system.

- Argonne National Laboratory provides technical support to BNL in the area of metal hydrides chemical heat pumps. ANL's early experience in the HYCSOS project is being applied to developing models and algorithms for Coefficient of Performance analysis and verification as well as in corroboration of selected metal hydrides with properties suited to CHP application.

- TRW Energy Systems completed a chemical heat pump cost and performance evaluation. This study provides a basis for projecting CHP commercialization prospects in terms of space conditioning and industrial heat pump market penetration potential. First-generation CHPs ($H_2SO_4/H_2O$; $CaCl_2/CH_3OH$) are characterized and compared to current and emerging technology as to likelihood for private sector commercialization. Results also point to possible new directions for second- and third-generation systems.

RELATED NATIONAL AND INTERNATIONAL ACTIVITIES

A number of research and development activities based upon the chemical heat pump principles are ongoing at the National and International level exclusive of DOE/BNL funding support.

At the National level:

- Exxon Research and Engineering has pursued development of a proprietary industrial chemical heat pump which could represent a lower cost second-generation system. Cooperative R&D with DOE as well as further independent pursuits by Exxon have been deferred pending clarification of the federal posture and budget;

- Zeopower, Inc. is currently marketing a zeolite/$H_2O$ cooling system;

- ARCLA currently markets solar-driven LiBr absorption coolers which require relatively high thermal input to achieve effective COP;

- Emerson Kumm, a private inventor, has patented a $NaOH/H_2O$ CHP chiller which promises high $COP_c$ (>0.7) with low thermal input ($\sim 160^oF$). Interest has been expressed by private sector investors in pursuing system development.

At the International level:

- Tepidus of Sweden has developed a $Na_2S/H_2O$ seasonal storage system which can be modified for CHP operation;

- Rutherford Laboratories and Open University of Great Britain have been sponsored by the Science Research Council to explore development of small $H_2SO_4/H_2O$ units;

- Munich Technical Institute in the Federal Republic of Germany has investigated $NH_3/H_2O$ absorption systems;

- Institute of Chemical Engineering in France has investigated isopropanol/hydrogen/acetone systems;

- Federal Institute of Reactor Research in Switzerland has done bench-top experimentation of ammoniated salt systems.

BENEFITS AND IMPACTS

The chemical heat pump must compete with current and emerging HVAC and industrial systems by virtue of the benefits to be derived as viewed from a cost/economic perspective. Chemical heat pump HVAC systems can become cost effective as conventional energy prices escalate; but applications in the paper and pulp, chemical and oil refinery industries alone promise savings greater than 0.2 Quads per year with equipment payoff estimated within periods of the order of two years. Recent analyses suggest chemical heat pump applications in traction vehicle refrigeration systems or possibly as automotive air conditioners. Currently large trailer trucks use 4 gallons of diesel fuel per hour to fuel their compression cycle refrigeration systems. A compact chemical heat pump driven by engine reject heat can displace this fuel consumption, representing a target of 0.1 to 0.2 Quad per year. Ultimately, advanced chemical heat pumps may penetrate the automotive air conditioning market; this can amount to energy savings of the order of 1 Quad per year.

ANNUAL CYCLE ENERGY SYSTEM PROGRAM
OAK RIDGE NATIONAL LABORATORY
W-7405-eng-26
ROBERT E. MINTURN

## OBJECTIVE

The objective of the Annual Cycle Energy System (ACES) Program is to incorporate in a practical system the outstanding energy conservation potential that exists when the uni-directional heat pump and the interseasonal storage of energy are combined to provide heating, cooling, and domestic hot water to buildings. The system is to be developed and tested, its applicability to different geographic areas is to be determined, and the methodology for designing and building such systems is to be made freely available to the private sector.

## CONCEPT OR BACKGROUND

When a heat pump operates, energy is drawn from an energy source and "pumped" to an energy sink. In the process, the source generally becomes colder and the sink warmer. In the normal course of operation, one or the other of these occurences is wasted; which is wasted depends upon whether the heat pump is operating in the heating or cooling mode. In the ACES, neither output from the heat pump is wasted. The compressor operates in the heating season to provide space and domestic water heating. The energy source is an insulated tank of water which grows colder as energy is extracted and eventually, by the end of the heating season, has turned to ice. The ice is subsequently used to cool the house in the summer, while compressor operation is limited to very short periods to provide hot water when needed. Fig. 1 is a schematic diagram of ACES operation.

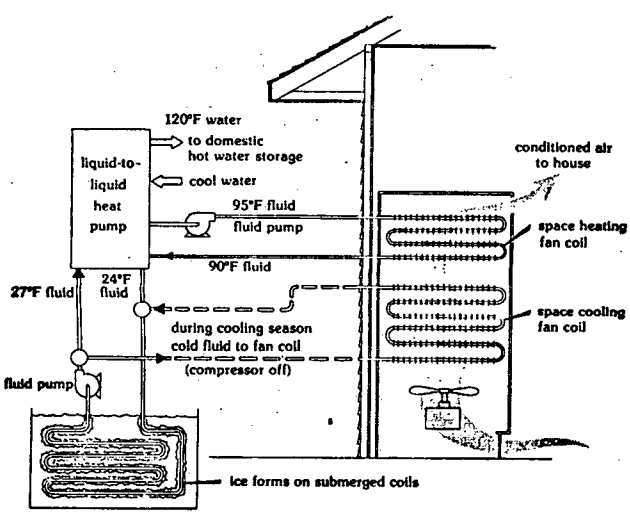



Fig. 1. Schematic diagram of ACES installation. The heat pump is uni-directional, and the energy is transported to and from the storage tank and to and from the conditioned space by means of a methanol-water anti-freeze solution.

## SUMMARY

An ACES has been designed, constructed, and field tested over a three-year period in a fully instrumented residential test complex near Knoxville, Tennessee. During that period, the system has been shown to be rugged, reliable, and appreciably more conservative of purchased energy than all practical alternatives (1-3). An ACES residential design methodology has been developed and published (4), and the performance characteristics of ACES and several alternative systems have been analytically estimated for 115 U.S. cities (5).

The ACES has a relatively high first cost that will likely discourage potential buyers or builders from utilizing ACES, although its life-cycle costs at present day electricity rates are competitive with those of other systems with the exception of those using natural gas. The ACES, however, offers some load management features that may be economically attractive to a number of electric power utilities. Its constancy in efficiency and capacity during winter operation, independent of extremes in weather, contributes to the reduction of peak demand and to increases in daily and seasonal load factors. Its significant reduction in summer demand may be a boon to utilities encumbered by summer peaking problems and low reserve margins. The ACES program is currently bringing these features to the attention of utilities.

## TECHNICAL ACCOMPLISHMENTS

- Field demonstration of ACES efficiency. Three residences of very similar plan and construction features have been constructed on the campus of the University of Tennessee near Knoxville. One of the residences utilizes an ACES to provide its space heating and cooling and hot water. A second house, called the Control House, is identical in design and orientation to the ACES house, has utilized three different conventional systems to provide the same services. A third house, sponsored by a University of Tennessee/Tennessee Valley Authority program, has been used to field test a number of solar energy features.

The ACES and Control House are unoccupied, but contain a programmed package that simulates the loads, including hot water use, normally imposed upon a house by a family of four. All three houses are very well instrumented, and hourly performance and weather data from all three are recorded by a computer in the ACES house.

During three years of operation, the ACES has consistantly outperformed by a wide margin the conventional systems used in the Control House (see Fig. 2). Fig. 3 demonstrates the field results for all four systems tested at the Knoxville site. For this comparison, the data has been adjusted for all systems to correspond to an average weather year.

- Evaluation of ACES' regional applicability. A computer program was developed and validated against field data from the Knoxville site to estimate the engineering performance of a number of systems including ACES in 115 U.S. cities. Fig. 4 illustrates the high annual COP's (ACOP) for ACES in almost all parts of the country. For comparison, equivalent ACOP's for a system based on a high-efficiency air-to-air heat pump are shown on the same figure. The numbers shown for the air-to-air heat pump are lower than the ARI-rated steady state

values because (1) the energy use includes water heating by electrical resistance, and (2) the field data include reductions in efficiency caused by cycling and defrosting losses, by decreases in efficiency as ambient temperatures drop, and by resort to strip-heater back-up as system capacity decreases at low ambient temperatures.

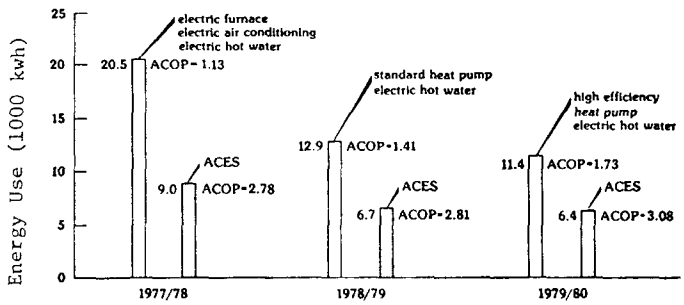



Fig. 2. Annual energy use of alternative systems during three-year test at Knoxville site. Space heating and cooling and domestic hot water provided by all systems. ACOP is annual coefficient of performance, defined as sum of all services delivered divided by energy purchased, all in equivalent units.

- Development of ACES residential design manual. A methodology was developed and published which will help designers and contractors gain an understanding of the fundamental concepts of the ACES for residences and to provide practical procedures and data for the selection and installation of ACES equipment. Methods for estimating the sizes of components and for computing monthly space-heating, water-heating, and air-conditioning loads are presented. Construction guidelines and procedures for estimating the performance and initial operating costs of the system are also discussed.

- Economic analyses. The initial and life-cycle costs for ACES and alternative systems have been computed and compared. The initial costs for ACES are appreciably higher than for non-solar alternative systems, but the life-cycle costs are competitive with non-gas-fired systems at today's energy costs. Fig. 5 provides cost information on the four systems tested at Knoxville, plotted as

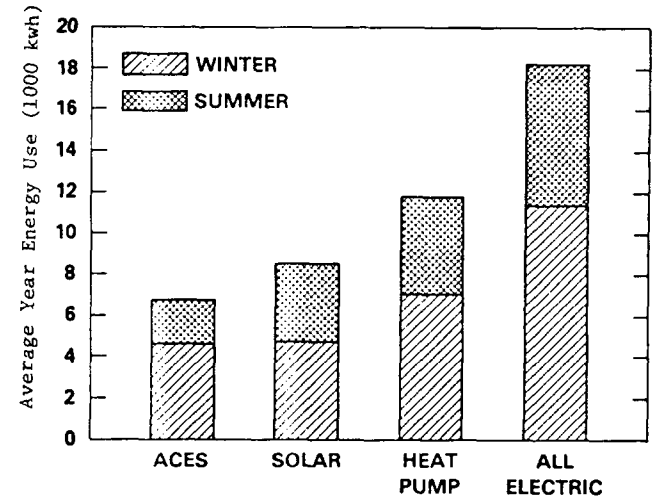
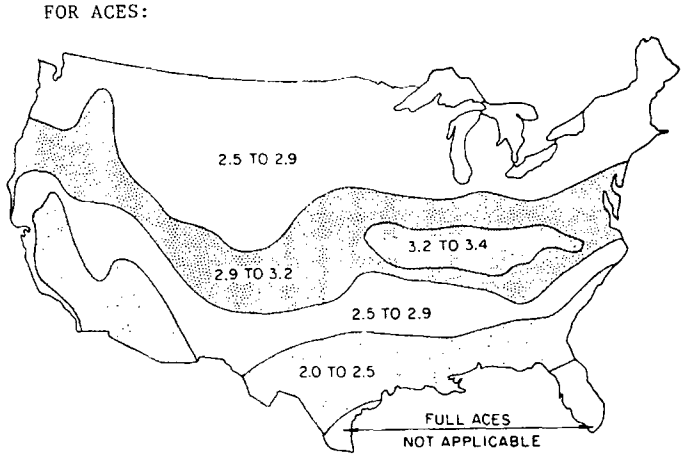



Fig. 3. Energy use for average weather year for systems tested at Knoxville site. All systems provide space heating and cooling and domestic hot water. Heat pump system was high efficiency unit with electric resistance water heater. Solar energy system was designed to supply 80% of space and water heating by solar energy; space cooling loads were estimated for hypothetical electric central air conditioner, since house had no active cooling system.

a function of annual energy use at the Knoxville site. The lower portion of each bar is the initial cost of the system, the central portion is the present value of maintenance costs (including replacement parts), and the top portion of each bar is the present value of operating costs over the presumed twenty-year life of the system. The costs are 1979 costs, and a present worth factor of 16.4 was assumed.

- Evaluation of ACES as a utility load-management option. Because of its high first costs, it appears unlikely that ACES will soon become an important factor in the residential HVAC market. However, storage, which is an integral part of the ACES concept, may make the promotion of ACES use an economically attractive option for utilities. Because of its constant efficiency and capacity, ACES' demand

FOR ACES:




FOR AIR-TO-AIR HEAT PUMP:

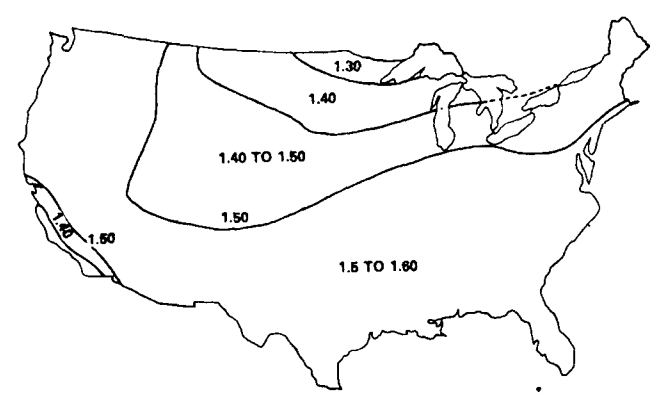



Fig. 4. Annual coefficients of performance (ACOP's) for ACES (left) and air-to-air heat pump ARI rated at 3.2 (right). Coefficients include heating, cooling, and water heating, the latter being done with resistance heating in the air-to-air heat pump system, and are calculated for an 1800 square foot, well insulated house.

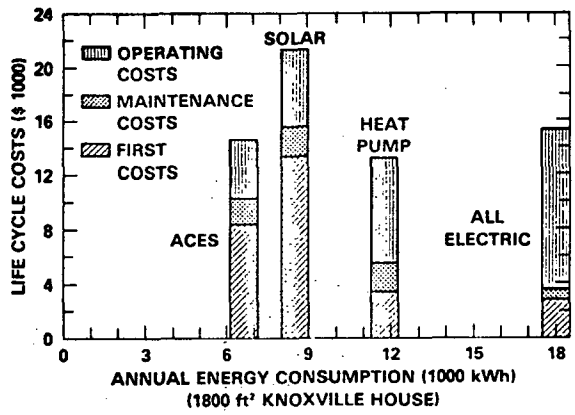



Fig. 5. Life-cycle costs and annual energy use for average weather year for four systems tested at Knoxville site. Present worth factor = 16.4, 20-year assumed lifetime.

for electric power is without sharp peaks and fluctuations, and this leads to high load factors. This is illustrated in Fig. 6, which shows the peak weekly one-hour integrated demand for both an ACES and a high-efficiency heat pump system for a complete test year at the Knoxville site. The loads illustrated are the measured total house loads, including internal as well as HVAC and water heating loads. The ACES curve is marked by the absence of the wild fluctuations which characterized the heat pump curve.

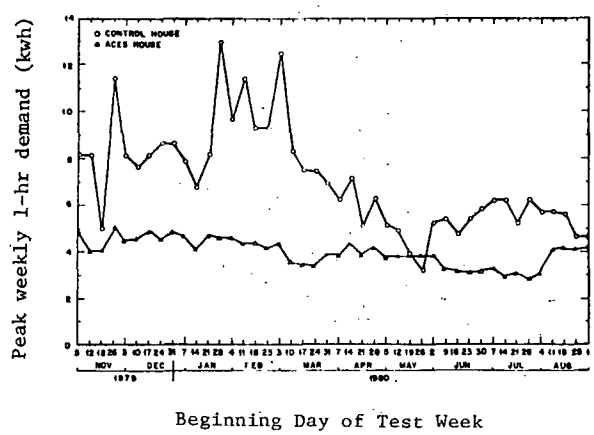



Fig. 6. Measured peak weekly one-hour integrated demand for two Knoxville systems during 1970-1980 test year.

Of interest to summer peaking utilities, the ACES exhibits substantially smaller summertime demand and electricity use than alternative systems studied. Fig. 7 shows calculated monthly energy use for three systems in an 1800 square foot well-insulated house in Philadelphia, PA. Similar data has been derived for 114 other U.S. cities (5).

## FUTURE ACTIVITIES

A slide presentation emphasizing the advantages that ACES might offer utilities has been prepared, and is currently being shown to selected interested utilities and organizations representing them. Engineering and design support is being offered to any utility or other entity interested in further field tests of ACES.

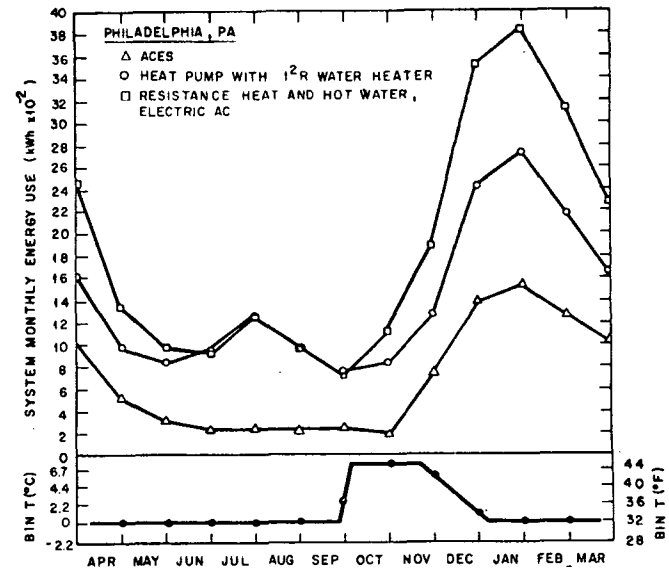



Fig. 7. Calculated monthly energy use by three systems during average weather year in 1800 square foot well-insulated house in Philadelphia, PA. The estimated bin temperature for the ACES is plotted at the bottom.

## REFERENCES


1. E. C. Hise, Performance Report for the ACES Demonstration House August 1976 through August 1977, ORNL/CON-19, March 1978.

2. A. S. Holman, et al., Annual Cycle Energy System (ACES) Performance Report November 1977 through September 1978, ORNL/CON-42, May 1980.

3. V. D. Baxter, et al., ACES - Final Performance Report December 1978 - September 1980, ORNL/CON-64, May 1981.

4. E. A. Nephew, et al., Theory and Design of an Annual Cycle Energy System (ACES) for Residences, ORNL/CON-43, May 1980.

5. L. A. Abbatiello, et al., Performance and Economics of the ACES and Alternative Residential Heating and Air Conditioning Systems in 115 U.S. Cities, ORNL/CON-52, March 1981.


## PUBLICATIONS


1. H. C. Fischer, et al., Summary of Annual Cycle Energy System Workshop I, held Oct. 29-30, 1975, Oak Ridge, TN, ORNL/TM-5243, Oak Ridge National Laboratory, Oak Ridge, TN, July 1976.

2. J. C. Moyers, et al., Design Report for the ACES Demonstration House, ORNL/CON-1, Oak Ridge National Laboratory, Oak Ridge, TN, October 1976.

3. E. A. Nephew, et al., The Annual Cycle Energy System: Initial Investigations, ORNL/TM-5525, Oak Ridge National Laboratory, Oak Ridge, TN, October 1976.

4. I. G. Cantor, Investigation of Alternate Types of Water Storage Components to be Utilized in Conjunction with the ACES System, ORNL/Sub-78/14233/1, Oak Ridge National Laboratory, Oak Ridge, TN, December 1977.

PUBLICATIONS cont'd

5. E. C. Hise, Performance Report for the ACES Demonstration House, August 1976 through August 1977, ORNL/CON-19, Oak Ridge National Laboratory, Oak Ridge, TN, March 1978.

6. H. C. Fischer, Development and Testing of a Single-Plate and a Two-Plate Ice-Maker Heat Pump, ORNL/CON-21, Oak Ridge National Laboratory, Oak Ridge, TN, April 1978.

7. A. S. Holman and V. R. Brantley, ACES Demonstration: Construction, Startup, and Performance Report, ORNL/CON-26, Oak Ridge National Laboratory, Oak Ridge, TN, October 1978.

8. V. D. Baxter, Intermediate Report on the Performance of Plate-Type Ice-Maker Heat Pumps, ORNL/CON-23, Oak Ridge National Laboratory, Oak Ridge, TN, October 1978.

9. A. S. Holman, et al., Annual Cycle Energy System (ACES) Performance Report, November 1977 through September 1978, ORNL/CON-42, Oak Ridge National Laboratory, Oak Ridge, TN, May 1980.

10. R. E. Minturn, et al., ACES 1979: Capabilities and Potential, ORNL/CON-48, Oak Ridge National Laboratory, Oak Ridge, TN, June 1980.

11. J. W. MacArthur, Economic Analysis of Annual Cycle Energy System, ORNL/Sub-7470/V1, V2, V3, Energy Resources Center, Honeywell, Inc., Minneapolis, MN, May 1980.

12. E. A. Nephew, L. A. Abbatiello, and M. L. Ballou, Theory and Design of an Annual Cycle Energy System (ACES) for Residences, ORNL/CON-43, Oak Ridge National Laboratory, Oak Ridge, TN, May 1980.

13. M. L. Ballou, et al., MAD: A Computer Program for ACES Design Using Monthly Thermal Loads, ORNL/CON-51, Oak Ridge National Laboratory, Oak Ridge, TN, March 1981.

14. V. D. Baxter, Ice-Maker Heat Pump Development: Final Report, ORNL/CON-50, Oak Ridge National Laboratory, Oak Ridge, TN, September 1980.

15. V. D. Baxter, Final ACES Performance Report, December 1978 through September 15, 1980, ORNL/CON-64, Oak Ridge National Laboratory, Oak Ridge, TN, May 1981.

16. L. A. Abbatiello, The Annual Cycle Energy System Characteristics and Performance, presented at the International Congress on Building Energy Management, Povoa de Varzim, Portugal, May 1980.

17. J. L. Bledsoe, et al., Data Collection and Processing for the ACES, ORNL/CON-59, Oak Ridge National Laboratory, Oak Ridge, TN, (in preparation).

# Overview of Active Solar Absorption/Rankine Cooling Program†

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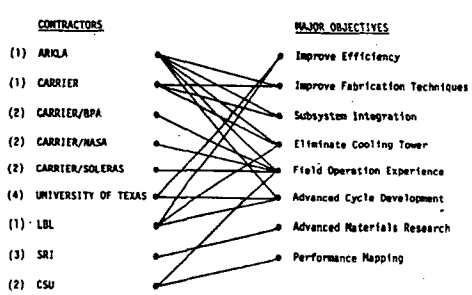
## OBJECTIVE

The overall objective of the DOE Active Solar Cooling program is to develop the basic solar cooling technology that industry can draw upon to engineer and produce solar cooling systems that will compete favorably with conventional cooling systems. By taking responsibility for the high-risk early research and development stages that private industry cannot yet justify undertaking, DOE will lay the foundation for technically sound solar cooling systems that the private sector can commercialize when market conditions are suitable.

## BACKGROUND

A number of specific major objectives have been identified as necessary to achieve the overall objective. To meet these major objectives, a number of projects are being supported in both the absorption and the Rankine cooling areas. The correspondence between the individual projects and the major objectives is shown in Figure 1 for the absorption program and in Figure 2 for the Rankine program. Brief descriptions of the technical content of each project are given below in the Technical Accomplishments section. This paper is an update of the review of project activities that was reported at the March 1980 Annual DOE Active Solar Heating and Cooling Contractors Review Meeting (1). This information covers the period March 1980 to May 1981.

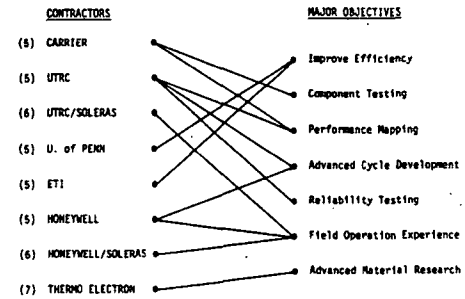
Figure 1.
ABSORPTION COOLING PROGRAM




## SUMMARY

The tasks being performed in the absorption and Rankine program areas run the gamut from basic work on fluids to development of chillers and chiller components, to field and reliability testing of complete cooling systems. In the absorption program there are three active component development projects (1), four systems field test projects (2), one advanced fluid study project (3), and one advanced cycle study project (4) currently funded by DOE. In the Rankine program, there are five active component development projects (5), two system field test projects (6), and one advanced fluid study project (7).

Figure 2.
RANKINE COOLING PROGRAM




Several of the solar cooling projects are managed by NASA/MSFC (Marshall Space Flight Center, Huntsville, Alabama): the Carrier/NASA absorption projects and the Honeywell Rankine project. The Honeywell project is a continuation of the earlier NASA/MSFC "404" program; two other 404 projects, by AiResearch and General Electric, have now been completed.

Some of the DOE solar cooling projects reported in last year's Overview (1) have also been completed since then, and several others are in the final report stage. Those completed are: an earlier ARKLA absorption project (Contract No. AC03-76SF-10507, Development of Third-Generation Three Ton Lithium Bromide-Water Absorption Chiller), the Brookhaven National Laboratory project (Contract No. C-02-0016, Development and Use of a Simulatory to Test Solar Cooling Subsystems), and the EIC Corporation project (Contract No. AC03-77CS-34537, Design of a Solar Air Conditioner Using Solid Phase Absorbent-Chemical Heat Pump). The projects in the final report phase are those by Southern Research Institute (Contract No. AC03-77CS-31586, Analysis of Advanced Conceptual Absorption Chiller Designs and the Institute of Gas Technology (Contract No. AC03-77CS-31439, Development of New Fluids for Solar Absorption Cooling). None of these projects will be included in the following section.

Three new projects have been started during the past year and will be reported on here: Determination of Properties of Fluids for Solar Applications, by SRI International; Fabrication and Installation at Field Test Sites of 120-ton and 75-ton Absorption Chillers, by Carrier Corporation (NASA/MSFC contracts); and Determination of the Thermal Stability of Organic Working Fluids Used in Rankine Cycle Systems, by Thermo Electron. In addition, much of the on going systems test work at Colorado State University is an integral part of the active solar cooling program and that effort will be reported here also.

† This work has been supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Solar Applications for Buildings, Active Heating and Cooling Division of the U.S. Department of Energy under Contract No. W-7405-ENG-48.

*Employees of Keller & Gannon, a subsidiary of Lester B. Knight Associates, Inc.

A joint US/Saudi Arabian effort has contributed to the development of Rankine and absorption solar cooling systems. Four projects have been funded by this (SOLERAS) activity, all to be installed and operated in Phoenix, Arizona. Each manufacturer has contributed some private money in addition to the 50/50 share by the U.S. and Saudi Arabia. Table 1 summarizes these projects.

Table 1.

U.S./SAUDI ARABIAN PROJECTS (SOLERAS PROGRAM)

| CONTRACTOR | TYPE SYSTEM CYCLE/COLLECTORS | COOLING CAPACITY (TONS) |
|---|---|---|
| UTRC | RANKINE/TROUGH TRACKING | 18 |
| HONEYWELL | RANKINE/EVACUATED TUBE | 25 |
| CARRIER | ABSORPTION (WATER COOLED)/TROUGH TRACKING | 15 |
| CARRIER | ABSORPTION (AIR COOLED)/FRESNEL DUAL TRACKING | 10 |

Some technical highlights of the program are presented in Table 2.

Table 2.

TECHNICAL HIGHLIGHTS

| CONTRACTORS | ACCOMPLISHMENTS |
|---|---|
| • HONEYWELL (RANKINE) | Continued successful field operation in Lawrence, Kansas and other locations. |
| • UTRC (RANKINE) | Prototype unit converted to air cooled operation, with successful start-up in Phoenix SOLERAS test project. |
| • U. of TEXAS (ABSORPTION) | Identification of potential for substantial technical performance improvements through development and use of computer models of double-effect absorption chillers. |

## TECHNICAL ACCOMPLISHMENTS

The individual absorption and Rankine projects are identified in this section, along with the main features and accomplishments/status of each.

### ABSORPTION PROJECTS

Contractor: ARKLA

Contract No.: AC03-77CS-34593

Project: Develop and test unitary 3-ton and 25-ton $LiBr/H_2O$ absorption cooling systems. Develop and test double effect, gas fired auxiliary absorption unit.

Features: Unitary package; fuel aux; manufacturing study; application and maintenance manuals; tooling improvements; lab reliability testing. Gas fired double effect absorption unit provides highly efficient COP (1.0) on non-solar operation.

Accomplishments:

- Two field test installations have been completed and performance predictions verified.
- 25-ton unitized packaged chiller has been lab tested and prepared for field installation scheduled for Building 71 at Lawrence Berkeley Laboratory.
- Tooling has been developed to produce absorber and condenser heat exchangers at reduced cost.

Future Plans:

- Complete design of double effect gas fired auxiliary absorption unit; fabricate and test.
- Assist in Field test of 3-ton system at CSU and 25-ton system at LBL.
- Complete laboratory reliability testing of 3rd generation 3-ton units.

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Contractor: Carrier Corporation

Contract No.: AC03-77CS-51587

Project: Development of air cooled absorption chillers.

Features: Air cooled; $120^{\circ}$ C - $130^{\circ}$ C input temperature; falling film approach using additives; manufacturing study and market evaluation.

Accomplishments:

- Prototype #1, a 10KW air-cooled unitized package chiller has been fabricated; performance testing is in progress.

Future Plans:

- Improve design of 10KW absorption chiller; fabricate and test three 10KW units.
- Design, fabricate and test a 70KW absorption chiller.

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Contractor: Carrier Corporation/BPA

Contract No.: AC79-79BP-10467

Project: Fabricate two 15-ton unitized absorption chillers and install them at two field sites: Dalles, Oregon and Tyler, Texas.

Features: Water-cooled, LiBr chillers; packaged; on-site testing; personnel training.

Accomplishments:

- Tyler, Texas and Dalles, Oregon installations have been successfully operated for a full cooling season.

Future Plans:

- The Tyler, Texas installation will continue to be operated during the summer of 1981 to obtain additional data on field test performance.

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Contractor: Carrier Corporation/NASA

Contract No.: DEN 8-000005 & DEN 8-000015

Project: Fabricate one 120-ton absorption unit to be installed at Frenchman's Reef, V.I. and two 75-ton absorption units to be installed at Houston and Las Vegas sites. Field test systems at each site.

Features: Water-cooled; LiBr chillers; packaged; on-site testing; personnel training.

Accomplishments:

- Frenchmen's Reef chiller has been installed and operated at the site and is meeting predicted performance.

Future Plans:

- Las Vegas and Houston installations are planned for completion in 1982.

---

Contractor: Carrier Corporation/SOLERAS

Contract No.: DE-FC03-80ET 20643

Project: Design, fabricate, install, and field test two active solar cooling systems using absorption chillers.

Features: System #1, 15-ton water-cooled absorption cshiller and trough tracking collectors (Acurex).

System #2, 10-ton air-cooled absorption chiller and Fresnel dual tracking collectors (E-Systems).

Accomplishments:

- The 15-ton water-cooled system has been installed and operated at Phoenix test site. Debugging is presently underway.
- The 10-ton air-cooled system is complete except for final placement of the chiller. The absorption unit is undergoing extensive lab tests at the Carrier facility in Syracuse, N.Y.

Future Plans:

- Both systems will be field tested and the data will be used in the active solar cooling program.

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Contractor: University of Texas at Austin

Contract No.: DE-AC03-79SF 10540

Project: Analysis of double effect absorption cooling system.

Features: Determine performance as a function of many parameters, including inlet water, cooling water and chilled water temperatures, location and size of heat exchangers; $LiBr/H_2O$ pair.

Accomplishments:

- Computer model has been successfully developed to simulate operating performance of double-effect absorption cycles.
- Final report is in draft form.

Future Plans:

- Apply the computer model developed for double-effect absorption chillers to investigate control schemes, parasitic power requirements, and heat exchanger optimization relative to system performance.
- Assist industry in using computer model.

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Contractor: Lawrence Berkeley Laboratory

Contract No.: W-7405-ENG-48

Project: Development of advanced cycle absorption chillers.

Features: Air-cooled; internally powered solution pump; tube-in-tube heat exchangers; single effect chiller with 3-ton capacity at 218°F using $NH_3/H_2O$; advanced cycles with COP increasing with temperature; heat pump operation also.

Accomplishments:

- Early testing of single effect chiller verify the condenser, generator, preheater and recuperative solution pump are operating according to design.
- Design drawings have been completed and fabrication is underway for the double effect regenerative absorption chiller.

Future Plans:

- Detailed testing will conclude on the single effect chiller.
- Fabrication will be completed, followed by testing of the double-effect regenerative absorption chiller.
- Cycle analysis of the single-effect regenerative absorption chiller will incorporate the results of the SRI measurements of fluids properties.

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Contractor: SRI International

Contract No.: DE-AC03-80CS 30221

Project: Determine pressure-volume-temperature (P-V-T) data and calorimetric data for organic refrigerant and absorbent fluids and their binary mixtures.

Features: Conduct a critical survey of thermodynamic data on absorption fluids and fluid pairs and measure properties of selected fluids and their mixtures.

Accomplishments:

- Literature search has been completed and findings documented.
- Experimental measurements of the properties of the fluids are underway.

Future Plans:

- In-depth experiments will be conducted and the data will be analyzed on selected absorption fluids and fluid pairs.

---

Contractor: Colorado State University

Contract No.: DE-AC03-81CS 30569

Project: Test and evaluate complete solar cooling systems incorporating absorption chillers, both air and liquid cooled, using CSU Solar Houses I and III.

Features: The CSU Solar Houses contain collectors, storage, chillers, pumps, piping and controls, all fully instrumented for collecting test data. The test systems are scheduled to operate with an ARKLA evaporatively cooled 3-ton chiller and a Carrier air-cooled 3-ton chiller.

Accomplishments:

- Instrumentation is in place and programmable controllers are being installed in the Solar Houses.
- Philips Mark I heat pipe evacuated tube collectors have been installed and operated on Solar House I.
- The ARKLA 3rd generation unit was installed in Solar House I and operated briefly at the end of the 1980 cooling season.

Future Plans:

- Extensive testing will be done on the ARKLA 3rd generation chiller using Solar House I during summer 1981.
- Evacuated tube collectors will be selected and installed on Solar House III during 1981.
- The Carrier air-cooled 3-ton absorption chiller will be installed in Solar House III and operated during the summer of 1982.

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Contractor: University of Maryland

Contract No.: DE AC03-79CS 30204

Project: Technical program support and special studies.

Features: General program support activities; optimization studies and parametric analysis.

Accomplishments:

- Report on refrigerant/absorbent pairs completed.
- Transient simulation of absorption cycles completed and compared favorably to experimental data.
- Simplified cooling design charts developed on a regional basis, and compared to detailed simulation.
- Parametric studies of PV/T systems have identified performance sensitivities to collector area, storage capacity and other parameters.

Future Plans:

- Technical support activities will continue.
- The cooling design charts and PVT analysis will be completed during the next year.

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## RANKINE PROJECTS

Contractor: Carrier Corporation

Contract No.: DE-AC03-77CS 31590

Project: Development of 25-ton Rankine chiller.

Features: Air-cooled; R-113 in both loops; 290°F; 20,000 RPM turbine directly driving compressor; electric motor share the load.

Accomplishments:

- Turbo-compressor assembly has passed the air test and is under R-113 test on prototype.
- Two stage boiler feed pump has been fabricated and successfully tested.

Future Plans:

- Operate T/C assembly to design conditions and determine performance map of the chiller.
- Continue development of chiller to include air cooled condensing coils and microprocessor based control system.

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Contractor: United Technologies Research Center (UTRC)

Contract No.: DE-AC03-77CS 34510

Project: Development of 18-ton Rankine heat pump.

Features: Air-cooled; R-11 in both loops; 290°F; 45,000 RPM turbo compressor; fossil fuel auxiliary; both heating and cooling operation.

Accomplishments:

- Endurance testing and component assessment has been completed.

- Advanced heat pump design has been completed to improve operating efficiency and reduce fabrication unit.

Future Plans:

- Fabrication and testing of advanced heat pump design.

---

Contractor: United Technologies Research Center/SOLERAS

Contract No.: DE-FC02-80ET 20642

Project: Design, fabricate, install and field test a Rankine cycle solar cooling system.

Features: 18-ton Rankine cycle heat pump; air-cooled; R-11 in both loops; 290°F; 45,000 RPM turbo-compressor; fossil fuel auxiliary; trough tracking collectors.

Accomplishments:

- Prototype unit converted to air-cooled operation with successful start-up in Phoenix SOLERAS test project.

Future Plans:

- Field testing of SOLERAS system through two cooling seasons.

---

Contractor: University of Pennsylvania

Contract No.: AC03-78ET-20110

Project: Development of 20-ton superheated steam Rankine chiller.

Features: Solar-boiled water at 260°F; fuel-superheated to 1100°F; 5 stage, 15,000 RPM radial outflow turbine and rotary expander design; stored water flashes to steam.

Accomplishments:

- Fabrication of turbine assembly is being completed and test loop is under construction.

Future Plans:

- Performance map of turbine will be measured using the steam power loop.

---

Contractor: Energy Technology, Inc. (ETI)

Contract No.: AC03-80CS-30214

Project: Development of a superheated steam Rankine turbine.

Features: Solar-boiled water at 300°F; fuel-superheated to 1000°F; 4 stage, 75,000 RPM radial outflow turbine; nominal 30 ton.

Accomplishments:

- Laboratory testing of turbine has been completed.
- Final report is in final draft stage.

Future Plans:

- Project completed.

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Contractor: Honeywell

Contract No.; NAS8-32093

Project: Development of Rankine cycle chillers and field installation for testing.

Features: 3-ton and 25-ton Rankine chiller design; water-cooled; R-113 power cycle loop; R-12 Refrigerant cycle loop; radial inflow turbine, 40,000 RPM; double reduction gear box to 1200 RPM; reciprocating compressor (centrifugal compressor optional); motor/generator auxiliary.

Accomplishments:

- Field test sites continue to be operated and monitored.
- High temperature Rankine unit fabricated and installed at NASA/MSFC.

Future Plans:

- Further technology development to improve efficiency of chiller and cooling system.
- Continued monitoring of operational test sites.

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Contractor: Honeywell/SOLERAS

Contract No.: DF-FE02-80ET 20645

Project: Design, fabricate, install and field test 25-ton Rankine cycle cooling system.

Features: Air-cooled; 300°F; R-113 power cycle loop, R-12 refrigerant cycle loop; radial inflow turbine, 40,000 RPM; double reduction gearbox to 1200 RPM; reciprocating compressor; motor/generator auxiliary.

Accomplishments:

- Components have been fabricated and system installed at Phoenix site. Operating tests are presently underway.

Future Plans:

- Continue field operating tests for two cooling seasons.

---

Contractor: Thermo Electron Corporation

Contract No.: DE-AC03-80CS 30220

Project: Conduct thermal stability on fluids used in Rankine cycle systems.

Features: Five identical dynamic test loops to be constructed to test decomposition of organic working fluids as a function of temperature and the materials in contact with the fluid.

Accomplishments:

- Loop designs are completed and construction underway.
- R-11 and R-113 selected as fluids to be tested in four of the loops.

Future Plans:

- Carry out dynamic thermal stability tests of fluids in 5 loops.
- Conduct supplemental capsule tests.

---

Contractor: Hittman Associates, Inc. (HAI)

Contract No.: DE-AC03-79CS 30202

Project: Technical program support and special studies.

Features: General program support activities; special system, component and economic analysis.

Accomplishments:

- Thermal storage for solar Rankine and absorption cooling systems report completed.
- Performance test plans completed for Rankine and desiccant cooling equipment.
- Hybrid solar/fossil Rankine cooling conceptual study completed and report written.
- High temperature solar cooling topical report completed.
- Preliminary plan for testing of cooling system control strategies was developed.

Future Plans:

- Technical support activities and special studies will continue.

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SYSTEM ANALYSIS PROJECTS

Contractor: Science Applications, Inc. (SAI)

Contract No.: XB-0-9145-1

Project: Performance and economic analyses of absorption and Rankine cooling systems.

Features: Develop computer programs to evaluate system performance and economics for residential and commercial active solar cooling systems; selected cities used.

Accomplishments:

- Completed detailed system modeling of 25-ton absorption and Rankine chillers.
- Annual system simulations and economic analyses completed for four cities.

Future Plans:

- Continued system evaluation for additional locations and system configurations.

---

In addition to the activities in the above contracts, extensive planning efforts are underway for long-range development efforts in the active solar cooling program. An important function for this planning effort was the active solar cooling workshop attended by representatives from the solar and HVAC industry, utilities and local government, financial and marketing organizations, and the architecture and engineering fields. The workshop concentrated on four major topics: (1) Technical, (2) financial, (3) codes and standards, and (4) marketing aspects of active solar cooling. The response from the 28 attendees provided a data base for follow-on presentations that will further contribute to DOE's planning efforts.

FUTURE ACTIVITIES

From a technical point of view, the Rankine and absorption projects have progressed from the very early design and development stage to operating prototypes undergoing performance evaluations. Operation of many of the early prototypes have identified key areas of concerns for Rankine units such as optimized system control, bearing design and cooling schemes, back-up system alternatives, and heat exchanger designs. For the absorption systems, concerns are fluid flow design in absorber section, heat exchanger optimization, control functions, and parasitic power reduction. Common concerns for both systems are cost reduction, collector and storage selection criteris, and fluid stability at high temperatures.

In the past year, work has begun to identify quantitative program goals. A method of system analysis has been developed to provide cost and performance values using computer simulation models. This effort is continuing.

It is expected that further development in the absorption and Rankine cooling technologies will ensue as a result of competitive solicitations for advanced cooling systems and components. In addition, it is planned to conduct systems engineering of developed components leading to design and application manuals that will be available for use by A&E firms and the HVAC Industry.

Publications/Reports/References

(1) Michael Wahlig, Al Heitz, and Barbara Boyce, Overview - Absorption/Ranking Solar Cooling Program, Proceedings of the Annual DOE Active Solar Heating and Cooling Contractors' Review Meeting, March 26-28, 1980.

# Absorption Heat Pumps

# HIGH EFFICIENCY ADVANCED ABSORPTION HEAT PUMP

COLUMBIA GAS SYSTEM SERVICE CORPORATION
GAS RESEARCH INSTITUTE

Edward A. Reid, Jr.

The Research Department of the Columbia Gas System Service Corporation and the Efficient Utilization Division of the Gas Research Institute are jointly funding a research program to design, fabricate and test a high efficiency absorption heat pump for the residential market. Design, fabrication and testing of the heat pump are being performed in the laboratories of the Columbia Gas System Service Corporation.

The performance targets established for this high efficiency absorption heat pump are a heating coefficient of performance of 1.5 and a cooling coefficient of performance of 0.8 at rating conditions, including parasitic electric power consumption. The resulting heat pump would have a space heating capacity of 68,000 BTU/hour, and a space cooling capacity of 36,000 BTU/hour at rating conditions.

Figure 1 is a very simplified schematic block diagram of the high efficiency absorption heat pump cycle. High temperature, high pressure, refrigerant vapor is produced in the refrigerant generator and heat exchange system, is condensed to a liquid in the condenser, expanded to a low pressure vapor in the evaporator, and mixed with and reabsorbed into the weakened solution returned from the refrigerant generator and heat exchange system in the absorber. The strong solution from the absorber is pumped into the refrigerant generator and heat exchange system. This refrigeration cycle is similar to absorption cycles which have been in use in industry for a number of years. The improved performance would be achieved through several stages of heat recovery within the refrigeration circuit.

The overall goals of Columbia's advanced absorption heat pump program are: design, fabrication and testing of the breadboard heat pump; demonstration of the target performance levels shown in the previous slide; and, preparation of a proposed future development program to advance the heat pump from a breadboard prototype to a packaged, self-contained prototype heat pump.

Columbia has completed fabrication of a breadboard prototype of the refrigeration circuit, including all of the components shown in the previous slide. The absorbent refrigerant solution has been charged into the prototype system, instrumentation of the prototype is complete, and shakedown testing is in progress in our laboratories.

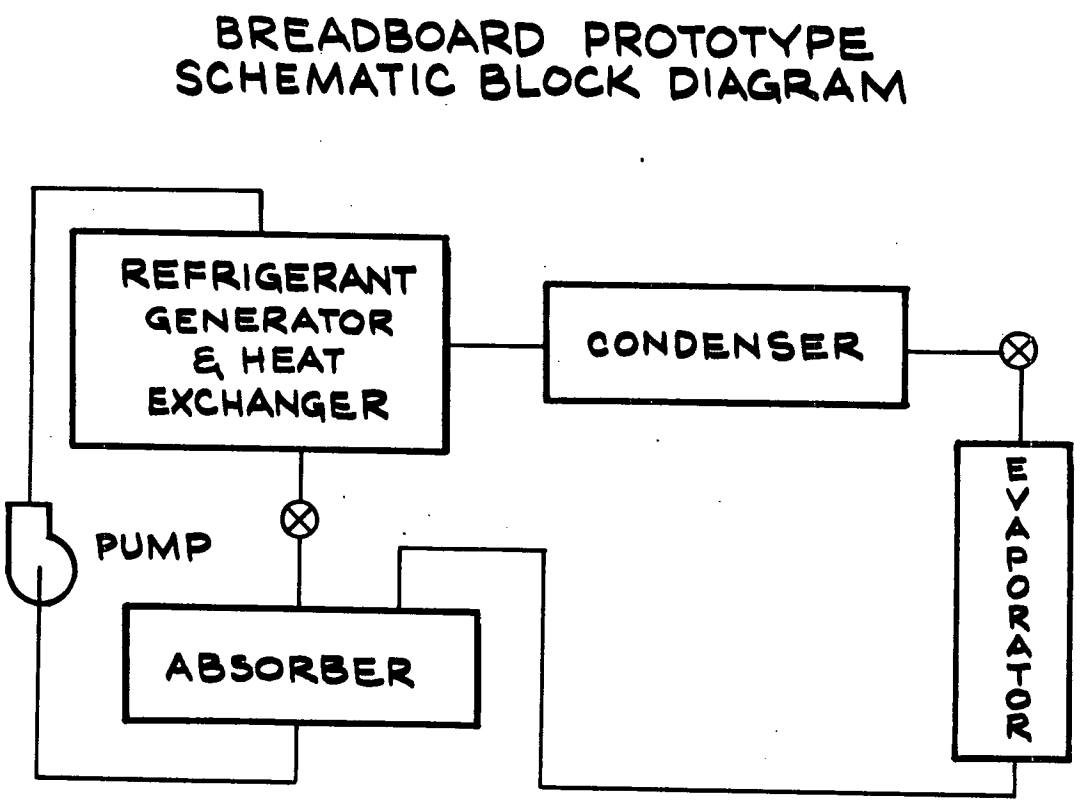


Figure 1

Support studies are underway to study the properties of the solution, and to design and fabricate recuperative and interface heat exchangers for the breadboard prototype.

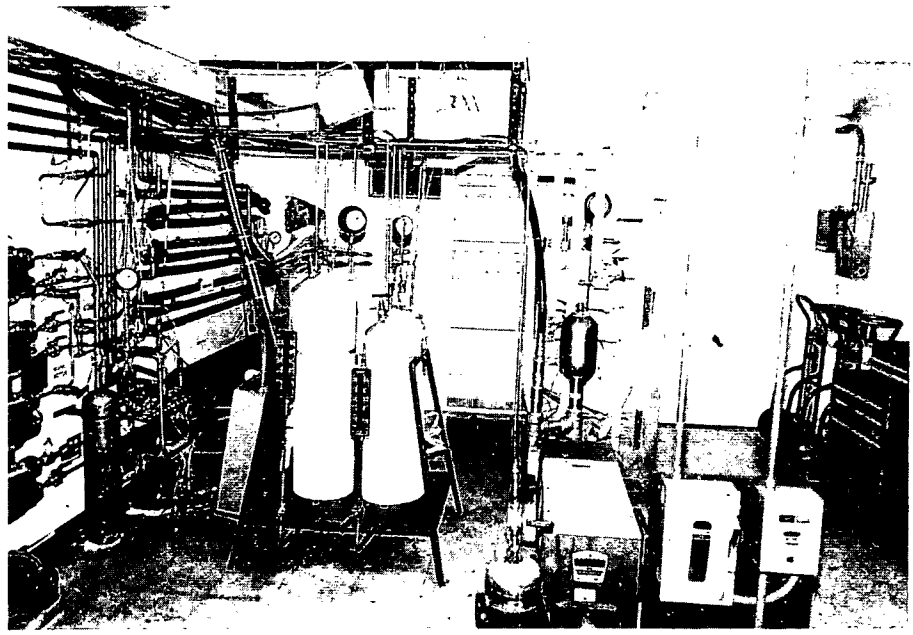

Figure 2

Figure 2 is a photograph of the environmental chamber showing the breadboard heat pump prototype. The breadboard currently uses temperature controlled water heat exchange loops to measure the quantity of energy acquired by the evaporator, to improve the accuracy of these measurements. The water heat exchange loops will be replaced by fan driven air heat exchangers later in this phase of the program.

Figure 3 shows the control and instrumentation panel for the breadboard heat pump testing. The primary data acquisition and analysis tool for this program is a digital data acquisition system which has been programmed not only to acquire data but to calculate component performance and instantaneous heat pump COP as the prototype is tested. The system is also equipped with analog temperature and pressure recorders which permit monitoring of trends in component temperature and pressure during testing.

Columbia's current activities on the program include adjusting the solution charge and its distribution throughout the prototype heat pump, establishing preliminary solution flow rates, checkout of the heat pump instrumentation and preparation of a preliminary performance map of the breadboard prototype. In parallel with these laboratory activities, we are also preparing a preliminary commercialization strategy for a high efficiency absorption heat pump at the request of GRI.

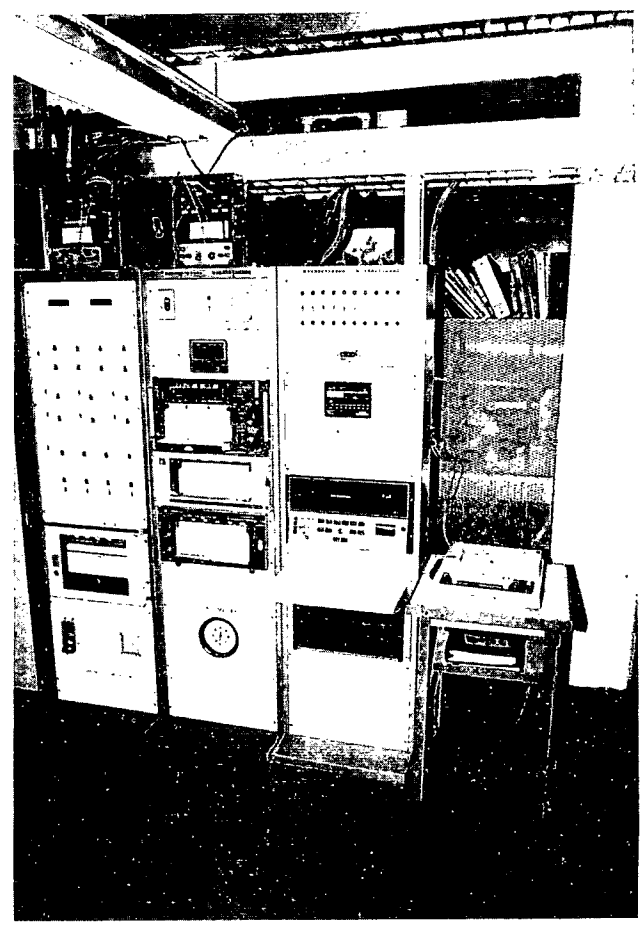

Figure 3

METAL HYDRIDE/CHEMICAL HEAT PUMP DEVELOPMENT PROJECT
SOUTHERN CALIFORNIA GAS COMPANY
BNL NO. 530498-5
SAMUEL J. CUNNINGHAM
PERIOD OF CONTRACT: FEBRUARY 1981 THROUGH MARCH 1984

## OBJECTIVE

The MMHP is a chemical heat pump containing two hydrides for the storage and/or recovery of thermal energy. It utilizes the heat of reaction of hydrogen with specific metal alloys. The MMHP design can be tailored to provide heating and/or cooling over a wide range of input and ambient temperatures. This system can thus be used with a variety of heat sources including industrial waste heat, solar energy or even a fossil fuel. The objective of this effort is the development of the technology which will lead to the demonstration of a cost effective metal hydride heat pump. The scope of this project includes a study of applications and market sectors, conceptual development, thermal and mechanical design, laboratory verification of design and material performance, cost analysis and the design, fabrication and testing of an Engineering Development Test Unit (EDTU). The major technical problems associated with the heat pump can be grouped under one general category: cost effectiveness. The MMHP can only be successful if it provides dependable, long life service to the user at a competitive price.

## CONCEPT

Many crystalline metals can absorb large amounts of hydrogen in their interstitial crystal lattice locations. When this absorption occurs, the resulting material is a metal hydride. The hydrogen can be readily dissociated from some of these hydrides by the application of thermal energy. When this is done, the metal exhibits its original properties. In principle, hydrogen can be reintroduced into the metal and removed from it an indefinite number of times with little or no metallurgical change in the host metal.

Metal hydrides have several common features which make them candidates for heat pumps; hydriding is a reversible reaction, hydriding is an exothermic reaction, dehydriding is an endothermic reaction, the diffusion rate of the hydrogen within the hydrides is rapid, and the pressure-composition isotherms display a plateau over a large range of hydrogen compositions.

The heat-of-formation of the metal hydrides is vastly different for different base metal systems. One alloy, $LaNi_5$, has a heat-of-formation of 7.3 kcal per mole of hydrogen, while cerium hydride has a heat-of-formation near 49.2 kcal per mole. For use in heat pumps, the hydrides must be relatively unstable. In practice, one can optimize hydride thermal properties for the temperature of the heat source used.

Basic Hydride Absorption Cycle. Hydride-forming metal alloys with different thermodynamic characteristics can be used as thermal working elements to provide heat pump operation. The heat pump cycle requires two different alloys chosen to operate together as a working pair. One can be designated as the warm side alloy (WSA) and the other the cold side alloy (CSA). These hydride alloys must be chosen to operate as a pair based on their natural pressure-temperature response characteristics.

Each hydride alloys has its own identity which is commonly described by the Van't Hoff mid-plateau desorption curves which are derived by plotting pressure points taken at mid-plateau from each of a family of desorption isotherms. These points form a straight line when plotted on semi-log axes. Figure 1 shows Van't Hoff curves for the $AB_5$ class of hydrides. The alloys must function as a working pair for a heat pump cycle operating at a specific driving temperature, $T_G$, an intermediate heat sink (condenser) temperature $T_C$, and a low (evaporator temperature $T_E$.

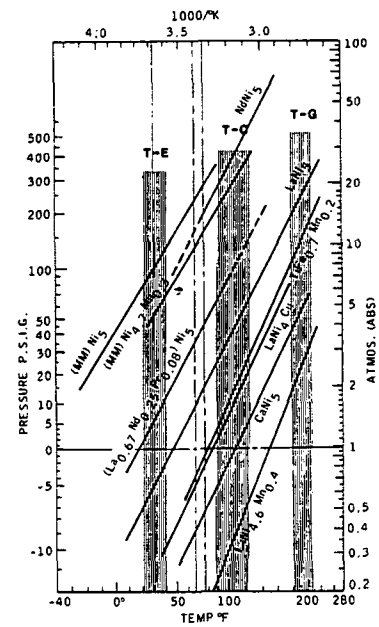


Figure 1. Pressure/Temperature Relationships at Mid-Plateau

The cycle is illustrated in Figure 2 and functions as follows. The fully hydrided WSA is exposed to $T_G$ which allows it to desorb hydrogen at a pressure $P_2$ while absorbing thermal energy from the heat source. Simultaneously the hydrogen-depleted CSA is exposed to a heat sink at $T_C$, which allows it to absorb hydrogen at pressure $P_3$ while rejecting heat to the heat sink. The two alloys are then connected by a flow path. Since $P_2$ is greater than $P_3$, the hydrogen flows from the WSA to the CSA. The flat character of the plateaus for the isotherms guarantees that the relative driving pressure differential between the two sides will remain fairly constant while the majority of the hydrogen passes from one side to the other. This action constitutes the charge half of the cycle.

The refrigeration half of the cycle occurs when the WSA is exposed to the heat sink $T_C$, lowering its equilibrium pressure to $P_1$. The cold side alloy is exposed to the chilled fluid loop and establishes equilibrium at $T_E$ and $P_4$ as heat is absorbed from the chilled fluid and hydrogen is desorbed. The hydrogen can then flow back to the warm side by virtue of the fact that $P_4$ is greater than $P_1$. Thus, to produce refrigeration ($T_E$), high tem temperature heat ($T_G$) is used to fire the system, while the ambient temperature heat ($T_C$) is rejected to the atmosphere.

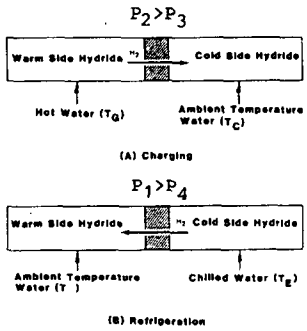


Figure 2. Simplified Heat Pump Diagram

Heating operation is identical to the refrigeration cycle; only the flow of heat used is altered. As in the refrigeration mode, the high temperature heat ($T_G$) is used to fire the system. The low temperature heat absorbed ($T_E$) that was previously used for refrigeration is obtained from the low temperature ambient conditions. The medium temperature heat ($T_C$), which was rejected to the atmosphere in the refrigeration cycle, is used for heating, such as space heating in a dwelling. The advantage of this mode of operation is that almost twice the energy used to fire the system is available for heating. A schematic of a solar-assisted gas fired system is shown in Figure 3 with both warmed and cooled air outlets.

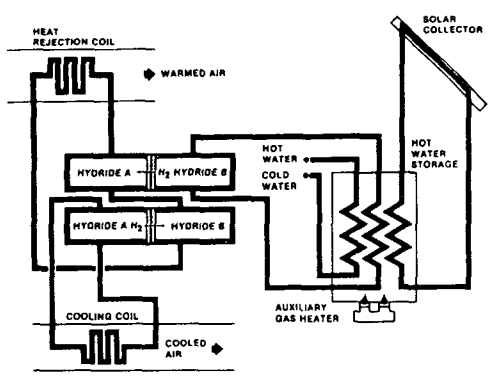


Figure 3. Solar-Assisted Gas Fired Hydride Heat Pump

The method of selecting a pair of alloys capable of cycling through a given set of temperatures begins by plotting mid-plateau Van't Hoff desorption curves of all candidate alloys. A typical set of such curves for some $AB_5$-type alloys is shown in Figure 1. The three operating temperatures for an air conditioning cycle are shown with vertical bars. Useful alloy pairs are identified by selecting a cycle in which certain pressure differentials exist; $P_2>P_3$ and $P_4>P_1$.

PROJECT SUMMARY

At the start of the program in Phase I, the conceptual design will be established followed by a national market survey and characterization to identify the most attractive applications and markets. This will be followed by a first-cut commercial feasiility analysis. Upon the completion of these tasks, an engineering devleopment test unit (EDTU) will be designed for the target market application.

In Phase II, the EDTU will be fabricated, tested, rebuilt as necessary and retested. This will lead to the detailed design of an Engineering Evaluation Test Unit (EETU). Additionally, based on the technical results obtained in this Phase, a more in-depth commercial feasibility analysis will be performed. During Phase III, the EETU will be fabricated and tested.

The Southern California Gas Company is currently under contract to proceed with Phase I of the program identified above. Funding for Phases II and III are not yet obligated. The conceptual design has been completed. The market survey has been initiated. Potential applications have been identified and characterized. Competing systems have also been characterized in each potential application. A screening criteria is currently being developed in order to evaluate the MMHP for the potential applications already identified. This will lead to a recommended application(s). Efforts to develop the thermal and mechanical designs, controls systems, and selected hydride materials have also been initiated. This work is of a general nature since to date the specific application has not been selected. Development of detailed designs will begin when the most attractive application is identified.

TECHNICAL ACCOMPLISHMENTS

- Program was initiated in February 1981.
- The conceptual design of the metal/chemical heat pump has been completed.
- A commercial feasibility program has been initiated by Booz Allen Hamilton. Potential applications in the residential, commercial, and industrial and transportation sectors have been identified and characterized. Screening criteria have been developed to evaluate the MHHP in each application. Technical criteria receive 40 percent, market criteria 50 percent and the "other" 10 percent of the weighting factors.
- A analytical model of the heat pump is being developed. Heat transfer, a critical element for acceptable performance is emphasized.

FUTURE ACTIVITIES

This program was initiated in early 1981. The projected three phases of contractual effort were built on several years of industrial experience in related technology. The three phases will be accomplished in 38 months. The first phase consists of concentrated efforts in technologies to support the development of the EDTU and commercialization efforts to select the optimum configuration and application for this device. In the second phase the EDTU will be built and tested extensively. Technology work will continue. The test results will be used to design an Engineering Evaluation Test Unit (EETU). That unit, to be built in Phase III, will be designed for non-laboratory operation. It will have a minimum amount of instrumentation and automatic controls.

In the post contract effort a unit similar to the EETU will be installed in a field location and

tested extensively. Performance will be monitored with automatic data acquisition equipment.

The commercialization plan development will restart in Phase II and continue through the post contract period. Technical performance and production cost information should be well developed at that time. The commercialization plan will delineate all activities subsequent to the EETU field test.

PUBLICATIONS

Because of the recent initiation of this work, no public reports have been prepared.

ORGANIC ABSORPTION GAS-FIRED RESIDENTIAL HEAT PUMP
ALLIED CORPORATION
CONTRACT NUMBER 86X-24610C
PRINCIPAL INVESTIGATOR: KEVIN P. MURPHY
PERIOD OF CONTRACT: BEGIN OCTOBER 1, 1979, END JUNE 1, 1981

## OBJECTIVES

The current objectives of this program are to develop an organic absorption gas-fired residential heat pump system to a production prototype level, analyze the market and business potential for the absorption heat pump market, evaluate the system first cost and actual annual performance and operating costs, develop computer simulation of the absorption system to more efficiently examine design variables, develop data for toxicity and product safety and operate test units to identify problems in actual service.

## CONCEPT

The use of the absorption cooling cycle is an old technology and, as with any cooling cycle, it can be made to perform as a heat pump. The system under development utilizes a new absorption pair, R133a ($CF_3CH_2CL$) as the refrigerant, and ETFE (ethyltetrahydrofurfurylether) as the absorber fluid. A diagram of the basic configuration is shown in Figure 1. In the cooling mode, refrigerant is evaporated chilling the glycol solution which circulates to the house coil. In the heating mode, the heat is rejected by both the absorber and condenser to the glycol solution which then goes to the home.

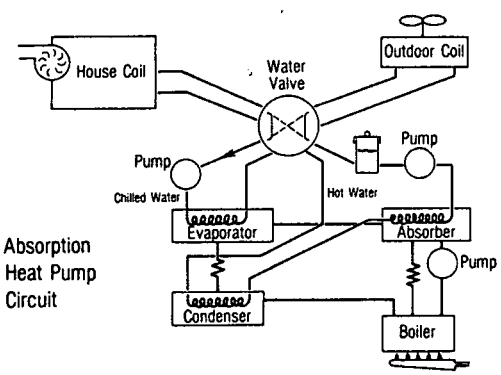


FIGURE 1.

## SUMMARY

The program has continued the development of the absorption heat pump using R133a/ETFE to a production prototype stage while meeting most performance goals. Included in this development was the demonstration of a long-lived reliable solution pump which had been a major problem in other absorption system designs.

Six units of an early hardware design were constructed and tested. Two of these units were placed in home heating service during the 1980-81 season.

A market evaluation of the business potential of the absorption system was made identifying location and size of the likely market for such a system.

A performance simulation analysis was performed for seven cities in the U. S. From these, general characteristics of the areas having the greatest performance benefits were established.

## TECHNICAL ACCOMPLISHMENTS

- Production Prototype Development. The use of new aluminum extrusions specifically designed for the absorption heat pump has allowed substantial reduction in the size of the major components while maintaining or improving performance. Final designs will have the outdoor package (all components except "indoor coil") in a 42"x42"x40" configuration. The size controlling component is currently the absorber. Figure 2. shows a mock-up of the production prototype configuration.

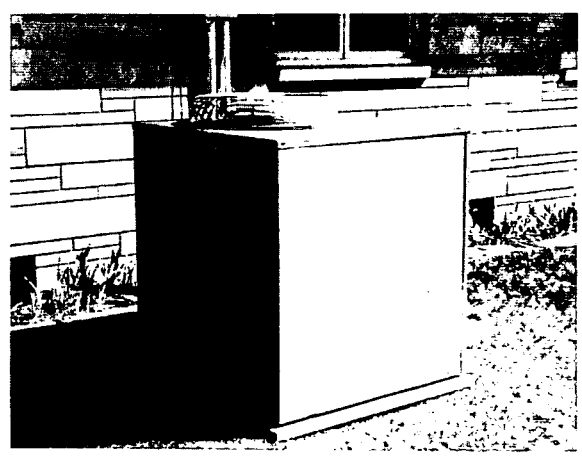

FIGURE 2. - MOCK-UP ABSORPTION HEAT PUMP

The overall system performance has been continually improved, and now in breadboard configuration meets prototype target performance in the heating mode. Table 1. summarizes the target performance for the prototype and early commercial systems (C.O.P.H. 1.25) and for future systems with continued development (C.O.P.H. 1.50).

- Solution Pump Development. One of the major problems in absorption system technology has been the development of a reliable solution pump to circulate the rich solution from the absorber to the generator. The pump under development in this program is of a regenerative turbine design, magnetically driven, of aluminum construction. A parallel program for the development of a vane type pump was stopped in its early stages because of the excellent performance shown by the turbine unit.

Table 2. shows the specifications and present performance characteristics of the regenerative turbine pump.

TABLE 1.

SYSTEM TARGETS AND PERFORMANCE

| | Prototype | Advanced Systems | Actual |
|---|---|---|---|
| Heat Input, BTU/HR | 72,000 | 60,000 | 72,140 |
| Evaporation Temp., °F | 36 | 36 | 36.2 |
| Heat Output, BTU/HR | 90,000 | 90,000 | 89,550 |
| Absorber Cooling Water, °F | 95 | 95 | 95.7 |
| Water to Indoor Coil, °F | 130 | 130 | 127.1 |
| C.O.P. Heating | 1.25 | 1.50 | 1.24 |

TABLE 2.

SOLUTION PUMP SPECIFICATIONS AND PERFORMANCE

| | Specifications | Performance |
|---|---|---|
| Flow Rate, GPM | 3.5 | 3.5 |
| Head, PSI P | 50.0 | 50.0 |
| Operating Lifetime, Hrs. | 50,000 | 33,000 |
| On/Off Cycles | 200,000 | 230,000 |
| Wattage (overall) | 400 | 390 |

The present tests have exceeded 33,000 hours of continuous duty and 230,000 on-off cycles. After the initial run in, no further wear has been seen. It is believed that with further detailed refinement, overall pump wattage to the motor can be reduced to the 350 watt level.

• Field Trial Tests. As part of the development program, two homes in the St. Joseph, Michigan, area were heated by test units of an earlier design for the 1980-81 heating season. The purpose of these tests was to establish feasibility, to identify problem areas not seen in laboratory testing and to evaluate real world cycling and defrost problems.

Unit #1 - This unit has been in operation since December, 1980, and has performed without service requirements, except for one malfunction in the defrost sensing switch. The unit has provided the complete heating requirement of the 1,850 ft.$^2$ home in ambient temperatures down to -10°F, as no supplemental heat was available or required.

One aspect of the unit performance which was unexpected was the almost total lack of defrost requirements. One case was recorded. This lack of defrost requirement appears to be due to the large outdoor coil surface giving minimal air temperature reduction over the coil. As presently installed, the unit will also provide complete summer cooling for the residence in St. Joseph. Figure 3. shows the installation of this unit.

Unit #2 - This unit was put in service in early January, 1981, and provides heating/cooling for a 1,400 ft.$^2$ home. Some initial difficulties with the unit operation were identified as a probable internal leak in the absorber. The unit was replaced and has now performed properly without further service. Again, in this installation, the lack of defrost cycling was seen. Despite an oversizing of these systems, preliminary results indicate cycling losses of only 10-12% from steady state values.

Unit #3 - This system was put into continuous operation using dummy loads in order to gain maximum hours of operation. Over 6,000 hours of operation have been achieved with only minor service requirements.

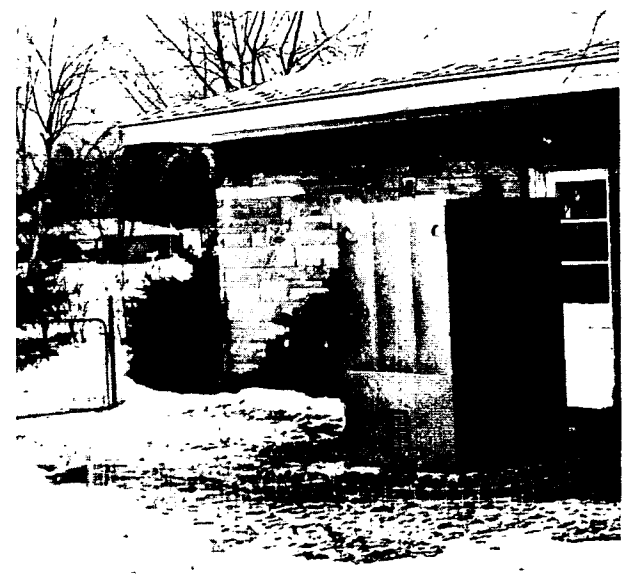

FIGURE 3. INSTALLED TRIAL UNIT

Units #4-5-6 - Three other test units were fabricated and are now in operation in various tests.

The field trial program has successfully demonstrated the actual operation of the system and shows indication of performance benefits in defrost operation.

• Market Evaluation. The most significant conclusion of the market study for gas-fired heat pumps is that these units must be sold through the existing HVAC distribution system in order to gain any significant market penetration. Estimated sales of a gas-fired heat pump at cost and performance levels of the Allied absorption system are 9,000 to 85,000 units annually, depending on the relative strength of the marketing support and distribution system utilized. Also, sales will be heavily weighted toward the northern climates due to the superior heating performance and relatively modest cooling performance of the system.

Among the other conclusions were that gas utilities could play a useful role in promoting gas-fired heat pumps, especially during their introductory period, but the utilities could not effectively replace the HVAC distribution system.

o System Manufacturing Cost. A detailed component-by-component engineering evaluation of the Allied absorption heat pump was carried out to determine the estimated manufacturing cost of the system. Results of this study indicate that the Allied absorption heat pump, while a premium priced product, is competitive with other comparable HVAC systems. Assuming a low-cost manufacturing facility, the factory cost of the 50,000th unit is estimated to be $1,170. Of this cost, 45-50% represents raw materials, 35-40% are vendor-supplied items, and the remainder is labor.

o Annual Performance Simulation. A study was undertaken to calculate the energy consumption and annual operating cost of the Allied absorption heat pump system in comparison with other alternative heating/cooling systems. Actual weather conditions and fuel costs from seven cities in different areas of the country were used to provide a realistic comparison. Major findings of this study are:

1. The primary annual energy consumption for the absorption heat pump is less than any of the competing systems in four of the seven cities.

2. The primary annual energy consumption of the absorption heat pump is less than that of an electric heat pump in all cities, except one.

3. The annual operating cost of the absorption heat pump is less than all of the competing systems in five of the seven cities.

4. The economic attractiveness of the absorption heat pump is dependent on relative fuel prices, heating to cooling load ratio, and total annual load, implying that market penetration will vary by geographic region.

FUTURE ACTIVITIES

Contract activities. The final production prototype system utilizing the improved component designs and extrusions will be completed and performance tested.

Analyses of the performance of other refrigerants in this system and their effect on hardware configuration will be carried out.

Post-contract activities. Commercialization plans for the absorption heat pump for both the domestic and overseas market are being developed. These would include the manufacture and marketing of this system by a hardware OEM and the manufacture of the working fluids by Allied Corporation. Current timetable for commercialization would be about 1985 for the first production models.

DEVELOPMENT AND DEMONSTRATION OF AN AMMONIA-WATER ABSORPTION HEAT PUMP
ARKLA INDUSTRIES INC.
86X-61603C
RICHARD H. MERRICK
JANUARY 1980 - MAY 1983

## OBJECTIVE

The objective is to develop a residential air source ammonia-water absorption heat pump designed to operate in the heating mode only and deliver its output as hot water.

## CONCEPT

Experience from gas air conditioning has demonstrated that absorption machines in small capacities will surpass alternative mechanical systems in approaching ideal cycle efficiency. Absorption has a further advantage over mechanical systems through insensitivity to refrigerant volume increases at low ambients which yields a flatter capacity vs outdoor ambient curve. The fewer moving parts leads to greater product reliability. It is, therefore, logical to pursue this development as there is every reason to believe that the experience in gas air conditioning will be repeated in the application of the heat actuated refrigeration cycle to heating.

When it comes to considering fluids for an absorption cycle, ammonia-water stands out because corrosion and thermal stability are proved up to 400F and the latent heat of ammonia at 550 Btus per pound is second only to water at 1000 (which is unsuited because of its freezing point). Organics have typically been limited by chemical stability to 300F in the absence of potential oxidizers. As refrigerants, their larger molecular weight places them in the 100 Btu per pound bracket which means there is a large mass circulating in the system which must be heat exchanged and propelled. This suggests that the ammonia-water cycle can achieve the same target of performance at a lower cost.

Utilizing the cycle non-reversing simplifies the design, improves reliability, makes it possible to optimize performance for cold climate operation and minimizes parasitic electric power. Though this product would be designed primarily for areas north of 40° latitude, cooling is still normal in the majority of these households. Integrating compression cooling with absorption heating is simply another example of the familiar hybrid system combination of a gas furnace and electric compression air conditioning.

Our program has been organized to develop, from the beginning, a prototype unit drawing heavily from experience in gas air conditioning to be followed later with an advance prototype that could include some performance and cost saving enhancements. This approach was selected to give hardware at the earliest date that would be reliable enough for field testing and life testing the overall program design concepts.

## SUMMARY

Cycle studies are complete and have indicated some variations from the cooling cycle are appropriate. Component testing in the breadboard system has been completed. We have tested three iterations of the prototype units and have arrived at a design which will be used for field testing during the winter. Figure 1 is a photograph of the third prototype which is designed for outdoor installation.

## TECHNICAL ACCOMPLISHMENTS

o Hydronic specifications were set at 125F supply and 105F return water temperatures. These values were chosen to give a reasonable sizing basis for the indoor heat exchanger. Obtaining these with a 17F outdoor ambient established the design parameters for the cycle.

o Our estimate of the requirements of a well insulated mass market home established that the prototype input should be 50,000 Btuh. We did not want to penalize cost or seasonal COP by oversizing.

o A breadboard system was built and tested. Experience there confirmed a choice of cycle configuration and allowed us to experiment with different component designs and defrost schemes.

o Prototype 1 was built and tested with the components organized into a product configuration which continues to be retained.

o Prototype 2 was built and tested. Here the current levels of COP were first attained. These were 1.25 at 47F ambient and 1.12 at 17F ambient. The defrost system was finalized which involves firing the generator while the evaporator fan is idle with the hot vapor by-passing the condenser and going directly to the evaporator.

o Prototype 3 was built and tested. This mainly introduced design refinements relating to handling the generator flue products and improving cold weather start-up reliability.

o Various payback studies have been made placing this product in the 4 to 7 year category based on natural gas against high efficiency furnaces of the future.

## FUTURE ACTIVITIES

Contract activities. Contract COP targets were 1.3 to 1.35 at 47F and 1.2 to 1.25 at 17F. While we hope for some improvement over current levels, these appear difficult to achieve.

No work has been done on the important subject of seasonal COP. Now that the prototype design is set, laboratory testing will address this subject. Three field test and three life test units will be built to the prototype design. The life test units will be installed on test pads outside the Arkla Engineering Department in Evansville, Indiana.

Sites for the field test units have not yet been selected. Integrating compression cooling into the product is another contract task. It is expected that some economies will derive from fin sharing in the indoor and outdoor coils as well as from the dual use of the outdoor cabinet and fan.

The contract calls for the design and development of an advance prototype and the construction of three field test and three life test units of this design.

Non-contract activities. It is anticipated that ten to twenty-five additional field test units will be put into operation during the coming winter. Construction and the installation of these units will not be charged to the contract. Obviously this program should hasten commercialization and broaden the technical exposure of participating contract personnel.

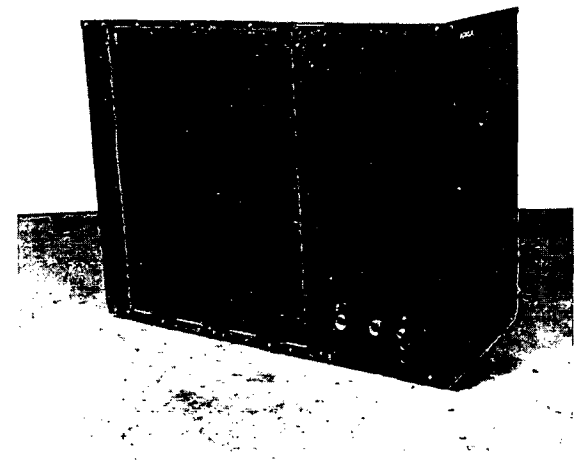

Figure 1. Ammonia-Water Absorption Heat Pump Prototype

# COMPOUND ABSORPTION COOLING FROM LOW-GRADE HEAT USING ADVANCED DESORBERS

BATTELLE, COLUMBUS LABORATORIES
SUBCONTRACT NO. 7805

FROM

UNION CARBIDE CORPORATION

WILLIAM T. HANNA, PH.D., PROGRAM MANAGER
WILLIAM H. WILKINSON, PRINCIPAL INVESTIGATOR

FEBRUARY 1980 TO OCTOBER 1981

## OBJECTIVE

The overall objective of this program is to demonstrate the commercial feasibility of an advanced absorption cooling system know as the Compound Absorption Chiller (CAC) which is motivated by low-grade heat. The program is organized into four phases having the following specific objectives. In the First Phase (Feasibility Study) the phase objectives were to show preliminary indications of technical and economic feasibility in recommended applications and to provide analytical performance estimates for a demonstration unit. In the Second Phase (Engineering Analysis and Design) the specific objectives were to design experimental equipment, and establish a design basis for an advanced desorber to be scaled up for use in the cooling system. The objective in the Third Phase (Experimental Study) was to conduct bench scale experiments on conventional and advanced desorber designs to determine heat transfer coefficients and other design information. In the current, Fourth Phase (Testing) the objective is to construct and test a nominal 25-ton capacity chiller, motivated by low-grade heat, to verify the performance predictions of the analytical model.

## CONCEPT BACKGROUND

In early 1978, W. H. Wilkinson of Battelle developed the CAC technology (which is proprietary to Battelle), to utilize low-grade heat, such as industrial waste heat, to provide chilled water for comfort or process cooling. In October 1979, the DOE approved for funding the demonstration of this novel system utilizing low-grade waste heat at temperatures between 130 F and 150 F to provide refrigeration. Battelle has since requested of DOE the patent rights to foreground technology under this contract, so that all related technology could later be transferred under license as a package to an equipment manufacturer for commercialization.

Absorption refrigeration is a class of thermally motivated systems in which heat energy is supplied to one thermodynamic cycle at a useful temperature so that, in rejecting a portion of that energy to the ambient "sink", work in the form of a solution potential difference is generated. This solution potential difference is the work input to a second thermodynamic cycle that pumps heat from the cold evaporator to the ambient sink. Except for a small direct mechanical input to solution and circulating pumps, all energy is derived from the heat source.

A simple cycle absorption chiller requires motivating energy in the form of steam or hot water at temperatures between 200 F and 300 F. The thermal efficiency or coefficient of performance of such simple cycle machines is about 65 to 70 percent.

Previous advancements in absorption chiller development have focused on increasing efficiency by increasing the thermodynamic availability (temperature) of the heat driving the cycle. Whereas this means more effective use of (higher temperature) heat, such advancements were not responsive to the use of low-grade waste heat. As part of the national interest for energy conservation, utilization of low-grade waste in the range of 130-150 F presents a significant challenge. Very limited direct use of this low-grade heat exists, since for most heating applications, the temperature level is too low. An alternative is to use this low-grade waste heat to drive thermally motivated refrigeration systems. The Compound Absorption Chiller is an absorption machine development that fulfills this objective of providing chilling from low-grade (130 - 150 F) heat. Although novel, advanced, and proprietary, the Compound Absorption Chiller development builds on a significant amount of commercial equipment.

In this program an existing commercial absorption chiller is being modified to operate on the advanced CAC cycle conceived at Battelle. The primary modifications involve the addition of a pair of heat/mass transfer surfaces, associated solution pumps and a heat exchanger to commercial absorption equipment. The resulting CAC:

- Will provide chilled water at conventional temperatures (40 - 45 F),
- Will operate using waste heat supplied at 140 F and leaving the simplest modified system at about 130 F,
- Requires no critical experiments for successful recovery of this waste heat, and
- Has growth potential for improved energy recovery through novel counterflow desorber developments.

The amount of energy recovery from the waste heat stream depends upon the desorber heat exchanger configuration. Battelle has developed, built and tested an advanced desorber design as an alternative to the standard type of heat exchanger configuration available in commercial units. This program includes an analytical and experimental development of an unique counterflow desorber which shows promise of futher enchancing the attractiveness of the absorption machine.

The Compound Absorption Chiller is an advance, based upon the extension of conventional, simple cycle absorption chillers to operate on low-grade waste heat. The CAC is a compound system consisting of the components of a simple cycle chiller integrated with an addition pressure vessel (or drum), recuperative solution heat exchanger and solution pumps. The key addition is the new drum which operates at a pressure level between those of the usual evaporator and condenser. The heat exchangers in this "intermediate

drum" provide for additional low-grade heat input to the CAC allowing the entire system to operate on low-grade heat. By staging the heat inputs to the intermediate drum and the original generator, not only can low-grade heat be recovered and utilized effectively, but the system can operate with good thermal performance.

The thermal performance of the CAC can be illustrated considering that for every Btu of waste heat added to the intermediate drum, 0.9 Btu of waste heat is added to the generator and 0.9 Btu of cooling effect can be accepted at the evaporator. This represents a COP of about 41 percent. The complexity is, of course, increased by a factor of about 50 percent as an extra drum and solution pumps are added.

To put this COP in perspective, idealized Carnot Cycles for the CAC and simple cycles are shown on Fig. 1 and 2 respectively. The CAC performance for a real fluid is about 59 percent of the ideal (Carnot) cycle. In contrast, simple cycle performance for a real fluid is about 57 percent of the associated ideal cycle. The CAC shows, therefore, an excellent approach to ideal thermodynamic limits.

$$C.O.P. = \frac{(141-95)/(141+460)}{(95-40)/(40+460)} = 0.70$$

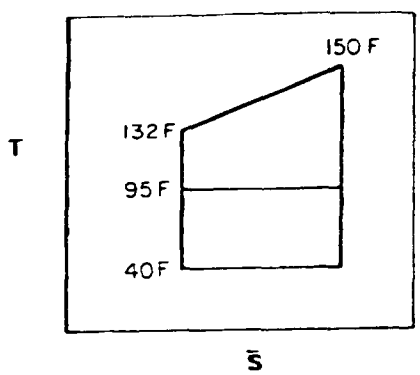


FIGURE 1. IDEAL THERMODYNAMIC CYCLE FOR COMPOUND ABSORPTION CYCLE

$$C.O.P. = \frac{(180-95)/(180+460)}{(95-40)/(40+460)} = 1.35$$

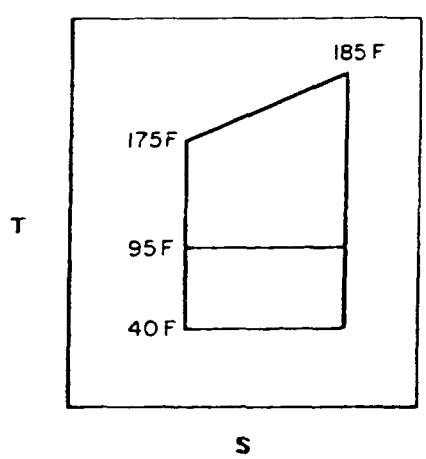


FIGURE 2. IDEAL THERMODYNAMIC CYCLE FOR SIMPLE ABSORPTION CYCLE

## SUMMARY

This project is organized into four Phases and the technical accomplishments are discussed for each phase. In the first (Feasibility) phase, the major results were the development of a detailed computer performance model for the CAC system and the use of that model, with absorption equipment cost data, to predict the relative cost and benefits of using CAC systems. The results of the second (Engineering Analysis) phase were: the definition of three desorber concepts and an experimental plan for ranking them, and the design of a heat exchanger using the best of these concepts for incorporation into the later test unit. In the third (Experimental) phase, the desorber tests, planned in Phase II, were carried out for one pool type and two advanced desorbers; with the result that one of the advanced desorbers, a counterflow unit, gave improved heat transfer (by 1400%) and improved cost (by 140%) as compared to a pool desorber. In the current Phase IV Testing effort, the CAC demonstration unit has undergone detailed design and construction is nearing completion on a 25 to 50 TR waste heat driven, 140 F input chiller. Testing is schedule for 2 months beginning in June. Post-contract activities include plans for commercialization via a licensed manufacturer and operation of the CAC demonstration unit in an industrial type application.

## TECHNICAL ACCOMPLISHMENTS

The technical accomplishments will be summarized for each of the four phases of the project on a task basis. In the Phase I Feasibility Study the three tasks were:

1.1 Analytical Model Development,
1.2 Applications Analysis, and
1.3 Definition and Ordering of Commercial Equipment.

- In Task 1.1 a flexible computer model of the CAC system was developed from relationships previously defined by Battelle to allow definition of design conditions for application cases. This model allows not only comparison of appropriate system options, but also evaluation of the impact of heat exchanger size on waste heat recovery and cooling capacity.

The computer model was developed in two stages of refinement. The first model anticipated a narrow range of CAC operation where average temperature differences were appropriate for determining the basic performance comparisons. These results have been the primary guide to understanding system sensitivities and application matches. The second system model resulted from insight developed in the preliminary economic/application evaluations.

Because heat exchanger surface area can be correlated as a major factor in determining unit cost, a primary design parameter for thermodynamic system analysis is the temperature difference for heat flow across each of the exchangers. It was thus appropriate in the second model to use realistic heat exchanger log-mean temperature differences (l.m.t.d) to properly reflect system sensitivities. To satisfy application evaluation needs, the computer program was also modified to force its output to match a given design point objective. This objective can be specified as an evaporator temperature, for more general cases, or as a set of chilled water temperatures (inlet and outlet) for an evaporator operating at the same l.m.t.d as the other system heat exchangers.

• In the Application Analysis Task, reported waste heat and cooling water temperatures were used to characterize the design specifications for applications typical of both government and industrial situations. The four Gaseous Diffusion Plants (GDP) (Oak Ridge, Savannah River, Portsmouth, and Paducah) were used to represent government application cases. Results from a general survey of industries were used to define an application case representative of a regime different from the GDP cases. Significant additional industrial use for CAC systems would match the GDP conditions.

Cost information on available (1980) commercial chillers was analyzed to provide extrapolations to the design options of the CAC. The results of the economical evaluations compare the incremental capital cost increase of using a CAC to replace a conventional chiller and save the fuel cost associated with the conventional chiller. The results of this comparison are shown in Fig. 3, for various design values of CAC l.m.t.d.($\Delta T$ = 6,7,9,11, or 13) and various cooling capacities from 100 TR to 2000 TR.

The economic analysis showed that even the most unfavorable of the application cases considered showed a simple pay-back period of about 2.5 years of comfort conditioning operation. Most cases showed shorter pay-back times. The Savannah River case is quite favorable because the cool river water available for heat rejection and its pay-back time is expected to be about 8 months. Installation size is also shown to be significant based on current commercial absorption equipment. The pay-back time for a 100-Ton chiller could be more than twice as long as for a 2,000-Ton chiller. This range is representative of current practice.

The economic picture shows a high likeihood of commercial success with a significant impact on the conservation of our fuel resources. In addition to the direct financial benefit to the user, indirect benefits exist through the increased productivity that will result from maintaining comfort conditioning standards we have become accustomed to without expending our scarce fuel resources.

From Fig. 3 it is reasonable to generalize that a CAC requring l.m.t.d $\leq$ 6.5 F would require justifications other than simple economics, i.e., greater than 2-year payback. Using that as a criterion, Fig. 4 shows how waste heat entering temperature and cooling water entering temperature define regions of "successful" and "difficult to use" applications. The four GDP cases are plotted and are well into the "successful" region. Extrapolations to more conservative cooling water temperatures are shown dotted and the potential for their use is attractive. Other potential application cases can be located on Fig. 4 to provide a preliminary indication of their economic attractiveness.

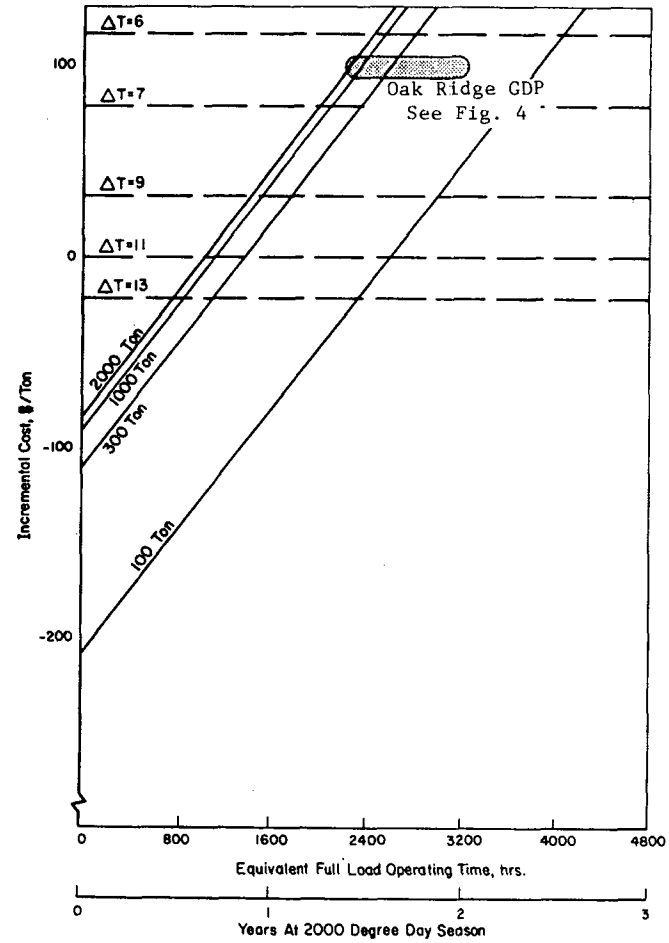


FIGURE 3. PAY-BACK CHARACTERISTICS OF COMPOUND CHILLER

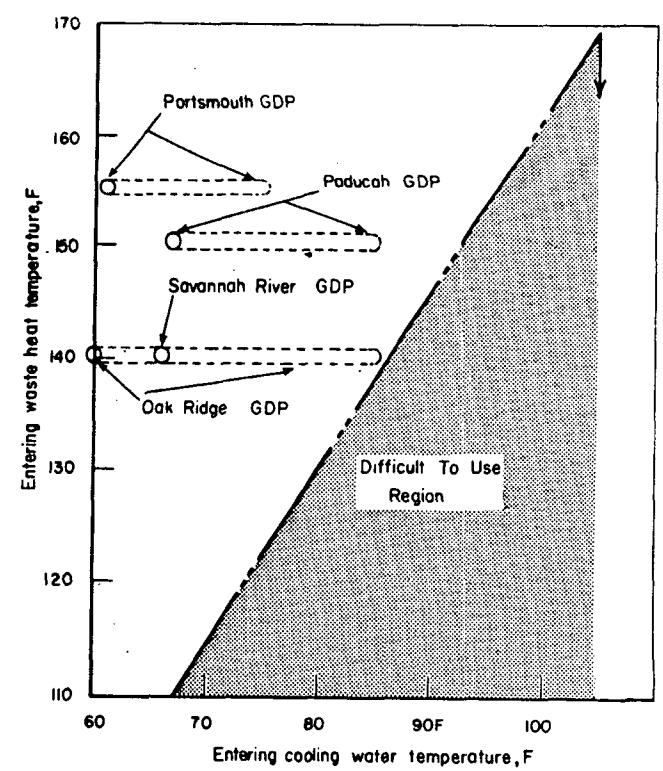


FIGURE 4. APPLICATIONS OVERVIEW FOR CAC

• In Task 1.3 the specifications were prepared for a commercial absorption chiller suitable for incorporation into the CAC demonstration unit. The specifications for the 100-ton capacity machine, a size deemed necessary for carrying out realistic verification studies, included requirements on water temperatures, water flow rates and pressure drops, etc.

The specifications were sent to three manufacturers of absorption chillers together with a request for bids. Although one unit was not suitable for modification, either of the other two units could be used. The lowest cost unit was selected as the most cost-effective unit to modify. Delivery of this machine was 6 months after receipt of order, thus meeting the program schedule requirements of delivery in November 1980.

The Phase II (Engineering Analysis and Design Phase) of this project was divided into the following four tasks:

2.1 Performance Characterization,
2.2 Design of Phase III (Desorber) Experiments,
2.3 Design of Intermediate Drum, and
2.4 Safety Review.

• Task 2.1 Performance Characterization resulted in the specification of the testing regimes for the Phase III desorber experiments and development of the design objectives for the Phase IV CAC demonstration unit. It was shown that a conventional commercial chiller should be operated de-rated by at least 50 percent when matched with a component for utilizing low-grade waste heat.

The computer model developed in Phase I was used to define the operating conditions for a CAC system. A realistic waste heat source was water at 140 F, and desirable recovery of waste heat would cool the stream to 120 F. The cooling water was assumed available at 70 F and would increase in temperature 10 F as it accepted the heat rejected from the CAC. The chilled water was assumed to return to the CAC at 55 F and be chilled to 40 F.

• In Task 2.2, Design of Phase III Desorber Experiments, the objective was to define alternative advanced desorber designs for the comparison testing of Phase III, and to define the experiments required for ranking these desorber types.

The designs defined were; a pool desorber (for a comparison baseline), a multiple pool design which approached a counter-flow configuration, and a uniflow (counter-flow) design.

Preliminary predictions showed the uniflow design as functionally far superior to the other alternatives although it could suffer from length limitations because of the required vertical tube configurations. Tube lengths on the order of 12 to 16 feet were anticipated.

• Task 2.3, Design of the Intermediate Drum, was carried out after the desorber testing of Phase III, and utilized the results of that testing.

The inventor correlated the experimental results and developed a rational procedure for applying the correlation results to investigate desorber design parameters such as length, temperature difference, and pipe size.

The results of correlating the test results for the uniflow design using the entering solution equilibrium temperature plotted against solution flow is

$$h_{ex} = 2428\ (\dot{w}_s)^{0.4} \tag{1}$$

where $h_{ex}$ = external film transfer coefficient in Btu/(hr-ft$^2$F), and

$\dot{w}_s$ = the solution flow in gallons/min.

This correlation provides a good fit to the experimental data. A lower correlation, where the correlation coefficient is reduced to 90% has been adopted as a conservative correlation for design specification.

The minimum tested flow rate case has a capacity of 0.73 tons of refrigeration (TR) for each tube made from 1/4-inch pipe with an active length of 12.4 feet; this case closely approaches the CAC design objective. Extrapolation of these correlations to various tube sizes can be made by adjusting the correlation so that

$$h_{ex} = k\ \left(\frac{\dot{w}_s}{d_i}\right)^{0.4} \tag{2}$$

where $d_i$ is the inside diameter of the tube. Based on this projection, this effect of changing pipe size is illustrated in Fig. 5 for designs based on the minimum tested solution flow (0.2 gpm for 1/4-inch pipe). Standard pipe sizes are indicated on Fig. 5 along with the associated cooling capacities.

• In Task 2.4, Safety Review, the objective was to review the potential impact of the modification (to a CAC) of the safety of the standard chiller. No significant factors were identified, although a tall vertical drum was considered to require different installation and structural foundation support techniques from convetional chillers.

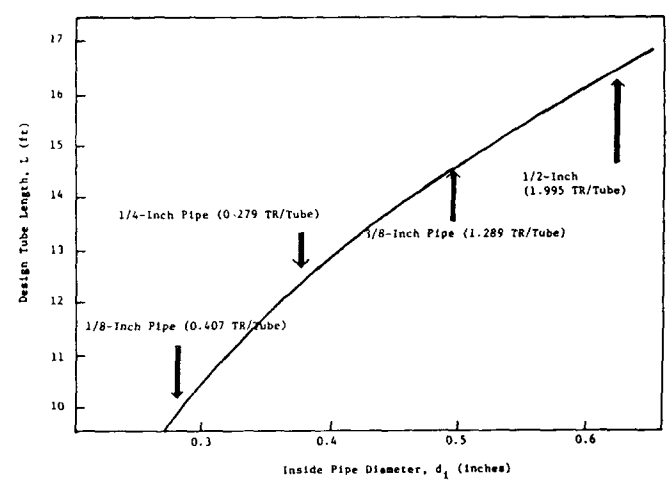


FIGURE 5. EXTRAPOLATION OF MINIMUM SOLUTION FILM THICKNESS DESIGNS TO OTHER PIPE SIZES

In Phase III, the Experimental Study, the overall objective was to rank the desorber designs on the basis of experimental heat-transfer performance and characterize the heat transfer for the "best" desorber design. This Phase includes three tasks:

3.1 Equipment Construction,
3.2 Conduct of Experiments, and
3.3 Analysis of Results.

- The major efforts were in Tasks 3.1 and 3.2. Three types of desorbers were built, installed in the test facility and tested. These desorbers were a pool type, used as the baseline for comparisons, a multi-pool type, and a uniflow type. The desorbers were tested over a range of conditions of varying solution flow rates, waste heat temperatures and heat exchanger temperature differences. Of primary concern was the external heat transfer coefficient, which reached the following maximum values for each desorber type:

pool - 101 Btu/hr/ft$^2$/F,

multi-pool - 598 Btu/hr/fr$^2$/F, and

uniflow - 1445 Btu/hr/ft$^2$/F.

Thus, the uniflow has a 14:1 advantage in performance over the baseline pool desorber. The cost of the uniflow tubing is ten times that of the smooth tubing used in the pool on a cost/unit area basis. Hence it appears that the uniflow desorber design offers an overall material cost advantage of 1.4:1 over a pool desorber.

The facility was designed to provide 0.4 gallons per minute of solution and about 12,000 Btu per hour of heat to the desorber section at a l.m.t.d. between the heating water and the solution of 10 F and a heating water temperature drop of 10 F. These operating conditions were selected based on analytical predictions of the performance for the Phase IV demonstration unit.

The apparatus consisted of the desorber and condenser sections, supply and receiver tanks, sources of heating and condensing water, and the instrumentation needed for control and measurement of desorber performance. The apparatus was designed so that the three desorber heat exchangers were readily interchangeable. The LiBr solution used in the facility was stored in the steam heated solution supply tank. Flow from the solution supply tank into the desorber section was driven by pressurizing the supply tank relative to the rest of the system. Heating of the solution in the desorber was done by circulating water in a closed loop in which the water was heated by a steam coil. The flow rates were measured using variable area flow meters. Within the desorber section boiling of the incoming saturated solution occurred, generating vapor which was subsequently condensed in the condenser section. The condenser was a multiple pass tube bundle through which the utility water, at about 78 F, was passed. The condensate that formed on the tube surface was collected in the bottom of the condenser section where it flowed to the drain. The concentrated solution leaving the desorber heat exchange surface flowed into the bottom of the desorber, through the drain, and into its reservior. A sample port was included in the drain line for use in determining the concentration of LiBr in the solution.

A strain gage absolute pressure transducer and thermocouples were used to measure the test data which were then used to quantify heat exchanger performance. Test data were recorded on mini-floppy disks using a programmable data acquisition-analysis system. With steady-state operation achieved, a series of sets of test data were recorded on a floppy disk at approximately 5 minute intervals.

The nominal operating conditions determined from analytical predictions of the performance of the Phase IV demonstration unit was as follows:

- Solution inlet flow rate 0.4 gal/min
- Solution inlet concentration 50% LiBr
- Solution inlet (saturation) temperature 120 F
- Heating water inlet temperature 135 F
- Heating water temperature decrease 10 F
- Log-mean temperature difference- heating water to solution 10 F
- Solution concentration increase 2.5% LiBr
- Condensate flow rate 0.016 gal/min

To quantify the performance of the three heat exchangers, the experimental runs were carried out using a range of operating conditions that were centered on the above nominal conditions.

- In Task 3.3, for each test run, the log-mean temperature difference (l.m.t.d.) between the heating water and solution was calculated from measured temperatures. The heat transfer rate ($\dot{Q}$) was calculated from the measured heating water flow rate and temperature difference. Using these values, an overall heat transfer coefficient (U), based on the internal heat transfer tube area, was then calculated where the definition of U was based on the internal tube area (A). This was done because the external heat transfer areas of the multi-pool and uniflow desorber configurations were difficult if not impossible to measure.

The external heat transfer coefficient ($h_e$) was calculated using the previously calculated value of U and the measured heating water flow rate, neglecting the tube wall flow resistance.

For comparison purposes, the external heat transfer coefficient for a pool desorber in a commercial absorption chiller was calculated. The value for the commercial chiller was 325 Btu/hr/ft$^2$ which lies between the values that were measured for the baseline pool desorber and the multi-pool desorber. The value for the commercial pool is that obtained with a temperature difference between the heating water and the pool of 45 to 50 F rather than the 10 F,so that was used in the laboratory studies and will exist in the Phase IV CAC demonstration unit.

The performance data developed in these experimental studies were used to determine the best heat exchanger configuration, to determine the effect of the configuration on overall system performance, and to guide the design of the intermediate drum in Phase II, Task 2.3.

The Construction and Testing Phase, Phase IV is the current activity. This phase is organized into the following six tasks:

4.1 Detailed Design,
4.2 Facility Construction,
4.3 Intermediate Drum Construction,
4.4 Startup and Shakedown,
45. Testing, and
4.6 Analysis of Results.

- The current status is that Tasks 4.2 through 4.4 are active, Task 4.1 having been completed. The Detailed Design Task provided specific information for facility layout, component sizing and ordering and construction plans. The Facility Construction Task included efforts on installing and modifying the basic chiller, installing cooling water piping and controls, waste heat piping and controls and instrumentation. This task is nearly completed, requiring only the delivery and installation of commercial water heaters for completion. The task of Intermediate Drum construction is essentially completed and some Tasks 4.4 items, such as leak checking have begun on the Intermediate Drum.

- The Intermediate Drum was designed and built at Battelle using commercially available tubing and custom designed and built headers and shell. The design of this drum will allow CAC system capacity of up to 50 TR under optimal conditions and a nominal capacity of 25 TR. The drum is extremely compact, containing two heat exchangers in a vertical arrangement with a shell o.d. of about 24 in. and a length (o.a.) of 17 ft.

FUTURE ACTIVITIES

Contract Activities. The future activities on this contract are the completion and check-out of the demonstration unit, its verification testing and analysis of those results in Phase IV. We anticipate the testing will begin in mid-June, or about 3 weeks later than our original schedule of February 1980.

The testing of the CAC demonstration unit includes a prioritized matrix of approximately 50 test conditions designed to verify the operation of both the system and the advanced desorbers in the intermediate drum. The results of these tests will be used to verify and improve the analytical model (of Phase I), so that the model can be used with greater confidence in performance predictions. The Results Analysis Task will also include an update of the Phase I activity on cost-benefit analyses for both government (GDP) and industrial applications of the CAC.

In most general terms, this program will bring the Compound Absorption Cycle technology to a state of demonstrated readiness for commercial applications to waste heat recovery. A sufficient range of predicted performance will exist so that applications to most real installations will be straightforward. Further, the computation model will be available to predict the performance of any unusual cases. The operating equipment will provide a correlation base for the operation of first-generation equipment. The experimental characterization of the counterflow desorber adds an empirical base for future growth toward optimum equipment.

Post-Contract Activities. There are two major and complementary activities for continued development of the CAC system. The most important of these is to develop a manufacturing licensee for commercializing the CAC. Primarily, this will involve searching for a well-suited U.S. based manufacturing company, probably one familiar with both absorption refrigeration machinery and also industrial waste heat recovery. Preliminary market studies indicate that a substantial market for CAC exists.

The second activity involves the demonstration operation of the Phase IV CAC unit, or its successor, at a DOE facility such as the EAST facility. Such an operational demonstration could be supported by joint funding involving commercial, industrial, trade association and/or federal interest and organization. An operational demonstration would provide operating experience in an industrial-type environment open to public scrutiny. Thus, potential users of CAC systems could avoid the nominal risks of being the "first" user for the system and commercialization would thereby be facilitated.

PUBLICATIONS/REPORTS/REFERENCES

The following reports have been prepared for ORNL under this project:

| | |
|---|---|
| Phase I Final Report | April 29, 1980 |
| Phase II Final Report | October 30, 1980 |
| Phase III Final Report | September 30, 1980 |

These reports are available on a limited basis through ORNL.

ACKNOWLEDGEMENTS

I would like to acknowledge the outstanding cooperative efforts of the project staff:

William H. Wilkinson, the inventor of the system and its crucial components; Sudheer M. Pimputkar, for the efficient computer model of the system, Lawrence J. Flanigan and Keith E. Herold, for the design and development of the laboratory and demonstration equipment, and our technicans, P. R. Fritter and J. E. Stets, for their often insightful suggestions.

I would also like to express appreciation for the concern and interest of Horacio Perez, Gershon Grossman and Harry Arnold from Oak Ridge National Laboratory. Largely through their efforts at ORNL, this program has become an excellent example of a cooperative effort to bring needed, advanced, energy conserving technology to commercial and industrial users.

CHEMICAL HEAT PUMP COST EFFECTIVENESS EVALUATION
TRW ENERGY ENGINEERING DIVISION
BNL CONTRACT NUMBER 519403-5
PRINCIPAL INVESTIGATOR: WARREN R. STANDLEY
PERIOD OF PROJECT: Begin June 1980 End May 1981

OBJECTIVE

The objective of this project is to compare the cost-effectiveness and energy-effectiveness of existing chemical heat pump (CHP) concepts with a baseline of conventional energy technologies and a group of near-term emerging energy technologies with which CHPs are expected to compete. The analysis is structured to evaluate these systems functioning as the primary space conditioning unit of both a "standard" single-family detached home and a "representative" commercial building. Each HVAC system and application is analyzed in each of two locations in the United States, the southwest (Albuquerque, NM) and the northeast (Boston, MA). In addition, the CHPs are evaluated in a "representative" industrial waste heat upgrading application, and compared to potentially-competitive technologies for industrial "heat pumping."

BACKGROUND

A chemical heat pump is a device that extracts thermal energy from a low-temperature source and rejects it to a higher-temperature sink. Unlike vapor-compression heat pumps which use the vapor-liquid phase characteristics of a single working fluid, chemical heat pumps make use of reversible physiochemical reactions between a working fluid (vapor) and one or more stationary phases (solid or liquid). Chemical heat pumps are being developed by DOE because they offer the promise of lower overall cost and greater efficiency than separate units for heating, cooling, and (in solar operations) storage. They are also being considered for industrial process heat upgrading applications.

Chemical heat pumps must compete with existing HVAC and industrial process-heat technologies; including gas and oil-fired furnaces, vapor-compression heat pumps, and vapor-compression air conditioners. In addition, they will compete with emerging technologies such as solar-thermal heating, solar-driven absorption air conditioning, and ground or aquifer-coupled vapor-compression heat pumps. An applications-specific comparison between the chemical heat pump systems, the baseline systems, and the emerging technologies will enable the promised benefits of the CHP's to be weighed against the cost of ownership and operation.

Methodology. A characterization of building type and pattern of use, as well as detailed location-specific climatic data, were used to determine the annual energy requirements for each HVAC application and location. The energy requirement in the industrial application was assumed to be one-million Btuh for each of two end-uses, process steam and process hot water. The annual fuel cost, system cost, and maintenance cost were determined based on the technical definition of a specific energy system in a given application. Levelized annual cost, based on constant-dollar calculations, was selected as the economic criterion for making cost-effectiveness comparisons between CHPs and competing systems. For the purpose of this study, the "constant-dollar" interest rates were taken to be 2 percent for homeowners and 6 percent for businesses; with annual fuel price escalation rates of 0.6 percent for electricity, 2.6 percent for distillate fuel, and 3.2 percent for natural gas.

Residential Space-Conditioning. The "standard" single-family detached home is assumed to be a single-story structure with 1500 square feet of floor space. The building meets the energy conservation standards of ASHRAE 90-75, and is considered to be typical of the well-insulated dwellings which will characterize the future new home market which CHPs might penetrate. The baseline space-conditioning systems for residential applications include:

- Gas-fired furnace with electric air conditioning
- Oil-fired furnace with electric air conditioning
- Electric heat pump with electric-resistance backup heating

The near-term, emerging technologies for residential HVAC include:

- Solar heating with electrically-driven air conditioning (gas, oil, or electric backup for heating)
- Solar-assisted heat pump with electric backup and electrically-driven air conditioning
- Ground-coupled heat pump
- Solar heating with gas backup and solar-absorption air conditioning

Commercial Space-Conditioning. The "representative" commercial building is assumed to be a three-story structure with 40,320 square feet of floor space. This building also meets the energy conservation standards of ASHRAE 90-75, with a well-designed building envelope to minimize solar loads and regenerative heat exchange to reduce ventilation-associated loads. The space-conditioning requirements of the commercial building were assumed to vary according to the pattern of use. The baseline space conditioning systems for the commercial application include:

- Gas-fired boiler with electrically-driven air conditioning
- Electric boiler with electrically-driven air conditioning

The near-term, emerging technologies for commercial HVAC include:

- Solar heating with electrically-driven air conditioning (gas backup or electric backup for heating)
- Solar-assisted heat pump with electric backup and electrically-driven air conditioning
- Ground-coupled heat pump
- Solar heating with gas backup and solar-absorption air conditioning

Industrial Waste-Heat Upgrading. Two industrial waste-heat upgrading applications were established as the basis for comparing CHPs to other industrial heat pumping systems. The primary application involves producing one-million Btu-hr of 300°F, 67 psia process steam, given a supply of 220°F waste water and 50°F cooling water. A secondary application involves a one-million Btu/hr demand for reheating a process-water stream from 240°F to 260°F, given a supply of 210°F waste water and 50°F cooling water.

The analysis of chemical heat pumps in applications for upgrading industrial waste heat to process heat requires comparison to some alternative energy system. The natural comparison is to a fired boiler of the same thermal rating or a competitive industrial heat pump system. For the upgrading of waste heat to process conditions, there are two industrial heat pump technologies which are feasible with state-of-the-art equipment: (1) the Westinghouse Templifier, and electrically-driven Rankine cycle heat pump with Freon working fluid, and (2) a steam vapor-compression system. For the purpose of this study, industrial CHPs are compared to four alternative systems, including:

- Fired steam boiler
- Westinghouse Templifier heat pump
- Electrically-driven vapor recompression
- Steam-driven vapor recompression

CHP Systems Evaluated. The Sulfuric acid CHP being developed by Rocket Research Company and the methanolated-salt CHP being developed by EIC Laboratories were compared to the baseline and emerging technologies described above. A hydrogen-metal hydride CHP which is under development by Southern California Gas Company was not evaluated because manufacturer-supplied performance and cost data are not available at this time.

SUMMARY

In residential HVAC applications, the role of either CHP concept in energy conservation is best served through solar-driven applications, the economics of which are somewhat better than baseline and emerging systems. Direct-fired CHPs are less expensive to own and operate, but are not a good approach to energy conservation. Given the increased complexity of the sulfuric acid system and the barriers to commercialization (safety, institutional, and customer-psychological), the methanolated salt system would probably be the most attractive CHP technology for residential use. Because of the dominance of the air conditioning load, use of either CHP for heating and cooling a commercial office building would be very expensive, competing only with the most expensive emerging systems. Since this use of CHPs is also not very attractive from an energy standpoint, there appears to be little incentive for CHPs in HVAC applications in commercial buildings.

The industrial CHPs were found to be more cost-effective to own and operate than an electrically-driven vapor recompression system because of the high energy cost for electricity. CHPs were more cost-effective than the self-driven vapor recompression system because of the high capital cost of the power-recovery turbine capable of exhausting to very low pressure. The only commercially-available industrial heat pump, the Westinghouse Templifier, is constrained to temperature levels below 230°F, and is not considered to be a serious competitor for process heat applications in the temperature range of interest to BNL.

TECHNICAL ACCOMPLISHMENTS

- In HVAC applications, not all CHP systems are energy-conservative when compared to the baseline (existing) technologies. The value of CHPs in energy resource conservation is both application and location specific. Each solar-driven CHP system consumes fewer energy resources than the competing baseline and emerging systems. (In this context, energy resources reflect the primary resource consumed to produce and deliver the energy requirement of the end-use system.) The residential application of a gas-fired CHP in a Northeast location (Boston), is more energy-conservative than the baseline systems, but not the emerging systems. The dominance of the air conditioning load for a commercial building in both locations and for the residence in the Southwest (Albuquerque) makes the gas-fired CHP systems much less energy efficient than the emerging technologies and somewhat less energy-efficient than baseline systems. The annual energy-resource consumption of both CHPs and competing systems is illustrated by Figure 1 for residential systems and Figure 2 for commercial systems.

- The TRW estimate of the cost-effectiveness of CHP systems in HVAC applications, using constant-dollar annual levelized cost as the criterion, shows that both the methanolated-salt and sulfuric acid systems can be competitive with baseline and emerging technologies in specific applications and locations. In residential applications, the gas-fired methanolated-salt system (EIC) is comparable to baseline technologies and is clearly more cost-effective than emerging systems, independent of location. The residential solar-driven methanolated-salt CHP is cost-competitive in the Southwest (Albuquerque) but not in the Northeast (Boston). In general, the solar-driven commercial CHP systems are more expensive to own and operate than all baseline systems and most emerging systems. The gas-fired commercial CHP systems are less expensive than the solar-driven CHP systems, but remain more expensive than the baseline and emerging technologies. The levelized annual cost of owning and operating both CHPs and competing systems is illustrated by Figure 3 for residential systems and Figure 4 for commercial systems.

- Both CHP concepts (methanolated salt and sulfuric acid) were evaluated based on performance data supplied by their respective developers. Each is technically able to operate in the industrial process-heat temperature range and there is no severe technical limitation which would preclude the use of either CHP for industrial waste heat recovery and upgrading. Both the sulfuric acid and methanolated salt systems would consume about the same amount of ancillary energy (electricity) during operation. The higher capital cost of the sulfuric acid system makes it apparently more expensive to own and operate, relative to the methanolated salt system. The capital cost of the methanol system is less well-developed than that for the sulfuric acid system. It is quite possible that a more detailed evaluation of the EIC system, equivalent to that done by Rocket Research for the sulfuric acid system, would indicate a higher real capital cost and would eliminate the apparent cost advantage of the methanolated salt system. The EIC system is being developed for space conditioning applications and has never been specifically configured for waste-heat upgrading. It has the disadvantage of a very large system envelope in the industrial applications, with 360 cubic feet of calcium chloride salt bed required for a one-million Btu/hr process

water reheating (from 240°F to 260°F) case, and over 1400 cubic feet required for a one-million Btu/hr process steam (300°F, 67 psia) case. The methanolated salt system is evaluated for the process reheating case, but is considered impractical at its current stage of development for the process steam application.

- Use of the sulfuric acid CHP for industrial waste heat recovery from a 220°F waste stream to produce 300°F, 67 psia process steam could be competitive with both baseline and emerging systems. Compared to a baseline system consisting of steam raised directly in a fuel-fired boiler, the CHP is both cost-effective and conservative of energy resources. The industrial CHPs were also found to be much more energy conservative than an electrically-driven vapor recompression system. A vapor-driven vapor-recompression system was the most energy-conservative system evaluated, although the practical application of a steam-driven turbine exhausting to very low pressure could reduce the energy efficiency of the system. The concept of vapor recompression is used commercially to conserve energy in evaporation and distillation operations. The concept has been specifically adapted to the industrial waste heat recovery application and, although there is no known commercial system in such an application, the engineering basis is considered valid. The annual consumption of energy resources and levelized annual cost of owning and operating the sulfuric acid CHP and its competitors in the process steam application are illustrated in Figure 5.

- The use of either CHP for industrial waste heat recovery from a 210°F waste stream to reheat a process steam from 240°F to 260°F could be competitive with both baseline and emerging systems. Compared to a baseline system consisting of steam raised directly in a fuel-fired boiler, the CHP are both cost-effective and conservative of energy resources. The industrial CHPs were also found to be much more energy conservative than an electrically-driven vapor recompression system. A vapor-driven vapor-recompression system was the most energy-conservative system evaluated, although the practical application of a steam-driven turbine exhausting to very low pressure could reduce the energy efficiency of the system. The annual consumption of energy resources and levelized annual cost of owning and operating both industrial CHP's and their competitors in the process water reheating application are illustrated in Figure 6.

PUBLICATIONS/REPORTS

The final technical report for this project is expected to be completed on or about 20 May 1981. It will be available through NTIS.

REFERENCES

1. R. Gorman, P. Moritz, and W. Standley, "Chemical Heat Pump Cost Effectiveness Evaluation Draft Final Report." TRW Energy Engineering Division for Brookhaven National Laboratory. March 1981.

2. E. C. Clark and D. K. Carlson, "Development Status and Utility of the Sulfuric Acid Chemical Heat Pump/Chemical Energy Storage System." AIAA Paper 809182. Proceedings of the 15th Inter-Society Energy Conversion Engineering Conference, August 18-22, 1980.

3. P. O'D. Offenhartz, "Methanol-Based Heat Pump for Solar Heating, Cooling, and Storage. Phase II." Draft Final Report EIC Corporation, October 1979. U.S. DOE under Contract to Sandia Laboratories #18,8423.

4. R. Gorman and P. Moritz, "Hydride Heat Pump, Volumes I and II." TRW Energy Systems Planning Division, McLean, Virginia, for Argonne National Laboratory, May 12, 1978.

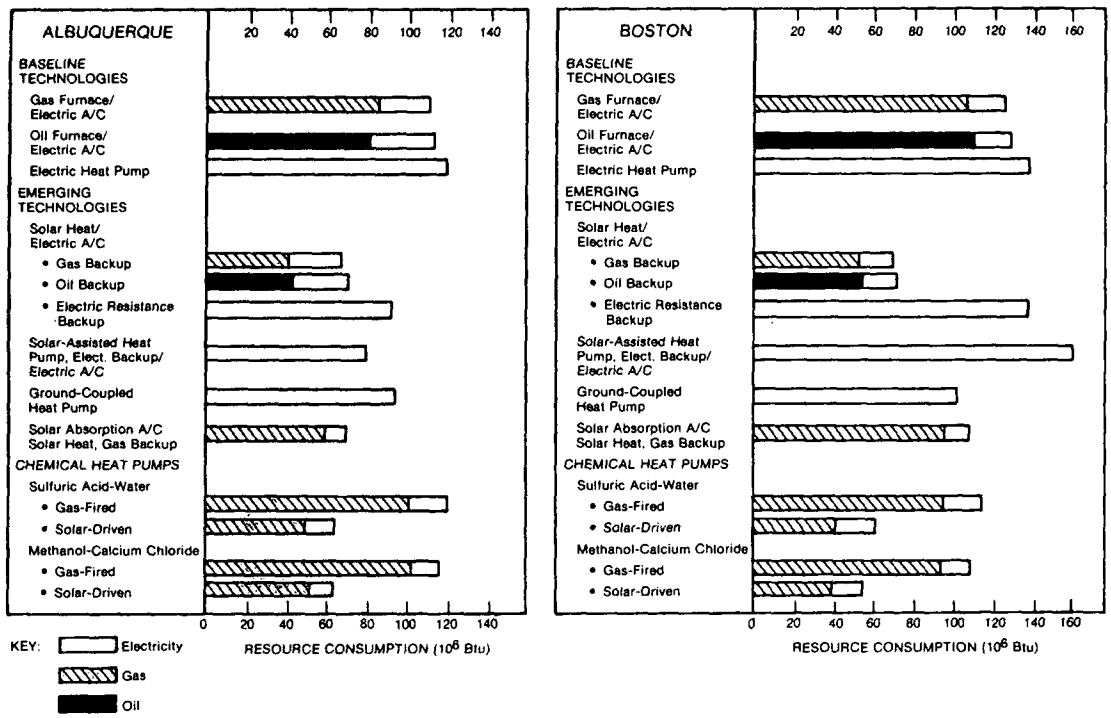


Figure 1. Annual Energy Resource Consumption for Residential HVAC Technologies

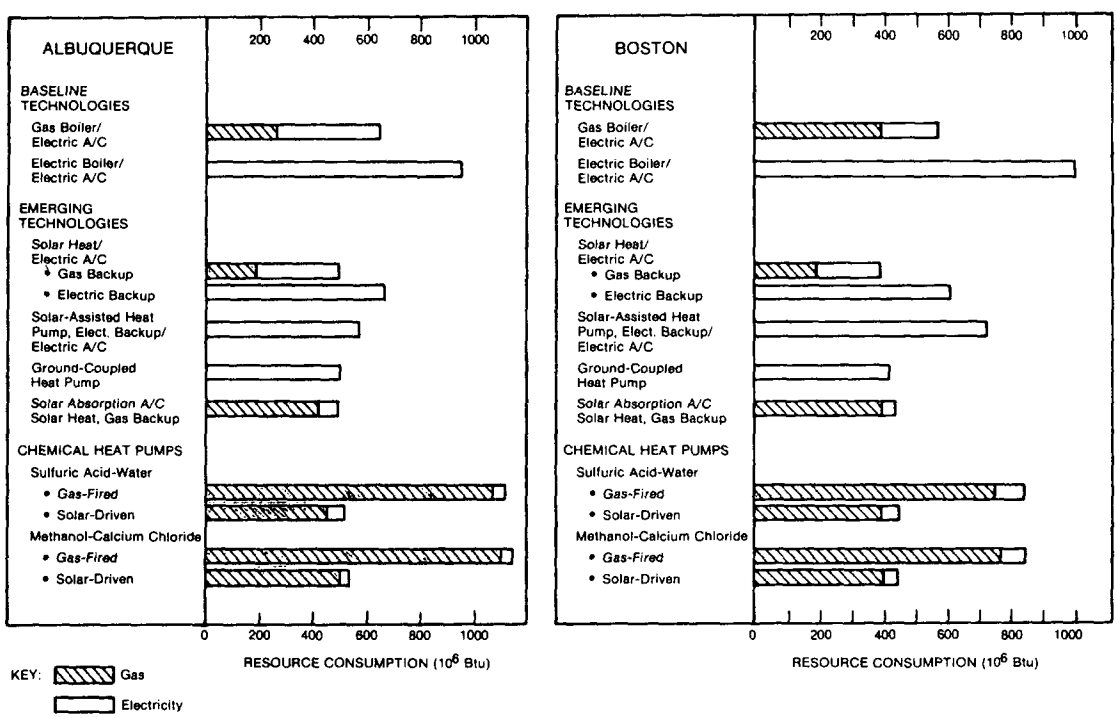


Figure 2. Annual Energy Resource Consumption for Commercial HVAC Technologies

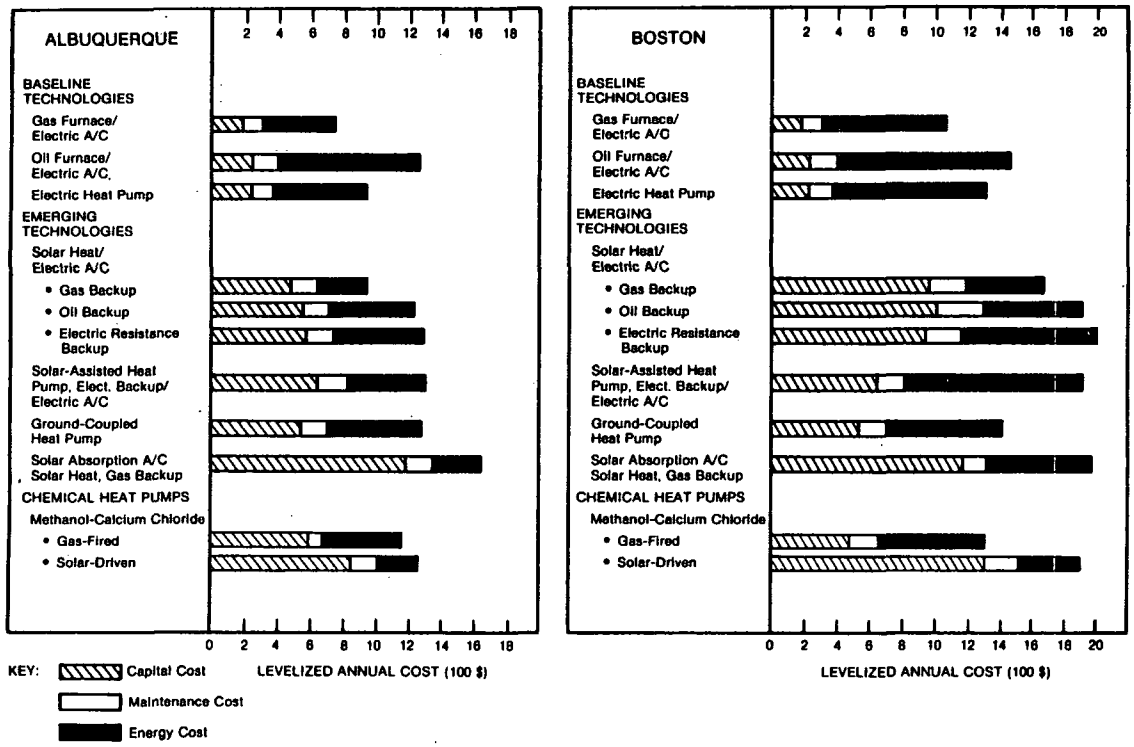


Figure 3. Levelized Annual Cost of Owning and Operating Residential HVAC Technologies

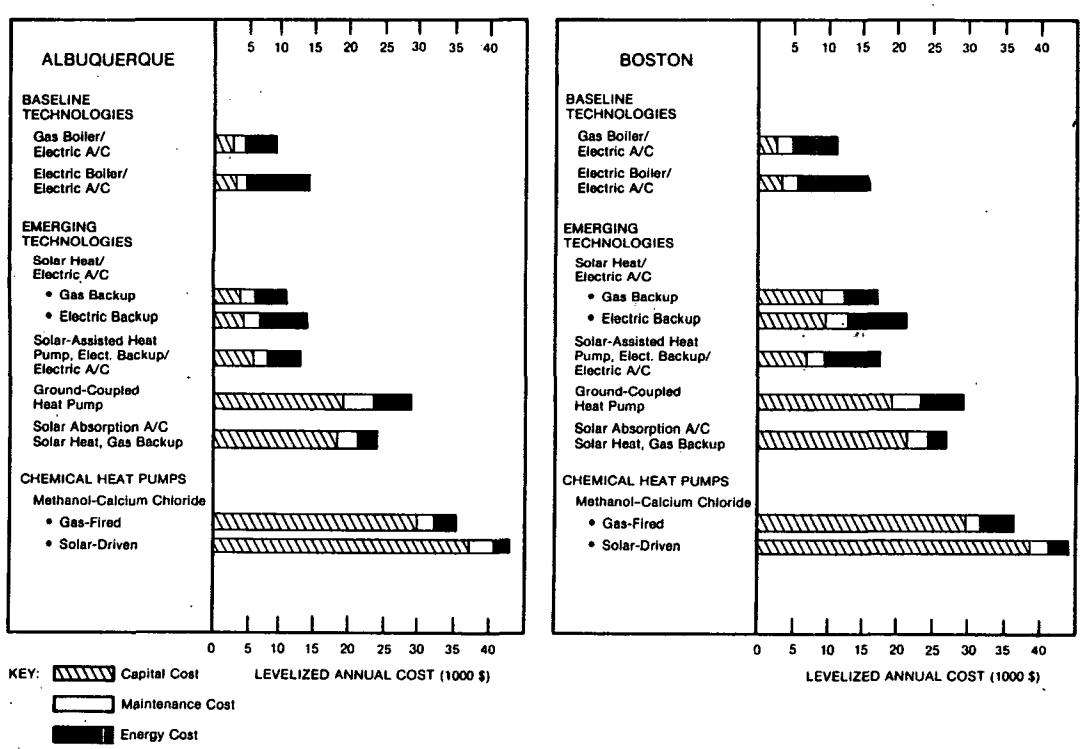


Figure 4. Levelized Annual Cost of Owning and Operating Commercial HVAC Technologies

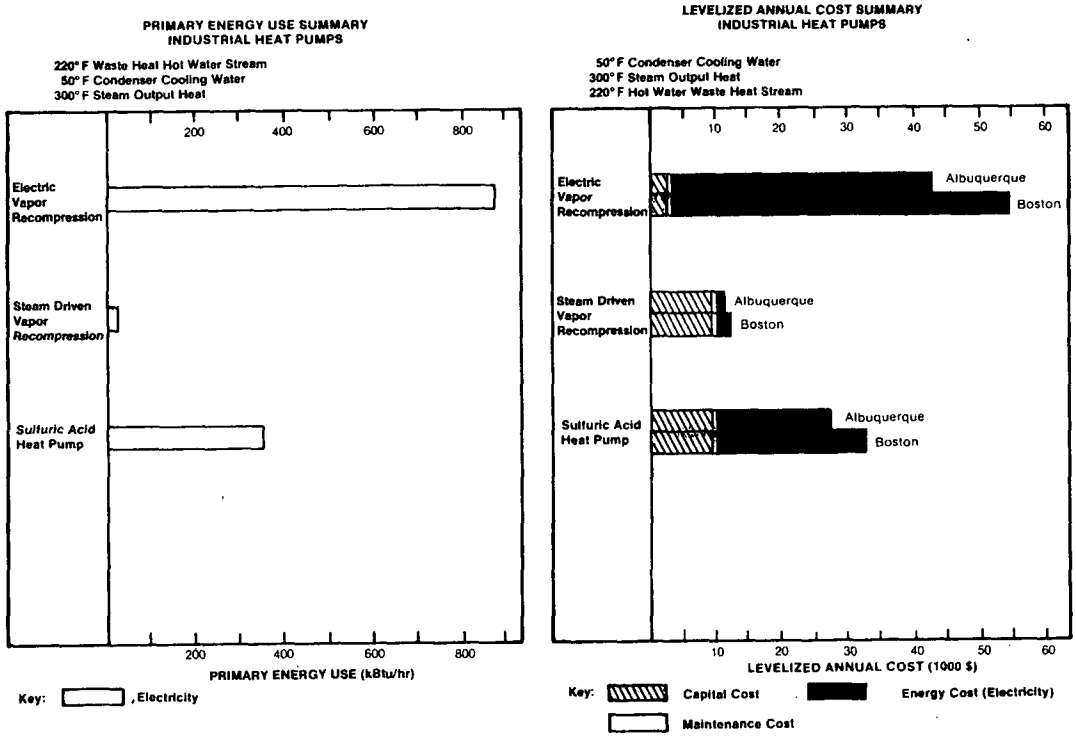


Figure 5. Energy Resource Consumption and Levelized Annual Cost for Several Industrial Heat Pumps Used for Raising 1.0 MBtu/hr of Process Steam

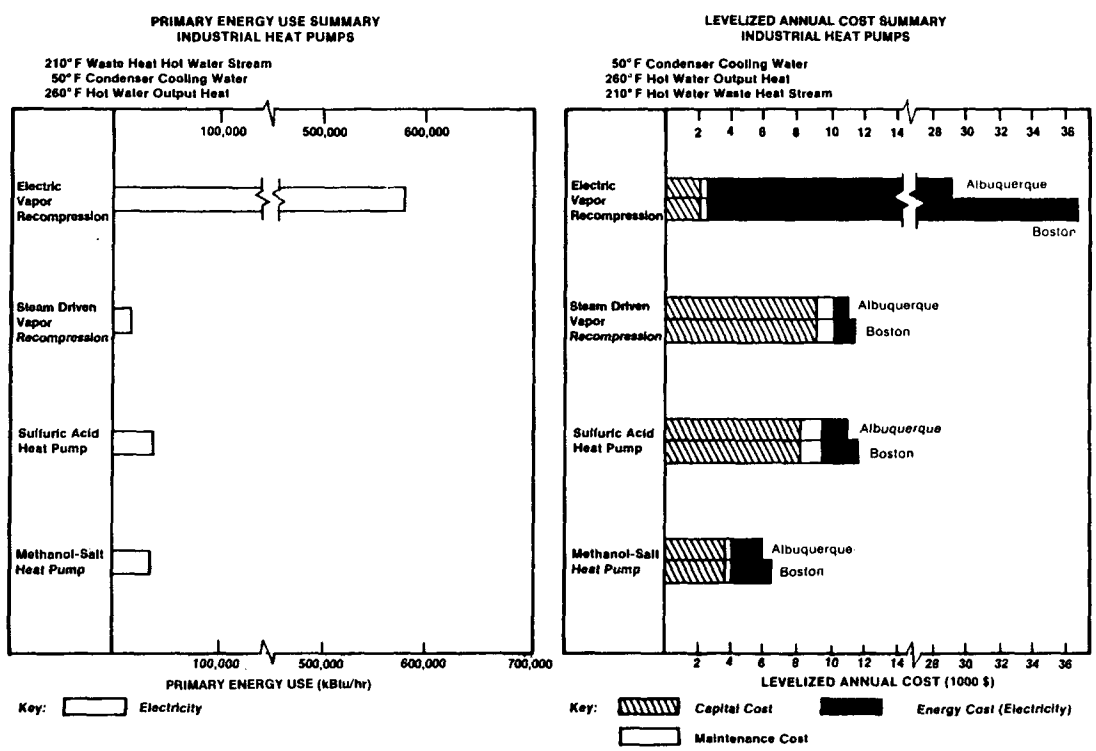


Figure 6. Energy Resource Consumption and Levelized Annual Cost for Several Industrial Heat Pumps Used to Provide 1.0 MBtu/hr for Process Water Reheating (240°-260°F)

SULFURIC ACID/WATER CHEMICAL HEAT PUMP
ROCKET RESEARCH COMPANY
494588-S
E. C. CLARK
PERIOD OF CONTRACT: BEGIN OCTOBER 1979, END SEPTEMBER 1981

## OBJECTIVE

The project is structured to verify the capability of the sulfuric acid-water system. The primary objective is to design, construct, and test a sulfuric acid chemical heat pump chemical energy storage, CHP/CES, verfication test unit (VTU) which incorporates as many commercial, mass producible design features as necessary to demonstrate system economic and technical feasibility. The VTU will be designed to function both as a storage HVAC unit and a temperature enhancement industrial heat pump. For purposes of gaining scaling criteria, the nominal storage design capacity shall be $10^6$ Btu with a nominal delivered heat rate of 150,000 Btu/hr.

## CONCEPT

The chemical heat pump system is a high technology device. It can be configured to operate as either an energy storage device or as a heat pump or both. In its operation, a working fluid of concentrated sulfuric acid as diluted with water yielding heat which is then used for space (comfort) heating or for industrial processing. The diluted working fluid is reconcentrated by distilling, using reject heat such as from an industrial source. Solar energy may also be used as an input energy source to drive the system. Because of the thermodynamics of the cycle, temperature amplification is achieved in its operation. THus, with the heat pump, low-quality reject heat streams can be upgraded and resorted to productiv euse, for example, in an industrial process. If the operational cycel is interrupted after the working fluid is reconcentrated, the chemicals can be retained in separate tanks until the user requires more heat. This latte case is an example of energy storage. In practice, heat pumping and energy storage can be provided in a single unit if required by the applications.

## SUMMARY

The current four-task contract is being managed by Brookhaven National Laboratories (BNL). In the present effort, Tasks 1, 2, and 3 haven been completed and consisted of a requirements analysis, detailed design of a 150,000-Btu/hr verification test unit (VTU) which stores $10^6$ Btu, and VTU fabrication. Generic applications have been identified which include industrial heat pumping using reject heat, space conditioning, air or gas moisture removal, steam accumulation, cogeneration, and energy storage. The VTU design permits evaluation of two of the most promising near-term applications: waste heat pumping and space conditioning. The Task 4 test program has begun.

## TECHNICAL ACCOMPLISHMENTS

- A requirements analysis was completed and resulted in the identification of the two most promising market applications: industrial heat pumping and space conditioning with storage. The industrial chemical heat pump, ICHP, operates in a continuous cycle to upgrade low temperature waste heat. As with most absorption cycles, four primary reaction components are required: a generator, absorber, evaporator, and condenser. Figure 1 provides a simplified schematic and operational description of the acid and water circuits. To provide energy storage and operate the unit for space conditioning requires additional components. Figure 2 presents a simplified schematic of a CHP/CES. The major components consist of acid storage, water storage, generator, evaporator, condenser, absorber, and heat collection and distribution systems (not shown). Also as a part of the requirements analysis, detailed performance calculations were completed for the ICHP and HVAC operational modes. Operational boundaries, both theoretical and practical, were determined. A variety of thermal sources were investigated as drivers for the CHP/CES. Safety and control requirements were outlined.

- An economic analysis was completed which showed the ICHP as having payback periods of 1 to 2 years depending on equipment power rating. A detailed cost estimate of an ICHP with a power rating of 1 x $10^6$ Btu/hr provided the basis for the economics.

- The VTU components were sized to provide 150,000 Btu/hr of power and be capable of storing up to 1 million Btu's.

- A verification test unit, VTU, was designed and is capable of two operational modes: 1) ICHP and 2) CHP/CES for space conditioning. The design incorporates low cost, readily available engineering materials which are packaged in a commercial type configuration. The unit is easily scaled and is designed with ease of maintenance and operator safety in mind. The nominal power rating and storage capacity meets the above project goals. Figure 3 presents a sketch of the system.

- The heat exchangers are an RRC design. Each contains 31 Pyrex tubes in a steel shell. Sealing is accomplished by means of an unique dual tube sheet and O-ring system.

- In support of the VTU design, the 25,000 Btu/hr engineering model constructed during a previous contract was used. A total of 618 hours of test have been accumulated since its inception in 1977, including 143 hours during the current program. A horizontally integrated reaction canister was added to the engineering model to provide proof-of-concept for the VTU design and accumulated 43 hours of test to charge 1 million Btu's.

- A materials evaluation program was begun and will continue through the end of the program with the primary objective being the evaluation of low-cost coatings. Of particular significance is the porcelain coating used to line the VTU components.

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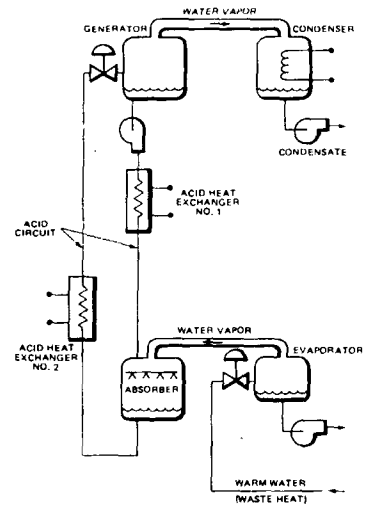
## INDUSTRIAL CHEMICAL HEAT PUMP OPERATION

### ACID CIRCUIT

1. **Generator** — Dilute acid is input into the generator where the acid/water separation takes place. Concentrated acid is removed from the bottom and water vapor is removed at the top of the generator. The concentrated acid is cooled by the distillation process.
2. **Acid Heat Exchanger No. 1.** — The concentrated acid from the generator is input into acid heat exchanger No. 1, where it is heated by the waste heat source to regain sensible energy lost in the generator.
3. **Absorber** — *Concentrated acid* leaving acid heat exchanger No. 1 is input into the absorber along with water vapor transported from the evaporator. The concentrated acid provides a significant chemical attraction for the water vapor, and *the water vapor is forced to condense* into the acid and dilute it. The resulting dilution process dramatically increases the temperature of the acid solution.
4. **Acid Heat Exchanger No. 2** — Hot dilute acid is routed to acid heat exchanger No. 2 where heat is removed and delivered to a process at a temperature significantly higher than the waste heat source. The dilute acid is then routed to the generator for reconcentration.




### WATER CIRCUIT

1. **Condenser** — Water vapor is *removed from the generator by the* condensation process. A low-temperature source such as industrial water, groundwater, etc. is used to remove the heat of condensation at ambient temperature. The liquid condensate formed is *either* routed to the *evaporator* or discarded, depending on the system design.
2. **Evaporator** — The waste heat source is used to input energy into the evaporator to produce water vapor which is transported to the *absorber. If the waste heat source is* water (as shown in the schematic), it is input directly into the evaporator where flash evaporation takes place without the necessity for a heat exchanger.

**Figure 1**

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## CHP/CES SYSTEM SCHEMATIC

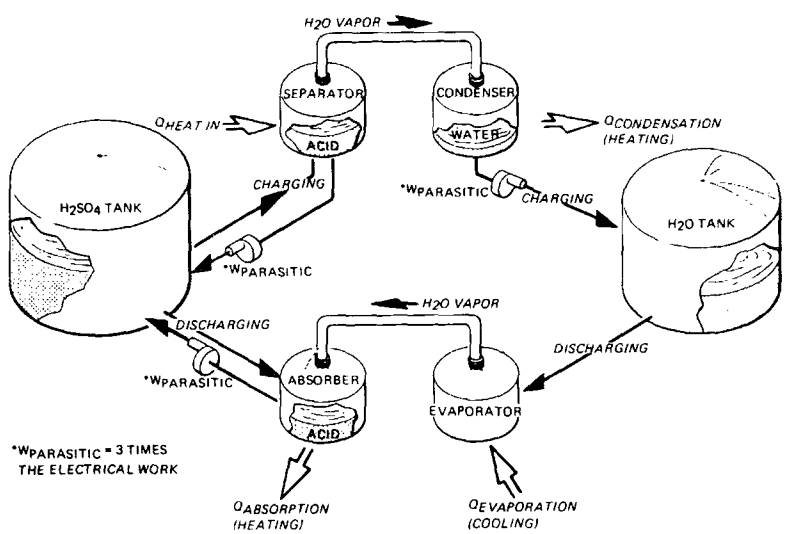


**Figure 2**

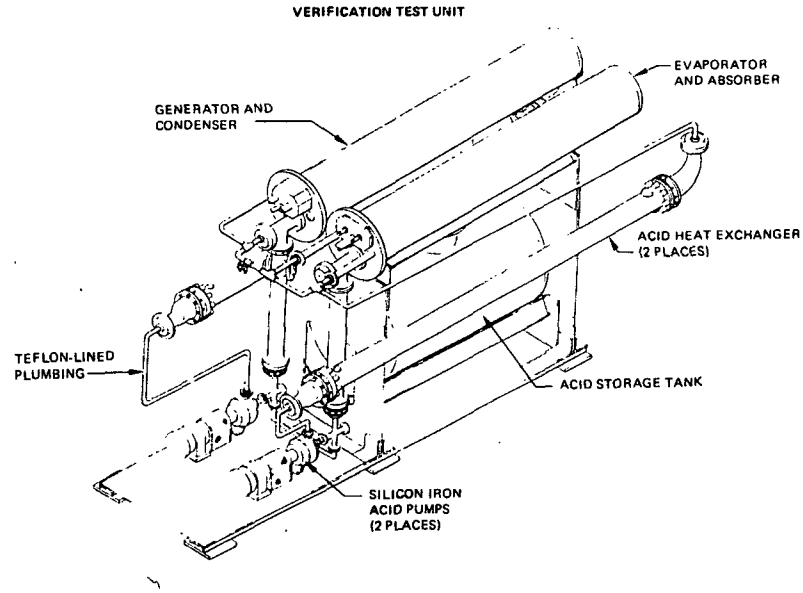


Figure 3

- Fabrication of the VTU has been completed. The assembly went smoothly with all subcontractors meeting delivery schedules. A low voltage, salt solution, non-destruction test was identified and used to verify porcelain coating integrity. The large components were fabricated by local small business organizations.

- Performance testing of the unit in the ICHP mode has recently begun.

## FUTURE ACTIVITIES

Contract Activities. A performance test program with the VTU will be completed by September 1981. A final report will be prepared and will include design changes as identified by the test program.

Post-Contract Activities. An experimental evaluation program is presently being planned for implementation during FY'82. The program will consist of the design, fabrication, and testing of a chemical heat pump having an industrial capacity of about 1 million Btu/hr. The test will be conducted at a representative industrial plant personnel. The objective of the proposed program is two-fold: 1) design a full-size industrial chemical heat pump using scaling parameters evolved during previous development and 2) evaluate the technical and economic performance of the sulfuric acid system at a representative industrial site. The acquired information will serve as the basis for technical and economic performance projections for a wide variety of industrial applications of chemical heat pump technology.

## PUBLICATIONS/REPORTS/REFERENCES

1. Rocket Research Company, "Phase I Final Report, Sulfuric Acid-Water Chemical Energy Storage System, ERDA Contract No. E904-3)-1185," Rocket Report No. RRC-76-R-530, August 1976. (Note: This report presents the heat of dilution concept only and does not include the chemical heat pump.)

2. Clark, E.C.: "Sulfuric Acid/Water Integrated Chemical Energy Storage System," Thermal Energy Storage Contractors Information Exchange Meeting, Gatlinburg, Tennessee, September 29—30, 1977.

3. Clark, E.C.: "Sulfuric Acid and Water Chemical Heat Pump/Thermal Energy Storage", Energy Technology: Third Annual Proceedings of Thermal Energy Storage Contractors' Information Exchanger Meeting, December 5—6, 1978.

4. Clark, E.C.: "Final Report — Phase II Sulfuric Acid-Water Chemical Heat Pump and Storage System, DOE Contract No. EY-76-C-03-1185," Rocket Report No. RRC-78-R-595, December 1978.

5. Clark, E.C.: "Final Report — Phase II-A, Sulfuric Acid and Water Chemical Heat Pump/Chemical Energy Storage Program, Sandia Contract No. 18-4958," Rocket Research Report No. RRC-79-R-627, September 1979.

6. Clark, E.C. and Hiller, C.C.: "Sulfuric Acid-Water Chemical Heat Pump/Energy Storage System Demonstration," presented at the ASME Winter Annual Meeting, December 10—15, 1978.

7. Clark, E.C. and Hiller, C.C.: "Chemical Heat Pump/Energy Storage System Demonstration using Sulfuric Acid and Water," published in the Proceedings of Solar Energy Storage Options, March 19—20, 1979, San Antonio, Texas (Conf. 790328).

8. Hiller, C.C.: "Development of Chemical Heat Pump/Chemical Energy Storage Systems for Heating and Cooling Applications," published in the Proceedings of the 4th Annual Heat Pump Technology Conference, April 1979, Oklahoma State University, Stillwater, Oklahoma.

9. Hiller, C.C. and Clark, E.C.: "Development of the Sulfuric Acid-Water Chemical Heat Pump/Chemical Energy storage System for Solar Heating and Cooling," published in the Proceedings of the 1979 International Solar Energy Society Meeting, Atlanta, Georgia, May 1979.

10. Clark, E.C. and Hiller, C.C.: "Development and Testing of the Sulfuric Acid-Water Chemical Energy Storage System." Presented at the 14th Annual Intersociety Energy Conversion Engineering Conference, Boston, August 1979.

11. Clark, E.C. and Morgan, O.M., "Sulfuric Acid and Water Chemical Heat Pump/Chemical Energy Storage System," published in the Proceedings of the DOE Chemical Energy Storage and Hydrogen Energy Systems Contract Review, November 13—14, 1979 (Conf. 791127).

12. Clark, E.C.: "Sulfuric Acid and Water Chemical Heat Pump/Chemical Energy Storage System presented at the Internal Seminar on Thermochemical Energy Storage, Stockholm, January 7—9, 1980.

13. Clark, E.C. and Carlson, D.K.: "Development Status and Utility of the Sulfuric Acid Chemical Heat Pump/Chemical Energy Storage System," presented at the 15th Intersociety Energy Conversion Engineering Conference, Seattle, Washington, August 1980.

14. Clark, E.C., Carlson, D.K., and Morgan, O.M.: "The Sulfuric Acid/Water Chemical Heat Pump/Energy Storage Program," published in the Proceedings of the DOE Thermal and Chemical Storage Annual Contractors Review Meeting, October 14--16, 1980.

15. Clark, E.C. and Morgan, O.M.: "Chemical Heat Pumps for Industry," to be presented at the 16th Intersociety Energy Conversion Engineering Conference, Atlanta, Georgia, August 1981.

AIR COOLED ABSORPTION CHILLERS FOR SOLAR COOLING APPLICATIONS
ENERGY SYSTEMS DIVISION, CARRIER CORPORATION
EG-77-C-03-1587
DR. WENDELL J. BIERMANN, ROBERT REIMANN
PERIOD OF CONTRACT: Begin SEPTEMBER 1977 End SEPTEMBER 1982

## OBJECTIVES

The purpose of this program is to identify the chemical composition of a "best" absorption-refrigerant system, capable of being air cooled to determine those properties of the system necessary to design hot water operated, air cooled chilling equipment and to design and operate air cooled chillers from single family residential sizes into the commercial rooftop size range.

## DESCRIPTION OF WORK

At previous Contractors' meetings we have discussed the considerations which led us to the selection of a mixture of lithium bromide and ethylene glycol as an absorbent, coupled with water as a refrigerant.

Figure 1 shows the pressure-temperature relationship of this system in the form of a Duhring plot. Superimposed on this plot is a parallelogram showing the approximate limits covered by an air cooled absorption cycle at the design point of 35°C (95°F) air. The absorbent ratio of LiBr:glycol was selected as 4.5, roughly optimized for the various considerations of vapor pressure, cycle pumping rate and heat transfer properties. It will be noted that the crystallization line, shown as a dashed line, is comfortably removed from the region occupied by the cycle. By contrast, the crystallization line for lithium bromide cuts very close to the working area, inviting a solidified system should there be a sudden perturbation of the operating conditions or should there be some minor errors in the information.

This system is useful only if it can be operated near the equilibrium conditions. Our major chemical innovation has been the identification of a chemical additive, 1-nonylamine, which lowers subcooling in the absorber to about 1°C from the 15°C range of the simple refrigerant-absorbent combination. Without this additive the gain in crystallization is obviously lost in increased subcooling.

For convenience we have coined the term "Carrol" to describe a system comprising an absorbent of lithium bromide and ethylene glycol in a weight ratio of 4.5:1, with water as refrigerant and 1-nonylamine as an additive.

## MACHINERY DEVELOPMENT

At the present writing three chillers have been constructed and two of them operated under test conditions. The first of these, a "breadboard" type construction, produced about 8 kW (2.3 t) of cooling at the design point of 110°C (230°F) hot water entering the generator, 35°C ambient air and 7.2°C (45°F) chilled water to the load. This served to vindicate the basic design concepts and refined some of the heat transfer coefficient data.

A second piece of equipment, a first prototype single family residential unit, was designed from the breadboard data with a projected cooling capacity of 10 kW (3 tons). This unit is shown, with the sheet metal case removed, in Figure 2. When placed in operation in a laboratory facility designed and built especially for the purpose, the initial cooling capacity at design conditions was about 8.7 kW, sustained for about ten days. Unfortunately the performance of this equipment soon deteriorated to about 6 kW at design conditions as a result of heat transfer surface contamination, due at least in great part to air leakage. After a series of unsuccessful attempts to resuscitate the prototype, the decision was made to abandon the machine and move on to the next program step. The first prototype experience emphasized to us the importance of designing all components with ease of leak detection and leak repair a major design consideration.

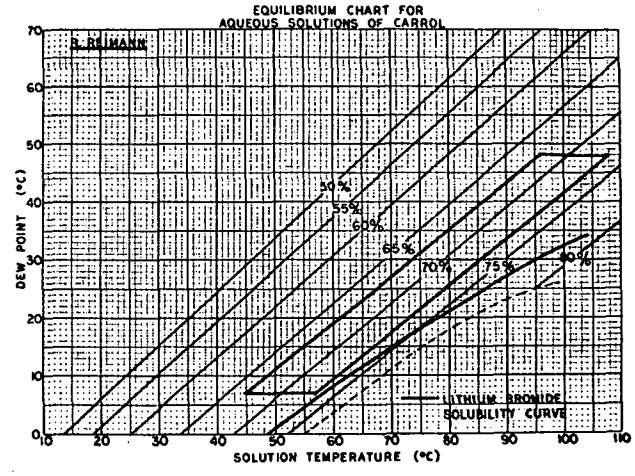


FIGURE 1. Equilibrium Chart for Aqueous Solutions of Carrol. For reference purposes, the crystallization line for lithium bromide-water has been superimposed as well as a typical cooling cycle path.

The next program step selected was not an immediate redesign of the 10 kW prototype but rather a move to a larger rooftop solar package built around a 35 kW (10 ton) air cooled chiller. Funding for this particular step has been supplied through the SOLERAS program administered by the Solar Energy Research Institute (Contract DE-FC02-80-ET20643). Figure 3 shows the 35 kW rooftop package which integrates the solar system pumps, controls and some data retrieval functions.

Selection of the larger equipment as the successor step to the first residential 10 kW prototype was based primarily on the belief that the larger size was, in some ways, less of a deviation from established designs. These considerations will be included in the following consideration of the major components of the chiller designs.

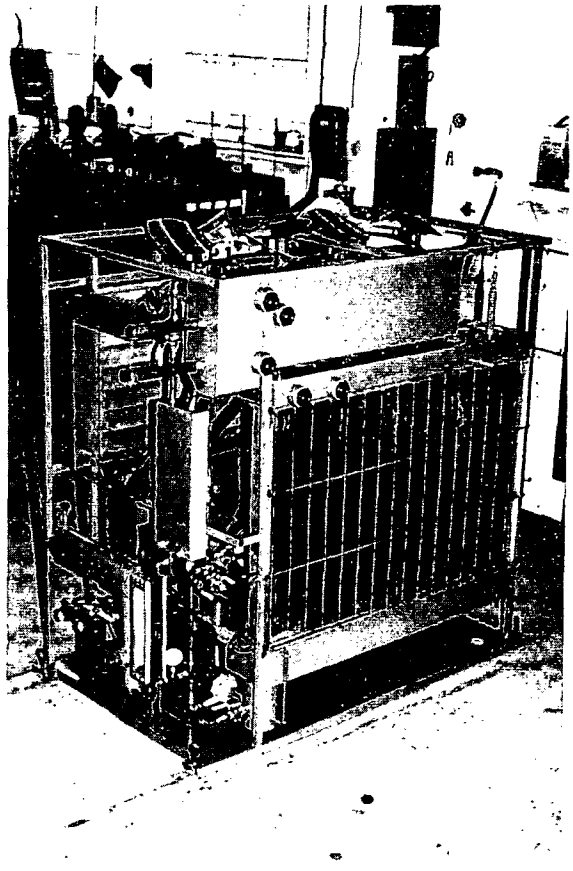

FIGURE 2. First 10 kW (3 ton) Prototype Air Cooled Absorption Chiller.

FIGURE 3. View of the 35 kW (10 ton) Air Cooled Solar Absorption Chiller Package.

Generator. In all cases a falling film generator is used, based on the very successful application of this design in our parallel development of water cooled chillers for low temperature application. The falling film generator is preferred over the submerged tube bundle because it permits counter-current heat transfer, eliminates submergence losses, minimizes the need for superheat to induce nucleate boiling and greatly reduces cycling losses by eliminating the large quantity of hot solution stored in the generator.

Distribution of the solution over the tube bundles in the 10 kW design was by means of a series of weirs which overflowed onto the exterior of the generator tubes through capillary drippers. In the larger 35 kW package there is sufficient room to use spray headers, which has been the basis of all our previous experience with the falling film generator.

Absorber and Condenser. Both are plate fin coils with one inch copper tubes and ten mil aluminum fins, 14 per inch. Since the last Contractors' conference report was written, the coil parameters were measured by the Carrier Research Division and a computer analysis made of this application. As a result, the condenser coil was redimensioned and recircuited and the air velocity through both condenser and absorber reduced. This will significantly reduce fan power from that required in earlier designs.

Evaporator. It has been most convenient thus far to work with a flash evaporator, in which return chilled water is flashed through a nozzle in a chamber communicating with the absorber. With the 35 kW rooftop package, however, we will return to the more conventional tube and shell chiller which will lessen the likelihood of leaks and will simplify the problem of freeze-proofing the units.

It is conceptually possible that the ethylene glycol component could evaporate in the generator and concentrate in the refrigerant loop. This does not happen and a steady state concentration of glycol in refrigerant of about 5% has regularly been observed.

Fans. Based on the new requirements for the redesigned coils, the Carrier Corporate fan selection program was used to make an optimum selection of fan motor and blades. The 35 kW rooftop unit will use six fans with about 215 watts per ton capacity, compared to about 300 watts per ton in the previous designs.

Solution Heat Exchanger. The solution heat exchanger is a major consideration in getting good thermal efficiency. Very little head is available on the strong solution side, a situation not helpful in achieving a high effectiveness ($\eta \cong 0.8$). Our flat plate design, constructed of a stack of simple shapes easily punched from sheet metal, has worked out well.

Purge. A small version of the jet purge that Carrier uses on its commercial 16JB absorption chiller series has given us effective purging of gases from leaks and corrosion. The maximum pump head in the 35 kW chiller is sufficiently high to permit emptying the purge storage tank by pump discharge pressure.

Controls. A low refrigerant level sensor protects the refrigerant pump by turning it off when refrigerant stores on the solution side. Should overconcentration of the Carrol threaten solidification, this is detected by a high refrigerant level sensor which activates a Cycle-Guard[R] solenoid valve which feeds refrigerant into the solution pump suction. To maintain constant load capacity as ambient temperatures increase, it is necessary to increase generator temperature by two degrees for each degree increase in ambient above the design point of 35°C.

A second solenoid valve will also be introduced into the 35 kW package and subsequent machines which will permit the output of the solution pump to be introduced into the suction side of the refrigerant pump as a means of preventing the refrigerant from freezing during winter shutdown. In conjunction with an additional level sensor in the refrigerant pump, it can also be used to extend machine operation to lower ambient temperatures.

## TECHNICAL ACCOMPLISHMENTS

A survey of possible air cooled absorbent-refrigerant combinations has identified a lithium bromide-ethylene glycol-water system, with 1-nonylamine as an additive, which was a "best" system with respect to cost, chemical stability, flammability, toxicity and projected development time.

All necessary engineering design information-vapor pressure, enthalpy, heat transfer coefficients, etc., has been measured.

Component designs have been tested in 10 kW (3 ton) breadboard and first prototype configurations.

A 35 kW (10 ton) rooftop solar package has been designed and built.

A test facility for running small air cooled absorption chillers over a variety of controlled conditions has been designed and built.

## FUTURE ACTIVITIES

The equipment resulting from this program are intended as prototypes of products aimed at a market still in process of development. With this in mind, our Commercial Products Division prepared a Fabrication Cost Estimate for a 10 kW (3 ton) product based on the first prototype design. A run of 1000 units would have a fabrication cost of $2,400 each.

Our corporate experience with similar products has closely followed an 80% experience curve in projecting cost reductions. Table I shows a possible future cost scenario which would be consistent with our previous experience in developing very similar new products for emerging markets. Cost to the consumer will be higher by distribution, profit and other business costs.

TABLE I

AVERAGE FABRICATION COST PROJECTION
3-TON ABSORPTION MACHINE (AIR COOLED)

| Year | Annual Production | Cumulative Production | 20% Initial Cost Reduction Average Fab. Cost/Unit 1980 Dollars |
|---|---|---|---|
| 1 | 100 | 100 | $1,916 |
| 2 | 500 | 600 | 1,886 |
| 3 | 2,000 | 2,600 | 1,782 |
| 4 | 6,000 | 8,600 | 1,561 |
| 5 | 12,000 | 20,600 | 1,299 |
| 6 | 13,200 | 33,800 | 1,111 |
| 7 | 14,500 | 48,300 | 992 |
| 8 | 16,000 | 64,300 | 905 |
| 9 | 17,600 | 81,900 | 838 |
| 10 | 19,300 | 101,200 | 783 |

## PUBLICATIONS AND REPORTS

"Candidate Chemical Systems for Air Cooled Solar Powered Absorption Air Conditioner Design." Part I - Organic Absorbent Systems; Part II - Solid Absorbents, High Latent Heat Refrigerants; Part III - Lithium Salts with Antifreeze Additives; Wendell J. Biermann (1978). Parts I and II are available from NTIS, Part III release expected shortly.

"Properties of the Carrol System and a Machine Designed for Solar Powered, Air-Cooled, Absorption Space Cooling," Revised Report May 1981.

# WATER COOLED ABSORPTION CHILLERS FOR SOLAR COOLING APPLICATIONS
ENERGY SYSTEMS DIVISION, CARRIER CORPORATION

DR. WENDELL J. BIERMANN, ROBERT C. REIMANN

## OBJECTIVES

The long term objective of the program which has been implemented by the contracts enumerated below, was the development of a broad line of absorption chillers designed to operate with hot fluids at as low a temperature as practical while rejecting heat to a stream of water.

A second objective was to develop a packaging concept for solar application in which controls, pumps, valves and other system components could be factory assembled into a unitary solar module.

## DESCRIPTION OF WORK

Energy Systems Division, Carrier Corporation, has been conducting programs in solar cooling with various agencies all of which have been tied together with a common group of assumptions and goals. One of these assumptions was that of the various methods of solar cooling possible, absorption cooling was the only technology which could become commercial on a relatively short time frame. Absorption cooling with heat rejection to a water stream, as might couple to a cooling tower, appeared to warrant the highest priority because of the large background of developed technology. This program is the subject of the present paper. A second program, development of a line of air cooled absorption chillers over the residential and smaller commercial sizes, appeared to be somewhat longer in term and our contributions in that area are described in a second paper in this Conference Proceedings.

A design point was selected on the basis of what might reasonably be expected from a system using good quality flat plate solar collectors. Specifically we chose:

Rating Conditions:

Hot water 82°C (180°F) and 0.060 l $s^{-1}$/kW capacity
(3.33 gpm/ton (c) )
Cooling water 29°C (85°F) and 0.126 l $s^{-1}$/kW (c)
(7.00 gpm/ton (c) )
Chilled water 7.2°C (45°F) and 0.043 l $s^{-1}$/kW (c)
(2.40 gpm/ton (c) )

Size Range:

50 - 500 kW (c) (15 - 150 tons)
Smaller units to be contained in preassembled solar packages.

In addition we wanted a basic design which would retain a large fraction of design capacity when the hot water temperature dropped to 65°C (150°F), other quantities remaining at the rating condition, and which would have minimum losses due to cycling.

The primary approach which was to be used was to increase surface in a fairly conventional tube bundle for evaporator, absorber and condenser design, but to increase the number of passes in the evaporator and absorber so as to approximate counterflow rather than cross flow heat transfer. Instead of the submerged tube bundle generator normally employed, we elected to use a falling film type generator which would reduce approach temperatures by permitting counterflow rather than cross flow heat transfer and avoid superheating effects associated with nucleate boiling and hydrostatic head. By eliminating the storage of the mass of solution required to cover the generator tube bundle, the heat losses on interrupted operation were very greatly reduced.

Table 1, on the following page, summarizes in tabular form the contracts and the several agencies which have made it possible to carry out the individual steps in the overall program. In each case, a brief note gives the specific objective of that portion of the program and the current status.

Typical performance data and illustrations of the 50 kW (15 ton) solar absorption chillers have been given at previous Contractors' meetings and in the publication list appended.

By way of update, three figures are included. Figure 1 shows the latest packaged version of the 50 kW chiller, presently installed at the BDP Distribution Center, Phoenix, Arizona, as part of the SOLERAS program. This package includes pumps for solar collector, chilled water distribution and for induced draft cooling tower, necessary control sensors and valves plus all needed control logic devices.

The second figure shows the largest unit thus far built in this series, a solar absorption chiller designed to produce about 130 tons of cooling at the program rating point, and was developed under sponsorship of Marshall Space Flight Center (NASA). This machine is semi-commercial in that it was produced down the same assembly line as the standard Carrier absorption chillers. To become fully commercial, it requires only enough volume to justify certain special tooling and setting in place a complete manufacturing plan.

Figure 3 shows the performance of the large absorption chiller over a range of hot water temperatures. These data result from performance testing done in the Engineering Test Laboratory, Machinery and Systems Division, Carrier Corporation.

## TECHNICAL ACCOMPLISHMENTS

A low temperature absorption chiller has been designed, laboratory tested and field tested. It is suitable for solar, geothermal or energy recovery applications.

The manufacturability of the low temperature chiller has been demonstrated on a commercial production line.

The low temperature chiller design includes 15 tons, 75 tons and 130 tons.

Packaging of the 15 ton size chiller into a simple unitary solar package has been done.

| Major Task Number | Technical Task | Status or Schedule |
|---|---|---|
| 1 | Design, build, factory test, field install in solar system with evacuated tube collectors and monitor for one year; a 50 kW (15 ton) solar absorption chiller. | Installation completed, Ross Control House (Vancouver, WA) March 1978. Currently operating. Preliminary Final Report submitted January 1980. Bonneville Power Administration to issue final final report about May 1981. Funding: Bonneville Power Administration. 14-03-6280N $848,959 |
| 2 | Select new solution and refrigerant pumps for 50 kW (15 ton) package developed in Task #1. Develop solar package for steam as auxiliary energy. Install with flat plate collectors and operate for one year. | Installation completed, Carrier factory in Tyler, Texas, October 1979. Operation monitored to October 1980. Presently inoperative due to collector problem. Funding: Bonneville Power Administration/ DOE DE-AC79-79BP 14067 $375,000 |
| 3 | Operate Tyler, Texas installation for cooling season, 1981. | Contract anticipated May 1981. To be followed by preliminary final report to Bonneville Power Administration November 1981. Funding: SERI Contract Pending $15,250 |
| 4 | Using the modified 50 kW (15 ton) solar absorption chiller of Task 2, develop package using electric driven heat pump for auxiliary. Install, using drainable, evacuated tube collectors, at Big Eddy Control House, The Dalles, Oregon. | Installation completed April 1980. Year's operation completed April 1981. Preliminary Final Report, June 1981. Funding: Bonneville Power Administration/DOE DE-AC79-79BP 14067 $375,000 |
| 5 | Using absorption chiller design proven in Tasks 2 and 4, develop improved packages. Install, using tracking concentrating trough collector, BDP Distribution Center, Phoenix, Arizona. Operate through two cooling seasons. | Installation completed April 1981. Preliminary Final Report November 1982. Funding: SOLERAS, Managed by SERI DE-FC02-80ET 20643 $815,000 (including a second installation) |
| 6 | Scale up the 50 kW (15 ton) design to meet needs of Frenchman's Reef Holiday Inn, American Virgin Islands. Under selected rating conditions, this would be about 450 kW (130 tons). Design will produce over 700 kW (200 tons) at higher temperatures. | Construction completed March 1981. Laboratory tested April 1981. Shipped April 1981. Installation and field testing will be contracted by NASA, scheduled about June 1981. Funding: NASA DEN8-000007 $266,318 |
| 7 | Using same design as Task 6, design, construct and laboratory test two solar absorption chillers of 260 kW (75 ton) capacity at design conditions. | Construction completed March 1982. Will be installed and field tested under separate NASA contract in 1982 at University of Houston and University of Nevada (Las Vegas). Funding: NASA DEN8-000015 $227,900 |
| 8 | Do cost reduction study of design in 50 kW (15 tons) to 150 kW (40 tons) range. Build and test one unit to prove cost reduced design. | Desirable to improved commercial value. |
| 9 | Using design concepts proven in past program, design a solar chiller with two stage generator to increase COP. Design hot water temperature to be not in excess of 115°C (240°F). | Logical step for technological advance of solar cooling. |

TABLE 1. Summary of Contracts Constituting the Water Cooled Absorption Chiller Program of Energy Systems Division, Carrier Corporation.

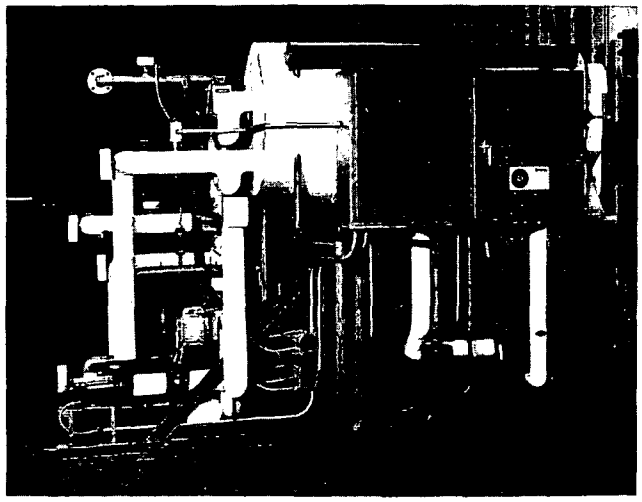

FIGURE 1. Solar Unitary Package, BDP Distribution Center, Phoenix, Arizona.

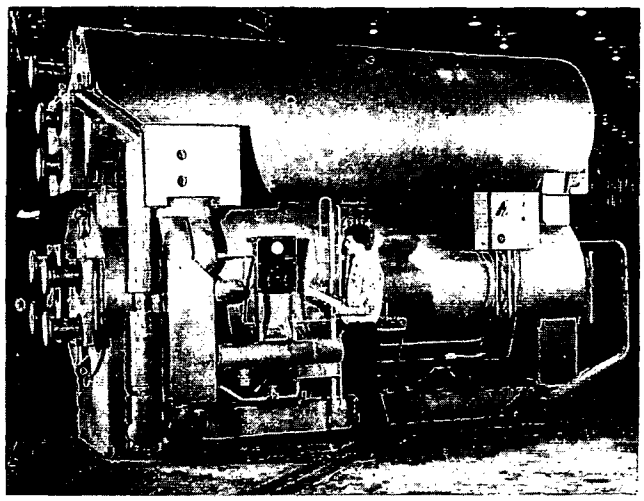

FIGURE 2. Scaled Up Solar Absorption Chiller, Frenchman's Reef Holiday Inn, American Virgin Islands.

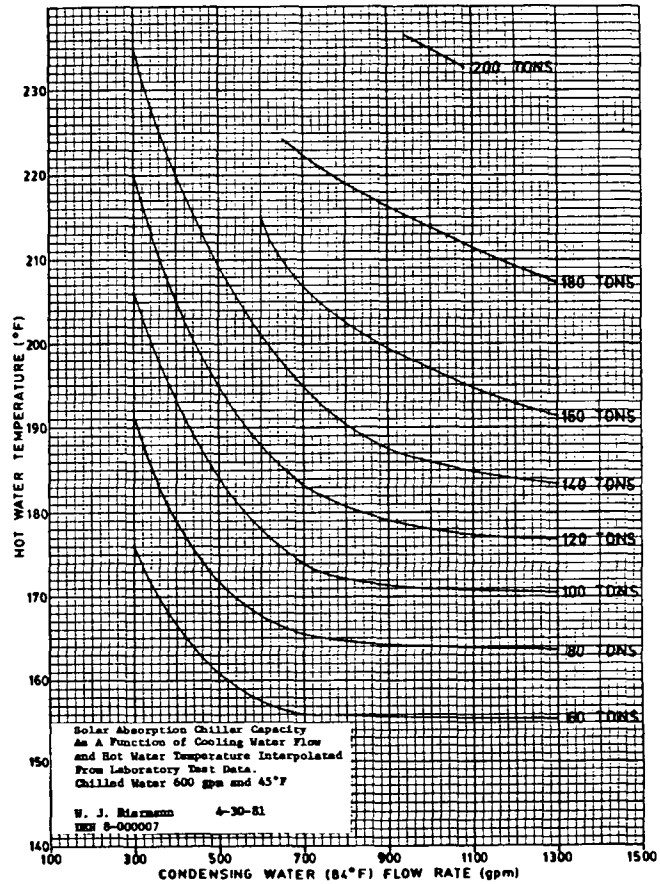


FIGURE 3. Capacity of Large Solar Absorption Chiller Over a Range of Hot Water Temperatures.

## FUTURE ACTIVITIES

Absorption chillers of this improved design are salable items. They are being offered for sale, potential customers have expressed interest but no actual private sales have been consummated. Cost is a serious obstacle -- capital investment is needed to reduce manufacturing cost (tooling, fixtures and procedures) but the volume of production needed to amortize the investment cannot be identified, at least not presently for the single stage machine.

## PUBLICATIONS


"Evaluation Report Prototype Energy Retrieval System," Submitted to Bonneville Power Administration, January 1980. (Final Report, Contract 14-03-6280N).

"The Prototype Energy Retrieval and Solar (PERS) System," April 1981, by Bob Guddat, Mechanical Section, Division of Substation and Control Engineering, Bonneville Power Administration, Portland, Oregon.

"An Absorption Machine for Solar Cooling," Dr. Wendell J. Biermann, ASHRAE Symposium Paper PH-79-3 No. 2, ASHRAE Transactions 85, Part I (1979).

MULTIPLE-EFFECT ABSORPTION CYCLE SOLAR COOLING

THE UNIVERSITY OF TEXAS AT AUSTIN: CENTER FOR ENERGY STUDIES/DEPARTMENT OF MECHANICAL ENGINEERING

CONTRACT: DE AC03-79SF10540

GARY C. VLIET

PERIOD OF CONTRACT: Begin September 1, 1979 End November 30, 1980

## OBJECTIVE

The purposes of this investigation were: to develop a dynamic computer model of the water-lithium bromide double-effect absorption cooling cycle; to use the code to assess the effect on performance of variations in the cycle's several design and operating parameters; and to briefly assess the cost of double-effect absorption cooling per unit of capacity for varying capacity. The work accomplished under this contract is documented in the contract final report [1].

## DESCRIPTION OF WORK

The absorption cycle is one of the alternatives for solar cooling, and the double-effect absorption cycle is a competitor to proposed Rankine cycle solar cooling systems. While the Rankine cycle has the specific advantage of mechanical/electrical output, the double-effect absorption cycle appears to have a potentially better COP when operating purely in the cooling mode. The double-effect cycle has been available for several years, although there has not been significant interest in it for application to solar cooling systems.

Auh [2] provides an excellent overview of absorption cooling. A preliminary assessment of the potential of double-effect water-lithium bromide absorption systems is presented in [3]. The first prototype double-effect unit was developed by Southwest Research Institute in 1956-58 under funding from the American Gas Association [4,5]. Later, the Iron Fireman Company of Cleveland, Ohio, purchased the manufacturing rights and began production of the "Iron Fireman" unit in 1963, a nominal 15-ton, gas-fired unit with a gross COP of 0.9 to 1.0 and a net COP of 1.2 to 1.3 near design capacity. Only a few units were manufactured and sold, and it is no longer in production. In 1973, the Trane Company introduced a line of steam-fired double-effect absorption units with capacities between 385 and 1,060 tons, and COPs in the range of 0.9 to 1.0 near design capacity. The Sanyo Company of Japan previously built a line of double-effect units; however, the line has been discontinued. Yazaki, also of Japan, has announced the introduction of a gas-fired 20-ton unit. Although double-effect units have been and still are available, they do not appear to have reached their performance potential, possibly because they were designed during the availability of cheap energy.

The major objectives of this project were to develop a fairly comprehensive dynamic computer code of the (water-lithium bromide) double-effect absorption cooling cycle, and to use the code to investigate the influence of the several design and operational variables on the cycle performance (coefficient of performance and capacity). Ultimately, it is hoped these results and/or the computer model would be used by manufacturers for optimizing the design of future systems.

## TECHNICAL ACCOMPLISHMENTS

- A physical model of the (water-lithium bromide) double-effect absorption cooling system was defined as shown in Figure 1. It includes seven (7) heat exchangers in all--including the two recovery heat exchangers, two regenerators, condenser, evaporator, and absorber—as well as a solution pump. The model allowed for either an orifice (pressure-driven flow) or a float (refrigerant vapor trap) in the refrigerant condensate stream between the second generator and condenser.

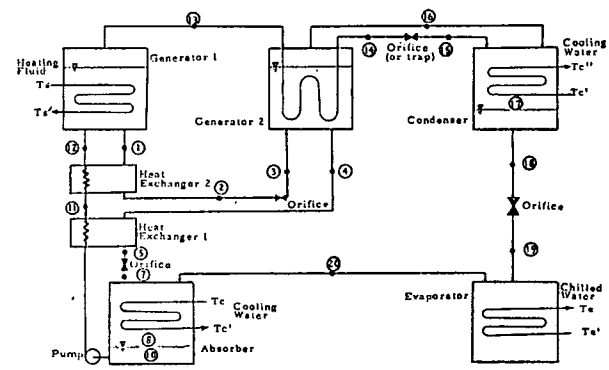


Figure 1. Schematic of Physical Model

- A dynamic computer code was developed around the above physical model. It includes mass, species, and energy balances as well as fluid flow, heat transfer, and mass transfer specifications for the various components and thermophysical property data for the water-lithium bromide system. The computer code has been used successfully to simulate cycle performance over ranges of the several individual design and operational parameters.

- The effects on cycle performance (COP and capacity) were determined for each of the design and operational variables, some of which are discussed briefly below. Most of the results were developed assuming a constant capacity solution pump and a refrigerant vapor trap (or float) between the second generator and condenser. A "nominal" design condition was selected (after several investigative runs) which resulted in reasonably optimal sizes for the various components at the nominal condition. The nominal condition for a 10-ton system is summarized in Table 1. Each of the parameters was then varied individually as all others were held at their nominal values. Table 1 also includes the ranges over which each of the parameters was varied.

  --Figure 2 compares the relative effects on performance of float (trap) control and orifice control over a range of source hot water temperatures (and constant-capacity solution pump).

Table 1. Nominal Conditions and Parametric Ranges

| | Investigated Condition | Nominal Condition* | Range |
|---|---|---|---|
| $T_s$ | Source hot water temperature | 280°F | 220-320°F |
| $T_c$ | Cooling water temperature | 85°F | 50-100°F |
| $T_{ch}$ | Chilled water temperature | 44°F | 34-74°F |
| $m_s$ | Source hot water flow rate | 5,000 lb/hr | 2,000-10,000 lb/hr |
| $m_c$ | Cooling water flow rate | 18,000 lb/hr | 8,000-28,000 lb/hr |
| $m_{ch}$ | Chilled water flow rate | 10,000 lb/hr | 5,000-15,000 lb/hr |
| m(10) | Solution flow rate | 900 lb/hr | 500-1,500 lb/hr |
| $A_{G1}$ | Area of first generator | 35 $ft^2$ | 5-105 $ft^2$ |
| $A_{G2}$ | Area of second generator | 20 $ft^2$ | 5-105 $ft^2$ |
| $A_C$ | Area of condenser | 20 $ft^2$ | 5-105 $ft^2$ |
| $A_E$ | Area of evaporator | 55 $ft^2$ | 5-105 $ft^2$ |
| $A_A$ | Area of absorber | 60 $ft^2$ | 5-105 $ft^2$ |
| $A_{HX1}$ | Area of first heat exchanger | 5 $ft^2$ | 0-20 $ft^2$ |
| $A_{HX2}$ | Area of second heat exchanger | 5 $ft^2$ | 0-20 $ft^2$ |
| c | Pressure drop coefficient between the evaporator and the absorber | $0.5 \times 10^{-6}$ | $0-5 \times 10^{-6}$ |
| | Orifice flow control (varying $T_s$) | | 220-320°F |
| | Centrifugal pump I (varying $T_s$) | | 220-320°F |
| | Centrifugal pump II (varying $T_s$) | | 220-320°F |
| | Orifice flow control with centrifugal pump I (varying $T_s$) | | 220-320°F |
| | Orifice flow control with centrifugal pump II (varying $T_s$) | | 220-320°F |

*The nominal condition corresponds to a cooling capacity of approximately 10 tons.

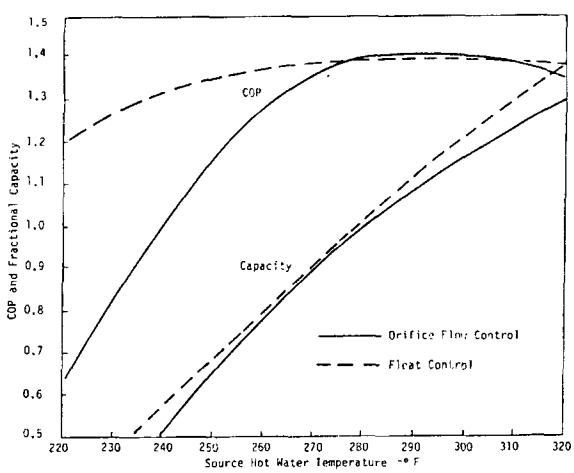


Figure 2. Effect of Source Hot Water on Performance: Comparison of Orifice and Float Control

It is seen that use of a vapor trap maintains the COP (in particular) at a high level over a wider range of source temperatures and also enhances capacity (but to a lesser degree) compared to the orifice flow design.

--Figure 3 shows the combined effects of refrigerant flow control and type of pump characteristics on performance. The use of a centrifugal pump somewhat further degrades the cycle performance, but principally at higher source temperatures. The reason is that for a centrifugal pump with its varying capacity (due to pressure head), the solution circulation rate is reduced at higher source temperatures.

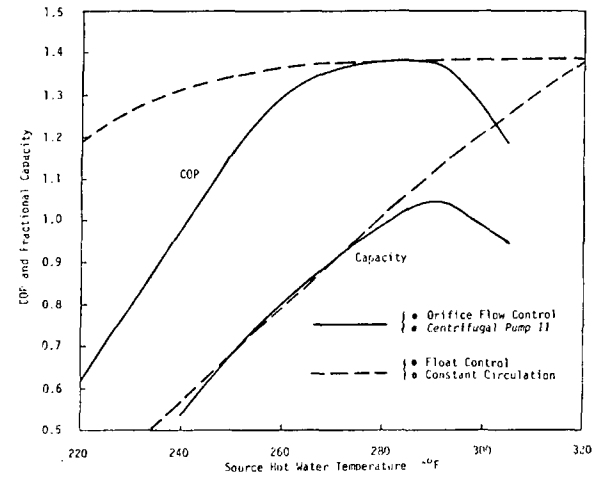


Figure 3. Effect of Centrifugal Pump (Varying Circulation Rate) and Orifice Control on Performance

--Figure 4 shows the effect on performance of the two regenerator heat exchanger areas. Note that the capacity is affected somewhat more by the area of the first generator than by the second, and that COP is not markedly affected by either.

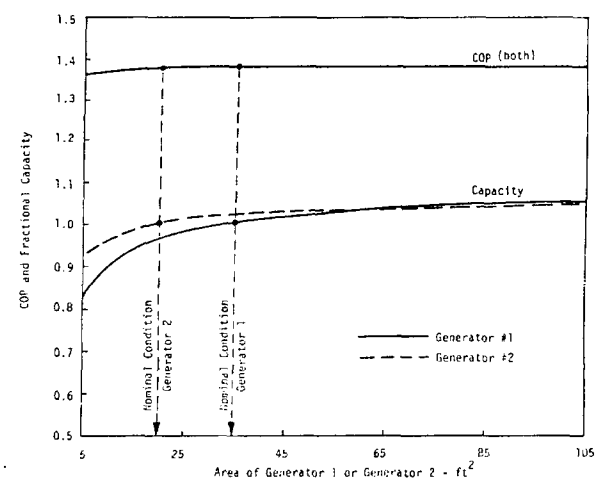


Figure 4. Effect of Each Generator on Performance

Sensitivity data were also obtained for varying condenser, evaporator, and absorber areas (not presented). Figure 5 shows the effect on performance of the two recovery heat exchangers. The capacity is seen to be little affected for either heat exchanger; however, the COP is significantly affected, and about equally by each.

--Figure 6 shows typical curves for performance (COP and capacity) for a specified 50°F exit chilled water temperature and varying source and cooling water temperatures. These data are for constant solution circulation and float (trap) control. The locus of the crystallization condition is indicated.

- Some specific conclusions about optimal system design evolve from these results. (a) The trap or float control is much preferred to the orifice

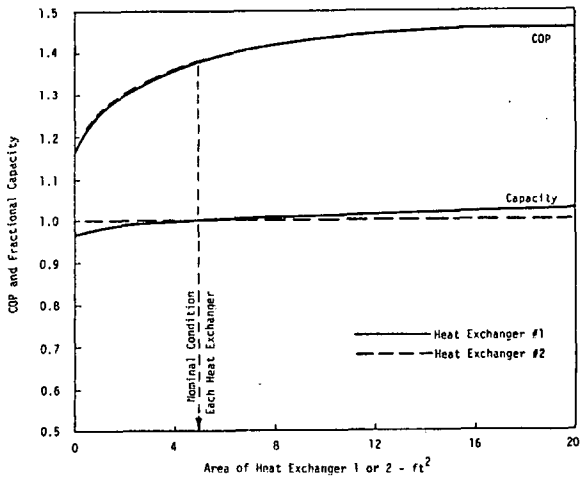


Figure 5. Effect of Each Recovery Heat Exchanger Area on Performance

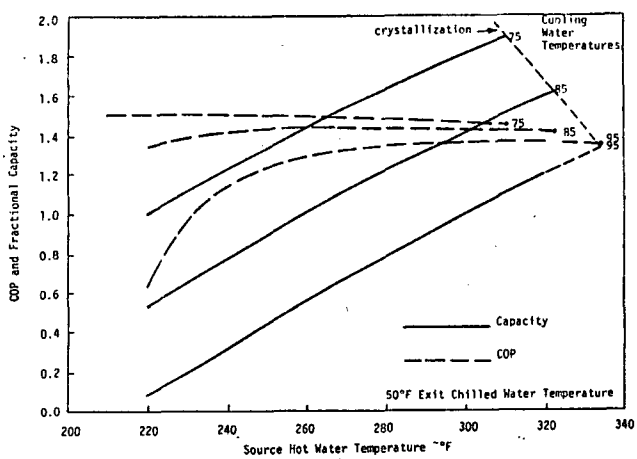


Figure 6. Cycle Performance for 50°F Exiting Chilled Water Temperature

flow design in terms of maintaining high performance over a wide range of source temperatures. This capability is particularly important for a solar-driven system, where the source temperatures will vary considerably. (b) The solution pump characteristic is important, particularly at high source temperatures, with one approaching constant displacement preferred. (c) The relative allocation of heat exchanger area to the seven heat exchangers is very important to cycle performance and cost, since a large portion of the cost of absorption systems is in heat exchange surface.

- A secondary objective of the study was to assess the cost of absorption cycle cooling per unit of design capacity. Figure 7 shows a summary of data for single- and double-effect units. The cost of small tonnage (single-effect systems) appears excessive, but it would appear that there is merit in considering intermediate capacity double-effect systems in the range of 20 to 100 tons. Modularizing, as done by Yazaki, has merit in flexibility, but unit cost is high. A comparison of the Trane single- and double-effect large capacity units is interesting. Their relative

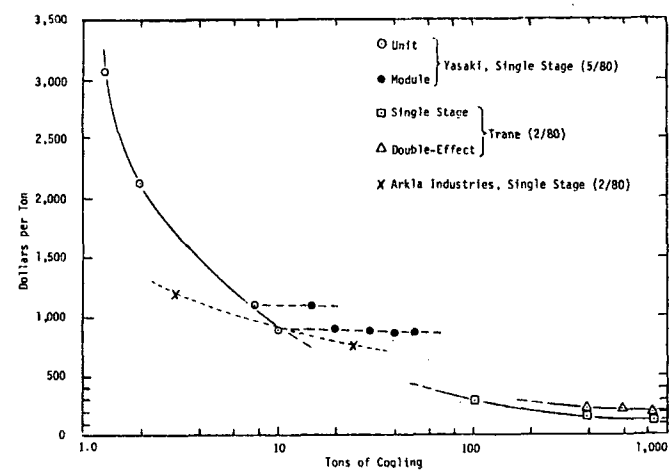


Figure 7. Unit Capacity Cost Versus Capacity of Water-Lithium Bromide Absorption Systems

costs are about equal to their relative COPs. Since the increased capital costs of the double-effect units (over single) are probably considerably less than the incremental collector cost for solar systems or life-cycle fuel costs for conventional systems, it would appear that the double-effect cycle is a viable choice.

## FUTURE ACTIVITIES

The availability of this dynamic computer code for the (water-lithium bromide) absorption cooling cycle provides a tool for manufacturing and systems analysts to develop more optimal designs.

It has been proposed that some additional effort be put into the computer code to ensure that it has the most current thermophysical property data, to provide an improved physical model for the absorber, and to better document the code. In addition, a more in-depth assessment of the allocation of heat transfer area to the different components should be made, and a direct comparison between the model's predictions and the performance of commercially available unit(s) is needed. It is also proposed that the model, results, and advice on using or modifying the model be made available to interested manufacturing firms.

## PUBLICATIONS/REPORTS/REFERENCES


1. Vliet, Gary C.; Lawson, Michael B.; and Lithgow, Rudolfo A., Water-Lithium Bromide Double-Effect Absorption Cooling Analysis Final Report on DOE Contract DE AC03-79SF10540, (Austin: The University of Texas at Austin Center for Energy Studies, December 1980).

2. Auh, Paul C., A Survey of Absorption Cooling Technology in Solar Applications, Report Number BNL50704 (Upton, New York: Brookhaven National Laboratory, July 1977).

3. Vliet, Gary C., and Saiidi, Mohammad J., "Double Effect Absorption Cooling with Solar Energy," Proceedings of the Third Workshop on the Use of Solar Energy for Cooling of Buildings, 1978.

4. Whitlow, E. P., and Swearingen, E. P., "An Improved Absorption Refrigerant Cycle," J. Gas Age, October 1958.

5. Versagi, Frank J., Technical Conversations in Air Conditioning and Refrigeration (Detroit: Business News Publishing Company, 1962).

LOW-GRADE-HEAT-ACTUATED ABSORPTION
HEAT PUMPS FOR TEMPERATURE BOOSTING

ENERGY DIVISION, OAK RIDGE NATIONAL LABORATORY *
W-7405-eng-26/AA 55 15 15 0
Gershon Grossman and Horacio Perez-Blanco
PERIOD OF CONTRACT: Begin October 1979 End Open

## OBJECTIVE

A conceptual design of an industrial-type double-stage absorption heat pump was developed as the first step toward the construction of a laboratory working system. The heat pump is designed to upgrade low-temperature waste heat by boosting its temperature typically from about 60°C (140°F) to about 120°C (250°F). The system uses part of the low-temperature heat as its energy source and requires no primary energy, except for small auxiliary equipment. A detailed computer simulation of the system was developed to perform this study.

## CONCEPT AND BACKGROUND

The U.S. industrial sector rejects annually $2.85 \times 10^6$ TJ of heat in the form of condensate, cooling water, and process water (at temperatures between 40 and 80°C). This amount is roughly 3% of the U.S. annual energy consumption. Although the thermodynamic availability of this reject heat is low, its large amount justifies studying means of recovering it for useful purposes. To this end, and particularly for use as process heat, it is often necessary to upgrade its temperature. Absorption heat pumps seem most attractive for this application since they rely on the waste heat itself as their source of power.

Several versions of the absorption heat pump cycle may be used to achieve different degrees of temperature boosting under various operating conditions. A schematic of a simple, one-stage closed cycle absorption heat pump is shown in Figure 1.

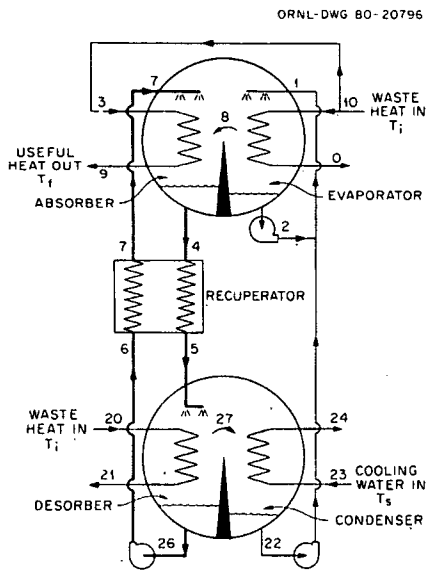


Figure 1. Schematic of a one-stage, closed-cycle heat pump.

Steam is generated in an evaporator (pts. 1 to 8), by applying waste heat at temperature $T_i$ as a heat source. The steam is absorbed by a concentrated LiBr-water solution, releasing useful heat at a higher temperature than $T_i$ (pts. 7 to 4). Transferred to a water stream (state 3), this heat boosts the stream temperature from $T_i$ to $T_f$. The diluted solution (pt. 4) passes through a recuperative heat exchanger, into the desorber where it is reconcentrated by evaporating water from it through the application of more waste heat at temperature $T_i$. The solution leaving the desorber (pt. 26), returns to the absorber via the recuperative heat exchanger (pts. 6 to 7). The steam leaving the desorber is condensed (pts. 27 to 22), and the condensate returned to the evaporator (pt. 1).

Generally, the temperature boosts needed for industrial applications are too large to be achieved in one stage. Larger temperature boosts, on the order of 60°C, may be achieved by means of a two-stage cycle, as illustrated in Figure 2. Here, the condensate (pt. 1) is evaporated in the first stage evaporator by means of waste heat (pts. 1 to 2). The steam produced (pt. 8) is absorbed by a LiBr solution, releasing heat (pts. 7 to 4). The dilute solution (pt. 4) returns to the desorber via a recuperative heat exchanger. The heat released in the first stage is used to heat the stream of water (pts. 3 to 9) which is divided into two fractions. One circulates in the second-stage evaporator where it releases heat to evaporate more condensate (pts. 11 to 12). The other is heated further in the second-stage absorber by the heat of absorption (pts. 13 to 19) and released as hot process water. The weak solution from the second stage (pt. 14) is returned to the desorber where it is concentrated using waste heat as a heat source (pts. 25 to 26). The evaporated water is condensed (pt. 22) and sent back to the evaporators.

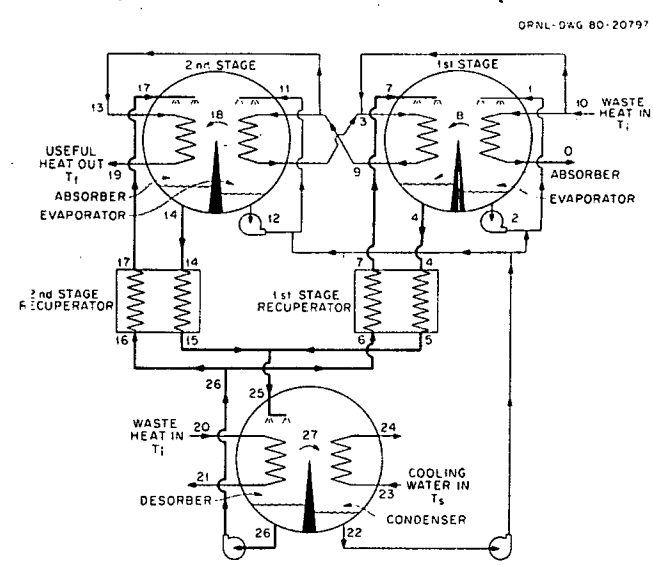


Figure 2. Schematic of a double-stage, closed cycle absorption heat pump.

*Research sponsored by the U.S. Department of Energy under contract W-7405-eng-26 with the Union Carbide Corporation.

In a real situation, a temperature drop is needed across heat exchangers in order to ensure good heat transfer. If instead of closed evaporators as in Figure 2, flash chambers are employed (Figure 3), a closer thermal equilibrium between the steam and the water leaving the chamber may be obtained. If the water is flashed from 60°C to 54°C, a mass of steam at state 8 equal to 1% of the water mass would be produced. The temperature boost possible with a flash chamber is, therefore, larger than the one possible with a closed evaporator.

Another limitation of the closed absorption cycle takes place in the desorber. Its operating conditions are determined by the heat supply temperature and vapor pressure in the condenser which determine the high concentration obtainable in it. When the waste heat temperature and cooling water temperature are close to each other, it is hard to obtain any regeneration effect.

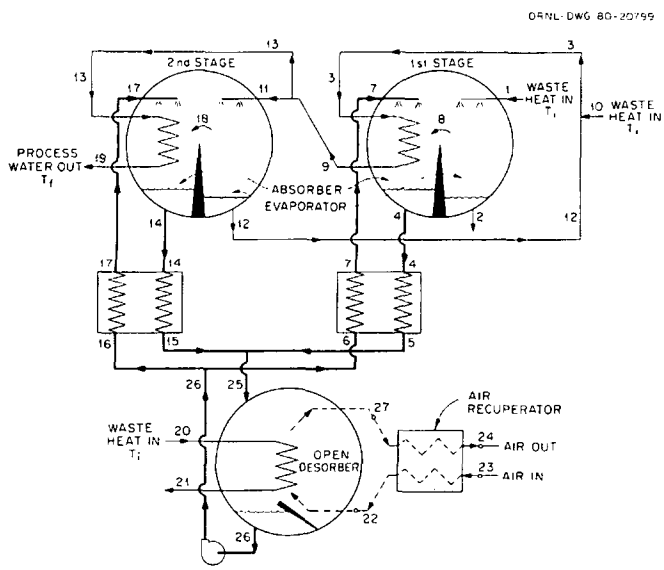


Figure 3. Schematic of a double-stage, open-cycle absorption heat pump.

If the vapor pressure in the condenser could be reduced, it would be possible to increase the maximum cycle concentration and preserve the temperature boost. When cooling water at a low enough temperature is not available, the solution water may be directly evaporated into atmospheric air whose water vapor pressure may be as low as 18 mm Hg on hot, humid summer days (32.2°C dry bulb and 29°C wet bulb).

The heat pump cycle with an open desorber may take on different configurations. A cycle with flash evaporators and an open desorber may be implemented as illustrated in Figure 3. The steam entering the device in the flash chambers is evaporated into a stream of air after serving its useful purpose. If cooling water at low enough temperatures is available, it is possible to replace the open desorber with a closed desorber/condenser.

While the absorption cycle has been used extensively in cooling systems, very little work has been done on its application for temperature boosting. Several absorption systems were studied for upgrading waste heat, which were powered by a high temperature heat source. Only two or three are known which were low-grade heat actuated. Hardly any work was done on multi-staging and other design aspects necessary for systems with a small temperature difference between the heat source and sink. The main effort in this work was directed toward this application.

## SUMMARY

The present study has resulted in a conceptual design of the heat pump system providing the information needed for the detailed design and construction of a laboratory working system. The basic cycle was studied for use with low-temperature waste heat (60°C). Multistaging of the heat pump was investigated for increased temperature boosting and a preferred two-stage configuration was identified. Open cycle evaporation and desorption was considered for cases of low temperature differential between the heat source and sink. Suitable working materials were considered and components performance was analyzed at the conceptual design level. The overall system performance was studied for varying operating conditions. Several improvements in the cycle were considered.

A complete computer model of the system was developed to perform the above study. The code is available for investigation of different configurations and operating conditions, and will be used in the detailed design of the laboratory system.

## TECHNICAL ACCOMPLISHMENTS

- Study of basic cycle and performance criteria: The performance of the absorption heat pump is characterized by several parameters which should be considered in the design of an efficient working system. These depend on two given operating conditions: The temperature of the source of heat to be upgraded ($T_i$) and the heat rejection environment as determined by the cooling water or ambient air, whichever is applicable.

The first and perhaps most important performance characteristic is the output temperature ($T_f$) which the system can deliver. Clearly, a large temperature boost is desirable. Another important criterion is the overall coefficient of performance (COP) of the system, defined as the ratio between the heat output and the energy input. The latter consists of two parts: Input from the heat source and parasitic power.

A key consideration in the design of the heat pump system is the price of the heat to be upgraded. If this heat is taken as being abundantly available and free, the system should be designed so as to minimize the parasitic power requirements, at the possible cost of a low thermal COP. A typical example of this case is waste heat, which not only is free, but often entails an additional cost to dispose of. With a source of heat such as solar or geothermal, the thermal COP is of greater importance.

The last, but not least, effectiveness criterion of the system is its initial cost. As in many similar cases, good thermal performance calls for the temperature differentials in the heat exchangers to be kept to a minimum. The designer must weigh this consideration against the cost of the heat exchangers which are a major part of the overall system's cost.

- Selection of working fluids: The properties required of refrigeration/absorbent combinations for use in absorption cooling systems have been discussed extensively in the literature. Many of those requirements are equally applicable to a heat pump

system of the type discussed here. The most important characteristics may be summarized as follows: (1) The equilibrium temperature of the absorbent/refrigerant solution should be as high as possible at the absorber conditions and as low as possible at the desorber. (2) The solubility of the refrigerant in the absorbent must be as high as possible. (3) The properties of the absorbent-refrigerant solution which affect heat and mass transfer should be favorable. (4) The absorbent should be nonvolatile, or considerably less volatile than the refrigerant, to prevent the transfer of absorbent into the condenser or regenerating air. (5) The saturation concentration of the absorbent/refrigerant solution should be as high as possible and beyond the range of operating concentrations of the system. (6) The working fluids should be chemically stable, nonflammable, nontoxic, and noncorrosive to common materials of construction, and available at a reasonable cost.

Of the many fluid pairs considered and experimented with for absorption systems, only a few were found practical, having satisfied most of the above criteria. Water-ammonia and lithium bromide-water have been used extensively in closed-cycle cooling systems, each with its own merits and disadvantages. In the former pair, ammonia is toxic and the absorbent (water) is volatile, requiring a rectifier between the desorber and condenser. The lithium bromide-water combination is corrosive to metals and the salt crystallizes out of solution at high concentrations. These problems have been dealt with effectively in closed-cycle cooling systems, and both fluid pairs seem suitable for absorption heat pumps. The lithium bromide-water pair seems more promising, since in the water-ammonia combination, the working pressures are high even in cooling applications, and become much higher at the heat pump range of working temperatures to the point where construction may become impractical.

Open cycle absorption cooling systems have invariably employed water as the refrigerant, with various absorbents including lithium bromide, lithium chloride, calcium chloride, ethylene glycol, as well as several types of solid absorbents of water. Again, of the many candidate materials, two emerge as being most suitable for the heat pump system. These are the lithium bromide and lithium chloride absorbents, both nonvolatile and satisfying most of the above criteria. Another possible candidate is calcium chloride, which although an inferior absorbent as compared to the other two, is less expensive and less corrosive than the lithium salts.

The present study considers LiBr-water and LiCl-water as working materials for the heat pump system, both in the closed- and open-cycle configurations.

- Evaluation of absorber/evaporator system performance: The basic unit of the absorber/evaporator system in a multistage absorption heat pump consists of one evaporator, one absorber, and one recuperative heat exchanger (see Figure 3 for description and notation). In the following analysis, the high and low absorbent concentrations ($C_H$ and $C_L$, respectively) are preselected. The former is determined by the conditions in the desorber; the latter is a variable for performance optimization, the effect of which will be evaluated and discussed. The temperature of the strong solution entering the recuperator (state 6) is also fixed by the desorber. The properties at all other states may be calculated as functions of the above parameters and of the temperature of the heat source to be upgraded ($T_i$). Heat exchanger effectivenesses $\varepsilon_a$ and $\varepsilon_r$ are assumed for the absorber and recuperative heat exchanger, respectiely. Two basic assumptions are made: (1) the weak solution leaving the absorber (state 4) is in vapor pressure equilibrium with the vapor coming from the evaporator (state 8) and (2) pressure drops in the system are negligible.

For the given concentrations and temperatures, a set of 15 equations may be written to describe the process in the absorber/evaporator system. This set consists of heat and mass balances for the evaporator, absorber, and recuperator and of temperature relations based on the heat exchange effectivenesses in the different units. The enthalpies of the solution at the different states may be expressed in terms of the temperature and concentration. With the inlet flowrates and inlet temperatures given, the above set of equations may be solved for the temperatures and flowrates at all states.

A similar system of equations may be written for the second stage of the absorber/evaporator system. With the flowrate of the output stream $m_{19}$ selected arbitrarily as a measure of size of the system, and with the given inlet temperatures $T_1 = T_{10} = T_i$, $T_5 = T_{16} = T_g$, there are altogether 36 equations for the 38 unknown temperatures and flowrates at points 1 through 19. Two of the latter can hence be selected as free parameters for optimization.

A computer program has been written to carry out the above calculation. In selecting $m_{19}$ as unity, the temperature boost and the heat gain of the output stream are given by the same expression. It makes physical sense to select as the free optimization parameters the evaporation temperatures in the first and the second stage, $T_8$ and $T_{18}$, respectively. These are also the saturation temperatures corresponding to the pressures prevailing in the two stages. Figure 4 shows an operating map of the absorber/evaporator system in terms of these parameters. The figure describes the temperature boost ($T_f$-$T_i$) as a function of the mass ratio between the rejected and useful streams ($m_2/m_{19}$). The working material is lithium bromide-water. The heat source temperature is fixed at $T_i$ = 60°C (140°F) and the solution from the desorber is at 54% and 57°C (134°F). Two sets of curves are shown for the low concentration at 50% and 52%.

It is clear from the operating map that the higher the evaporation temperature at both stages, the higher the temperature boost. For a given $T_8$, raising $T_{18}$ results in a significant rise in temperature boost without greatly increasing the mass of the rejected stream. The opposite is not as desirable: increasing $T_8$ at fixed $T_{18}$ raises the temperature boost only slightly, and increases the mass rejected considerably. However, while $T_8$ and $T_{18}$ can be selected independently, there is an upper limit on each. The upper limit for $T_8$ is the heat source temperature $T_i$, and bringing the former close to the latter results in a rapid growth of the rejected stream $m_2$. For each $T_8$, the upper limit on $T_{18}$ is set by the condition that the water temperature in both absorbers is always lower than that of the solution. A good choice would therefore be to select $T_8$ somewhat below $T_i$ so as to have a reasonable $m_2$, and then select $T_{18}$ somewhat below its limit corresponding to $T_8$ so as to have a practical $m_{12}$.

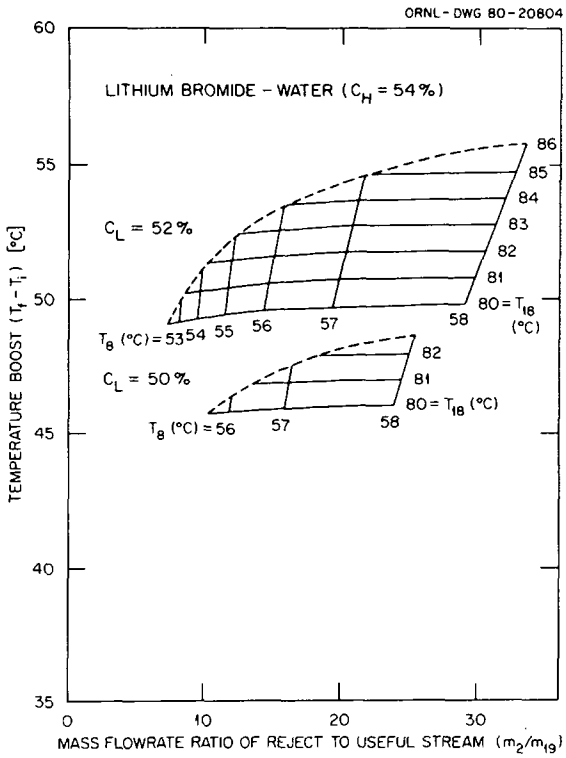


Figure 4. Operating characteristics of the evaporator/absorber system.

The effect of the low concentration is also evident from Figure 4. Lowering the concentration from 52% to 50% reduces the temperature boost by about 3.5°C, for the same values of $T_8$ and $T_{18}$. It also reduces the maximum $T_{18}$ attainable for each $T_8$. The rejected stream $m_2$ is reduced somewhat, indicating a better heat gain of the upgraded stream under the lower $C_L$.

An analysis similar to the above has been performed for the system with lithium chloride-water as the working fluid. The high concentration of this solution has been selected at 41%, which has the same vapor pressure at 57°C as a 54% LiBr solution when leaving the desorber. The lithium bromide solution gives a larger temperature boost than the lithium chloride when desorbed under the same conditions.

- Evaluation of desorber performance: The performance of the desorber depends on the heat supply temperature and on the heat rejection environment as manifested by the vapor pressure and the corresponding saturation temperature. Ideally, the former temperature should be as high as possible and the latter as low as possible to achieve maximum concentration. In many conventional absorption units where the heat supply temperature is not limited, a closed desorber is used in conjunction with a condenser. This is generally a good arrangement with many advantages. However, as explained before, when the heat supply temperature is low as is often the case with low grade heat, and when the ambient temperature is not sufficiently low to provide adequate cooling for the condenser, one would have to resort to an open desorber.

In the following analysis (refer to Figure 2 for notation), heat exchanger effectivenesses $\varepsilon_d$ and $\varepsilon_c$ are assumed for the desorber and condenser, respectively. A set of 11 equations may be written to describe the process in the desorber/condenser system. The set consists of heat and mass balances for the desorber and condenser, and of temperature relations based on the heat exchanger effectivenesses. The enthalpies of the solution at the different states may be expressed in terms of the temperature and concentration. By specifying the inlet conditions of the solution ($m_{25}$, $T_{25}$, and $C_L$), the high concentration required ($C_H$), and the inlet hot and cooling water temperatures ($T_{20}$ and $T_{23}$), the equations may be solved for the flowrates and temperatures of all states, with one free parameter for optimization. It makes physical sense to select as that parameter the condensing temperature $T_{22}$, as was done in the analysis of the absorber/evaporator system.

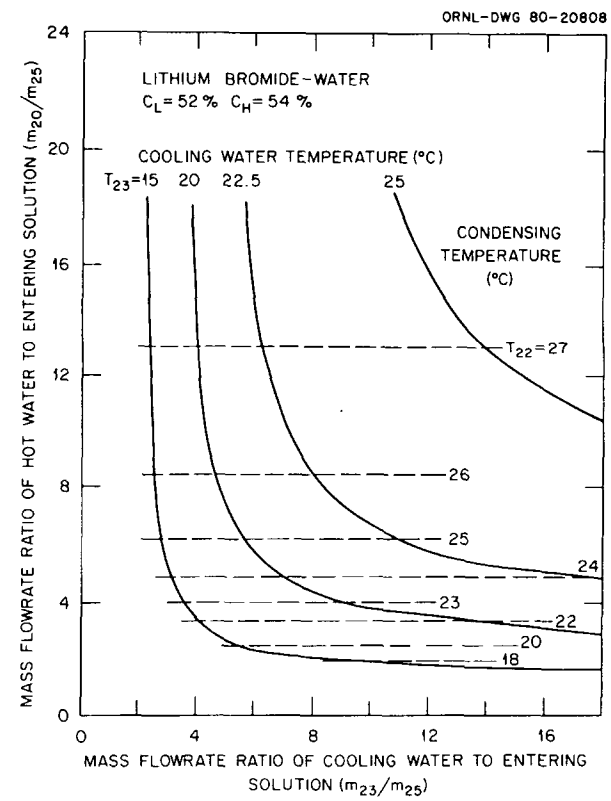


Figure 5. Operating characteristics of the closed desorber/condenser system.

Figure 5 shows an operating map of the desorber with LiBr-water, obtained from a computer solution of the equations. The relative flowrates of hot water and cooling water are plotted for different cooling temperatures (the waste heat temperature is fixed at 60°C). The inlet low concentration and required high concentration are 52% and 54%, respectively, and the effectiveness of both the desorber and condenser is 0.80. It is clear that the higher the cooling water temperature, larger and larger flowrates of both cooling and hot water are required to produce the desired change in concentration. The upper limit on the condensing temperature is set by the required high concentration and the waste heat temperature available. Figure 5 also shows the hot water flowrate to be dependent directly on the condensing temperature $T_{22}$, and to increase very rapidly as the limit on $T_{22}$ is approached. The higher $T_{22}$, the smaller the amount of cooling water required.

An operating map similar to that of Figure 5 was calculated for a LiCl-water solution. Here again, as in the absorber/evaporator analysis, the high concentration of the solution has been selected at 41%, which yields the same vapor pressure at 57°C as LiBr at 54%. For the same condensing temperature, larger quantities of both cooling and hot water are required to produce a 2% change in concentration for LiCl than for the LiBr. This is to be expected since the amount of water desorbed is higher in the case of LiCl, being 0.0488 Kg per Kg of entering solution as opposed to 0.0370 Kg for LiBr. The ratio $m_{20}/m_{23}$ remains about the same for both working materials.

When an open desorber is used instead of a closed one, performance calculations become more involved because a two-phase, three-component heat and mass transfer process takes place. In the simplified model shown in Figure 3, the solution enters the desorber, where water evaporates from it in direct contact with the air, flowing in counter current to the solution. Waste heat is added continuously to the solution in order to keep both its temperature and vapor pressure high so that evaporation can proceed. If waste heat was not added during this process, the solution temperature and vapor pressure would decrease rapidly until equilibrium with the air vapor pressure was reached. At this point, all mass exchange would cease. The situation idealized in this model may be achieved in practice in a packed tower with heat-exchanging coils immersed in the packing, or in a falling film reactor with counterflow of air.

The open desorber calculations show the performance to depend on the following parameters for a given concentration and waste heat temperature: the inlet air moisture, the approach temperature between the air and solution, and the effectiveness of the air-to-air recuperator. The moisture determines the vapor pressure against which water is being desorbed from solution. The above parameters determine the required air-to-solution and hot water-to-solution mass ratios required for a given concentration change.

The results of the open cycle desorber calculations are not fully described in this paper. They indicate, however, that in the absence of cooling water or when cooling water temperature is too close to that of the waste heat, the open cycle desorber can offer an alternative to the closed one.

- Evaluation of overall cycle performance: A computer program similar to those described previously was used to evaluate the overall performance of the system. In varying the low and high system concentrations, several behavior patterns of interest were observed. As expected, the temperature boost is almost independent of $C_H$, and determined primarily by the low concentration. The temperature boost increases with $C_L$, and, of course, with the heat exchanger effectiveness. The coefficient of performance turns out to be sensitive to the concentration difference. If $C_H$ is kept constant, one can achieve a significant increase in COP by lowering $C_L$, at the expense of a reduced temperature boost. The change in COP is more significant for a system with low heat exchanger effectiveness. There the losses due to solution circulation are highest, and therefore, a larger gain is achieved by increasing the concentration difference.

Figure 6 shows the effect of the waste heat temperature and cooling water temperature on the performance of the heat pump. The COP is plotted as a function of the temperature boost for different values of the above temperatures, and the tradeoff between

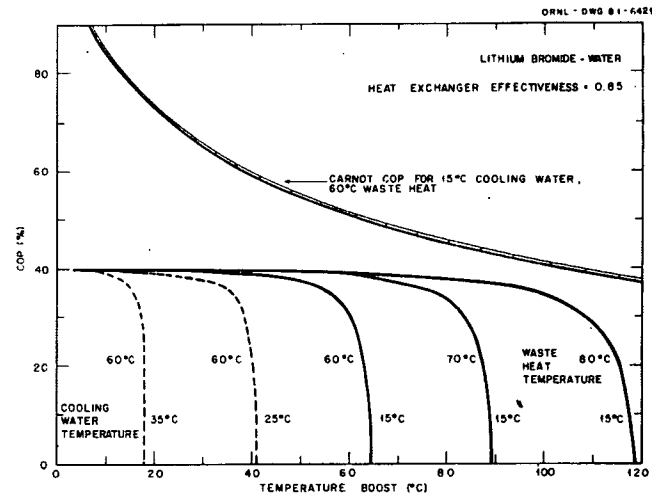


Figure 6. Performance characteristics - COP and temperature boost - for a two-stage system for different heat source and sink temperatures.

these two performance criteria is evident. The working fluid here is lithium bromide-water and the effectiveness of all heat exchangers is 0.85. The condensing temperature is assumed to be 5°C higher than the cooling water temperature; similarly, the high solution temperature in the desorber is 5°C lower than the waste heat temperature. It is clear from the figure how raising the waste heat temperature increases performance and raising the cooling water temperature reduces it. Trading off a lower temperature boost against a higher COP is achieved by reducing the low concentration. There is a limit to that, as the COP reaches an asymptotic value of about 40%. The theoretical COP obtainable with a Carnot heat pump operating with 60°C waste heat and 15°C cooling water is shown for comparison with the corresponding absorption heat pump curve. Figure 6 may be regarded as a summary of all previous calculations, describing the overall system performance for different operating conditions.

## FUTURE ACTIVITIES

With the conceptual design study essentially completed, several refinements are being introduced in the computer program enabling it to perform more detailed calculations of the open cycle regeneration.

The next step involves a detailed design leading to the construction of a laboratory working system. The system will use in its first version, a closed desorber coupled with a condenser, which will operate with low-temperature cooling water such as may be available in the winter or from a cold well. Next, an open desorber will be incorporated in the system.

## PUBLICATIONS/REPORTS/REFERENCES


Publications related directly to present study:
Perez-Blanco, H., and Grossman, G., Cycle and Performance Analysis of Absorption Heat Pumps for Waste Heat Utilization, ORNL/TM report, 1981 (in preparation).

1. Blue, J., and Arehart, J., "A Pocket Reference of Energy Facts and Figures," Oak Ridge National Laboratory, Energy Division, December 1980.

2. Chiogioji, M. H., Industrial Energy Conservation, Marcel Dekker, Inc., 1979.

3. Perez-Blanco, H., Heat Pump Concepts for Industrial Use of Waste Heat, Oak Ridge National Laboratory, ORNL/TM-7655, 1981.

4. Reay, D. A., Industrial Energy Conservation, Pergamon Press, Oxford, 1977.

5. Isshiki, N., "Study on the Concentration Difference Energy System," J. Non-Equilib. Thermodynamo, 2, 85-107 (1977).

6. Shwarts, I., and Shitzer, A., "Solar Absorption System for Space Cooling and Heating," ASHRAE J., 51-54 (November 1977).

7. Knoche, K. F., and Stetmeier, D., "Absorption Heat Pumps for Solar Space Heating Systems," Studies in Heat Transfer, Hemisphere Publishing Corp., Washington, 1979.

8. Sauer, K. L., "Absorption Type Heat Pump," Sonnenenergie Waermepumpe, 4 (2/3), 15-19 (1979).

9. Anon, "Heat Transformer Makes Waste Heat Efficient Source," Des. News (Oct. 22, 1979).

10. Cohen, G., Salvat, J., and Rojey, A., "A New Absorption-Cycle Process for Upgrading Waste Heat," Proceedings of the 14th Intersociety Energy Conservation Conference, 2, 1720-1724, Boston, Mass., Aug. 5-10, 1979.

11. Perez-Blanco, H., and Chen, F., "Some Heat Pump Concepts for Residual Heat Utilization," Third Miami International Conference on Alternative Energy Sources, Miami Beach, Fla., Dec. 15-17, 1980.

12. Best, R., "Methanol-Lithium Bromide, Zinc Bromide Mixture Applied to Solar Absorption Heat Pumps," Proceedings of the International Conference on Solar Building Technology, 2, 437-443, London, England, July 25, 1977.

13. Hiller, C., "Development and Testing of the Sulfuric Acid-Water Chemical Heat Pump/Chemical Energy Storage System," Proceedings of the 14th Intersociety Energy Conversion Engineering Conference, 1, 510-515, Boston, Mass., Aug. 5-10, 1979.

14. Offenhartz, P., and Brown, F., "Methanol Based Heat Pumps for Storage of Solar Thermal Energy," Proceedings of the 14th Intersociety Energy Conversion Engineering Conference, 1, 507-509, Boston, Mass., Aug. 5-10, 1979.

15. Factor, H. M., and Grossman, G., "A Packed Bed Dehumidifier/Regenerator for Solar Air Conditioning with Liquid Dessicants," Sol. Energy, 24, 541-550 (1980).

16. Collier, R. K., "The Analysis and Simulation of an Open Cycle Absorption Refrigeration System," Sol. Energy, 23, 357-366 (1979).

17. Buffington, R. M., "Qualitative Requirements for Absorbent-Refrigerant Combinations," Refrig. Eng., 57, 343 (1949).

18. Hainsworth, W. R., "Refrigerants and Absorbents," Refrig. Eng., 48, 97 (1944).

19. Wilbur, P. J., and Karaki, S., Solar Cooling, The Franklin Institute Press, 1977.

20. Grossman, G., and Shwarts, I., "An Open Absorption System for Solar Air Conditioning," Proceedings, the Conference on Heat Transfer in Buildings, 2, 641-647, Dubrovnik, Yugoslavia, August-September 1977.

SOLAR HEATING AND COOLING WITH THE $CaCl_2$-$CH_3OH$ CHEMICAL HEAT PUMP

EIC LABORATORIES, INC.
55 CHAPEL STREET
NEWTON, MASSACHUSETTS 02158
CONTRACT NUMBER 490105-S
PETER O'D. OFFENHARTZ
APRIL 1977-SEPTEMBER 1980

## OBJECTIVES

A chemical heat pump based on the reaction of calcium chloride and methanol is being designed and optimized for solar heating and air conditioning, primarily for the residential and light commercial market. The performance requirements for this application are quite stringent. For example, to minimize maintenance, a cooling tower should not be used, and the solar collectors should be fixed rooftop flat plates or evacuated tubes. The chiller should be capable of reaching 45°F on a 95°F day in order to provide effective dehumidification. Energy storage, for late afternoon and early evening cooling, as well as night time winter heating, must be provided. Heating operation at ambient temperatures well below freezing must be possible, and at the same time heating air temperatures should be kept above 100°F so that fan power consumption and heating air velocity are reduced. Use of "parasitic" electrical power must be minimized so that operating costs are low, and electricity must not be used as a backup fuel, which would place an unreasonable burden on the electric utility peak load profile. Above all, system costs must be sufficiently low to make the system economically attractive to both homeowners and investors. Taken together, these requirements define the objectives of EIC's research and development program.

## CONCEPTUAL BACKGROUND

The $CaCl_2$-$CH_3OH$ chemical heat pump belongs to the class of intermittent absorption cycles based on solid-gas reaction. The absorber, anhydrous $CaCl_2$, reacts with $CH_3OH$ vapor to form the solid-phase compound $CaCl_2 \cdot 2CH_3OH$. Unlike liquid-based absorption cycles, the absorber affinity for vapor does not decrease as the reaction takes place, so that heating or cooling performance stays substantially constant while the storage capacity of the cycle -- the quantity of unreacted $CaCl_2$ -- decreases. However, because a solid-phase absorber cannot be readily recirculated, the absorption and desorption reactions are carried out in a batch process fashion, using two separate beds of solid. While one bed absorbs vapor, producing the desired heating or cooling effect, the second bed uses solar energy to desorb vapor, producing anhydrous $CaCl_2$ and $CH_3OH$, which condenses in a separate tank. When either the absorption or desorption reaction reaches effective completion, the roles the two beds are reversed, with absorption and desorption functions swapped. A schematic diagram of the basic cycle is shown in Fig. 1.

Methanol is an excellent refrigerant for absorption cycles. The heat of vaporization per unit mass is exceptionally high, 1.75 kJ/g, yet the vapor pressure is not so low as to cause problems due to pressure drops, as is often the case with water. Furthermore, no freeze protection is needed. Flammability and toxicity are potential problems, but since the system vapor pressure is subatmospheric, there can be no substantial vapor escape. Of course, the liquid inventory must be placed outdoors, as is conventionally done with LPG.

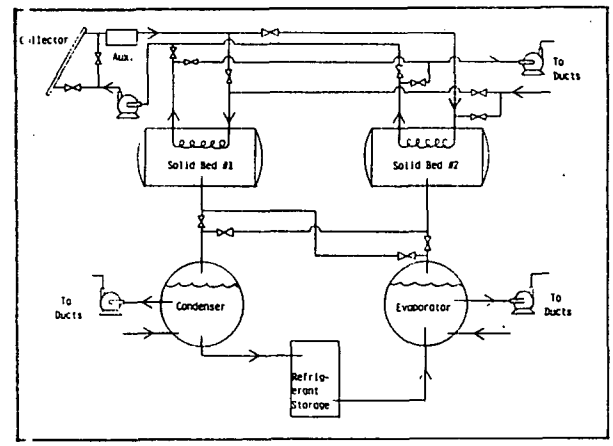


Fig. 1. Schematic design of the $CaCl_2$-$CH_3OH$ chemical heat pump.

Calcium chloride is an equally excellent absorbent. The thermodynamics of the $CaCl_2$-$CH_3OH$ reaction are such that heat can be pumped "uphill" over a gradient in excess of 110°F. Absorption rates are sufficiently rapid that practical heat pumping can be achieved with an absorber-evaporator temperature difference in excess of 100°F. Three practical consequences are immediately apparent: (1) Heat can be pumped in winter even on cold days, and while maintaining air duct temperatures in excess of 100°F; (2) Heat rejection in the cooling mode does not require a cooling tower or other "wet" system; (3) Required solar collector temperatures are higher than for other absorption cycles with less temperature upgrading capability. This last point is a potential disadvantage of the $CaCl_2$-$CH_3OH$ cycle, which requires solar collector temperatures around 250°F. Nevertheless, such temperatures are well within the range of fixed evacuated tube collectors, which appear to be by far the most cost-effective type of collector for residential absorption cycles.

Calcium chloride and methanol are entirely noncorrosive toward aluminum and copper. Thus all components can be constructed using standard materials. This is a decidedly important feature of the cycle. Furthermore, the $CaCl_2$-$CH_3OH$ reaction occurs without by-products or side reactions, so that maintenance associated with purging the system should be small or non-existent.

## SUMMARY

EIC Laboratories began its absorption cycle investigations in 1977 with a study of the thermodynamics and kinetics of a number of salt-methanol reactions. Early in the program, $CaCl_2$ was selected for detailed investigations. The work has now progressed to the design, construction and successful operation of a 1/3 scale engineering development test prototype with 100,000 BTU storage capacity. The cooling coefficient of performance (COP) of this unit is 0.52, and

the heating COP is about 1.5. These are full-cycle "round-trip" COPs, not the usually reported "steady state" COPs, which are generally higher. Ultimately, COPs of 0.6 for cooling and 1.6 for heating should be possible.

During the course of development, extensive work has been carried out on the reaction thermodynamics and kinetics, on heat transfer and heat exchanger design, on cycling rates and temperatures; on detailed computer simulations of expected performance in different climates; and on economic analyses. These studies indicate that the $CaCl_2$-$CH_3OH$ cycle should prove highly cost-effective when used in mid-latitude U.S. residences. In the southern tier of states, when residential energy costs are dominated by cooling, water-based cycles with higher cooling coefficients of performance will probably prove more competitive. In the northern tier, all heat pump cycles encounter problems due to extreme winter cold. The market for the solar-driven $CaCl_2$-$CH_3OH$ cycle lies in between, where the costs of heating, cooling, and domestic hot water all are important factors.

The next step in development is the construction and testing of a full-scale residential unit using evacuated tube solar collectors. At the same time, some work must be done on optimization of the $CaCl_2$ heat exchanger in order to minimize ultimate manufacturing costs and increase performance closer to the ultimate thermodynamic limits. In addition, other potential applications of the $CaCl_2$-$CH_3OH$ cycle need to be investigated. Because the electrical parasitic power requirement of a solid-phase absorption cycle is inherently low, the $CaCl_2$-$CH_3OH$ cycle may be well suited to industrial waste heat recovery.

## TECHNICAL ACCOMPLISHMENTS

- The thermodynamics of the $CaCl_2$-$CH_3OH$ reaction have been established. The reaction proceeds in a single step to form $CaCl_2 \cdot 2CH_3OH$.

- Corrosion studies have shown that, in the absence of air, none of the three reaction components, $CaCl_2$, $CaCl_2 \cdot 2CH_3OH$, nor $CH_3OH$ vapor, is corrosive toward aluminum or copper.

- Chemical studies indicate that the $CaCl_2$-$CH_3OH$ reaction occurs reversably, without side reactions. Below 150°C, $CH_3OH$ does not decompose, even in the presence of catalysts such as platinum.

- Kinetic studies have established the rate of reaction in a pellet-bed heat exchanger with a maximum pellet-to-fin distance of 0.25 cm and a maximum vapor transfer distance of 8 cm. (Optimum pellet size is in the range 20-40 U.S. mesh.) With the heat transfer fluid to the absorber bed set a 40°C, the rate of absorption was found to fit the equation

$$\text{Rate (moles } CH_3OH \text{ per mole } CaCl_2 \text{ per hour)}$$

$$= 0.0878 \exp((58.1-\Delta T)/11.3)^2$$

where $\Delta T$ is the temperature difference between the absorber heat transfer fluid inlet and the evaporator. Even at the highest $\Delta T$ examined, 55°C (evaporator at -15°C), the rate is nearly 10% of storage capacity per hour, more than adequate for solar heating.

- Computer simulations of expected performance have been carried out using the University of Wisconsin TRNSYS program. These studies indicate that in a climate such as that of Washington, D.C., the $CaCl_2$-$CH_3OH$ heat pump should be capable of meeting of 90% of the residential cooling load using only 25 $m^2$ of evacuated tube collectors. Fuel consumption for heating would be reduced by about 80%, and for domestic hot water by about 70%. Part of this fuel reduction is due to solar collection, and part due to the high coefficient of performance of the heat pump operating in the backup mode. Even without a solar collector, the heat pump COP reduces heating fuel consumption by about 40%.

- A detailed thermodynamic analysis of the expected heat pump performance has been carried out. The calculated coefficient of performance for cooling is 0.6, and for heating, 1.6.

- A 1/3 scale engineering development test system aimed at testing the heat exchanger design has been constructed and tested. The cooling coefficient of performance of the test system was 0.52, in excellent agreement with the thermodynamic analysis. Long-term cycling of the heat exchanger established that the $CaCl_2$ pellets retain their integrity. Some warpage of the original heat exchanger, due to expansion and contraction of the salt, was observed. However, a redesigned unit, constructed with additional bracing, showed no distortion.

- A cost study and economic payback analysis was carried out. Under the standard criteria for customer acceptance -- number of years required to achieve positive cash flow, and number of years required for accumulated savings to offset initial cash payments and early cash flow losses -- the system looks highly attractive.

- As part of the cost study, a conceptual design of a full-scale residential system was prepared. Specifications were prepared for all pumps, valves, heat exchangers, fans, and other components.

## FUTURE ACTIVITIES

All activities under the present contract have been successfuly completed. The next stage is the construction and field testing of a full-scale solar-driven system. Licensing, production engineering, marketing and commercialization must await completion of a full year of field testing, so that performance and costs can be fully evaluated. Because the $CaCl_2$-$CH_2OH$ cycle is designed as a complete single-package system (except for solar collection), incorporating heating, cooling, domestic hot water, and energy storage, the degree of technical risk is high compared to systems based on separate components. However, the potential benefits of an integrated system, aimed directly at solving the problem of solar cost-effectiveness, is an answer to fossil energy dependence of our residential and commercial sector.

## REFERNCES


1. P. O'D. Offenhartz, F. C. Brown, R. Mar and R. W. Carling, J. Solar Energy Engineering (ASME Transactions), Vol. 102, p. 59 (1980).

2. P. O'D. Offenhartz, "TRNSYS Simulation of Chemical Heat Pumps for Solar Heating, Cooling, and Storage," Proc. Second Symposium on Solar Simulation and Economic Analysis, San Diego, January, 1980 (SERI/TP-351-431, p. 251-257).

3. P. O'D. Offenhartz, "TRNSYS Simulation of Chemical Heat Pumps for Solar Heating, Cooling, and Storage: Part II: Simulations of Washington, D.C. and Fort Worth, Texas Including Window Effects," Proc. Annual DOE Active Solar Heating and Cooling Contractors' Review Meeting, Lake Tahoe, March, 1980. (CONF 800340, p. 11-18).

4. P. O'D. Offenhartz and F. C. Brown, "Methanol Based Heat Pumps for Storage of Solar Energy," Proc. 14th IECEC, Vol. 1, p. 507, Boston, August, 1979.

5. P. O'D. Offenhartz, D. Schwartz, R. E. Malsberger and T. V. Rye, "Engineering Prototype Studies on the $CaCl_2$-$CH_3OH$ Chemical Heat Pump for Solar Air Conditioning, Heating, and Storage," Proc. 15th IECEC, Seattle, August, 1980.

6. P. O'D. Offenhartz, D. Schwartz and R. E. Malsberger, "A Chemical Heat Pump Based on the Reaction of Calcium Chloride and Methanol for Solar Heating, Cooling and Storage," Proc. DOE Chemical Energy Storage and Hydrogen Energy Systems Contracts Review (CONF-791127), p. 80, February, 1980.

7. R. W. Carling, A. T. Wondolowski and D. C. Macmillan, "Heat of Formation of $CaCl_2 \cdot 2CH_3OH$ and $CaCl_2 \cdot 2C_2H_5OH$ by Solution Calorimetry," SAND80-8689, June, 1980. Submitted to J. Chemical Thermodynamics.

8. P. O'D. Offenhartz, "Design of a Chemical Heat Pump Using Calcium Chloride and Methanol for Solar Heating, Cooling and Storage," Int. Seminar on Thermochemical Energy Storage, Stockholm, 1980.

9. Ron Dagani, "Chemical Heat Pump Cools as Well as Heats," Chemical and Engineering News, October 20, 1980, p. 36.

10. P. O'D. Offenhartz, "A Chemical Heat Pump Based on the Reaction of Calcium chloride and Methanl for Solar Heating, Cooling and Storage," Proc. DOE Thermal and Chemical Storage Annual Contractors' Review Meeting, McLean, VA, October, 1980 (CONF 801095), pp. 154-159).

# Electric Driven Systems

# ADVANCED ELECTRIC RESIDENTIAL HEAT PUMP

WESTINGHOUSE ELECTRIC CORPORATION

Contract No. 86X24712C

S. E. VEYO

PERIOD OF CONTRACT: Begin Oct. 1, 1979; End Dec. 31, 1982

## OBJECTIVE

The Advanced Electric Residential Heat Pump research and development program has as its objective the development of a preproduction prototype, commercially and economically viable, advanced heat pump that can provide satisfying performance with an annual energy efficiency at least 20 percent better than the most efficient unit available in the market place in early 1979. A concommitant objective is that the degree of efficiency improvement realized combined with the projected units sold result in a significant national energy savings within 15 years of product introduction.

## BACKGROUND

In response to the twin prods of escalating energy price and portending government regulated minimum efficiency standards, the HVAC industry is providing an increasing number of electric heat pump models with increasing efficiency. In order to have a significant impact upon national energy consumption it is inadequate to merely produce a heat pump with a label indicating superior performance. Required, in addition, is customer satisfaction and viability in the market place.

Commercial viability demands that improved efficiency be achieved with a unit cost permitting successful competition in a free market against reasonable alternatives and with sales volume adequate to support a business venture with reasonable return on investment.

Economic viability demands that the annual cost of heat pump ownership computed on a life cycle basis be less than its reasonable alternatives.

Customer satisfaction demands that increased heat pump efficiency be realized in practice without sacrifice of user perceived comfort and reliability.

As identified in Ref. 1 and elsewhere the key to realizing significantly improved seasonal or annual efficiency in a marketable heat pump is modulatable compressor capacity coupled with optimal control and system proportions. Improvements in motor and air mover efficiency are also necessary to meet our objective.

The availability today of several multiple or variable capacity heat pumps in no way compromises the objectives of this program since none can meet simultaneously the necessary efficiency and cost targets.

## CONCEPT

The heat pump system concept that Westinghouse is developing uses the well known vapor compression refrigeration cycle with R-22 as the working fluid. In order to achieve the target efficiency an improved reciprocating compressor with modulatable capacity is being developed along with higher efficiency air mover a microprocessor based control system and higher effectiveness heat exchangers. The relative proportions of the compressor, blower, fan and heat exchangers are specified through system optimization to minimize annual ownership cost while constrained to provide comfort. The key to the potential commercial success of this system is a low cost compressor capacity modulation means. Our preferred means is unique and is disclosed in U.S. Patent 4,236,874. Capacity modulation is achieved through the use of an eccentric cam between crankpin and connecting rod. The action of the cam is such that the stroke of the compressor is either a minimum or maximum value dependent upon the direction of rotation of the crankshaft. The ratio of minimum to maximum stroke, hence minimum to maximum compressor capacity is a parameter of optimization.

## SUMMARY

The Advanced Electric Heat Pump program is meeting its technical goals on schedule and within budget. Based upon computer analysis the estimated heating season performance factor (HSPF) for our advanced heat pump when installed in a newly constructed 1600 square foot house located in Minneapolis, Minnesota is 8.00 Btu/Wh. The cooling season SEER is estimated to be 11.23. When similarly sited in Nashville, Tennessee the HSPF is 9.52 and the SEER is 11.33. The performance improvement relative to a currently available two speed heat pump exceeds 20 percent while the average installed cost will be approximately 5 percent less. The installed cost premium relative to low cost builders line hardware will be approximately 25 percent. The estimated annual market for this heat pump in 1990 under a pessimistic relative energy price scenario is 680,000 units when the installed price is considered to be ten percent higher than estimated.

The dual stroke mechanism has been reduced to practice and has been incorporated in several 3.5 ton developmental compressors. Capacity modulation has been demonstrated in addition to a modest increase in efficiency due to compressor improvements. Two developmental indoor blowers have been designed and built. The first yields a measured 55 percent static efficiency which contrasts well with the 35 percent efficiency blower used in current practice. The predicted static efficiency of the second blower is 75 percent with testing now underway. An improved outdoor fan assembly has been designed, built and tested. The measured static efficiency of 52 percent contrasts well with the 30 percent efficiency available in current practice. Fan and blower motors have been produced that yield high speed electrical efficiencies in excess of 65 percent. A breadboard heat pump system of nominally 3.5 ton cooling capacity has been built to test the integration of these advanced concepts and develop the control system software. The breadboard unit is currently under test. The package design has

been completed and detail drawings for preprototype construction are in progress. A preprototype will be produced in sheet metal and tested before the end of 1981.

TECHNICAL ACCOMPLISHMENTS

• Dual Stroke Compressor. The stroke changing mechanism of Ref. 2 has been incorporated into four Westinghouse CD048 refrigeration compressors to date and testing for both performance and endurance continues. The initial unit has accumulated over 4000 hours of run time and over 15,000 cycles of capacity mode reversal. Based upon periodic tear down and inspection there exists no evidence to suggest abnormal wear or any form of mechanical deterioration. Thermodynamic performance remains stable. The measured performance of the dual stroke compressor is shown in Fig. 1 for the high capacity mode of operation and in Fig. 2 for the low capacity mode of operation. The magnitude of the minimum stroke is bounded between zero and the maximum stroke with its precise value a subject for system optimization. The particular stroke ratio embodied resulted from system optimization. The significance of this dual stroke compressor is threefold. First, the capacity ratio can be tailored to the system and its duty. It is not fixed as in the two-speed pole changing motor case. Second, it is inexpensive. Our estimates indicate that relative to a single capacity hermetic compressor the dual stroke unit will have an incremental cost one third the incremental cost of a two-speed motor driven hermetic compressor. Third, the physical size and weight of the dual stroke compressor are nearly equivalent to those of a conventional, single capacity compressor.

• Indoor Blower. The indoor blower for the advanced electric heat pump is a direct driven, single entry design with multiple, backwardly inclined, curved, airfoil blades. An approximately correct design for application at 1650 rpm was designed and built by Westinghouse prior to this contract. It was tested and then scaled for application to the 3.5 ton breadboard heat pump (1575 scfm at 0.85 inches of water external pressure drop). The performance of this advanced design relative to the conventional squirrel cage blower with motor in inlet is shown in Figs. 3 and 4. Note that the 55 percent efficiency of the backwardly inclined air foil blower is roughly 50 percent better than that of the conventionally applied blower. As applied in the breadboard heat pump this blower was noisier than desired. Consequently, a larger diameter wheel for application at 6 pole speed has been designed and built with testing now in progress. The expected efficiency is 75 percent at the same design flow and pressure drop.

• Outdoor Fan. The outdoor air mover designed, built and tested by Westinghouse is an axial fan with eight twisted, cambered plate blades designed to operate at 8 pole speed. Specifications called for 2900 scfm at 0.2 inches of water pressure drop. A photograph of the prototype fan is shown in Fig. 5. The measured performance of this fan coupled with a diffuser stack, hub afterbody, inlet flare, and a stator vane row is shown in Fig. 6. As can be seen, it has more pressure rise capability than specified. At 2900 scfm the efficiency is 52%. This is approximately twice the application efficiency of current heat pump fan practice. Simplification of the air mover subassembly to just the inlet flare, fan, and diffuser ring as in current practice would yield an air mover with about 43 percent static efficiency, but with a somewhat lower pressure flow capability.

• Controls. The control system for the advanced heat pump will use a microprocessor. The breadboard heat pump is equipped with an umbilical cord to a remote 12 channel relay closure output device that is driven by a Hewlett Packard HP9825 programmable calculator. The calculator in turn receives a stream of data from the supervisory computer for the testing complex. Control algorithms using inputs from a variety of sensors and sensor locations can thus be developed easily using software for the control system. The software now being exercised controls two stages of compressor capacity, four stages of resistance heat, two levels of indoor air flow, two levels of outdoor air flow, two circuit variations in the indoor heat exchanger, mode selection (heating, cooling, ventilating, off), demand defrost, and anomaly detection.

• Packaging. A matrix type decision making process has been applied to evaluate the packaging options devised for both indoor and outdoor sections of residential split system heat pumps. The configuration for the preprototype has been selected. The indoor section encompasses the compressor, and all controls and actuators. The outdoor section contains only the outdoor heat exchanger, air mover and a limited number of sensors.

• Optimization. The optimization procedure first reported in Ref. 1 has been expanded to consider both the heating and cooling season, a specified histogram of hourly temperature occurrence, the use of resistance heat, a more cost effective gradient follow technique, and a means to follow constraints. This procedure systematically perturbs the relative size of heat exchangers and air flows so as to minimize or maximize some measure of system performance such as first cost, or efficiency, or annual ownership cost. In this program, optimization is taken to mean the minimization of annual ownership cost per unit of heat transported beneficially. In constant 1980 dollars ownership cost includes the cost of energy, the amortization of the heat pump system based upon a 15 year expected life and 3 percent true interest, and an allowance for maintenance as a percentage of unit sales price. The heat transported beneficially is calculated using a specific histogram of hourly temperature occurrence, a building load line heating based upon a judiciously selected balance temperature, a cooling load line based upon a scaled reflection of the heating load line, and the capacity capability of the system. The energy consumed by the system is calculated using the energy consuming characteristics of the system and a cyclic degradation rationale and defrosting rationale similar but not identical to the DOE procedure of Ref. 3. A comfort constraint is imposed that places a lower limit on delivered air temperature heating as a function of air flow rate and an upper limit on the sensible to total capacity ratio cooling. A brief summary of results can be found in Table 1. Typically a system optimized on an annual basis is more efficient than one optimized on a seasonal basis because the increased cost of higher efficiency can be amortized over more heat that is transported. Units optimized in a northern climate are less costly on a first cost basis per unit of nominal capacity than units optimized for southern duty. This is because a northern unit can use capacity to displace resistance heat whereas a southern unit, applied optimally at a higher balance temperature heating must use higher system efficiency to displace already efficiently used power. Note that the optimum SEER values for a heat pump do not exceed 11.7, even in the south.

• Breadboard System. In order to verify analytical estimates of performance as well as to develop our control system, a breadboard unit was optimized for heating in DOE Region V, constructed, instrumented extensively and is currently being tested.

Since samples of the dual stroke compressor were not available at the start of the effort, the system was configured using the Copeland TED1-0350-PFV twin compressor, and set to run at either minimum or maximum capacity. The heating performance of this unit is shown in Fig. 9. The HSPF is estimated to be 8.50 in Albany at $7.90/MMBTU. The efficiency is very good, but because the projected OEM cost of this twin compressor is considerably higher than the estimated OEM cost of the dual stroke compressor, the cost of delivered heat is higher. The twin compressor system as configured is not acceptable as an air conditioner in low capacity mode because the compressor capacity ratio and system proportions do not provide adequate humidity control.

- Market Analysis. The estimated performance of the optimized dual stroke system, Figs. 7 and 8 for DOE Region V, was used in the sizing study to select the best nominal capacity to apply to a 1600 square foot residence of three different insulation levels (see Ref. 1) in various parts of the United States. The unit cost and installation cost for various system capacities was scaled from the 3.5 ton size on the basis of current practice while the performance was held independent of capacity. The estimated annual cost per million BTU as a function of unit capacity was then plotted as shown in Fig. 10 and the best application size selected. Note that the load is a function only of the climate and insulation level. This was done similarly for builders line hardware and for a competitor's dual speed unit. The performance and cost of these three units for the three different levels of insulation in four representative cities is shown in Table 2.

Having selected the best size for three alternative heat pumps in four representative cities, a market analysis was performed to gauge the market penetration of these three heat pumps against each other and against alternatives using projected energy costs. A customer acceptability function based upon payback of incremental cost between alternatives was generated, tested against recent experience and the market projected out to the year 2000.

Data published by Data Research Incorporated (DRI), Ref. 4, was used as a baseline. In addition to using DRI data for the relative escalation of electric, gas, and oil rates, a more pessimistic scenario for the electric heat pump was also considered. If we assume that the electric rates escalate at a rate two percent faster than DRI projections and that gas rates escalate three percent slower than DRI projections and if we further assume that our estimates of installed cost for the Westinghouse/DOE advanced electric heat pump are ten percent low then assuming product introduction in 1985, the 1990 market, new plus replacement, is divided as shown in Fig. 11 with thousands of units shown. Adoption of DRI energy price projections and our original cost estimates yields a 1990 market for the advanced heat pump in excess of 2.5 million units.

## FUTURE ACTIVITIES

Contract Activities - The major tasks to be completed in 1981 are continued compressor testing, the completion of breadboard heat pump testing, the final development of control software, the construction of a breadboard microprocessor based control system, and the design, construction, and testing of a preprototype advanced electric heat pump. Scheduled for 1982, but at hazard due to government budget policy, is the development of a prototype advanced electric heat pump with the inherent final trade offs between performance and unit cost.

Post Contract Activities - Westinghouse will cooperate to the fullest with the government and other agencies in order to insure that the technology developed under this contract is carried to field test and ultimate commercialization.

## PUBLICATIONS/REPORTS/REFERENCES


1. Kirschbaum, H. S., Veyo, S. E., An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate, EM-319 (Research Project 544-1), Electric Power Research Institute, Palo Alto, CA, 1977.

2. Sisk, F. J., "Dual Capacity Compressor with Reversal Motor and Controls Arrangement Therefore," United States Patent No. 4,236,874, December 2, 1980.

3. "Test Procedures for Central Air Conditioners, Including Heat Pumps,", Federal Register, Vol. 44, No. 249, Thursday, December 27, 1979, Rules and Regulations, page 76700.

4. Data Review from Data Resources Incorporated, Lexington, MA.


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Table 1 - Estimated Performance of Optimized Systems

| Compressor | Optimization | DOE Region | Balance Temp. | HSPF | SEER | APF | $/MMBTU* |
|---|---|---|---|---|---|---|---|
| W CD048 | Annual | IV | 24°F | 7.63 | 7.61 | 7.63 | 7.55 |
| W Dual Stroke | Annual | III | 36°F | 10.00 | 11.68 | 10.62 | 5.76 |
| W Dual Stroke | Annual | IV | 24°F | 9.34 | 11.58 | 9.70 | 6.24 |
| W Dual Stroke | Annual | V | 16°F | 8.38 | 11.33 | 8.88 | 7.06 |
| W Dual Stroke | Heating | V | 16°F | 8.35 | - | - | 7.78 |

* Includes only system selling price, effective interest @ 3 percent, annual maintenance @ 5 percent of selling price, electricity @ 4.97 ¢/kWh.

Table 2 Comparative Heat Pump Performance (Optimum Heat Pump Size @ 4.97 ¢/kWh in 1600 sq. ft. dwelling)

| | DOE Region | II | | | III | | | IV | | | V | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | City | PHOENIX | | | NASHVILLE | | | PITTSBURGH | | | MINNEAPOLIS | | |
| | Insulation | 1 | 2 | 3 | 1 | 2 | 3 | 1 | 2 | 3 | 1 | 2 | 3 |
| | Htg. Load MBTU | 47.08 | 31.56 | 20.18 | 102.59 | 68.77 | 43.97 | 163.77 | 109.78 | 70.19 | 191.84 | 128.60 | 82.22 |
| | Cool Load MBTU | 125.20 | 83.92 | 53.66 | 59.48 | 39.87 | 25.49 | 24.78 | 16.61 | 10.62 | 31.14 | 20.87 | 13.34 |
| Low Cost | Nom. Cap. (Tons) | 5* | 3.5 | 2.5 | 4 | 3 | 2 | 4 | 2.5 | 1.5 | 5 | 3.5 | 2.0 |
| | Normalized Cost | 1.40 | 1.13 | 0.97 | 1.26 | 1.05 | 0.87 | 1.30 | 1.00 | 0.77 | 1.53 | 1.20 | 0.91 |
| | HSPF | 7.58 | 7.33 | 6.92 | 6.86 | 6.79 | 6.62 | 6.32 | 6.18 | 6.00 | 5.82 | 5.79 | 5.62 |
| | SEER | 6.58 | 6.48 | 6.30 | 6.81 | 6.62 | 6.37 | 6.77 | 6.59 | 6.29 | 6.75 | 6.57 | 6.33 |
| Competitor 2 Speed | Nom. Cap. (Tons) | 5* | 4 | 2.5 | 4 | 3 | 2 | 4.5 | 3 | 2 | 5 | 3.5 | 2 |
| | Normalized Cost | 1.87 | 1.64 | 1.27 | 1.67 | 1.39 | 1.14 | 1.81 | 1.43 | 1.14 | 1.97 | 1.57 | 1.14 |
| | HSPF | 8.84 | 8.44 | 7.98 | 7.36 | 7.38 | 7.23 | 6.78 | 6.70 | 6.64 | 6.09 | 6.11 | 5.89 |
| | SEER | 8.31 | 8.13 | 7.95 | 9.04 | 8.75 | 8.36 | 9.05 | 8.70 | 8.13 | 9.02 | 8.71 | 8.34 |
| W/DOE | Nom. Cap. (Tons) | 5* | 4 | 2.5 | 4 | 3 | 2 | 5 | 3.5 | 2 | 5 | 3.5 | 2.5 |
| | Normalized Cost | 1.81 | 1.59 | 1.24 | 1.62 | 1.35 | 1.11 | 1.85 | 1.46 | 1.11 | 1.92 | 1.49 | 1.24 |
| | HSPF | 11.12 | 10.57 | 9.88 | 9.35 | 9.52 | 9.31 | 9.18 | 9.24 | 8.62 | 7.90 | 8.00 | 8.19 |
| | SEER | 10.54 | 10.57 | 10.23 | 11.74 | 11.33 | 10.71 | 11.54 | 10.93 | 10.15 | 11.73 | 11.23 | 10.42 |

*Cooling Deficiency

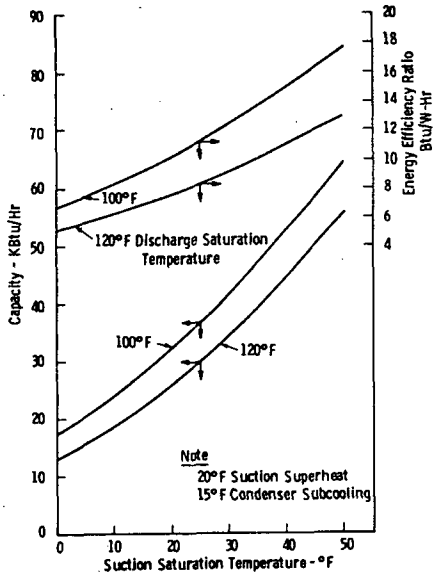


Figure 1 Dual Stroke Compressor Performance High Capacity Mode

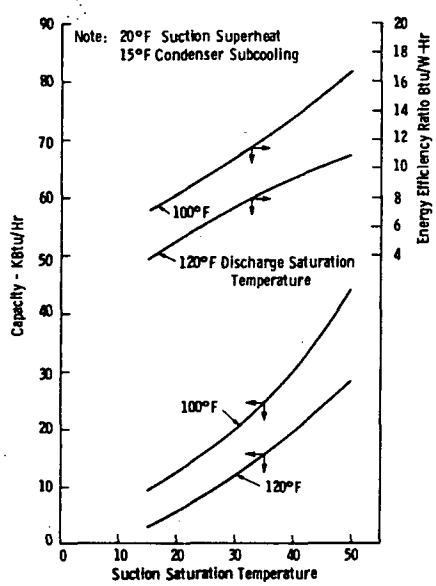


Figure 2 Dual Stroke Compressor Performance Low Capacity Mode

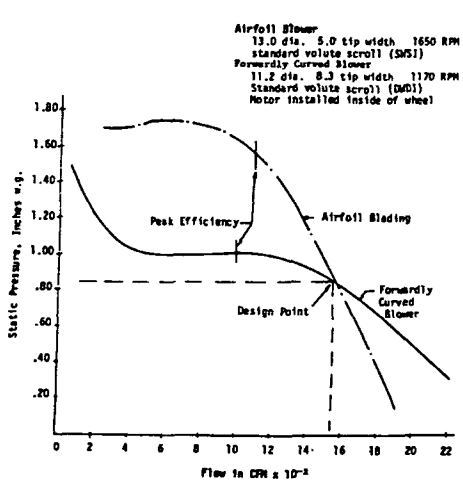


Figure 3 Comparison of Performance Characteristics for an Airfoil and Forwardly Curved Blower

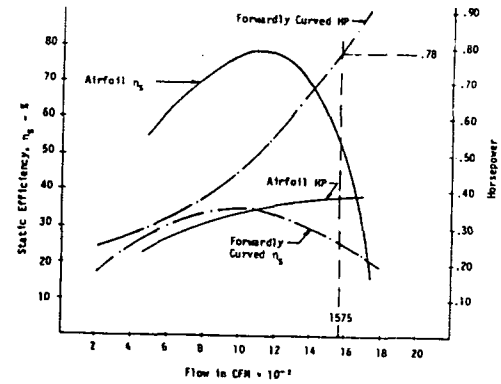


Figure 4 Comparison of Efficiency and Power for an Airfoil and Forwardly Curved Blower

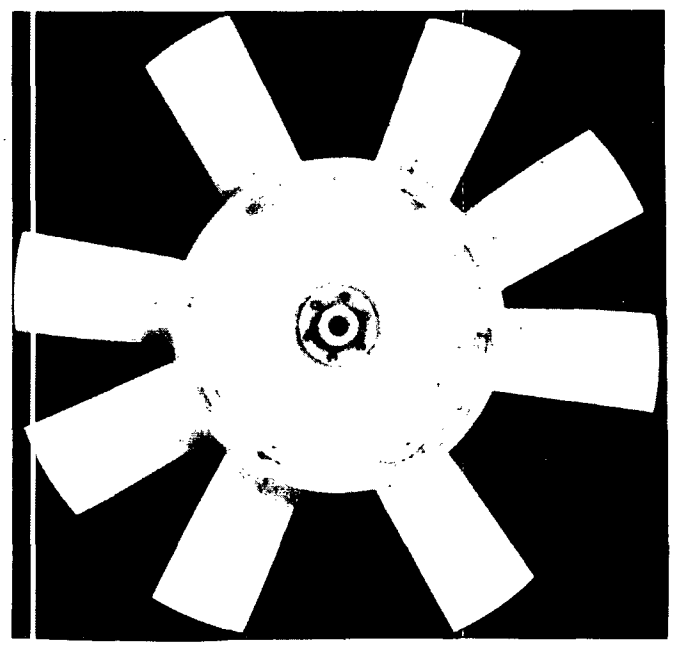

Figure 5 Prototype Fan

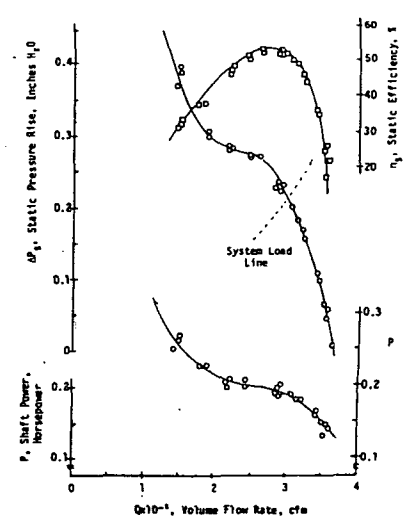


Figure 6 Outdoor Fan Performance at N = 850 rpm, $\rho$ = .075 lb/ft$^3$

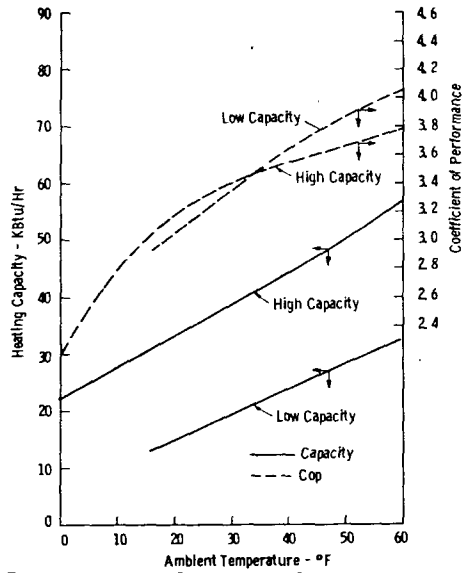


Figure 7 Heating Performance of Optimized Dual Stroke Compressor System

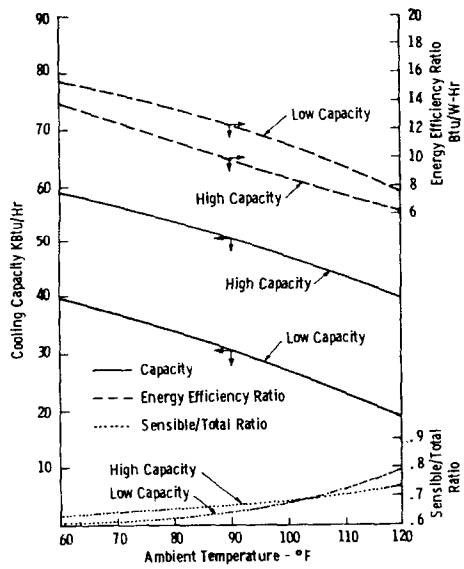


Figure 8 Cooling Performance of Optimized Dual Stroke Compressor System

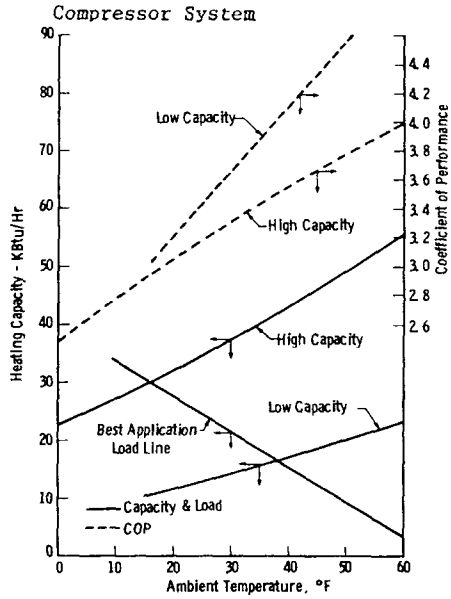


Figure 9 Heating Performance - Breadboard

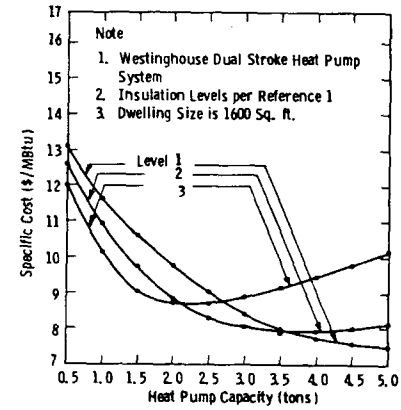


Figure 10 Specific Cost versus Nominal Heat Pump Capacity in DOE Region IV - Pittsburgh

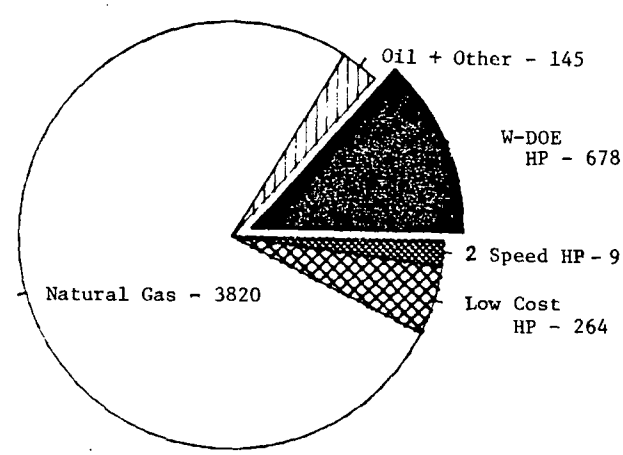


Figure 11 Conservative Projection of Annual Advanced Heat Pump Market in 1990

DEVELOPMENT AND DEMONSTRATION OF AN ELECTRIC HEAT PUMP FOR WASTE HEAT RECOVERY IN INDUSTRY

WESTINGHOUSE ELECTRIC CORPORATION

DE-AC01-77CS40327

W. C. MORELAND

PERIOD OF CONTRACT: Begin July 1977 End July 1982

## OBJECTIVE

Demonstrate technical feasibility and economic viability of using heat pumps to reclaim low temperature waste heat energy from industrial processes. The specific technical goals of the development program are:

- Delivery temperatures of 250 to 350°F
- Source temperatures of 180 to 250°F
- Coefficients of performance of 3 or greater
- Paybacks of four years or less.

## SUMMARY

The Westinghouse R&D Center, under contract to the Department of Energy, is developing a high temperature heat pump which will extend the technology more than 100°F above that of available commercial units. This is a two phase program in which the first phase covers the development, design, fabrication, and laboratory testing of a prototypical unit. A second phase, if contracted, would cover the actual installation and demonstration of this heat pump at an industrial partner's plant.

Under the present contract the critical aspects of the basic technology required for extending the high temperature capability of heat pumps have previously been identified and investigated (Ref. 1 and 2). These technical investigations have been supplemented by a number of application studies (Ref. 3) which indicate that electrically driven heat pumps can be cost effective in reducing fossil fuel consumption in particular applications. These studies have concentrated on the pulp and paper industry. This industry is one of the major consumers of fossil energy and has both process energy requirements and waste heat sources within the thermal capability of the high temperature heat pump.

## CONCEPT

The basic concept for the heat pump is to utilize a closed cycle system with two stages of compression, single or dual expansion and liquid spray intercooling. Use of a closed cycle provides a clean and totally controllable environment for the compressor and freedom to select working fluids with the most desirable thermodynamic and transport properties in the temperature regime of intended application. When compared to conventional steam vapor recompression, this results in smaller compressor sizes and, in general, fewer stages required for a given thermal lift.

The schematic in Fig. 1 depicts such a system design for an application in which the heat source is contaminated atmospheric pressure steam and the output 50 psig steam with 20°F superheat. The COP of the system is calculated to be 4.5 based on the compressor manufacturer's rated power requirements for this service, with inclusion of additional losses for system pressure drops, a rated compressor motor efficiency of 96 percent, and an allowance of 5 kW for auxiliary electrical power. The heat pump design capacity is 30 x $10^6$ Btu/h.

Since methanol (methyl alcohol) is the working fluid, each component in the following discussion will be identified by its function in the methanol loop. For example the methanol condenser/steam boiler will be referred to as the condenser. The thermodynamic states of the methanol can be followed on the pressure-enthalpy diagram in Fig. 2. The thermodynamic states at different points in the heat pump system are indicated by the eight Δ's on both the schematic and pressure-enthalpy diagram.

Heat is added to the system when low pressure waste steam enters the tubes of the methanol evaporator (1) and condenses. The heat of condensation is transferred to the methanol liquid surrounding the tubes and the methanol evaporates. The evaporated methanol enters a superheater (2) where its temperature is raised five degrees above its saturation temperature with heat from additional condensing steam. This superheating is required to prevent partial condensation of the vapor when it accelerates into the compressor inlet at Point 1 on the pressure-enthalpy diagram. The superheated vapor entering the first stage compressor (3) is then compressed to approximately 2-1/2 times inlet pressure (Point 2). The compression process (with an anticipated isentropic efficiency of 75 percent) adds a significant amount of superheat, which, if left unchanged, would decrease the efficiency of the second stage of compression and cause undesirably high discharge temperatures. Correcting this situation requires the addition of an intercooler (4) where liquid methanol of lower enthalpy but higher pressure is sprayed into the superheated vapor from the first stage discharge. Evaporation of the methanol liquid spray brings the methanol vapor to within 5°F of the saturation temperature. Any overspray is separated from the vapor in the knockout drum and returned to the evaporator. The intercooler discharge corresponds to Point 3 on the enthalpy pressure diagram. The vapor with five degrees of superheat enters the second stage compressor (5) where it is compressed to approximately 2-1/2 times the interstage pressure. The exit condition corresponds to Point 4 on the enthalpy-pressure diagram. As before, the methanol vapor is superheated during the compression process. The superheated methanol is used to superheat the steam exiting the condenser (6). This is accomplished in the desuperheater (7). The vapor exiting from the desuperheater corresponds to Point 5 on the enthalpy-pressure diagram. The exiting vapor enters the tubes of the condenser where it condenses to the saturated liquid state at the discharge pressure. The saturated methanol is then subcooled by 5°F in a partitioned section of the condenser to an enthalpy corresponding to Point 6 on the diagram. The methanol

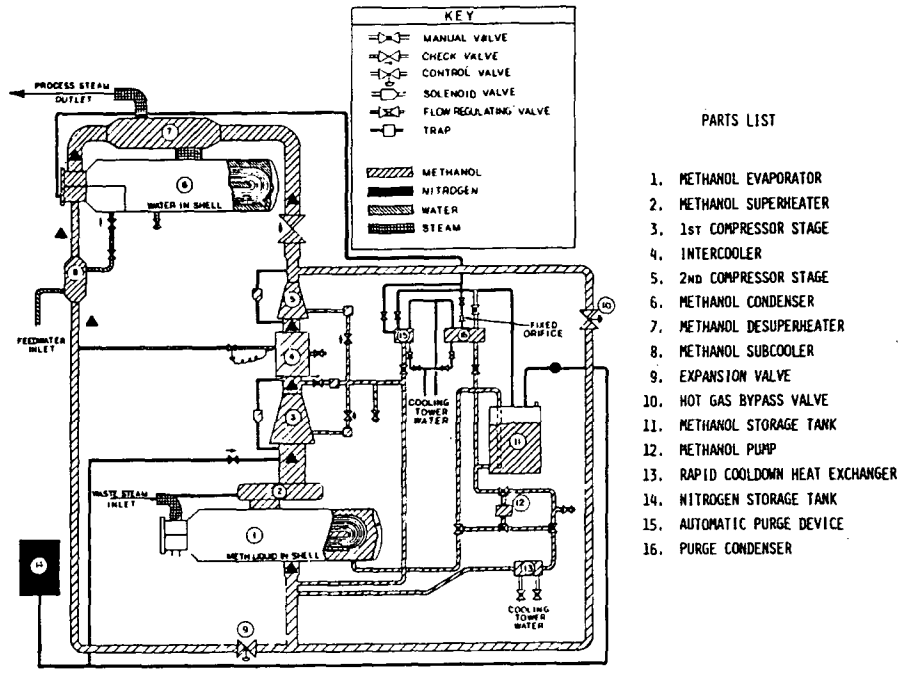


Figure 1 Schematic of Heat Pump System

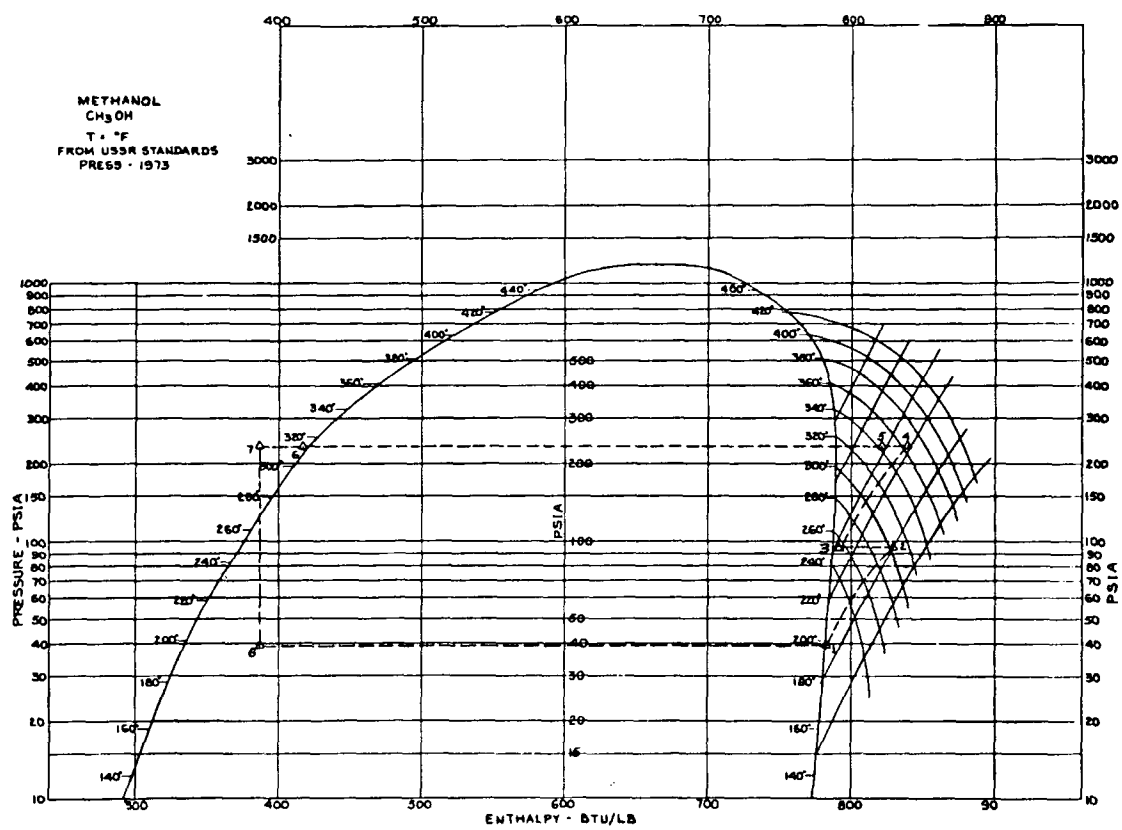


Figure 2 Pressure-Enthalpy Diagram Indicating Thermodynamic States of Heat Pump Cycle

liquid then enters the subcooler (8) where it is further subcooled by transferring additional heat to the boiler feedwater. After its exit from the subcooler at Point 7 on the enthalpy-pressure diagram the methanol flow is split into two streams. Roughly 10 percent of the liquid flow goes to the intercooler (4) as discussed previously, while the remainder of the methanol is admitted to the evaporator (1) through a float controlled expansion valve (9). The methanol enters the evaporator at a condition corresponding to Point 8 on the enthalpy-pressure diagram. With a methanol flow of approximately 66,000 lbm/h this thermodynamic cycle has a net output of 30 million Btu/h.

In addition to the basic heat pump system there are several subsystems which are required for operation of the high temperature heat pump. These subsystems are the hot gas bypass system, the methanol storage system, the nitrogen injection system and the nitrogen purge system. While most of these subsystems are not required during normal operation, they serve important functions in protecting the equipment and providing desired levels of safety when the system is either not running or not operating normally.

The hot gas bypass system has two main functions: a) to prevent the compressor from surging under abnormal system conditions such as loss of source heat, and b) provide some measure of capacity control. Surging is prevented by modulation of the hot gas bypass valve (10) to lower the discharge head when the pressure ratio across the compressors increases and brings them close to their surge line. The bypassed gas is throttled and returned to the evaporator. The gas enters the evaporator through a special diffuser and baffle system to prevent violent bubbling which could cause liquid carry-over into the compressors. In addition to protecting the compressors, the hot gas bypass system can be used to maintain operation even when the heat source is reduced in capacity or cut off. Although this mode of operation reduces the COP of the system significantly, it does provide flexibility in handling capacity variations for short periods of time.

The methanol storage tank (11) serves to maintain a reservoir of liquid methanol for make-up of losses by the compressor seals and purge devices. It is large enough to hold the total system's capacity in case it is necessary to remove the methanol from the heat pump for servicing. This capability is provided by a separate methanol pump (12), a heat exchanger (13) for cooling the methanol before returning it to storage, and appropriate valves.

Because methanol is potentially explosive when mixed with air and because methanol with water in it is highly corrosive, the methanol and the heat pump system must be kept dry and separated from the atmosphere at all times. The methanol is kept dry while in storage by the use of a dry nitrogen blanket which is maintained over the liquid methanol surface. The dry nitrogen is supplied from a liquid nitrogen storage tank (14) through pressure reducing valves. A nitrogen injection system is also used to keep the methanol side of the heat pump system dry and above atmospheric pressure in case of prolonged shutdown. As with the methanol storage tank, nitrogen is automatically injected into the heat pump system if the system is not operating and the pressure drops below a fixed set point. This prevents air from leaking into the system at any time.

While the injection of nitrogen has obvious benefits in protecting the heat pump and providing safer operation, it does present a problem if present when the heat pump is started. Nitrogen is a non-condensible gas at system temperatures and its presence in any significant amount in the heat pump system, in particular in the methanol condenser, would have a serious effect on the operation of the system's heat transfer and performance. As a result, a system for purging the nitrogen from the heat pump is required. Actually two systems are required, one for the high purge rate at start up (15) and one for a small continuous purge during normal operation (16). Both of these systems will condense methanol from the nitrogen-methanol mixture, returning the major portion of the methanol to the heat pump system. The remaining vapor, having a reduced methanol content, will be vented to atmosphere through the storage tank vent.

Throughout its operation the high temperature heat pump system's temperature, pressure, and liquid levels will be monitored. At the same time, the surrounding environment will be monitored for the presence of methanol leakage. Should significant levels of methanol be detected, alarms will be sounded and the system automatically shut down.

A much more detailed description of the thermodynamic cycle and the proposed heat pump hardware is provided in the design report (Ref. 4) than is given in this summary. Derivations of the performance requirements of the major components are provided and the various codes and standards to which they must be designed, manufactured, and tested are given. The test plan and safety procedures are also described in the design report which should be read if more detailed information is desired.

## TECHNICAL ACCOMPLISHMENTS

- Identification and Selection of Candidate Working Fluids. A study was undertaken to identify acceptable working fluids (refrigerants) for a heat pump design which would meet the program goals. After an initial screening the fluid candidates were examined extensively as to their merit in terms of heat pump cycle performance, system and component cost related to fluid choice, fluid hazard potential, and fluid stability and corrosivity. Although several fluids were identified in the study as being acceptable for use in a prototype heat pump, methanol was shown to be significantly better than the others.

- Experimental Evaluation of Four Working Fluid Candidates. Accelerated thermal stability and chemical compatibility tests were run on five candidate materials—methanol, thiophene, ethanol, methylene chloride, and Fluorinol$^R$. In these tests samples of the working fluid and selected engineering materials were sealed in glass tubes and aged at elevated temperatures, the highest test temperature being 500°F. Both ethanol and methylene chloride showed unacceptable stability and/or compatibility at all test temperatures. The other three were acceptable but methanol again showed more desirable characteristics.

- Design of Heat Pump. A design for a specific application of the heat pump has been completed. Cost quotations for all equipment items have been obtained in anticipation of parts procurement and fabrication of the heat pump and test loop. Assuming an installation cost equal to the equipment cost and using the fossil energy and electric costs at a specific paper mill targeted for the demonstration phase, the payback for the heat pump system is calculated to be less than two years.

## FUTURE ACTIVITIES

Contract Activities. The next contract tasks in Phase 1 involve the procurement of hardware items, fabrication of the heat pump and test loop, and verification testing of the system for both performance and

reliability.

Post-Contract Activities. This would involve submission to DOE of a detailed proposal in conjunction with an industrial partner to initiate Phase 2 of the project. It is anticipated that satisfactory integration into an industrial process will make possible the commercialization of high temperature heat pumps for wide spread use.

REFERENCES


1. Snyder, P. H. and Moreland, W. C., "The Selection of Working Fluid Candidates for Use in High Temperature Industrial Heat Pumps," Westinghouse Research Report No. 79-9E9-SOTEM-R1.

2. Gainer, G. C., Luck, R. M. and Moreland, W. C., "Thermal Stability and Materials Compatibility Tests of Five High Temperature Working Fluid Candidates," Westinghouse Research Report No. 79-9E9-SOTEM-M1.

3. Wolfe, R. W. and Moreland, W. C., "High Temperature Electric Heat Pumps for Waste Heat Recovery in Industry," Westinghouse Research Report No. 81-9E9-SOTEM-R1.

4. Moreland, W. C., Eder, J. S. and Wolfe, R. W., "Preliminary Design Report - Development and Demonstration of an Electric Heat Pump for Waste Heat Recovery in Industry," Westinghouse Research Report No. 81-9E9-SOTEM-R2.

# THE DESIGN OF AN OPEN RANKINE-CYCLE INDUSTRIAL HEAT PUMP

MECHANICAL TECHNOLOGY INCORPORATED

CONTRACT NUMBER 86X-24713C

D.W. Chaudoir
H. M. Leibowitz

October 1979 through September 1981

## ABSTRACT

An open Rankine-cycle heat pump is ideally suited for producing low-pressure industrial process steam. Because steam serves as both the heat pump motive fluid and process fluid, the system achieves a unique simplicity and versatility. No intermediate refrigerant fluid exists for which to construct a process interface or impose a temperature limit. Interface components such as the heat pump condenser are not required. Moreover, the use of water vapor eliminates toxicity and flammability risks inherent with most closed-cycle heat pump fluids.

The control strategy is simple. Low-pressure (subatmospheric) water vapor, generated by flashing steam at a temperature below that of the waste stream, is compressed to the process pressure and temperature by an electric-motor-driven, multistage compressor train. This strategy permits the heat pump to accommodate upsets such as sudden changes in the waste stream flow and/or temperature, as well as fluctuation within the process stream.

## INTRODUCTION

In an effort to reduce fuel costs in the industrial environment, more and more emphasis is being placed on low-temperature waste heat streams. These streams, normally varying between 130°F and 200°F, offer an excellent heat source for the generation of low-pressure process steam (10-50 psig). As shown in Figure 1 [1], there are approximately 10 quads (10 x $10^{15}$Btu/yr) of heat being discarded by the industrial sector. Of that, approximately 20% is in the 130 to 200°F range. This represents a sizeable "feedstock" for recently developed energy utilization techniques.

To effectively utilize low-temperature thermal energy, an economical and technically reliable method must be applied that will convert the waste heat stream to a usable form. The conversion method proposed and explained in this paper is an open type Rankine-cycle industrial heat pump. This system utilizes the available waste stream as a source of heat with which to generate subatmospheric water vapor. High-performance compression equipment then boosts the vapor to the temperature and pressure that is required by the specific process.

This paper describes the design of an industrial heat pump capable of generating 5000 pounds per hour of 249°F, 10-psig process steam from a simulated 150°F clean waste water stream. The plant is based on the Open-Open™ heat pump concept, i.e., no system evaporator or condenser. The water vapor is generated by a flash evaporation process. In this heat pump system, shown schematically in Figure 2, the waste stream supplies the heat energy for the system and also serves as the heat pump working fluid.

The waste water stream is directed to a flash evaporator, where approximately 0.5 to 1% of the stream is vaporized at subatmospheric pressures [2].

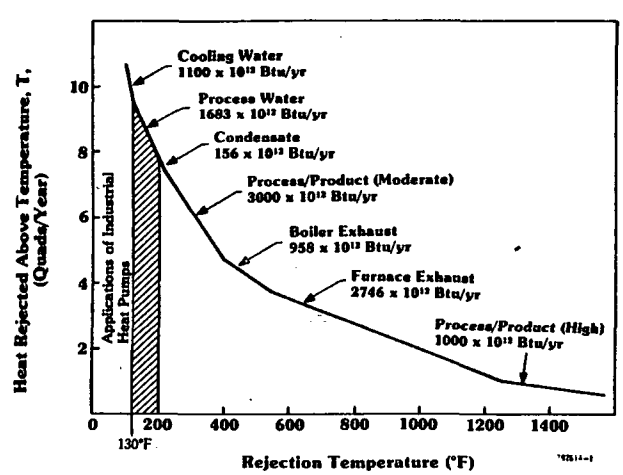


Figure 1 Distribution of Industrial Waste Heat Flows

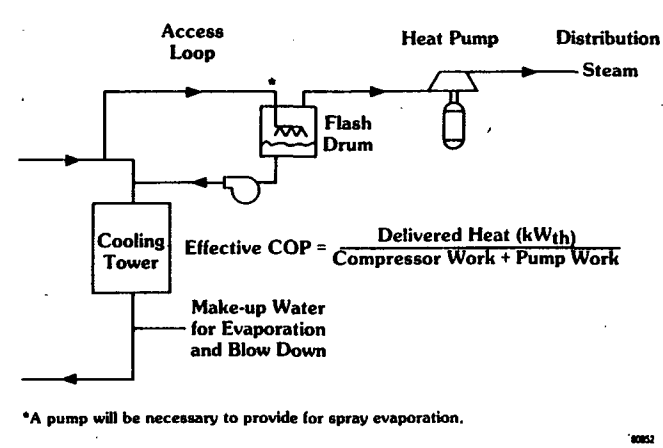


Figure 2 Schematic for Open-Open™ Heat Pumps

This subatmospheric water vapor is then compressed to the desired process pressure by mechanical vapor compression. The system is restricted to a relatively clean waste water stream because the same water is used in vapor form in the heat pump and then delivered to the process. For the case of contaminated streams, a "kettle" type reboiler replaces the flash tank in the heat pumping scheme to provide a means of separating the contaminants in the water stream from the process water stream.

The open-type, Rankine-cycle heat pump offers significant advantages over the conventional closed-cycle heat pumps (see Figure 3).

- It eliminates the need for heat exchangers, and therefore reduces the approach $\Delta T$, increasing the system's COP and reducing the capital cost.
- It eliminates working fluid problems. The heat pump fluid is the process fluid, water vapor.
- It provides simple control strategies, i.e., all control strategies are proven control techniques.

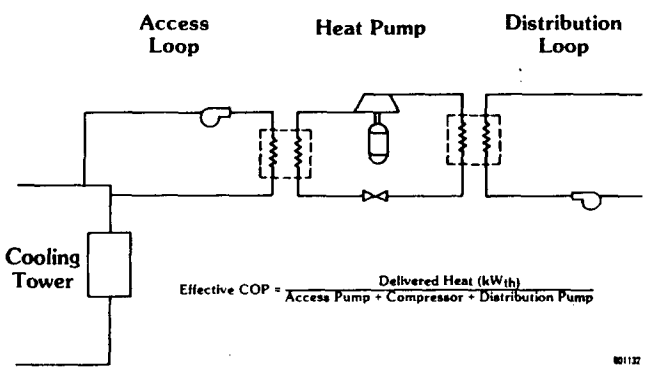


Figure 3 A Conventional Industrial Rankine-Cycle Heat Pump

Although these advantages are profound, there are areas where this system is weaker than the closed-type heat pumps. The major disadvantage of the Open-Open™ system is the effect of the system's working fluid on the compression requirement. Because of the much lower density of water vapor as compared to refrigerants, the size and quantity of the water vapor compressors are considerably greater than that required for a similar closed Rankine-cycle heat pump.

## HEAT PUMP SYSTEM DESIGN

The heat pump plant design presented here has the specific objective of demonstrating the Open-Open™ heat pump concept and providing the equipment/instrumentation necessary to fully evaluate the system's operation. A completely different design strategy would be utilized for industrial applications. The later strategy would be directed primarily toward the most cost-effective design, whereas the demonstration pilot plant presents a design to yield the maximum system operational knowledge and thus, as expected, is heavily instrumented.

The actual pilot plant size, defined in terms of the plant's steam output, was determined by stipulating the minimum equipment required to demonstrate all system technology while, at the same time, addressing the issue of potential scale-up problems. This decision mandated that the design include the following features:

- A flash evaporator capable of handling source stream transients, both in flow rate and temperature.
- A series compression train with a minimum of two desuperheaters between compression stages.
- A control system with monitoring, recording, annunciation and dynamic control capabilities.

The following system description details the manner in which these features were incorporated into the heat pump design.

Process/System Description. The definition of the heat pump process commenced with the generation of a process flow sheet, shown in Figure 4. This schematic denotes the significant process flow state points and their corresponding pressure and temperature levels. Prior to the detailed system descriptions, the original pilot plant concept was designed to generate 5000 pounds per hour of 10-psig dry saturated steam from a simulated 150°F waste water stream. The waste stream was simulated because of the unavailability of a suitable stream. The equipment designation referred to is consistent with that designated on the process flow sheet (Figure 4).

The heat pump pilot plant was designed to operate with a 150°F clean waste water stream as the heat source. To accomplish the simulation, a 6500-gallon capacity storage tank (T-1) is initially heated to 150°F by a 40-hp makeup steam generator. The system feed pump (P-1) pumps the heated water from the storage tank to the flash/drain tank (V-1) where subatmospheric water vapor is produced by a flash evaporation process. The flash tank is maintained at 3.28 psia, or the vapor pressure of 145°F water, by a liquid ring vacuum pump (P-3) during the start-up mode and by the first-stage compressor suction during the steady-state operational mode.

The flash/drain tank combination provides the required volume for flashing, for vapor/liquid separation and for the final settling of the nonflashed water. The specific design of the flash tank is critical to the system's performance and is described in more detail later. The system return pump (P-2) pumps the water from the drain tank back to the water storage tank. Since the water in the drain tank is at its vapor pressure (3.28 psia), the return pump must be able to operate without cavitating, at an available net positive suction head of 7 feet.

Prior to returning to the water storage tank, the 145°F water from the drain tank is mixed with the heat pump's discharge steam to recover the heat lost in the flash evaporation process, 4.42 million Btu per hour. By reinjecting the process steam into the water return, the system becomes self-sustaining after the initial system preheating is accomplished. The storage tank is equipped with a 480 Volt AC immersion heater to maintain the storage tank tem-

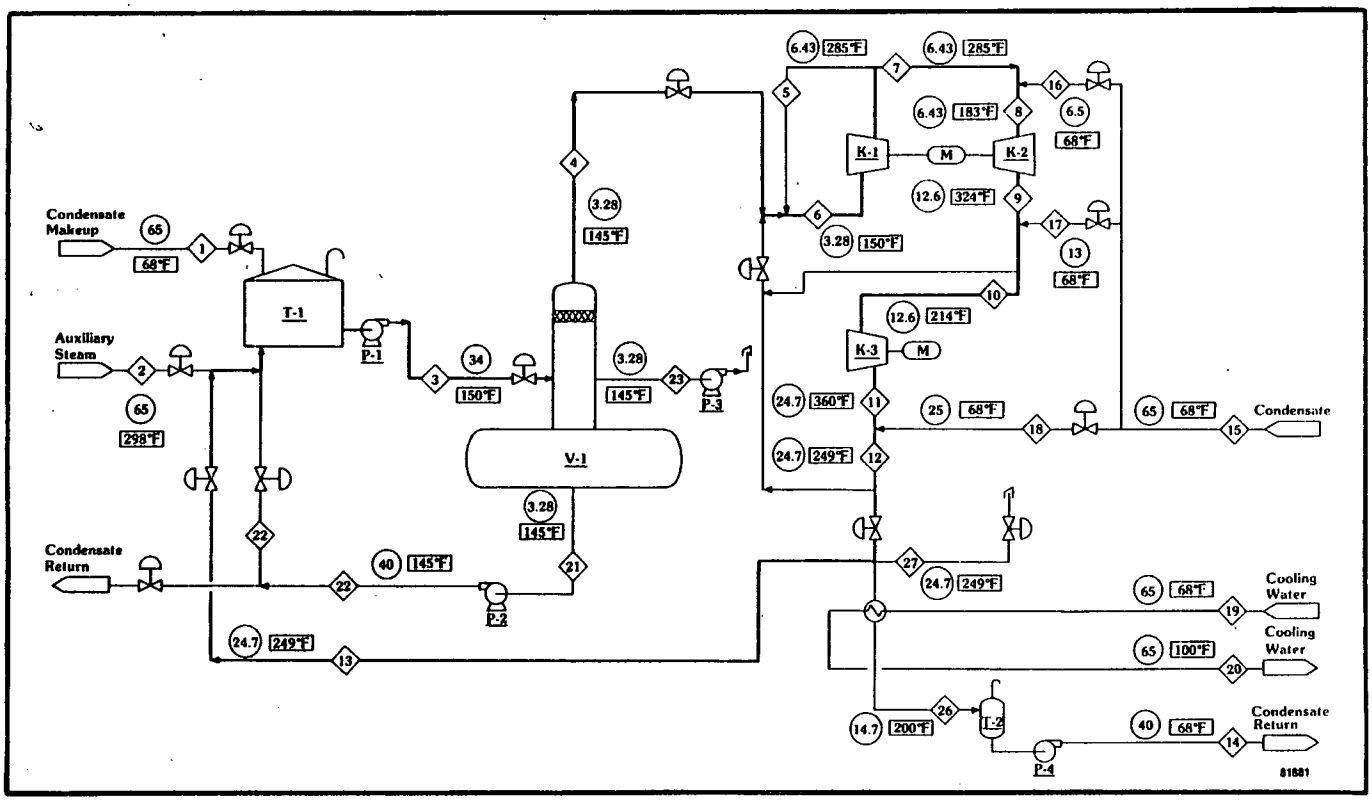


Figure 4 Open-Open™ Heat Pump Pilot Plant Process Flow Sheet

perature during short periods of system shutdown to maximize system availability.

The subatmospheric water vapor generated during the flash evaporation process is piped to the suction of the first-stage compressor. The first-stage compressor is equipped with a small bypass or recycle line which provides sufficient first-stage discharge steam at the compressor suction to maintain a 5°F level of superheat. This level of superheat is required to ensure that water droplets are not introduced into the compressor impeller.

The vapor compression process consists of three stages, followed by a desuperheating station after each stage. The purpose of this compression/desuperheating process is to boost the subatmospheric water vapor to a usable process temperature, 249°F for the pilot plant, at the lowest power consumption level possible. The compression/desuperheat process is described on the T-S diagram, Figure 5.

Subatmospheric saturated water vapor at 3.28 psia is piped to the three-stage compression train. Roughly 4% of the first-stage compressor discharge is utilized to provide 5°F of superheat at the first-stage suction. This process is the A-B process shown in Figure 5.

The first-stage compressor boosts the steam pressure from 3.28 to 6.43 psia with an adiabatic efficiency of 80%. This process (B-C) results in an increase of the superheated steam temperature from 150°F to 285°F. At a discharge temperature of 285°F from the first stage, a second stage of compression

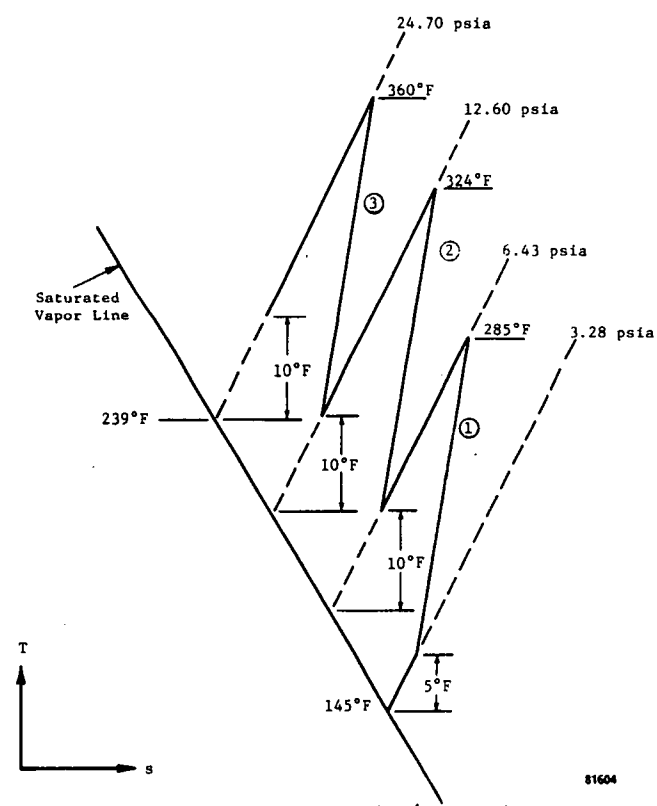


Figure 5 Pilot Plant Compression/Desuperheat Process

would increase the temperature beyond the practical limits of standard materials. Therefore, it is mandatory to provide some form of intercooling or desuperheating to limit temperature rise.

The method of desuperheating selected for the heat pump system is a direct spray of cooling water into the superheated steam flow. This method allows for desuperheating by a direct means of heat exchange without the significant pressure drop penalty associated with indirect heat exchange techniques. A temperature sensor, or RTD, located 30 to 40 feet downstream of the point of injection senses the stream's temperature and controls the amount of cooling water injected. Commercially available desuperheaters allow for temperature control down to 10°F above saturation and, thus, somewhat limit the potential power savings associated with the desuperheating process. The closer the desuperheater can control the stream to 100% quality, dry saturated vapor, the greater will be the power savings. The practical limit, with today's technology, is 10°F above this optimum point. This desuperheating process is shown as C-D.

The remaining two stages of compression and desuperheating follow exactly the same process; refer to Figure 5. For processes in the industrial arena, where temperature limitations are not a constraint, the final desuperheating station is deleted from the system to maximize the process heat availability. But for the pilot plant design, the process steam is desuperheated to 249°F. At this point the plant output is 5000 pounds per hour using 68°F cooling water. If higher temperature water is available the steam output of the heat pump can be increased to provide additional process heat simply due to the extra heat energy available in the warmer water.

Pressure drop or line losses represent an important design consideration that needs careful analysis because excess pressure drop ultimately means more compression power and can ultimately dictate an additional stage of compression. For example, the pilot plant is designed to boost water vapor from 3.28 to 24.70 psia, an overall pressure ratio of 7.53; 1.96 per stage. If a pressure drop of 0.5 psi is assumed for each piping length, it would increase the overall pressure ratio to 9.24; 2.10 per stage. The maximum pressure ratio available on steam is 2 to 1 for the single-stage compressor selected using standard materials of construction. To provide for the 9.24 overall pressure ratio would require four stages of compression, each with a pressure ratio of 1.74 or three stages with the much more expensive titanium impellers. For this reason, care must be exercised in the system design to minimize pressure losses. For the pilot plant Table 1 details the critical pressure drops for the design under consideration. Incorporating these pressure drops results in an overall pressure ratio requirement of 7.77; 1.98 per stage.

TABLE 1

PRESSURE DROP THROUGH COMPRESSOR TRAIN

| | Pressure Drop (psi) | Pipe Size (in.) |
|---|---|---|
| Flash Tank to 1st Stage | 0.08 | 14 |
| 1st Stage to 2nd Stage | 0.07 | 12 |
| 2nd Stage to 3rd Stage | 0.09 | 10 |

Four specific components and processes, which make up the heart of the heat pumping system, deserve special attention. These components are the flash tank, vapor compressors, motor control center and microprocessor-based control system. A detailed description of the equipment and the specific process involved is presented in the next several paragraphs.

Flash Tank - In theoretical terms, the amount of steam ($\dot{m}_v$), in pounds per hour, that can be generated during a flash evaporation process is described by the mass/energy balance:

$$\dot{m}_L C_p T_L = \dot{m}_v h_{fg} \quad (1)$$

where $\dot{m}_L$ is the liquid mass flow (lb/hr), C is the liquid specific heat (1 Btu/lb-°F), $\Delta T_L$ is the liquid temperature drop (°F) and h is the latent heat of vaporization at the lower liquid temperature. Using equation (1), 884,018 pounds per hour of 150°F water is required to generate the required 4372 pounds per hour of water vapor at 3.28 psia and 145°F. See Figure 6. It should be noted that the flash process is initiated by a critically sized, sharp-edged orifice located at the entrance to the flash tank.

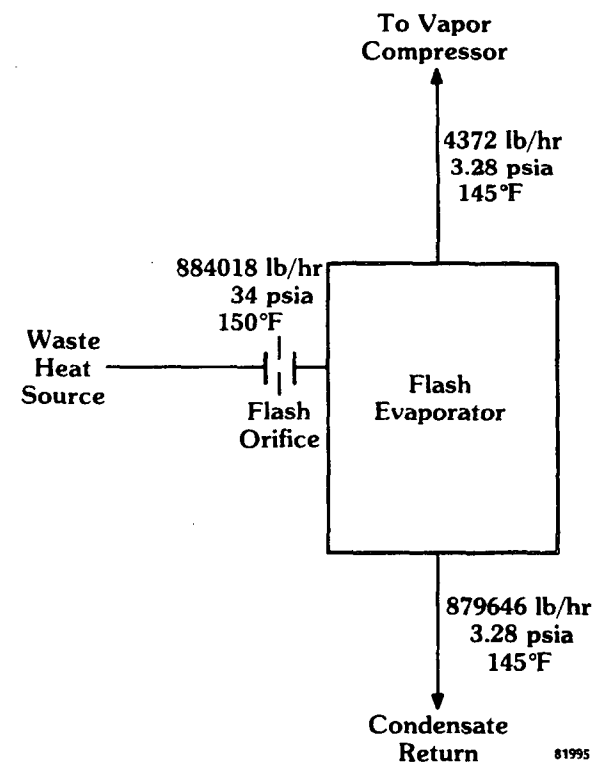


Figure 6 Theoretical Flash Evaporation Process

The above process definition does not include two "real world" factors. Flash tank sizing and vapor/ liquid separation are closely related and are critical influences on the generation of water vapor by low ΔT flash evaporation. The design of the flash tank includes provisions for:

- Generation of 4372 pounds per hour of water vapor at 3.28 psia and 145°F
- An additional vapor/liquid separation mechanism (either a vaned or wire mesh demister)

- 10% additional pumping capacity to facilitate flash inefficiencies (not yet determinable and will vary with every application)
- Sufficient time for evaporation
- Sufficient surface area to stimulate evaporation.

The above features were designed into the system to allow for variances from the ideal flash process. Limited experience does not allow a clear definition of the efficiency of the flash evaporation process for low $\Delta T$ applications. To optimize the flash efficiency, the 150°F water enters the flash tank through a 10-inch nozzle immediately downstream of the flash orifice. The water flow is directed down and distributed into the tank with the aid of a spray dispersal cone. This breaks up the water flow into droplets, which maximizes the water surface area and enhances the heat transfer required for effective evaporation. See Figure 7.

The need for proper liquid/vapor separation is provided by the proper sizing of the internal diameter of the flash tank, plus the inclusion of a vaned or wire mesh demister. The sizing of the flash tank diameter (D), is determined by the following equation:

$$D = \sqrt{\frac{4q}{\pi k\left(\frac{\rho_L - \rho_v}{\rho_v}\right)^{1/2}}} \qquad (2)$$

Here, q is the vapor flow rate ($ft^3$/sec); k is a vessel sizing coefficient (0.35 dimensionless); $\rho_L$ is the density of the liquid ($lb/ft^3$) at flash temperatures; and $\rho_v$ is the vapor density ($lb/ft^3$) at flash conditions. For the pilot plant, the calculated diameter is 2.4 feet. Therefore the design of a 3-foot-diameter flash tank would be more than adequate for proper liquid/vapor separation and will provide a 25% safety factor.

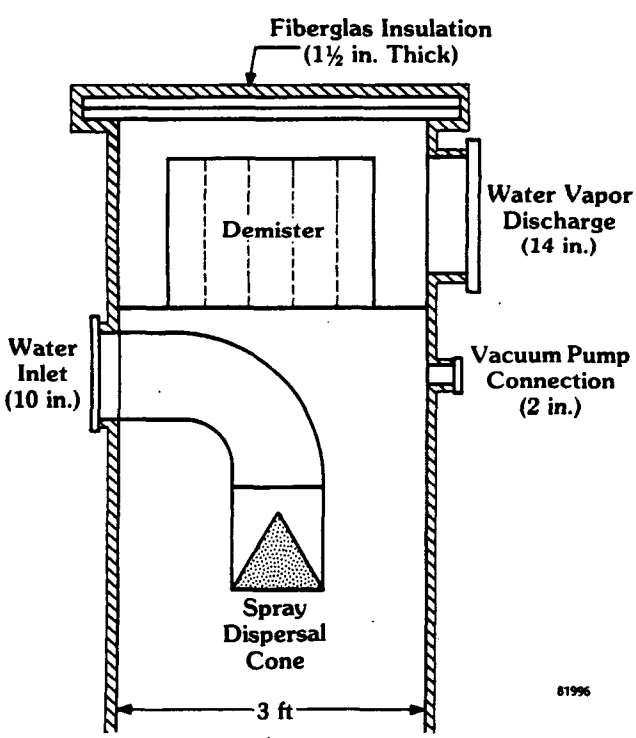


Figure 7 Flash Tank/Process Detail

Vapor Compressors - The most technology-intense components of the open-type industrial heat pumps are the vapor compressors. The compression equipment must be capable of developing high head capacity without a sacrifice in operating efficiency. This combination of performance parameters is effectively met with high-speed, single-stage centrifugal compressors such as that shown in Figure 8. These compressors are electric-motor-driven units with a single step-up gearbox, which also provides the structural support for the compressor assembly. This basic design has resulted in a family of compressors capable of generating up to 45,000 feet of head, or a 2 to 1 pressure ratio with steam, at an efficiency level of 80%. Their maximum flow capacity is 60,000 acfm measured at the inlet.

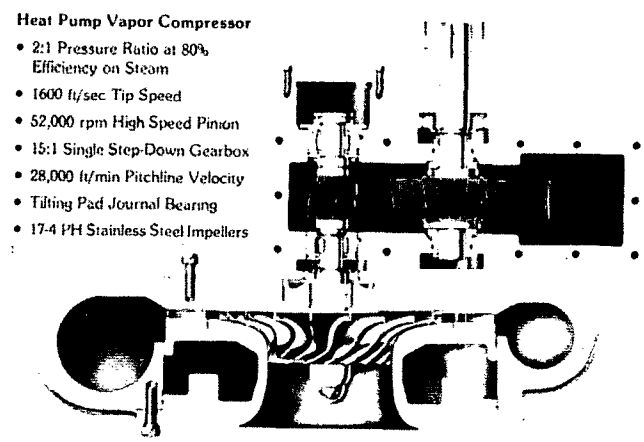


Figure 8 Atlas Copco Turbonetics Single-Stage Centrifugal Compressor Design

The mechanical design of the vapor compressors allows for the implementation of the high-performance strategy and incorporates features such as:

- High-strength, 17-4 PH stainless steel impellers which are standard (titanium alloys are optional)
- Tilting-pad journal bearings for high-speed pinion stability [3]
- Comprehensive rotor-dynamics analysis, both lateral and torsional, for stable, vibration-free operation [4]
- Standard AGMA A13 high-quality gears specified for all gear tooth profiles using case carburized 9310H or nitrided 4140 ground gears.

These design features have been field proven for over ten years. A recent survey [5] examined the service records of 270 Atlas Copco Turbonetics single-stage centrifugal compressors which collectively accumulated 3.2-million operating hours from 1973 to 1979. The survey was conducted to determine the reliability of the compressors, using availability, mean-time-between-failure (MTBF), and mean-time-to-repair (MTTR) as evaluation criteria.

The operating characteristics for the compressors involved in the study were:

- Operating speeds: 8,500 to 52,000 rpm
- Impeller diameters: 6 to 28 in.
- Flow capacity: 1,500 to 30,000 acfm
- Power levels: 250 to 2500 hp.

A total of 168 failures were reported over the seven-year evaluation period. Twenty-five failures were attributed to the compressor/gearbox section of the unit. Based on 3.2-million operating hours, the total on-line availability was 99.80% or 99.97% when only the compressor/gearbox failures were considered. For the study referenced, availability was defined as:

$$\text{Availability} = \frac{\text{Operating Hours} - \text{Repair Hours}}{\text{Operating Hours}} \quad (3)$$

The MTBF for the total system and the compressor/gearbox subsystem was 2.2 years and 14.5 years, respectively, defined as:

$$\text{MTBF} = \frac{\text{Operating Hours}}{\text{Number of Failures x 8760 Hours}} \quad (4)$$

In addition to the data presented above, the survey indicated no gear failures have been reported since 1971.

The strategy required to achieve optimum performance is to operate at high rotational and blade speeds. For water vapor, impeller blade tip speeds on the order of 1500 ft/sec are required. As shown in Figure 9 [6] the blade tip speed and the rotational speed are selected to yield maximum efficiency, based on Balje's specific speed ($N_S$), specific diameter (Ds) and head coefficient ($q_{ad}$) correlation. It suggests the manner in which the compressor's speed, flow and head rise should be arranged to achieve optimum efficiency. For example, a 28-inch diameter impeller rotating at 13,000 rpm (1580 ft/sec blade tip speed) can accommodate 32,000 acfm. Reducing the flow to 2,000 acfm results in a 7-in. wheel rotating at 52,000 rpm to achieve the same 1580 ft/sec tip speed. Both designs result in 80% aerodynamic efficiency.

All the performance and reliability characteristics presented were considered in the vapor compressor's selection. Based on these data and the pilot plant's operating characteristics, Atlas Copco Turbonetics models TP (Twin Pinion)-14/10 and SC-10 were selected. The TP-14/10 was selected in order to minimize space consumption and because it represents a more cost-efficient selection than two single-pinion machines. The cost savings are the result of utilizing a single lubrication system, single drive motor and a common bull gear to operate two single-stage compressors.

Due to the inherent surge characteristics of turbocompressors and the constant operating speed, the system is limited in turndown. As shown in Figure 10, the heat pump's operating range is approximately 30% of the design flow, i.e. 3500 to 5000 lb/hr. The turbocompressor's surge line is roughly 60% of the design flow at the operating speed of the unit. Basic antisurge protection of turbocompressors utilizes 10% of the flow range for control which leaves roughly a 30% range in off-design operation.

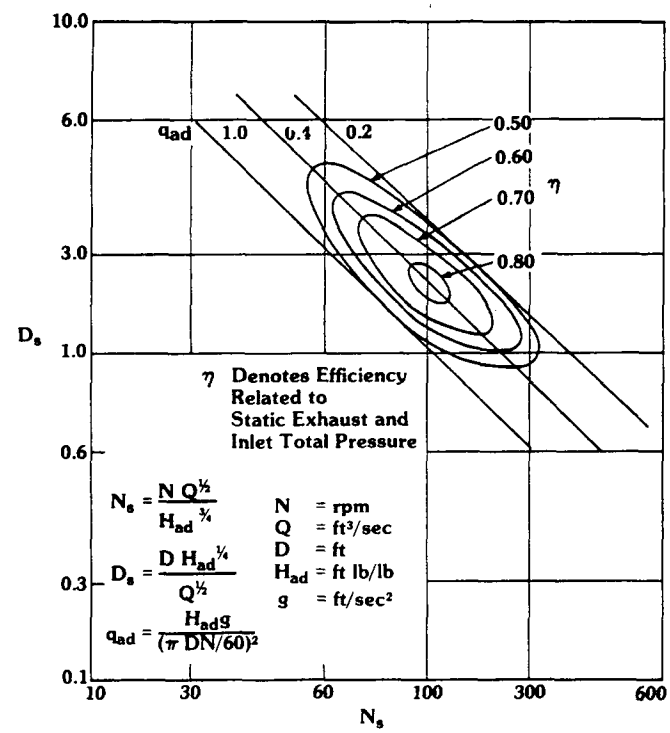


Figure 9 Specific Speed ($N_S$), Specific Diameter ($D_S$) Relationship for Centrifugal Compressors

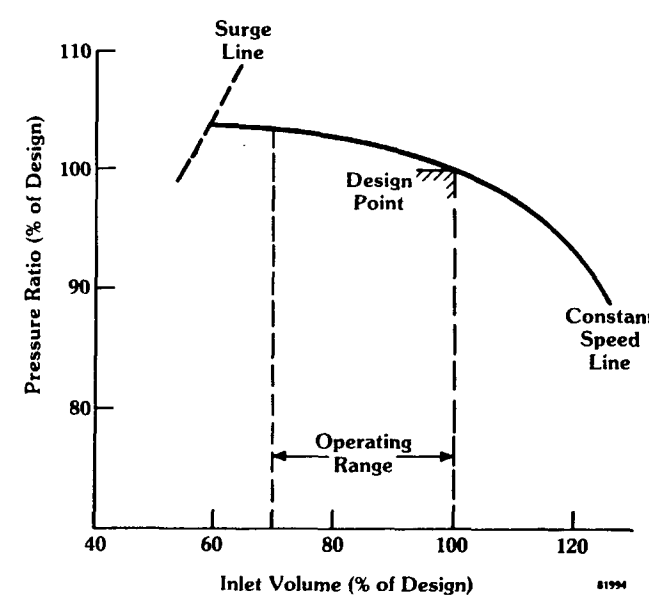


Figure 10 Typical Operating Characteristics of Centrifugal Compressors

Motor Control Center (MCC) - The pilot plant features a single point utility power connection. The system is designed to utilize 480-Volts AC electric power as the input to the MCC. The MCC houses the system's power measurement instruments, motor

starters and contactors in a common location to facilitate a prepackaged design and maintainability. The MCC requirements for the pilot plant include:

- Two 50% reduced voltage starters
- Six standard NEMA motor starters
- Three sets of electric heater controls
- Two step-down transformers (480 Volts AC to 120 Volts AC)
- Power measurement equipment.

Microprocessor-Based Control System - The heat pump control strategy follows a typical pilot plant format. All subprocesses of the heat pump are instrumented to such a degree as to provide information in both steady-state and transient operating modes. In order to accomplish this task, the control system is able to provide for:

- Start-up sequencing
- Dynamic process control
- System alarm annunciation
- Trend mapping of specific parameters.

The accomplishment of the control tasks will be handled by a microprocessor-based control system. This system is capable of handling all dynamic process controls, such as compressor antisurge control, without any manual interface requirement utilizing instrument-generated, discrete and analog, inputs and outputs. The system allows the monitoring and control functions to be located at a central operator interface console which includes:

- CRT monitor (color)
- Video trend controller
- Limited-access operation interface panel
- Output printer.

The CRT monitor has the capability to display any control loop and/or parameter of interest along with providing the visual record of set points, operating point, and alarm points. The video trend controller monitors and records specified parameters for historic trends over a specific time interval. The video trend controller also has the capability of providing hard copies of the trended parameters with the inclusion of the output printer in the system.

Operator interface with the control system is provided with a limited-access panel. The panel (keyboard) allows the operator to change various parameter setpoints and alarm points, but does not allow the operator to change the control algorithm. This function can only be performed with the use of an access key. With this limited-access feature, the security of the control function can be easily maintained.

## SUMMARY

The heat pump design presented in this paper represents the system needs for a pilot demonstration plant. The plant's objective is to verify, through strenuous steady-state and transient modes of operation, the open-type heat pump concept. The system includes instrumentation capabilities to provide informational data on normal operation and on possible system upsets that could be expected during the heat pump's operation.

The vapor compressors selected provide both high-performance capability and reliable operation. High-performance vapor compression enhances the overall heat pump performance (COP) and, thus, its commercial attractiveness. The compressors selected are of a common generic family which reduce scale-up uncertainties and have proven field reliability.

The control philosophy used for the pilot plant has direct application to the existing industrial world of process control. Microprocessor-based control systems, similar to the one presented, are gaining great acceptance in the industrial markets for process plant control applications. The control feature, as defined, will provide additional interface expertise with near state-of-the-art control equipment and techniques. Larger heat pumping systems will employ the same control features with the only difference being a duplication of strategies utilized on the pilot demonstration plant.

## ACKNOWLEDGEMENTS

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## REFERENCES

1. "Industrial Applications Study," DOE Report HCP/T 2862-01, March 1978.

2. Leibowitz, H.M. and Colosimo, D.D. "An Evaluation of Industrial Heat Pumps for Effective Low Temperature Heat Utilization." Presented at the Second Annual Conference on Industrial Energy Conference Technology and Exhibitions, April 1980.

3. Pennick, H. "The State of the Art of High-Speed Overhung Centrifugal Compressors for the Process Industry." Presented at the 7th Turbomachinery Symposium, December 1978.

4. Malanoski, S.B. "Rotor Bearing System Design Audit." Proceedings of the 4th Turbomachinery Symposium, Gas Turbine Laboratories, Texas A&M, October 1975

5. Leibowitz, H., Rochow, K., and Bryant, T. "The Reliability of High-Speed Turbomachinery: A Survey of Single-Stage Centrifugal Compressors." Mechanical Technology Incorporated Report 80TR34, June 1980 (unpublished).

6. Balje, O.E. "A Study of Design Criteria and Matching of Turbomachines." Journal of Engineering for Power, January 1962.

# OPEN CYCLE HEAT PUMP FOR INDUSTRIAL WASTE HEAT UTILIZATION

GENERAL ELECTRIC COMPANY
CORPORATE RESEARCH & DEVELOPMENT
SCHENECTADY, N.Y. 12345

CONTRACT NUMBER 85X24717C
C. R. SNYDER
CONTRACT BEGAN MAY 12, 1981
PHASE I COMPLETE OCT. 12, 1981

## OBJECTIVE

The objective of this contract is to provide information on open cycle industrial process heat pump which would lead to a demonstration in an industrial setting. The demonstration site was to be chosen from sites in New York state on the basis of greatest technical and economic merit. The selection of New York State sites reflected preliminary indications from site management for cooperation and New York State Energy and Development Authority's involvement.

## BACKGROUND

The open cycle industrial process heat pump technology is a cost-effective method of utilizing an industrial plant's waste heat. Phase I of this contract concentrated on the evaluation of three sites and four process configurations. The basis for the initial screening provided opportunities to exploit the features of the open cycle arrangement.

Site visits yielded process information on flow rates, process temperatures, pressures, dynamic behavior, energy balances, control functions and plant operating philosophy. Additional information was obtained relating to plant utilities and equipment location. Comparative evaluations were made considering energy savings, cost savings, value of operating experience and the potential for wide application.

Additional studies in the area of energy usage were completed for New York State Energy Research and Development Authority. NYSERDA was interested in assembling energy cost data on a state-wide basis, and using the application criteria from site visits in determining the potential for other similar applications.

The site study conducted at a New York City food processing plant provided two potential applications. The first of these applications was to recover from a liquor evaporator 2.4 MB TU/Hr of energy at a level of 3000 lbs/hr of 233$^{o}$ F steam. This application provided constant and near continuous operation. However, the facility in close proximity to the evaporator did not allow sufficient room for equipment installation.

The second application had the potential of reducing the plant's waste treatment costs by flashing the cooling water discharge from a barometric condensor. This application unfortunately varied in operating temperature from 110$^{o}$ to 160$^{o}$ and would be operated less than 4000 hours a year.

A second site at a large chemical processing plant provided a very large waste stream (5000 GPM) which could yield 25000 lbs/hr of process level steam. This site was an excellent opportunity for a very large demonstration. The application of an industrial process heat pump would be very attractive economically in replacing oil-fired steam capacity and at the same time closely match process steam requirement levels.

A third site in a paper processing plant provided a fourth application in providing process level steam from a hot condensate return system. The 35000 lb/hr paper machine application would allow a small demonstration of the equipment with a marginal economic return.

Budgetary equipment cost information was obtained from various compressor, motor, gear, flash evaporator, and pump manufacturers to justify technical and economic feasibility. Installation estimates were made at the paper and chemical plant locations to further identify costs related to construction and inter-connection requirements.

## SUMMARY

Phase I indicated the potential at two locations. The larger application has the most attractive economic return and opportunity for operational experience.

The evaluation of energy data bases yielded many potential application opportunities. Studies of industrial waste streams have long been identified as applications for heat pump technology. The site specific activities allowed for a better realization of the actual opportunities.

## TECHNICAL ACCOMPLISHMENTS

- The technical and economic feasibility of the open cycle industrial process heat pump was established.
- Compressor hardware exists to exploit the opportunities of heat pumping.

## FUTURE ACTIVITIES

Contract Activities - Presently a change in work scope has been requested by Oak Ridge. This change has been returned to Oak Ridge with our comments. A meeting will be held in the near future to discuss the possibility of continuing this contract.

Post Contract Activities - Two activities are needed to assure acceptance of this technology. First the performance of flash evaporators must be better understood. Much literature has been published on similar flashing systems but little information exists on the process dynamics, directly related to its utilization in an open cycle industrial process heat pump.

The second is the demonstration of a complete system. Until the industrial sector recognizes this technology and a unit has demonstrated reliable, economic operation, the technology will remain in a study mode.

## PUBLICATION

Draft Report issued October 12, 1980 to Oak Ridge National Laboratory.

Conceptual Design of Large Scale Heat Pump for Central Energy Management System

United Technologies Research Center

R. W. Bass, A. M. Landerman, and F. R. Biancardi

Subcontract DE-AC09-77ET12866-Sub-15

## OBJECTIVE

The objective of this study was to select systems comprised of current and near-term equipment capable of economically recovering 110 to 150 F waste heat from federal facilities, utilities and industries to produce electric power and thermal energy at useful temperatures. At federal facilities and utilities where large amounts of waste heat are available, recovery and distribution of energy is handled by a Central Energy Management System (CEMS) and utilized in an adjacent industrial park. For industrial waste heat recovery, recovered energy is utilized on-site.

## CONCEPT

A CEMS is defined as the utility serving an industrial park and which will purchase or generate on-site all electrical and thermal energy needs of the industries in the park from waste heat and/or fossil fuel. The CEMS for a particular source of waste heat is determined by matching the desired electrical and thermal requirements of the industrial park (or other energy recipient) to the characteristics of the recovery technologies and dividing the source between the technologies to obtain the greatest return on investment. During this study, both heat pump and power generation systems capable of recovering 110 to 150 F waste heat were studied and optimized separately in a manner which permits economic assessment of joint operation with a source insufficient to meet the entire desired energy output. Emphasis will be placed on discussing the heat pump concepts although the power system concepts will also be examined.

The heat pump design applications were defined as shown in Table 1. These applications reflect the finding of this and earlier studies (Refs. 1-3) that most available nongaseous waste heat is at temperatures at or below 160 F and most industrial processes require steam at approximately 300 F.

## SUMMARY

An analytical study was undertaken to identify specific power generation and heat pump requirements for low-temperature waste heat recovery from government, utility and industrial sources, develop techniques needed to evaluate the performance and cost characteristics of various power system and heat pump options, and perform economic analyses on selected designs. Recently completed, DOE-sponsored studies as well as other available literature, were reviewed to define waste heat sources and industrial thermal end-uses. Distinct waste heat sources and industrial end-uses were selected as representative applications and used for heat pump design specifications. Two of these applications typify large sources of waste heat and co-located industrial park end-uses while the remaining four represent in-plant industrial sources and end-uses. Rankine-cycle heat pump configurations and working fluid options were assessed and four configurations were selected on the basis of cost and performance for further study. Five different prime movers were examined as potential compressor drivers and gas turbines and diesels were selected on the basis of high overall performance. A similar design assessment was performed for power recovery systems. A conceptual design of an economically attractive central energy management system for use at an industrial park located nearby the Portsmouth gaseous diffusion plant is presented.

## TECHNICAL ACCOMPLISHMENTS

- Heat Pump Configuration Selection. Four heat pump configurations were examined: closed, semi-open, open, and dual fluid. The most common heat pump configuration uses a single working fluid in a closed loop. When the required output is steam, it can be used as the working fluid in a semi-open configuration and thereby eliminate the cost and performance penalty associated with the condenser. Furthermore, if the waste heat stream is clean water, it can be flashed to steam and used in an open-

| WASTE HEAT SOURCE | | | END-USE APPLICATION | | |
|---|---|---|---|---|---|
| LOCATION | FLOW RATE $10^6$ lb/hr | TEMP. F | LOCATION | HEAT REQUIREMENTS $10^6$ Btu/hr | TEMP. F |
| FED. FACILITIES | 20 | 140-165 | IND. PARK | 400-1000 | 300 |
| UTILITY COOLING TOWER | 20 | 105-121 | IND. PARK | 400-1000 | 300 |
| INDUSTRY | 0.2 | 95-120 | IN PLANT | 10 | 300 |
| INDUSTRY | 2.0 | 95-120 | IN PLANT | 100 | 300 |
| INDUSTRY | 0.3 | 130-160 | IN PLANT | 30 | 300 |
| INDUSTRY | 0.5 | 170-200 | IN PLANT | 100 | 300 |

**Table 1 Candidate heat pump applications**

cycle configuration thereby also eliminating the evaporator. A fourth configuration can be made using two fluids: one with characteristics suited to the low temperature portion of the heat pump and the other, steam, which can be heat pumped to achieve the desired temperature.

Table 2 shows the results of the analysis of a wide variety of working fluids in these configurations for application at the Portsmouth GDP. Performance is indicated by COP for the condition where 150 F waste heat is used to provide 300 F saturated steam. The cost of the major components of each system assuming a fixed heat input is given relative to the least costly system. Furthermore, the ratio of this cost to the fuel savings assuming an electric motor drive is given relative to the most cost effective system. These values can only be used to compare systems with similar configurations. They cannot be used to make a final system selection since the units have different outputs, the driver is not optimized and the installation costs are not included. The use of these results provided a simple method of narrowing down the number of potential working fluids for further evaluation. For example, heat pumps employing toluene, hexafluorobeneze, Fluorinol-85 and R216/steam are too costly. The R113 system has low COP resulting in overall economics that are less favorable than other options. The poor COP is due in part to the low critical temperature of the fluid relative to the desired output temperature of the heat pump. Finally, closed steam, propanol and R133a/steam systems show no advantage over the selected systems. The remaining systems (methanol, semi-open steam, open steam and R11/steam) were selected for further consideration.

WASTE HEAT FLUID = WATER AT 150F
OUTPUT = 300 F STEAM
HEAT INPUT = 200 x $10^6$ Btu/hr

| OPTIONS | COP | RELATIVE MAJOR EQUIPMENT COST | RELATIVE COST TO ENERGY SAVING RATIO |
|---|---|---|---|
| SINGLE-FLUID, CLOSED | | | |
| STEAM | 3.30 | 1.80 | 1.68 |
| R113 | 2.80 | 1.05 | 4.57 |
| METHANOL | 3.16 | 1.13 | 1.29 |
| PROPANOL | 3.16 | 2.93 | 3.36 |
| TOLUENE | 3.06 | 4.95 | 5.67 |
| HEXAFLUOROBENZENE | 2.88 | 11.90 | 25.27 |
| FLUORINOL-85 | 3.06 | 2.78 | 3.81 |
| SEMI-OPEN | | | |
| STEAM | 3.37 | 1.56 | 1.33 |
| OPEN | | | |
| STEAM | 3.52 | 1.37 | 1.00 |
| DUAL-FLUID | | | |
| R11/STEAM | 3.14 | 1.04 | 1.24 |
| R133a/STEAM | 2.93 | 1.00 | 1.84 |
| R216/STEAM | 2.80 | 3.70 | 10.35 |

**Table 2 Comparison of heat pump options for federal facility application**

● Heat Pump Driver Selection. Shaft power can be supplied to the heat pump by a; (1) gas turbine, (2) diesel, (3) steam turbine, (4) electric motor or (5) waste heat recovery Rankine system. Each is characterized by a power conversion efficiency (i.e., the percent of input fuel convertible to useful shaft work) and a heat recovery efficiency (i.e., the percent of input fuel recoverable as useful (i.e., 300 F) process heat) (see Table 3). These characteristics are used with typical heat pump COP values to determine the ratio of useful heat output to fuel input as a function of output temperature for a 150 F waste heat input (Figure 1). Gas turbines and diesels offer high fuel conversion ratios. Back pressure steam turbines have lower fuel conversion ratios and condensing steam turbines and electric motors have even lower ratios since no useful prime mover heat recovery is permitted. Waste heat driven-Rankine cycle systems do not use fuel and their economics must be evaluated on a site-specific basis.

| | POWER EFFICIENCY | HEAT RECOVERY EFFICIENCY |
|---|---|---|
| GAS TURBINE | 0.32 | 0.43 |
| DIESEL | 0.42 | 0.16 |
| STEAM TURBINE | | |
| • BACK PRESSURE | 0.17 | 0.68 |
| • CONDENSING | 0.30 | 0 |
| ELECTRIC MOTOR | 0.33 | 0 |
| WASTE HEAT DRIVEN RANKINE CYCLE | ∞ | UNDEFINED |

**Table 3 Prime mover characteristics**

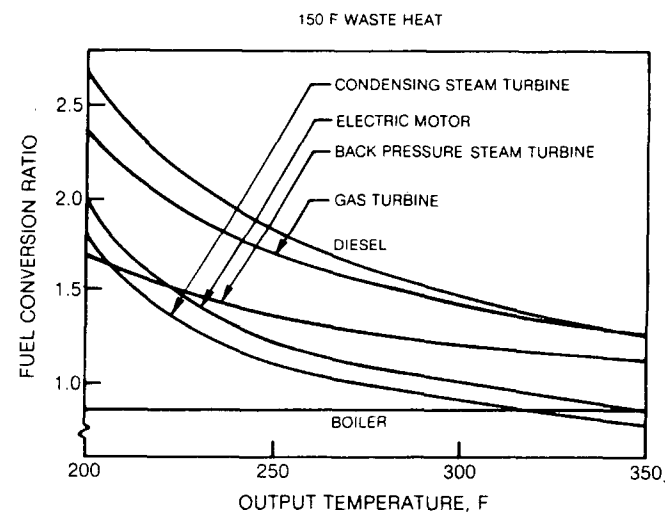



**Figure 1 Heat pump driver comparison**

● Waste Heat Power Recovery. The only thermodynamic cycle capable of efficiently generating power from the low-temperature waste heat, (at 150-200 F) is the Rankine cycle. As part of this study, the economic viability of Rankine power systems was determined for such working fluids as $NH_3$, R11, R113, R114, R500, and steam. The results show that $NH_3$ is the preferred working fluid for these low-temperature applications because of its superior heat transfer, thermodynamic and turbomachinery characteristics relative to the other fluids. An ammonia system would be 50 percent less costly in the 1 to 30 MWe size range than its nearest competition--an R500 dual pressure system. Figure 2 shows a conceptual design for a 30 MWe $NH_3$ system for the Portsmouth GDP.

● Heat Pump Economic Analysis. Economic comparisons of the various heat pump configurations which passed the initial screen were made. Table 4 shows the before-tax, rate-of-return for the four configurations with a gas turbine drive. In each case the cost and performance is compared to a boiler providing the same thermal output. The before-tax return-on-investment shows that the dual-fluid configuration has a slight advantage over the other configurations. Analysis of an application where the waste heat is at 110 F (typical of utility

cooling tower water) and 300 F steam is required, shows that the dual-fluid configuration is superior relative to the others although the levels of the return on investment are lower than for the 150 F waste heat case. Analysis of the various heat pump drivers shows the gas turbine can be superior to either the diesel or the steam turbine depending upon steam delivery temperatures.

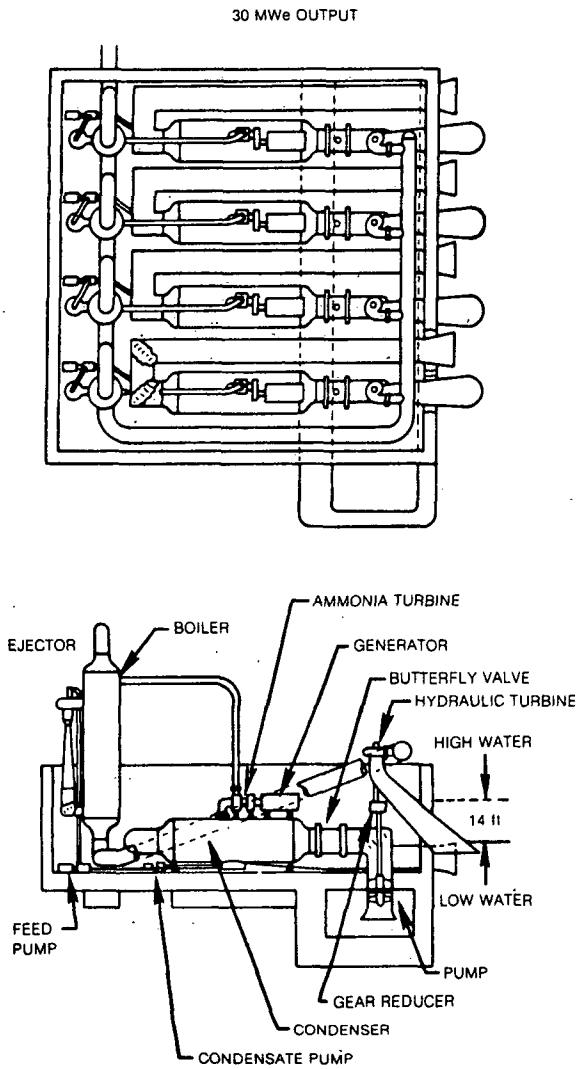



**Figure 2 Power recovery system conceptual design**

GAS TURBINE DRIVE
WASTE HEAT TEMPERATURE = 150 F
OUTPUT = 400 x $10^6$ Btu/hr OF 300 F STEAM

| CONFIGURATION | COPF | ANNUAL FUEL CONSUMPTION, $10^{12}$ Btu | TOTAL SYSTEM INSTALLED COST, $\$10^6$ (1980) | BEFORE TAX RETURN ON INVESTMENT, % |
|---|---|---|---|---|
| BOILER | 0.85 | 3.76 | 3.2 | BASE |
| R11/STEAM | 1.41 | 2.27 | 20.5 | 36.2 |
| METHANOL | 1.42 | 2.26 | 21.4 | 34.8 |
| SEMI-OPEN STEAM | 1.48 | 2.16 | 24.6 | 32.0 |
| OPEN STEAM | 1.52 | 2.10 | 22.5 | 35.7 |

**Table 4 Comparison of heat pump systems with boilers for Portsmouth application**

• CEMS Economics. A demand of 29 MWe and 400 x $10^6$ Btu/hr of 300 F saturated steam is typical of an industrial park that could be located near the site of the Portsmouth GDP. Figure 3 shows the effect of dividing the available flow between a gas turbine-driven heat pump and a waste heat recovery Rankine power system. The maximum power recovery (zero waste stream to heat pump) is about 20 MWe. The total process heat requirement can be met by using only 30 percent of the available heat. Since the rate of return is greatest at this point, the CEMS should be matched to these conditions. Figure 4 shows a conceptual design of a CEMS for an industrial park at the Portsmouth GDP where all electricity is generated by gas turbines and all process heat produced by heat pumps and exhaust heat recovery.

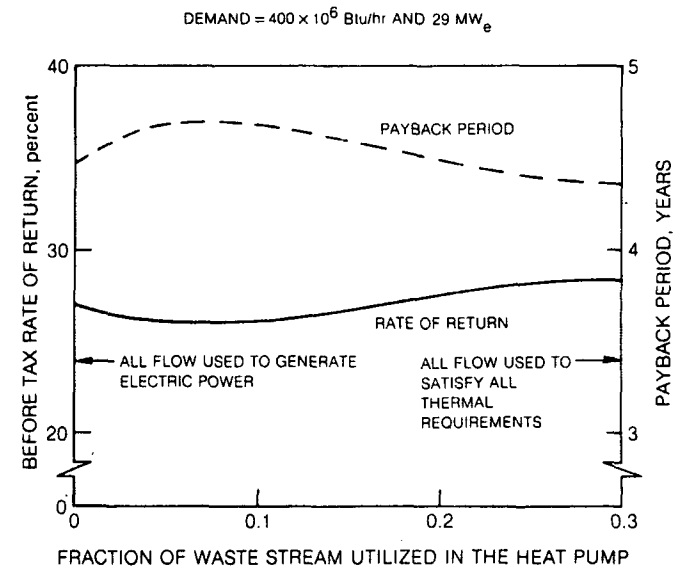



**Figure 3 Effect of flow split on the economics of a CEMS at the Portsmouth GDP**

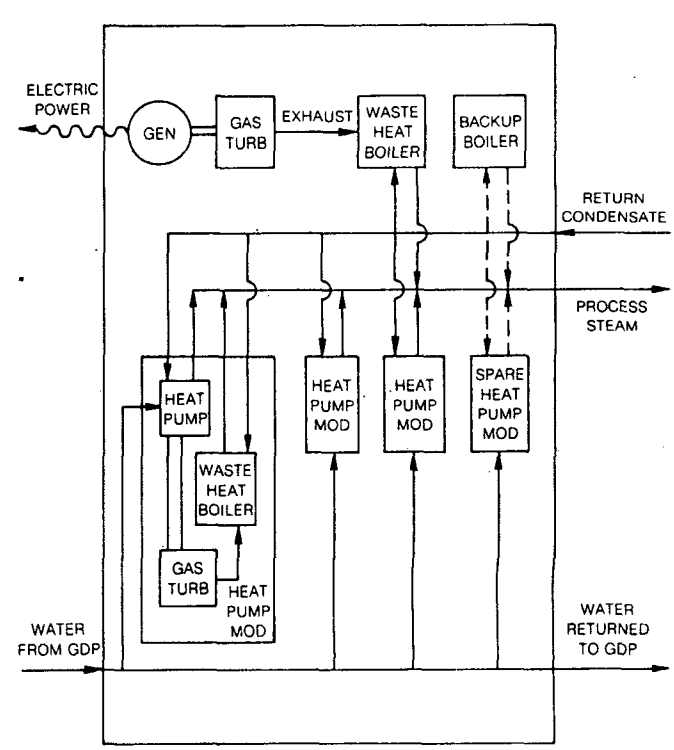



**Figure 4 Schematic of Portsmouth CEMS facility**

## CONCLUSIONS

- Various heat pump configurations using 110-150 F waste heat to provide 300 F steam show acceptable return on investment. However, the highest investment value is associated with a gas turbine-driven, dual-fluid system.

- Very large output (>100 x $10^6$ Btu/hr) heat pumps show the most attractive energy savings and investment benefits.

- Both heat pump and power generation concepts can be implemented to recover and utilize federal, utility and industrial waste heat.

- A Central Energy Management System (CEMS) provides an attractive approach to waste heat recovery by permitting economics of scale to enhance system return on investment.

- Fuel costs and availability, application requirements must be assessed in detail to determine the best approach for the CEMS energy conversion equipment.

## PUBLICATIONS/REPORTS/REFERENCES


1. Residual Energy Applications Program - Systems Analysis Report, South Carolina Energy Research Institute for U.S. Department of Energy, Report DOE/ET/12866-3, October 1980.
2. Preliminary Assessment of Waste Energy Recovery from the Savannah River Plant - Industry Characterization, United Technologies Research Center for South Carolina Energy Research Institute, Report R78-954137-20.
3. Industrial Applications Study, Drexel University for U.S. Department of Energy, Report HCP/T2862, March 1978.


## ACKNOWLEDGMENT


This study was performed by the United Technologies Research Center under Subcontract No. DE-AC09-77ET12866-SUB-15 to provide assistance to the South Carolina Energy Research Institute for their prime contract with the United States Department of Energy, Power Systems Division.

BRAYTON-CYCLE HEAT PUMP FOR SOLVENT RECOVERY

AIRESEARCH MANUFACTURING COMPANY
2525 W. 190TH STREET, TORRANCE, CALIFORNIA 90509

CONTRACT NUMBER DE-AC03-78C540005

GILBERT E. LIMBERG

PERIOD OF CONTRACT:
BEGIN SEPTEMBER 1978 END MAY 1983

## OBJECTIVE

The current objectives of this program are to develop a Brayton-cycle heat pump solvent recovery system to a production-prototype level, analyze the market and business potential for adaptation to this solvent recovery system, evaluate the system first cost and actual annual performance and operating costs, and field-demonstrate a system to obtain performance/cost data and to identify problems in actual service.

## CONCEPT

The inherent cooling and heat pumping capabilities of the Brayton cycle lend themselves to solvent recovery systems. The fundamentals of the cycle are shown in Figure 1. The solvent-laden gas supplied to the system may range in temperature from 70° to 500°F, but usually the range will be from 150° to 450°F. Some precooling may be used either to reduce the regenerated gas (air) temperature delivered back to the process or to reduce the motor input power by lowering the compressor inlet temperature. The power-driven turbine/compressor is designed for a pressure ratio across the compressor of approximately 2.5, and the turbine operates at a pressure ratio that is slightly less because of pressure losses. The heat of compression in the compressor discharge is removed in the regenerator, which provides regenerative heat to the gas stream flowing to the process emitting the solvent vapor. The condenser cools the high-pressure solvent-laden gas until a portion of the solvent vapor condenses on the cold walls. Depending on the application, the condensing temperatures may be below the freezing temperature of water; in these instances special designs to circumvent icing problems will be required.

The turbine inlet temperature is further reduced by the expansion process. The power developed by the turbine is absorbed by the compressor, which also requires supplemental power from the motor. Depending on the system/solvent, all of the condensed solvent may be collected in the condenser at the high pressure, or a portion may be collected in the turbine discharge; the latter would be in the form of very fine droplets. A coalescer separator is incorporated to provide for separation in these cases. For the closed-loop (oven recirculation) systems, less cooling is used (to save power), and the return gas to the process has a solvent concentration of approximately 500 ppmv. For the open-loop oven systems, where the gas (air) is discharged to ambient, much more cooling is required from the turbine to reduce the solvent concentrations to 20 to 50 ppmv.

## SUMMARY

The solvent-using industries are lagging in recognizing the impact of the recently accelerated fuel price increases, and studies show that solvent recovery is an excellent means of reducing these costs. The Federal Clean Air Act requiring that more stringent clean air emission levels be implemented by December 31, 1982, is another incentive for these industries to enhance solvent recovery. The versatility of the Brayton cycle allows it to be integrated into a variety of processes, and considerable energy and cost savings can accrue over those obtainable with the normally used carbon-steam desorption solvent recovery systems. The annual energy savings to be realized by installing an 8000-cfm Brayton-cycle system at the demonstration site is approximately 56 x $10^9$ Btu/yr and simple cost paybacks of less than 1 yr are achievable. The estimated market potential for the Brayton cycle in 1980 in the U.S. is the removal of 6.3 million tons of volatile solvents (56 percent of the estimated recoverable solvents). The potential net energy savings is 0.11 quads better than for a carbon-steam desorption system.

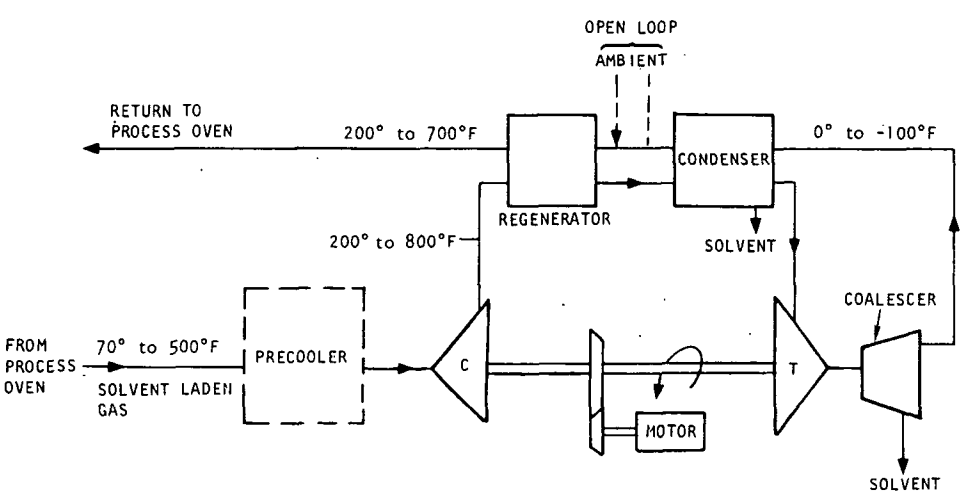



Figure 1. Brayton-Cycle Concept

TECHNICAL ACCOMPLISHMENTS

- The flexibility of the Brayton cycle is illustrated in Figure 2, which shows that any one of three different condensing processes may be utilized in place of the carbon adsorption systems utilizing steam desorption that are presently the most common approach. Because many solvents are not compatible with carbon, which can cause acids to be formed, removal techniques other than those presently used are desirable.

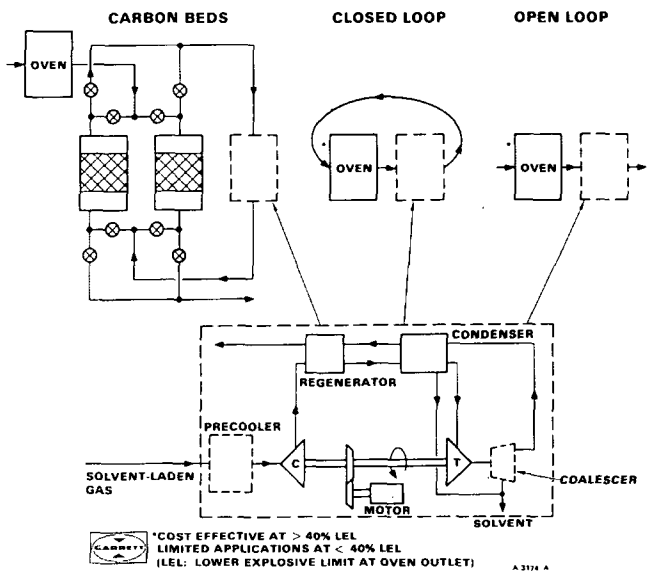



Figure 2. Flexibility of Brayton-Cycle Solvent Recovery

When operating the equipment in the closed loop, or oven recirculation mode, the flammability limits of the solvent are the criterion for design. Insurance underwriters limit the solvent concentration to 25-percent lower explosion limit (LEL) for systems without adequate instruments. With good instrumentation, the allowable maximum concentration is 50-percent LEL. Good instrumentation requires continuous vapor concentration indicators and controllers, proper maintenance, and automatic shutdown or startup of auxiliary equipment at a vapor set point not exceeding 50-percent LEL.

When the Brayton-cycle equipment is operated in the open-loop mode, very low operating temperatures are required to minimize the solvent concentration in the air vented to the atmosphere. For the Brayton-cycle equipment to perform as effectively as the existing carbon absorption-steam desorption systems, the LEL of the oven exhaust must be at least 40 percent.

Where carbon is compatible this equipment can also be used to desorb carbon beds that have been used to absorb the solvent contained in the air that leaves the ovens. In this method, the gas carrier (nitrogen) is used to strip the solvent instead of steam, as is the present method. Because the Brayton system provides all the heat necessary to increase the absorbing bed temperature to a desorb level (approximately 400°F) and the heat to desorb the solvent physically (somewhat greater than the solvent latent heat of vaporization), a relatively high flow of nitrogen is required. The Brayton-cycle system provides better carbon desorption than does steam and will, therefore, provide a weight savings of the carbon on an average of 50 percent. For a carbon-bed system originally designed with 30,000 lb of carbon per bed and which cycles from absorb to desorb once every 6 hr, the average removal rate is 500 lb/hr; by comparison, when desorbed with the Brayton system, the capacity is increased to 750 lb/hr. It is expected that the coabsorption of water will be approximately 2 percent, or a removal rate of 100 lb water/hr.

Table 1 shows the average energy utilization for using the Brayton cycle to separate a pound of solvent as compared with the industry standard method of steam stripping carbon beds. At today's energy costs, it is expected that the Brayton-cycle system will pay for itself in energy savings in less than one year.

Potential applications are shown in Table 2.

- A plot of expected market trends based on a growth rate of 3 percent per year is presented in Figure 3.

Total emissions for 1980 in the U.S. were 19.3 million tons, and it is estimated that 11.3 million tons (57 percent of total emissions) could have been collected and recovered. The estimated market potential for the Brayton cycle is 6.3 million tons of solvents (56 percent of the estimated recoverable solvents); this potential is based on application of the Brayton cycle as shown in Table 3.

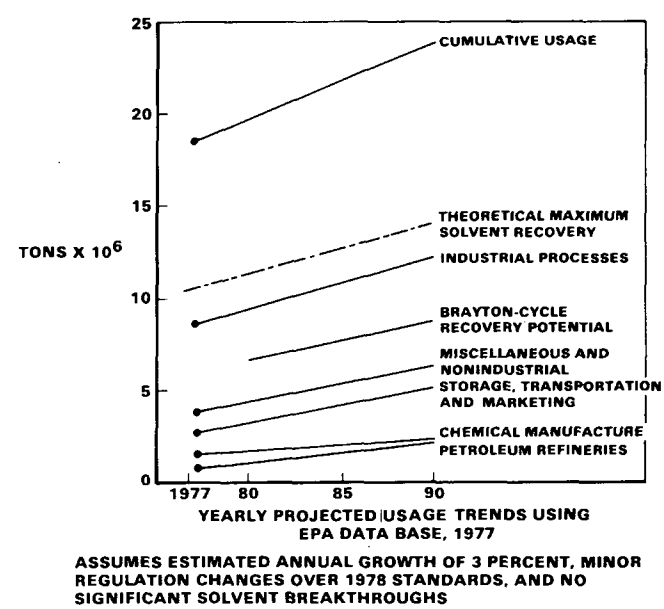



Figure 3. Volatile Organic Usage Trends from Stationary Sources

TABLE 1

UTILIZATION OF PRIMARY ENERGY

| System | Primary Energy Used to Separate 1 lb of Average Solvent,* Btu/lb solvent |
|---|---|
| Carbon absorption, steam desorption (reference system) | 10,400** |
| Closed-loop Brayton cycle | |
| <40-percent LEL | 5,500 |
| >40-percent LEL | 2,500 |
| Open-loop Brayton cycle | |
| <40-percent LEL | 10,000 |
| >40-percent LEL | 5,000 |
| Brayton cycle, carbon desorption | 1,400 |

*Electrical energy conversion of 10,500 Btu/kwhr used to take in account electrical power generation losses
**Assumes steam distillation is utilized

TABLE 2

BRAYTON-CYCLE POTENTIAL SOLVENT RECOVERY APPLICATION MATRIX

| System | Lower Explosive Limits Normally Less Than 40 Percent | Lower Explosive Limits Greater Than 40 Percent |
|---|---|---|
| Closed-loop Brayton cycle | Varnish cooking (undesirables present)<br>Pharmaceutical synthetic drying<br>Can coating<br>Coil coating | Oil extraction--soybean/cotton/lemon<br>Plastic polymer drying<br>Varnish and ink cooking potential<br>Adhesive label<br>Paper coating<br>Tape coating<br>Fabric coating<br>Dry cleaning<br>Synthetic fiber and film (polymers, dry and melt spinning)<br>Degreasing--hot vapor<br>Natural fiber cleaning |
| Open-loop Brayton cycle | Paint manufacture<br>Varnish thinning<br>Auto/metal coating | Petroleum storage and loading<br>Plastic polymer production<br>Pharmaceutical synthetic production<br>Degreasing, cold and conveyorized |
| Brayton cycle, carbon desorption | Paint manufacture<br>Food processing<br>Auto/metal coating<br>Wood coating<br>Dry cleaning<br>Rotogravure and flexography<br>Letter press and lithography<br>Rubber product manufacture | |

TABLE 3

SUMMARY OF ANNUAL VOLATILE ORGANIC EMISSIONS SUITABLE FOR RECOVERY ($1 \times 10^6$ TONS/YR)

| Industry Sources (Emissions) | Brayton Closed | | | Brayton Open | | | Brayton Desorb | | |
|---|---|---|---|---|---|---|---|---|---|
| | Recovered | Percent of Industry | Percent of Total | Recovered | Percent of Industry | Percent of Total | Recovered | Percent of Industry | Percent of Total |
| Petroleum | | | | | | | 1.2620 | 30.0 | 6.3 |
| <40 percent LEL | | | | | | | | | |
| (4.206) >40 percent LEL | | | | 0.5410 | 12.9 | 2.7 | | | |
| Organics | | | | | | | 0.5410 | 30.0 | 2.7 |
| <40 percent LEL | | | | | | | | | |
| (1.803) >40 percent LEL | | | | | | | | | |
| Industrial Processes | | | | | | | 0.4871 | 34.0 | 2.4 |
| <40 percent LEL | 0.0522 | 3.7 | 0.3 | 0.0114 | 0.8 | 0.1 | | | |
| (1.427) >40 percent LEL | 0.3627 | 25.4 | 1.8 | 0.1625 | 11.4 | 0.8 | | | |
| Surface Coating | | | | | | | 0.5164 | 22.0 | 0.3 |
| <40 percent LEL | 0.2018 | 8.7 | 1.0 | 0.2736 | 11.7 | 1.4 | | | |
| (2.3298)>40 percent LEL | 0.6482 | 27.8 | 3.3 | | | | | | |
| Other Solvent Uses | | | | | | | 0.7122 | 46.0 | 3.6 |
| <40 percent LEL | | | | 0.0459 | 3.0 | 0.2 | | | |
| (1.5485)>40 percent LEL | 0.3274 | 21.1 | 1.6 | 0.1159 | 7.5 | 0.6 | | | |
| Totals | | | | | | | 3.5187 | 31.1 | 17.7 |
| <40 percent LEL | 0.2540 | 2.2 | 1.3 | 0.3309 | 2.9 | 1.7 | | | |
| (11.32) >40 percent LEL | 1.3383 | 11.8 | 6.7 | 0.8194 | 7.2 | 4.1 | | | |
| Grand Total: 6.27, or 55.4 percent of industry and 32 percent of totals | | | | | | | | | |
| Totals = $19.927 \times 10^6$ tons/yr | | | | | | | | | |

- The results of site surveys indicate that

  (a) New technology is needed to recover a portion of the annual volatile organic solvents used in the U.S.

  (b) The Brayton cycle is a viable recovery system and has preliminary industrial acceptance

The industries surveyed are becoming more aware of the financial potential for recovering solvents and saving energy through the application of engineering technology. There is no single solvent recovery solution, i.e., carbon is not always compatible and incineration should be limited to more exotic particulate contamination applications. The versatility of the Brayton cycle allows it to fit nicely as a primary solvent recovery system or as a secondary energy saving carbon desorption system.

The three major industries that have potential for a Brayton-cycle solvent recovery system were found to be:

(a) Industrial surface coatings

(b) Industrial processes

(c) Other solvent users

FUTURE ACTIVITIES

Presently the program is in the design phase. Except for the motor-driven turbocompressor, the system consists of commercially available components. The system is being designed for installation in a Minnesota Mining and Manufacturing Company magnetic tape production facility where it will operate to receive solvent-laden air from either of two ovens. It is a recirculating type system with a design flow of 8,000 cfm. At maximum capacity it will remove solvent at a rate exceeding 500 lb/hr, and the average removal rate will be 380 lb/hr.

All of the removed solvent will be collected in the condensers, where the lowest temperature is approximately -60°F. The water vapor that accompanies air leakage into the oven accumulates as ice in the condensers. This ice is removed by utilizing two condensers and thermally cycling them. The system utilizes a precooler (cooled by cooling tower water) for controlling the regenerative heat that is supplied in the return flow to the oven.

The motor-driven turbocompressor is being designed so that it can be applied to a wide range of systems (closed-oven loop, open-oven loop, and carbon desorb); this flexibility is achieved by allowing gear ratio and motor power modifications for each specific application. Design and development are scheduled to be completed in time for the first unit to be installed in the 3M facility in July 1982, and demonstration will follow.

Follow-on marketing is planned as the first system becomes operational and its performance demonstrated to potential users. This system offers good economical advantages over the carbon steam-desorb system. It also is very competitive with mechanical (vapor/Rankine cycle) refrigeration condensing systems, particularly in those instances where condensing temperatures are in the -50°F range. It is noted that little development has occurred in the mechanical refrigeration systems for subfreezing operations involving ice accumulation.

DEMONSTRATION OF A HEAT PUMP WATER HEATER*
ENERGY UTILIZATION SYSTEMS, INC.
ORNL SUB-7321
ROBERT P. BLEVINS
AUGUST 1978 THROUGH JUNE 1981

## OBJECTIVE

The objective of this project is to demonstrate the reliability and operating efficiency of a residential heat pump water heater by installing 85 units in climatically dispersed single-family homes and monitoring these installations for a period of one year.

## BACKGROUND

Energy Utilization Systems, Incorporated (EUS), in a project sponsored by the Buildings and Community Systems Division of the United States Department of Energy (DOE), through the Oak Ridge National Laboratory (ORNL), has developed a heat pump water heater (HPWH) for residential applications. The scope of work for this contract (UCC number 7321) was divided into two phases. Phase I covered the development of the heat pump water heater, including conceptual design and engineering, and was completed in August 1978.

Under Phase II of the contract, a pilot run facility for assembly and testing of 100 units was designed and constructed. After assembly and shipment, 85 of these units were installed in customers' homes by 20 electric utility companies as part of a field demonstration designed to test the performance and reliability of the device.

A heat pump water heater is a device that works like a room air conditioner except that it pumps heat from a room, basement or garage into an attached water tank.

The HPWH developed under Phase I of this program (Figure 1) uses a modified 82-gallon tank manufactured by Mor-Flo Industries, Johnson City, Tennessee. The tank is 52 inches high, 25 inches in diameter and has a four-inch access hole in the top to allow the insertion of the condenser. The heat pump assembly is mounted directly on top of the tank. This unit is the same diameter as the tank and is approximately 13 inches high. The cold water inlet is at the bottom of the tank and the hot water outlet is on the side of the tank near the top. This configuration eliminates any problem of interference with the heat pump unit during installation.

The heat pump assembly houses the compressor, evaporator, thermal expansion valve, filter drier, electrical controls and fan. Figure 2 shows a cutaway view of the system.

The compressor is a Copeland JRL4 model rated at 1 hp for R-22 refrigerant. The refrigerant used in the HPWH is R-12 because of the high discharge pressures associated with the system's discharge temperatures which can exceed 200°F. The use of R-12 derates the compressor by approximately one third. The immersion condenser has a dual-tube construction to provide double separation between refrigerant and potable water. The space between the tubes is filled with water (dyed red with food coloring) to improve heat transfer between the refrigerant and the water in the tank. In the event of a tube failure, a plastic cap is blown off the top of the condenser and the red dye gives a visible indication of the failure.

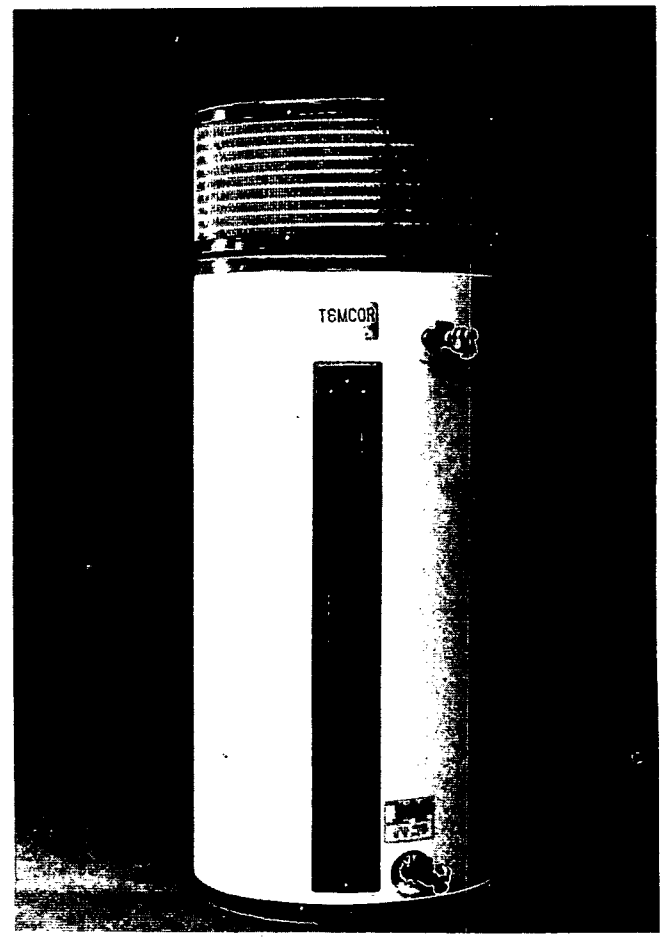



Figure 1. Heat Pump Assembled to 82-Gallon Tank

Phase II of the program involved the field demonstration of the HPWH in actual use conditions. Units were installed in 85 single-family homes located in the service territories of 20 climatically dispersed electric utilities. The units were automatically controlled to operate in the heat pump mode and in the resistance heating mode on alternate weeks. Each test side was instrumented to record kWh, water consumption, and inlet water, delivery water and ambient air temperatures. These operational data were forwarded to EUS by utility personnel each month. The data were then analyzed and the coefficient of performance (COP) was calculated for each unit on a monthly basis.

The field demonstration program was designed to test the reliability and performance of the HPWH and to determine if and how the system affects the heating and cooling loads of the home.

* Research sponsored by the U.S. Department of Energy under contract with the Union Carbide Corporation.

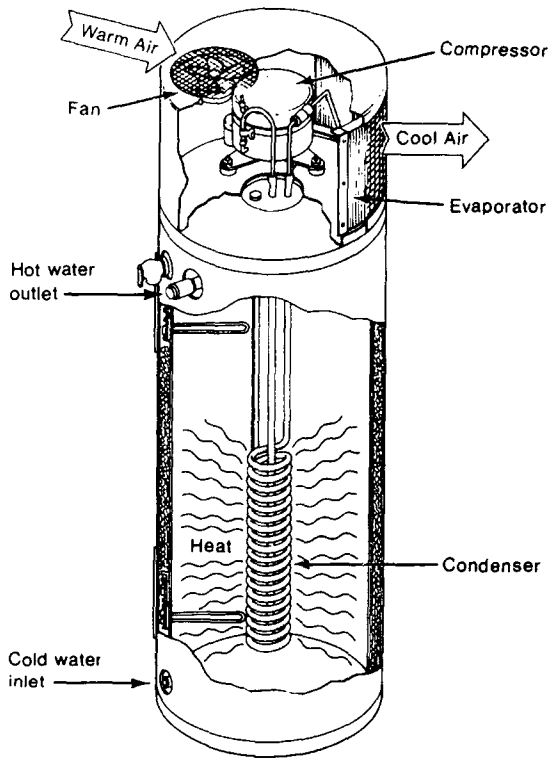



Figure 2. Cutaway View of Heat Pump Water Heater

SUMMARY

In the period between March 1979 and January 1980, 85 prototype heat pump water heaters were installed in single-family residences. Each system was monitored for a period of one year and total program monitoring was concluded at the end of December 1980.

The field demonstration provided a total of 643 unit-months of usable operational data which showed an average COP of 1.93, or an average 48% operating savings compared to resistance water heating. Average operating conditions were 73 gallons of 140°F water consumed each day with an average inlet water temperature of 61°F and an average ambient air temperature of 71°F. Despite a high initial failure rate for the prototypes, which resulted in a protracted debugging period, consumer reaction to the system was extremely positive. The data suggests that the HPWH would save the average consumer in the test program 2917 kWh per year. Measurable impacts on heating/cooling systems were detected in only 8% of the test homes.

Based on early results of the field demonstration program, EUS committed itself to the commercialization of the system and is now manufacturing and distributing these heat pump water heaters under the brand name TEMCOR.

TECHNICAL ACCOMPLISHMENTS

The research and development efforts covered under Phase I of this program led to the design and testing of several early prototype heat pump water heaters. The results of Phase I were then applied to Phase II, in which the design of the test prototype was finalized. Significant accomplishments occuring in and concurrent with Phase II include:

- A 3500-square foot pilot run manufacturing facility was designed and built at the EUS plant in Pittsburgh, Pennsylvania. The facility was equipped with specially designed tooling, including a hydraulically operated evaporator bending press and a motor-driven device for winding condenser coils, standard refrigeration system and metal-forming tooling, and a testing laboratory. Ninety-four heat pump water heaters were built and tested during the pilot run; 85 were sent to field test participants, 2 were put on life test in the EUS lab, 2 were sent to ORNL for testing, and 5 were retained as spares. Eighty-seven instrumentation packages were also designed, built, and tested at the pilot run facility.

- A laboratory test program was developed to prove the design and to assure that final design performance would meet expectations prior to running the field tests. Most of the tests were designed to determine the overall coefficients of performance of the unit as a function of ambient air, delivery water, and source water temperatures, based on twelve-hour, 64.3-gallon withdrawal tests. Results of the tests indicated that the design was capable of producing COPs greater than 2.0, and that the unit's performance is most sensitive to ambient air temperature (the performance improves as ambient temperature increases).

- Another major accomplishment under Phase II was obtaining Underwriters Laboratories (UL) approval and listing of the prototype production design. One of the original units produced in the pilot run facility was shipped to Underwriters Laboratories in Chicago, Illinois for pre-approval testing and evaluation. Several minor design modifications were required, but approval was granted on August 5, 1980.

- The most significant accomplishment within Phase II was the successful demonstration of the HPWH's operating performance under actual field use conditions. A total of 643 unit-months of operating data were obtained and evaluated in the field test. The average COP obtained in the program was 1.93, which is approximately 92% of targeted design performance. The validity of the test sample is enhanced by the fact that the average water heating requirements for the families in the test program were within 0.4% of average water heating requirement estimates made by the National Bureau of Standards [1].

  Measurable negative heating and cooling system impacts, which tend to slightly derate the overall performance of the units, occurred in only 8% of the test sites. Despite a high initial failure rate for the prototypes, which resulted in a protracted debugging period, consumer reaction to the system was extremely positive. The field data suggests that the HPWH would save the average consumer in the test program 2917 kWh per year and offer an annual performance factor of 1.90.

- A final major accomplishment of the field demonstration phase of the contract was that it led to the early commercialization of the heat pump water heater. The initial manufacturing, shipping, and

operating experience gained from the demonstration, in conjunction with the marketing studies begun in Phase I, allowed the company to make design improvements resulting in a marketable, energy-efficient product. A further discussion of the commercialization effort is presented in the Future Activities section of this paper.

## FUTURE ACTIVITIES

Contract Activities. The majority of Phase II work has been completed. Preparation of the final report is in progress and a draft will be sent to the ORNL-TM sometime in May 1981. The report summarizes the field demonstration program and presents operating and performance data.

One other significant task, originally scheduled to be performed during Phase II, has been postponed. This task involves the destructive examination of one of the life test units that has been installed and operating in the EUS testing facility for approximately two years, and is intended to provide information on compressor wear, tank life, and condenser fouling. The task will now be incorporated into the next phase of work.

Post-Contract Activities. One final follow-on phase is planned, pending ORNL approval. This phase consists of an expanded mechanical analysis of selected field test units to obtain an indication of expected system life and any reliability problems that may need to be resolved. The work as planned involves the selection of twenty of the field test units, based on water quality data for the installation sites. Because the chemical composition (particularly hardness and pH) of the water affects the amount of scaling or corrosion within the tank and on the condenser, several units will be selected from each of four water quality areas. The units will be obtained from the participants, through the participating utilities, in exchange for new factory-built TEMCOR's.

All of the units will be inspected, photographed, and disassembled at the EUS testing facility in Pittsburgh. Compressors will be returned to the manufacturer (Copeland) for examination and evaluation of remaining useful life. Tanks and condensers will be examined for corrosion and scaling, and amounts of such corrosion or scaling will be correlated with water quality. Results of the examinations should help indicate what, if any, further design modifications may be necessary to produce a reliable device.

During the course of the field demonstration, Energy Utilization Systems decided that the operating results and market projections were sufficiently promising to warrant commercialization of the TEMCOR heat pump water heater. A manufacturing plant site and Small Business Administration loan guarantees were obtained in Johnson City, Tennessee. A small production line, capable of producing up to 50 units per day (single shift), was built and production began in July 1980. In addition to the original 82-gallon unit, the factory produces 66- and 50-gallon units, which use basically the same heat pump assembly and smaller tanks. EUS has also developed, and will soon produce, a retrofit-type HPWH to use with existing installations.

The original marketing plan called for using electric utilities as the major distributors of the heat pump water heaters. However, many utilities do not actively market appliances and those that do are very hesitant to accept new products. As a result, EUS has established its own network of regional sales representatives across the country. These representatives, in turn, have recruited local distributors and dealers to sell directly to the public.

Through several independent EUS tests, it was determined that the 82-gallon TEMCOR is suitable for use in fast-food restaurants, typically as a water preheater installed in mechanical equipment areas. In this configuration, in addition to improving the overall water heating efficiency, the TEMCOR provides space cooling for the refrigeration condensing units, allowing them to operate more efficiently. The actual temperature drop is dependent on the installation configuration, but, because the heat pump generally runs continuously throughout the day due to the relatively high water consumption, some amount of cooling is always available during high condensing unit load periods.

EUS has also begun investigating the development of a larger (3-ton or greater) capacity heat pump water heater for use in full-service restaurant kitchens. In this type of application, the space cooling effects will be as important as the water heating. A proposal for development and testing of commercial prototypes has been submitted to the Electric Power Research Institute.

Other related applications of heat pump water heating in which the company either is involved or is considering are swimming pool and spa (hot tub) water heaters, and a solar water heater backup using the heat pump water heater rather than electric resistance during periods of insufficient solar gain. The solar application is particularly attractive in light of federal solar tax credits because it may be possible to reduce solar collector requirements as a result of the high-efficiency backup, and lower the overall net cost of the solar system.

## PUBLICATIONS

1. Research and Development of a Heat Pump Water Heater, Volume 1, Final Summary Report, ORNL/Sub-7321/1, Energy Utilization Systems, Inc. for ORNL, August 1978.

2. Research and Development of a Heat Pump Water Heater, Volume 2, R & D Task Reports, ORNL/Sub-7321/2, Energy Utilization Systems, Inc., for ORNL, August 1978.

3. Demonstration of a Heat Pump Water Heater, Volume 1, Design Report, ORNL/Sub-7321/3, Energy Utilization Systems, Inc., for ORNL, December 1979.

4. Demonstration of a Heat Pump Water Heater, Volume 2, Final Project Report, ORNL/Sub-7321/4 Energy Utilization Systems, Inc., for ORNL. Report is presently in draft form, scheduled for release June 1981.

## REFERENCES

[1] NBS water heating requirements estimates, Federal Register, vol. 42, no. 192, Tuesday, October 4, 1977.

SOLAR HYBRID ENERGY PROGRAM
LINCOLN LABORATORY
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
DE-AS01-77-CS-34577
E. KERN
PERIOD OF CONTRACT: Begin 9/1/77 End 12/31/80

## OBJECTIVE

The objective of the Solar Photovoltaic/Thermal (Hybrid) Energy Project was to design, install, and experimentally evaluate residential solar heating and cooling systems which utilize a heat pump and either photovoltaic/thermal (PV/T) or exclusively photovoltaic (PV) collectors. PV/T collectors are solar collectors in which PV cells and flat-plate thermal collectors share a common aperture. PV power was provided by a utility-interactive PV power system. This power system is one in which a PV or PV/T array and an inverter operate in parallel with, and supply power to, a utility line. The inverter converts DC power obtained from the PV array to AC power which can be supplied either to the residence or the utility. Based on life-cycle cost considerations, recommendations were made concerning the most favorable heat pump and PV/T or PV collector system configurations for northern and southern climates. Based on these recommendations, residential and laboratory experiments were carried out to accomplish the following:

1. Identify factors affecting system operation which were not included in the design phase analyses, and to re-evaluate the results of these analyses,

2. Evaluate the effects of modifying commercially available heat pumps,

3. Evaluate the compatibility of a two-speed air-source heat pump and utility-interactive PV power system, and

4. Establish the reliability of a prototype system.

## BACKGROUND

If PV cost reductions[1] can be achieved, a combined heat pump and utility-interactive PV power system will be a strong candidate for residential heating and cooling applications.[2-4] However, the performance and reliability of such a combination are as yet uncertain. Concern arises from the fact that heat pump compressor failure rates are higher than compressor manufacturers think they should be, and this failure rate seems to be rising. In 1979 residential heat pump compressor failure rates were approximately 10%; an acceptable rate would be about 5%.[5] This failure rate is unrelated to utility-interactive PV power systems; however, these power systems are only now undergoing the development necessary to achieve adequate performance and reliability. Heat pump manufacturers are particularly concerned about the characteristics of utility-interactive power systems because of these systems' possible effect on compressor operations.

In order to design a combined heat pump and residential PV power system with as high an efficiency as possible, it is necessary to determine whether a water- or air-source heat pump should be used. It is also important to determine if commercially available heat pumps can be easily modified to efficiently accommodate electrical and thermal energy provided by solar collectors.

## SUMMARY

During the design phase of the project a preferred residential heat pump and collector configuration was identified for both northern and southern climate locations. The preferred collectors [PV or PV/T] were identified for each location. Experimental data were used to quantify a temperature "defect" which may exist in previous simulations involving air-source heat pumps. Laboratory data were obtained which show that the steady-state operations of an air-source heat pump and a utility-interactive PV power system are compatible. Substantial data from a residential prototype system were obtained. These data establish the reliability and effectiveness of the combined heat pump and PV power system. It was demonstrated that the secondary heat pump functions of "boosting" and domestic hot water heating are difficult to perform effectively at low compressor speed.

## TECHNICAL ACCOMPLISHMENTS

Annual system performance simulations using the TRNSYS computer program were used to show that: (1) in northern climates both the electrical and thermal components of PV/T collectors make significant contributions to combined heat pump and collector systems, (2) in southern climates an exclusively PV array is more effective than a PV/T array, and (3) at moderate collector areas [less than 60 $m^2$] a "parallel" air-source heat pump and collector configuration is preferred for both locations. In a "parallel" system, the thermal energy obtained by the collectors is transferred directly to the residence. In a "series" system this thermal energy is collected and stored at a lower temperature, and a water-source heat pump is used to transfer energy from storage to the residence. To be effective in the heating mode, series systems require relatively large collector areas. Series systems lack the capability of transferring heat directly from the residence to the atmosphere when cooling operation is required.

A residential experiment at the University of Texas at Arlington established the annual effectiveness and reliability of a utility-interactive PV power system and a two-speed air-source heat pump. The Arlington location is characterized by a dominant cooling load with accompanying dehumidification requirements. A commercially available air-source heat pump was successfully modified to chill water when insolation was adequate or off-peak electrical rates applied. Stored chilled water was used to cool the residence in the late afternoon and early evening when insolation was not available and peak utility rates applied.

A laboratory experiment established that a heat pump's steady-state performance when powered by a utility-interactive PV power system is nearly indistinguishable from its performance when conventionally powered. This was shown to be true in the heating mode for a wide range of outdoor temperatures (-22 °C to +15 °C) and for a zero to 100% PV power contribution. The equivalence of the heat pump's thermal performance when driven by the new and conventional power sources was established by volumetric airflow and psychrometric data. The equivalence of the heat pump's electrical performances was established by power transducers and electrical waveform photographs.

Experimental data were obtained which revealed a weakness in many previous simulations involving air-source heat pumps. Not only is supplemental resistance heating required at ambient temperatures below the balance point, supplemental resistance heating is also required when the temperature of the residential return air falls below 32.2 °C (90 °F). Residential air-source heat pumps can develop a temperature "defect" in the return air-stream at outdoor temperatures as high as +5 °C (41 °F) unless resistance heating is used to "top-off" the return air temperature. A temperature defect was not included in the design analyses; however, including this defect would not have altered the conclusion that the parallel configuration is superior to the series configuration.

Experimental data showed that using stored thermal energy to "boost" the temperature of an indoor coil's supply air can severly degrade heat pump performance. It has been suggested in the literature[6] that "boosting" in conjunction with low-speed heat pump operation would be an economical procedure. Experimental data, together with the heat pump computer program available from Oak Ridge National Laboratory, were used to confirm that "boosting" can lead to inadequate supercooling of refrigerant leaving the condenser. Experimental data were obtained which show that using a heat pump desuperheater (to heat domestic hot water) also degrades heat pump operation at low compressor speeds. This is significant because in the southern climate cooling application it is important to heat domestic hot water using the heat pump. Otherwise it would be necessary to install thermal collectors or burn fossil fuels. Low-speed compressor operation is important in this application because it is more efficient and permits nearly continuous dehumidification. Higher speed compressor operation usually requires excessive heat pump cycling and humidity control is lost when the heat pump is off. Special controls need to be developed to better match hot water heating and space cooling [and dehumdification] requirements with the availability of insolation.

## POST-CONTRACT ACTIVITIES

A house whose heating and cooling system is generically similar to the prototype Arlington residence will soon be offered for sale.[7] This house, which contains a 7 kW (peak) utility-interactive PV power system, will be used to assess the occupants' acceptance of a combined heat pump and photovoltaic system and to familiarize utilities, builders, developers, town building officials and others with issues concerning photovoltaic installations. The heat pump is a Carrier dual-compressor model which was designed specifically for northern climate operation. The house is located on a two-acre lot in Carlisle, approximately twenty miles northwest of Boston.

A residential heat pump presents the utility with a load which has start-up current transients which may exceed the utility service rating; as a result, severe disturbances of the residence's utility voltage may occur during compressor start-up. In order to operate properly, a utility-interactive inverter relies on the presence of a relatively undistorted 60-Hz voltage waveform provided by the utility. Consequently, compressor start-up transients might affect the normal operation and long-term reliability of a utility-interactive inverter. Further work is required to establish that current transients associated with compressor start-up do not adversely affect inverter operation.

In order to most efficiently utilize available insolation, air-source heat pumps need to be developed which have special capacity modulation controls. Examples of capacity modulation are the two-speed heat pumps now being offered by the major manufacturers; one manufacturer offers a heat pump with a continuously variable-speed compressor. The difficulty with these units is that they are designed exclusively to fulfill space heating or cooling requirements. Possible secondary uses of the heat pump, such as boosting the temperature of the domestic hot water supply line, are not compatible with lower speed compressor operation. The more general requirement is for a heat pump which heats [or cools] water while heating [or cooling] residential air. When adequate insolation is available, the heat pump should be able to perform its secondary function of heating water [by desuperheating the refrigerant leaving the compressor] or chilling water [by circulating water around the evaporator]. Otherwise the capacity of the heating [or cooling] system will not be most favorably matched to the peak output power of the PV array.

In view of the fact that the peak cooling load lags the insolation peak by four to six hours, there is a need for the development of a cold storage system which will allow a heat pump to cool the stored material (e.g., water or ice) when adequate insolation is available. This chilled material can then be used to extract heat from the residence in the late afternoon and evening when the cooling load persists and insolation is not available.

## REFERENCES


1. Department of Energy PV cost goal for 1986 is $1.60 (1980 dollars) per peak watt, installed system price to the homeowner.

2. Russell, M. C., "Solar Photovoltaic/Thermal Residential Systems," MIT Lincoln Laboratory Report No. COOO-4577-9, December, 1979.*

3. Sheldon, D. B., and Russell, M. C., "A Review of Combined Photovoltaic/Thermal Collector-Solar Assisted Heat Pump System Options," Proceedings of the Fifth Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, April, 1980.

4. Kush, E. A., "Hybrid Photovoltaic/Thermal Systems with a Solar-Assisted Heat Pump,"

American Section International Solar Energy Society, 1980 Annual Meeting, Phoenix, Arizona, June 1980.

5. Moore, P.B, "A Historical Look at Heat Pumps," Proceedings of the Fourth Annual Heat Pump Conference, Oklahoma State University, Stillwater, Oklahoma, April 1979.

6. Bessler, W. F. and Hwang, B. C., "Solar-Assisted Heat Pumps for Residential Use," ASHRAE J., 59-63 Setpember, 1980.

7. Nichols, B. E. and Strong, S. J., "The Carlisle House: An All Solar Electric Lived-In Residence," Proc. 15th IEEE Photovoltaic Specialists Conf., Orlando, FL, May, 1981.

8. Hendrie, S. D., "Evaluation of Combined Photovoltaic Collectors," Proc. International Solar Energy Society, May, 1979.*

9. Hendrie, S. D. and Raghuraman, P., "A comparison of Theory and Experiment for Photovoltaic/Thermal Collector Performance," Proc. 14th IEEE Photovoltaic Specialists Conf., San Diego, CA, January, 1980.*

10. Hendrie, S. D. and Raghuraman, P., "An Advanced Combined Photovoltaic/Thermal Collector: Design and Testing," Proc. 1980 ISES Annual Meeting, American Section, Phoenix, AZ, June, 1980.*

11. Darkazalli,G., "Phase I Solar Photovoltaic/Thermal Residential Experiment Final Report," DOE/ET/20279-103, Lincoln Laboratory, M.I.T., Lexington, MA, July, 1980.*

12. Kugle, S, Leith, J., and Svane, M., "Phase II Solar Photovoltaic/Thermal Residential Experiment Final Report," COO-4577-10, Lincoln Laboratory, M.I.T., Lexington, MA, April, 1981.*

13. Green, S. T., "The Effect of Thermal Storage Temperature on the Performance of a Parallel Solar-Assisted Heat Pump Heating System," Master's Thesis, Dept. of Mech. Eng., University of Texas at Arlington, Arlington, Texas, December, 1980.*

14. Sheldon, D. B., "Heat Pumps and Utility-Interactive Photovoltaic Power Systems," Proc. 1981 ISES Annual Meeting, American Section, Philadelphia, PA, May, 1981.*

15. Sheldon, D. B., "Solar Photovoltaic/Thermal Final Report, Report No. 4577-11, Lincoln Laboratory, M.I.T., Lexington, MA, March, 1981.*

* Project Publication

# Heat Engine Systems

# ASSESSMENT OF I.C. ENGINES AS DRIVERS FOR HEAT ACTUATED HEAT PUMPS

ARTHUR D. LITTLE, INC.
Contract No. 62B-13836C
J. C. Burke, W. P. Teagan, P. G. Gott
PERIOD OF CONTRACT: Begin June 1980 End December 1980

## OBJECTIVE

The objective of this program is to assess the present suitability and future potential of gas-fired and oil-fired internal combustion (I.C.) engines for use in heat-actuated heat pumps for space conditioning applications. Results of this study will be used as a resource in future program planning and will include the definition of R&D tasks that could be undertaken to improve the suitability of I.C. engine heat pump applications. The study will focus on drives for residential and light commercial machines up to 100 kw output—or approximately 50 HP engine input.

## BACKGROUND

The Department of Energy is sponsoring a variety of programs directed toward development of fossil fuel-fired heat pumps. These systems, if successful, can result in substantial energy savings relative to conventional heating and cooling equipment. Currently, the work in the United States, while spaning a broad range of technology, does not include any programs involving the use of conventional I.C. engines as heat pump drivers, although I.C. engine heat pump systems could provide a near term approach with minimal development. Improvements in I.C. engine operating characteristics indicate that issues, such as maintenance and engine operating characteristics indicate that issues, such as maintenance and engine life, which have heretofore limited interest in I.C. engine heat pump drives can be satisfactorily resolved. There are already relatively advance programs in Europe and Japan many provide additional motives for U.S industry to establish a capability in this field.

## SUMMARY

A review of prior experience with I.C. engine applications, such as total energy systems, has both highlighted many operational problems and indicated that long life reliable operation could be achieved with proper maintenance and easily incorporated design changes. Based on a review of the various generic engine types, the most promising candidate appears to be a robustly constructed engine operating at reasonably high loads (high BMEP) whcih combines favorable maintenance characteristics with high efficiency. In general, both the engines and accessory equipment are currently commercially available.

I.C. engine heat pump systems incorporating heat recovery can achieve design point heating thermal COPs of the order of 1.6. Seasonal heating thermal COPs will be somewhat lower and dependent on climate, system design and load matching. Design point cooling COPs will be approximately 1.0 to 1.2. Weighing the energy savings achieved against the incremetal first cost and maintenance cost, the economic feasibility of I.C. engine heat pumps seems very promising for larger sizes, especially 50 hp and above. A seasonally steady hot water load, provides better economic results than seasonally variable space heat load. The economic performance of small systems ( 3 hp) is not as good, particularly when competing against advanced furnaces.

On balance, I.C. engine heat pump systems appear sufficiently promising to warrant further development, particularly in the larger sizes applicable to multi-family and light commercial applications. In addition to its intrinsic merits, this near term technology could provide a standard or bench mark by which more advanced system concepts could be evaluated. Developmental options recommended included: system demonstration, annual performance simulation, production cost projections and market analysis.

## TECHNICAL ACCOMPLISHMENTS

- A conceptual design consisting of a block diagram (Fig. 1) identifying major generic elements was prepared to serve as a basis for estimating performance first cost and maintenance costs. The major components are: an I.C. engine with starter and extended maintenance features; a vapor compression heat pump cycle with reversing valve and outdoor and indoor coils; a heat recovery loop extracting heat from water jacket, oil sump, and exhaust; an exhaust system with muffler and emission controls as needed; an acoustic enclosure providing noise attenuation and incorporating the outdoor air fan; inlet air and fuel systems; and a backup heating system—indicated as a fossil furnace, although resistance stream heaters might also be adequate.

- A review of prior experience with I.C. engines in applications similar to heat pumps was conducted to understand the prior success and/or problems and the approach for minimizing such problems in future applications. The review was accomplished primarily by contact with manufacturers, packagers, and users of engine driven equipment supplemented, particularly in the case of European activities, by various leterature sources. The major categories of experience included: heat pump and air conditioning applications; total energy systems; and other applications such as mobile refrigeration and light house operations. The results indicated that many of the maintenance and reliability problems encountered with early I.C. engine driven systems were amenable to correction; and, in fact, subsequent improvements in engine design and accessory systems have greatly minimized the sources of the difficulties experienced. There are also a number of applications where modern, robust I.C. engines are performing satisfactorily in long life service.

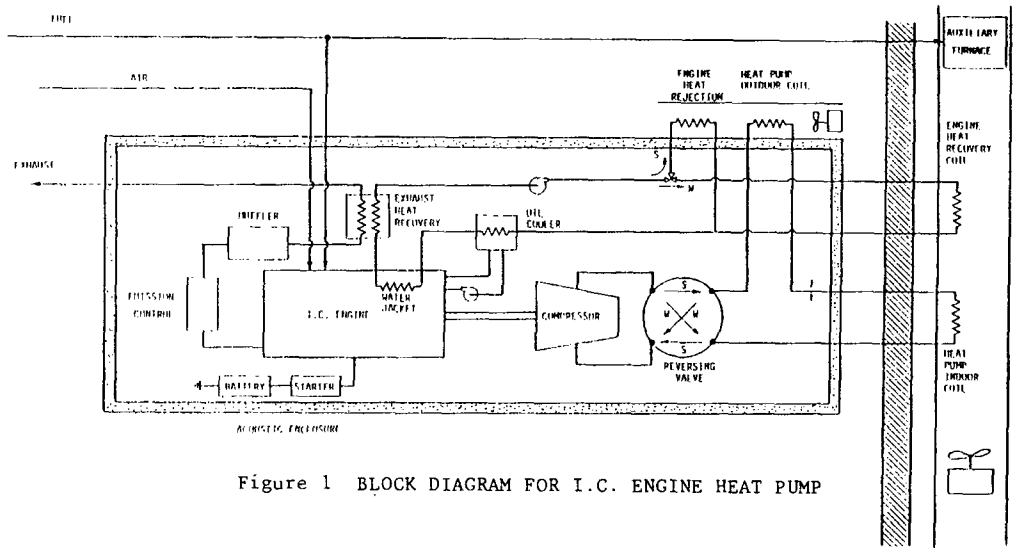



Figure 1 BLOCK DIAGRAM FOR I.C. ENGINE HEAT PUMP

- Manufacturers and users of I.C. engines were contacted to establish the cost, performance, maintenance and operational features (including data on noise and emissions) for various engine types. From our evaluations, it appeared that the more robust, relatively heavily loaded, stationary engines provide a better combination of low overall ownership costs and high efficiency than more lightly constructed lightly loaded types such as derated automotive engines. Accessory equipment, such as large oil reservoirs extra capacity oil and air filters are extremely important for extended maintenance intervals and reliable operation. In general, both the engines and accessory equipment are currently available.

- Capital costs and maintenance costs were estimated for spark and diesel engine driver subsystems based on highly loaded long life engines designed for maximum efficiency and durability with the necessary accessories and including provisions for heat recovery and sound isolation. Basic engine and accessory capital costs reflected current over the counter costs; annual maintenance costs were based on detailed maintenance schedules developed with manufacturers; and allowances for heat recovery and sound isolation were projected costs. Spark ignition engine driver subsystem capital costs were $3,100 and $5,100 for 3 HP and 50 HP sizes, respectively. Annual maintenance costs were $239 and $329, respectively. Diesel engine costs were slightly higher.

- Diesel and spark ignition engine efficiencies were determined for the range of speed and load which would be experienced for a heat pump driver coupled to a two speed air-to-air heat pump. The variation of efficiency with BMEP was determined for a speed of 1500 rpm, the midpoint in a 2 to 1 speed range over which the efficiency of the engines studied is relatively constant. Manufacturers full load data, and some diesel part load data, supplemented by estimates of natural gas part load efficiency were used for these estimates. (See Fig. 2.) At a BMEP of 100 psi, reasonable as a system design point, the efficiencies of the spark ignition and diesel engines were 27% and 32% respectively.

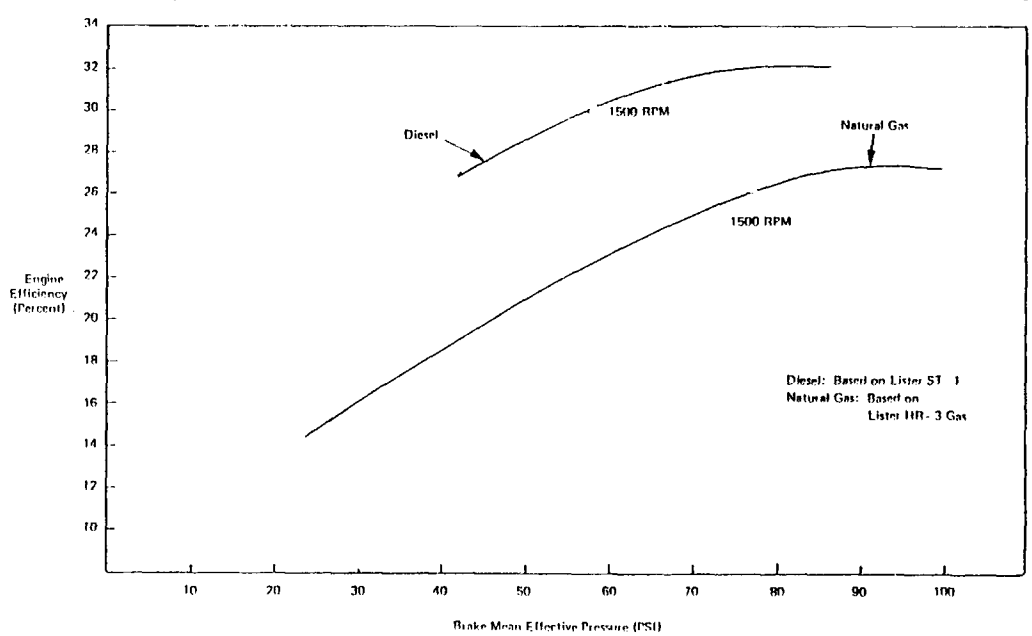



Figure 2 CHARACTERISTIC ENGINE EFFICIENCIES

- I.C. engine heat pump systems with heat recovery using current engines and vapor compression cycles can achieve design point thermal COPs of the order of 1.6 for natural gas engines (or about 1.7 for diesel engine). (See Fig. 3.) Seasonal thermal COPs will be dependent on climate, system design, and load matching, and, although lower than design point (particularly in cold climates), they will still produce substantial energy savings relative to gas furnaces.

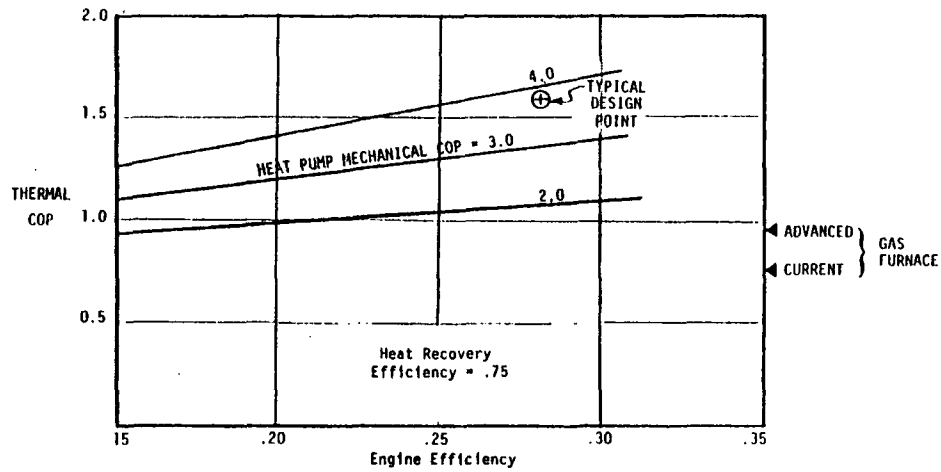



Figure 3 I.C. ENGINE-DRIVEN HEAT PUMP HEATING PERFORMANCE

- Design point cooling thermal COP will be about 1.0 for natural gas engines (or about 1.2 for diesel engines) which makes the I.C. engine driven heat pump roughly competitive with electric air conditioning. Since the energy savings appeared to be reasonably small and strongly dependent on regional energy rates, cooling was not quantitatively treated in the economic analyses.

- Simple economic payback, defined as the ratio of the incremental first cost of the I.C. engine heat pump system to the net annual savings resulting from both energy savings and maintenance, was computed for a range of variables including: locations; load types (space heat only, space heat and hot water and hot water only); engine size; efficiencies of conventional furnaces; and space heat balance point temperatures. The results, although not fully optimized in terms of either system design or cost indicated some very short paybacks (1 to 3 years) for large systems (50 HP) particularly in hot water usage. The paybacks of the small systems (3 HP) were longer and, in some instances, were infinite when incremental maintenance costs were in excess of the energy savings.

- Institutional and operational considerations which were identified but not quantitatively assessed included utility interaction, noise isolation requirements, emission controls, space requirements, and regulatory considerations.

- A development program was outlined which stresses near term field testing and market evaluation, since the I.C. engine heat pump system could be implemented in the near future using demonstrated technologies. This program, illustrated in Figure 4, has four elements:

-- Field Demonstrations of one or more I.C. Engine/heat pump systems in typical applications. The availability of all important subsystems would allow for installing a system within 16 months of program initiation.

-- Performance Simulations which quantify the technical and economic performance of various system options.

-- Manufacturing Cost studies to determine system economics as a function of production level and design alternatives.

-- Market Analyses to estimate the sales penetration into various market sectors.

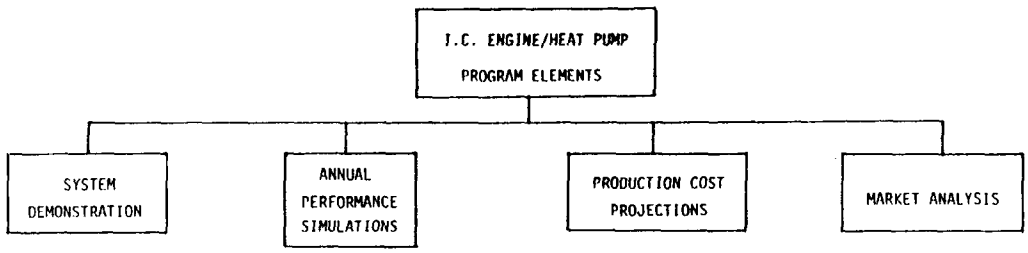



Figure 4 Development Options

This program would accelerate the development of a fuel fired heat pump system which shows good commercial promise, and thereby, help establish a strong position for U.S. industry in this emerging technology area. In addition, it would contribute to the larger range DOE heat pump technology development programs both by establishing system cost/technical goals which can set a standard by which more advanced engine drives heat pump systems can be evaluated, and by providing information relative to institutional, market, and environmental issues which will be common to all engine drive heat pump systems.

## FUTURE ACTIVITIES

### Contract Activities

The work on the present contract has been completed.

### Post-Contract Activities

Development options have been defined in the present contract.

## PUBLICATIONS/REPORTS/REFERENCES

"Assessment of I.C. Engines As Drivers for Heat Actuated Heat Pump", Final Report Prepared by Arthur D. Little, Inc. under subcontract 62B-13836C for Oak Ridge National Laboratory, May 1981.

TEST AND DEMONSTRATION OF A HERMETIC COMPRESSOR SEAL AND PHASE I, DEVELOPMENT OF A BRAUN LINEAR ENGINE-DRIVEN HEAT-ACTUATED HEAT PUMP

HONEYWELL INC., TECHNOLOGY STRATEGY CENTER

SEALS: 86X 61613C; PHASE I: 86X 61619C

A.T. BRAUN (TECTONICS RESEARCH, INC.)

PERIOD OF CONTRACT: BEGIN MARCH 26, 1980 END MARCH 31, 1982

## OBJECTIVE

The objectives of the two projects are to develop and demonstrate a viable hermetic seal; to advance the state of development, through analyses, simulation, design, fabrication and testing, of the key elements and components of the Braun linear engine/compressor so that it becomes a sound basis for a reliable light-commercial-sized heat pump system; to develop and evaluate integral heat recovery and defrost systems; to develop necessary controls; and to develop preliminary prototype heat pump system specifications.

## CONCEPT OR BACKGROUND

The proposed system employs an on-site Braun linear engine to drive the integral refrigerant compressor, with an outdoor unit (evaporator in heating mode), an indoor unit (condenser in heating mode) and a thermostatic expansion valve to complete the refrigeration circuit for Rankine-cycle heat pump operation. Heat from the outdoor air is absorbed by the low-temperature refrigerant, raised to a high temperature during the vapor compression phase by the engine work input to the compressor, and finally released for heating purposes. A schematic of the basic circuit is shown in Fig. 1. This figure also shows the utilization of engine exhaust and coolant heat to supplement the refrigeration cycle. Cooling is achieved by simple reversal of the refrigerant flow through the indoor and outdoor units by means of the four-way valve.

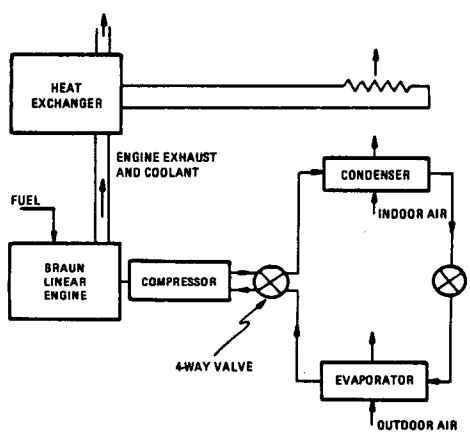



Figure 1. Heat Pump Circuit, Heat Mode

The heart of this advanced heat-actuated heat pump concept is the Braun linear engine--a two-stroke, loop-scavenged, free-piston engine with an integral head and no valves. When a compressor is driven by the engine, as in this heat pump application, the compressor provides the necessary energy to return the piston on the compression stroke. The balance mechanism between the engine and compressor results in smooth vibration-free operation. Fig. 2 shows the basic concept of the Braun linear engine.

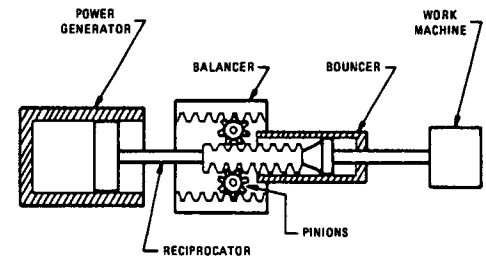



Figure 2. Braun Linear Engine Concept

The engine/compressor is characterized by simplicity, durability, reliability and high efficiency. Fig. 3 is a photograph of a sectioned Braun Linear engine/air compressor.

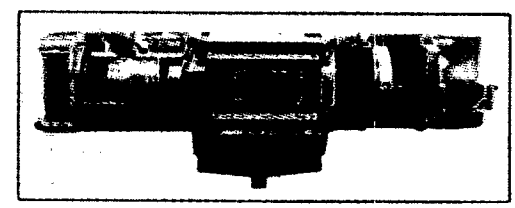


Figure 3. Sectioned BR-105 linear engine compressor. Left, power piston; right, compressor piston; center, balancer. Pinions inside of box.

The basic engine has two moving parts excluding the pinion gears in the balancer mechanism. There are no valves and the cyclinder and head are of one piece construction, thereby eliminating cylinder leakage and distortion caused by a head gasket and bolts.

The engine tends to operate at its natural speed. The total piston stroke varies with varying load conditions, since the end of the compression stroke is determined by the firing pressure and the end of the power stroke is determined by the compressor or bounce pressure. The piston completes its natural stroke and is reversed gently by the gas pressure. The pure linear motion and varying stroke reduce cylinder wear and ridge buildup. an engine with 15,000 hours of operation showed less than 0.001 inch cylinder wear, and no failures occurred within the basic mechanism of the engine. Braun linear engine air compressors have accumulated over 45,000 hours of laboratory and field running time. In 5 days of testing conducted on the Braun linear engine, 40,000 consecutive starts and stops were made without a miss.

The brake thermal efficiency of the engine operating in the spark ignition (SI) mode has been measured at 34 percent. This result was obtained after accounting for all the compressor losses up to the piston itself. It is anticipated that greater efficiency is achievable with additional improvements.

An early version linear engine/Freon compressor has been operated in both the heating and cooling mode with a commercial 20-ton split heat pump system. Exceptionally high COP, even at low outdoor temperatures, is expected through the utilization of exhaust, engine and compressor heat combined with the high engine efficiency.

Previous work (Ref. 1) has evaluated the performance of residential gas-fired heat pumps by simulation and compared them with conventional heating systems. Comparisons included both the characteristics of a conventional Otto cycle engine and a Braun linear engine. Comparisons were made on the basis of seasonal efficiency using binned U.S. average temperatures and various seasonal efficiency concepts. Table 1 defines the efficiency concepts and presents simulation results for various conventional and advanced heating systems. Table 2 presents a matrix of results for the Otto cycle engine (Type A) and the Braun linear engine (Type B). The most favorable comparison potential is made by reference to Table 2 using:

- Braun linear engine (Type B) with speed Modulation.
- High isentropic compressor efficiency.

Table 1. Comparison of Various Seasonal Efficiency Concepts[a] for Several Space Heating Systems Compiled with the HFLAME and HFROST Simulation Programs

| SYSTEM | CONVENTIONAL SYSTEMS | | | | | ADVANCED SYSTEMS | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | E/F | $\eta_{SP}$ | $\eta_{SE}$ | $\eta_{SF}$ | $\eta_{SC}$ | E/F | $\eta_{SP}$ | $\eta_{SE}$ | $\eta_{SF}$ | $\eta_{SC}$ |
| GAS-FIRED FWA FURNACE | 0.015 | 0.572 | 0.606 | 0.600 | 0.570 | 0.027 | 0.768 | 0.841 | 0.837 | 0.757 |
| GAS-FIRED FWA FURNACE RETROFITTED WITH IID AND SD[b] | 0.017 | 0.715 | 0.743 | 0.756 | 0.709 | – | – | – | – | – |
| OIL-FIRED FWA FURNACE | 0.028 | 0.693 | 0.736 | 0.767 | 0.681 | 0.031 | 0.770 | 0.824 | 0.858 | 0.731 |
| PULSE COMBINATION BOILER/FURNACE[c] | 0.013 | 0.863 | 0.888 | 0.900 | 0.842 | 0.042 | 0.854 | 0.933 | 0.972 | 0.797 |
| ELECTRIC HEAT PUMPS | 0.0 | 0.60 | 2.00 | – | 0.379 | 0.0 | 0.90 | 3.00 | – | 0.569 |
| RESISTANCE HEAT | 0.0 | 0.30 | 1.00 | – | 0.190 | – | – | – | – | – |

[a] Weather: Average U.S. weather.
[b] IID = Intermittent Ignition Device; SD = Stack Damper.
[c] Pulse combustion furnace is in development stage.

LEGEND: E/F = Ratio of seasonal electric to fuel utilization.
$\eta_{SP}$ = Seasonal prime energy efficiency (assuming 30% efficiency of generating and transmitting electric energy).
$\eta_{SE}$ = Seasonal energy efficiency at the site.
$\eta_{SF}$ = Seasonal fuel utilization efficiency.
$\eta_{SC}$ = Seasonal cost efficiency based on $R_c$ = 5.272.

Source: Reference 1.

Table 2. Comparison of Various Seasonal Efficiency Concepts[a] for Gas-Fired Heat Pumps of Different Configurations with Typical 3-Ton Refrigerant Circuit

| ENGINE TYPE[b]: | | TYPE A | | | | | | | | | | TYPE B | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| OPERATION MODE[c]: | | CONSTANT SPEED | | | | | SPEED MODULATION | | | | | CONSTANT SPEED | | | | | SPEED MODULATION | | | | |
| SYSTEM | ELECTRIC FAN + BLOWER (kW) | E/F | $\eta_{SP}$ | $\eta_{SE}$ | $\eta_{SF}$ | $\eta_{SC}$ | E/F | $\eta_{SP}$ | $\eta_{SE}$ | $\eta_{SF}$ | $\eta_{SC}$ | E/F | $\eta_{SP}$ | $\eta_{SE}$ | $\eta_{SF}$ | $\eta_{SC}$ | E/F | $\eta_{SP}$ | $\eta_{SE}$ | $\eta_{SF}$ | $\eta_{SC}$ |
| 7-hp ENGINE AND TYPICAL 3-TON HIGH-EFFICIENCY COMPRESSOR OF $\eta_{isentro}$ = 50.3% AT 47°F AMBIENT AND 3000 rpm | 1.2 | 0.087 | 0.76 | 0.90 | 0.93 | 0.67 | 0.141 | 0.69 | 0.88 | 0.92 | 0.58 | 0.173 | 0.90 | 1.21 | 1.31 | 0.74 | 0.560 | 0.80 | 1.47 | 1.93 | 0.58 |
| | 0.9 | 0.068 | 0.79 | 0.91 | 0.93 | 0.71 | 0.108 | 0.73 | 0.89 | 0.93 | 0.63 | 0.133 | 0.97 | 1.23 | 1.32 | 0.82 | 0.417 | 0.92 | 1.55 | 1.93 | 0.69 |
| | 0.6 | 0.049 | 0.82 | 0.91 | 0.93 | 0.76 | 0.075 | 0.79 | 0.90 | 0.93 | 0.70 | 0.094 | 1.05 | 1.25 | 1.32 | 0.92 | 0.276 | 1.10 | 1.66 | 1.94 | 0.86 |
| | 0.3 | 0.031 | 0.86 | 0.92 | 0.94 | 0.82 | 0.042 | 0.84 | 0.92 | 0.93 | 0.78 | 0.054 | 1.14 | 1.28 | 1.32 | 1.05 | 0.140 | 1.38 | 1.78 | 1.94 | 1.16 |
| | 0.0 | 0.012 | 0.90 | 0.93 | 0.94 | 0.88 | 0.009 | 0.91 | 0.93 | 0.94 | 0.89 | 0.016 | 1.26 | 1.31 | 1.33 | 1.23 | 0.010 | 1.68 | 1.92 | 1.94 | 1.65 |
| 5-hp ENGINE AND TYPICAL 3-TON HIGH-EFFICIENCY COMPRESSOR OF $\eta_{isentro}$ = 50.6% AT 47°F AMBIENT AND 3000 rpm | 1.2 | 0.125 | 0.85 | 1.07 | 1.13 | 0.72 | 0.185 | 0.73 | 1.00 | 1.06 | 0.50 | 0.261 | 1.03 | 1.53 | 1.77 | 0.81 | 0.727 | 0.85 | 1.68 | 2.43 | 0.60 |
| | 0.9 | 0.097 | 0.90 | 1.08 | 1.13 | 0.78 | 0.141 | 0.78 | 1.01 | 1.07 | 0.56 | 0.200 | 1.14 | 1.58 | 1.77 | 0.92 | 0.539 | 1.00 | 1.81 | 2.44 | 0.73 |
| | 0.6 | 0.069 | 0.95 | 1.10 | 1.14 | 0.86 | 0.097 | 0.85 | 1.03 | 1.07 | 0.75 | 0.139 | 1.27 | 1.63 | 1.78 | 1.07 | 0.358 | 1.22 | 1.97 | 2.45 | 0.93 |
| | 0.3 | 0.042 | 1.02 | 1.11 | 1.14 | 0.95 | 0.054 | 0.94 | 1.05 | 1.08 | 0.86 | 0.078 | 1.44 | 1.69 | 1.78 | 1.29 | 0.181 | 1.60 | 2.17 | 2.45 | 1.31 |
| | 0.0 | 0.014 | 1.09 | 1.13 | 1.14 | 1.07 | 0.010 | 1.05 | 1.07 | 1.08 | 1.03 | 0.018 | 1.68 | 1.76 | 1.78 | 1.62 | 0.012 | 2.37 | 2.43 | 2.46 | 2.31 |

[a] Average U.S. weather.
[b] Engine types A and B have characteristics similar to those of Briggs and Stratton engines and Braun linear engines, respectively; heat recovery efficiency = 70%; crankcase heater power = 50 W.
[c] Constant speed operation at 3000 rpm; speed modulation in the range of 1000 to 4000 rpm to minimize cycling losses.

LEGEND: E/F = Ratio of seasonal electric to fuel utilization.
$\eta_{SP}$ = Seasonal prime energy efficiency (assuming 30% efficiency of generating and transmitting electric energy).
$\eta_{SE}$ = Seasonal energy efficiency at the site.
$\eta_{SF}$ = Seasonal fuel utilization efficiency.
$\eta_{SC}$ = Seasonal cost efficiency ($R_c$ = 5.272).

Source: Reference 1.

## SUMMARY

The major part of the seal program has been completed. Proprietary hermetic seals of infinite life design have been investigated and demonstrated for practical stroke lengths and cyclic speeds.

During the recently initiated Phase I program, parallel development and interfacing of the various components are being carried forward, such as hermetic seals; refrigerant compressor (with valves, seals, etc.); BR-105 engine (fuel injection, ignition, starting, speed control, stroke control, etc.); controls (engine and compressor interface, system response to environment, etc.).

## TECHNICAL ACCOMPLISHMENTS

- The refrigerant compressor components have been re-analyzed and sized for the rather specific requirements of the Braun linear engine. This development work will continue to optimize the overall compressor concept to obtain the best valve, ring and seal performance. This effort integrates the results of the seal program into the Phase I program.

- Preliminary sound suppression work has been completed. Exhaust muffling and acoustic enclosure panels have reduced noise to levels less than 1978 EPA standards for portable air compressors.

- Simulation of heat pump performance, including heat recovery concepts, has been initiated to support engine/compressor component development and heat recovery concepts.

## FUTURE ACTIVITIES

Contract activities. The major tasks to be completed in this program relate to proof-of-concept elements of the Braun linear engine/Freon compressor.

Research and development will continue on the refrigerant compressor. Modeling and simulation analysis will be done to obtain optimum performance under the various heat pump system requirements in heating and cooling modes. The engine will be adapted to and integrated with the optimized compressor.

The results of the seal program will be integrated and tested with the most compatible engine/compressor configuration and demonstrated under operational conditions.

Engine components and systems such as fuel injection, lubrication, ignition, starting speed control, noise, emission control, etc., will be further developed and tested.

Present engine/compressor controls will be improved, and additional controls will be developed to integrate the engine/compressor with the heat pump system.

Heat exchangers will be adapted or developed for utilization of exhaust and engine heat. Prototype heat pump packaging configurations will be defined.

At the conclusion of Phase I, all key proof-of-concept components of the Braun linear engine heat pump system will have been developed through the testing stage. Preliminary specifications of a prototype heat pump system will be prepared.

Post-contract activities. Phase II consists of extending the work in Phase I through a series of tasks leading toward product commercialization:

- Operating breadboard system with performance evaluation,

- Market analyses,

- Production prototype development.

The present program activities, as well as those projected for Phase II, have had a significant effect in leading to a recently negotiated license agreement with a major manufacturer of commercial-sized heat pumps. This agreement, in turn, will have a major impact in furthering the commercialization of the Braun engine-driven heat pump.

## PUBLICATIONS/REPORTS/REFERENCES


1. A. Patani and U. Bonne, "Modeling the Performance of Gas-Fired Heat Pump Systems," 14th IECE Conference, Boston, Massachusetts, August 5-10, 1979.

2. A.T. Braun and Paul H. Schweitzer, "The Braun Linear Engine." International Automotive Engineering Congress, Paper No. 730185, Detroit, Michigan, January 8-12, 1973.

3. Brian J. Hogan, "Lightweight Free Piston Linear Engine Drives Compressor," Design News.

DEVELOPMENT AND DEMONSTRATION OF A STIRLING/RANKINE HEAT ACTIVATED HEAT PUMP
GENERAL ELECTRIC COMPANY
CONTRACT DE-AC02-77CS20043
PROGRAM MANAGER: RICHARD C. MEIER
PERIOD OF CONTRACT: DECEMBER 1976 TO JANUARY 1982

## OBJECTIVE

The objectives of the current contract for Phase II of the Stirling/Rankine Heat Activated Heat Pump project are to:

- Demonstrate that the performance predictions established in the Phase I program can be achieved.
- Assess whether the product cost goals are attainable.

## BACKGROUND

The Stirling/Rankine Heat Activated Heat Pump (HAHP) concept is schematically illustrated in Figure 1. The heart of the system is the combustor/Stirling engine/linear inertia compressor assembly which replaces the conventional motor/compressor unit employed in electric heat pumps. Heat input provided by a gas combustor provides thermal energy to the Stirling engine working fluid, helium. The engine displacer and the engine power piston operate with a prescribed phase and displacement relationship to create the highly efficient Stirling thermodynamic cycle.

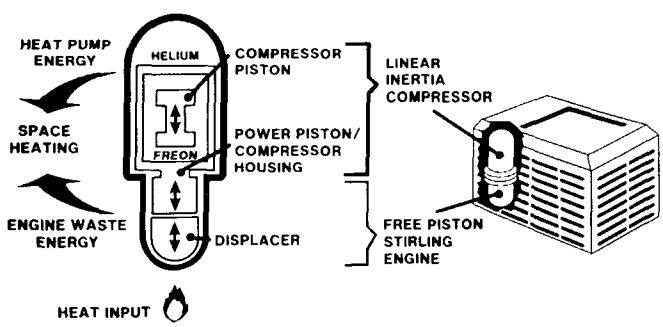



Figure 1. Stirling/Rankine Heat Activated Heat Pump Concept

Power delivered by the power piston is transferred to the heat pump refrigerant through a high efficiency linear inertia compressor. The power piston is directly connected to the compressor housing with the refrigerant being compressed by a free piston reciprocating within the housing - also with a prescribed phase and displacement relationship. The energy provided by the combustor/Stirling engine/linear inertia compressor is thus controlled by the rate of heat input, the phase and displacement relationships between the displacer, the power piston, and the compressor piston, and by the external conditions imposed on the refrigerant coil by the load (building) and outdoor ambient air temperature.

The Stirling engine has a high operating efficiency, since its thermodynamic cycle approaches that of a Carnot cycle. Because of this, the overall heat pump COP potential is high. Performance benefits are also obtained by using the waste thermal energy from the engine. In the Stirling/Rankine HAHP, exhaust gas waste heat is utilized to preheat incoming fresh air for the combustor. In addition, the engine coolant (water/glycol) is circulated through a hydronic coil in the indoor air handler unit to provide additional heating. Figure 2 illustrates that over 40 percent of the heating load is satisfied by the engine waste heat on a cold (0°F) day.

The use of waste energy from the engine, in conjunction with the high efficiency engine, makes the Stirling/Rankine HAHP a competitive product for cooling applications and an exceptional performer for heating applications.

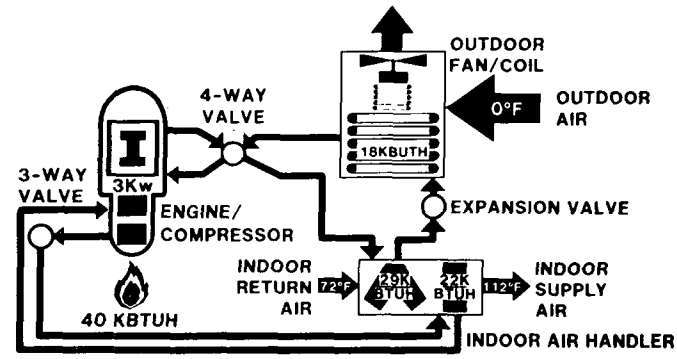



Figure 2. Heat Activated Heat Pump-Heating Operation

The design concept for the gas fired Stirling/Rankine HAHP is shown in Figure 3. The product is intended to be similar in packaging, styling, and installation to a conventional electric heat pump. As shown in the figure, the engine/compressor assembly, complete with natural gas combustor, will be located in one section of the outdoor unit cabinet, either surrounded by, or next to, the refrigerant and hydronic coils and outdoor fan unit. The outdoor cabinet will include the heat pump controls and the pump necessary to circulate the engine coolant from the engine to the indoor air handler (for winter heating) or to the outdoor hydronic coil (in the summer). The indoor unit is a conventional air handler with a standard hydronic coil added for the engine cooling function.

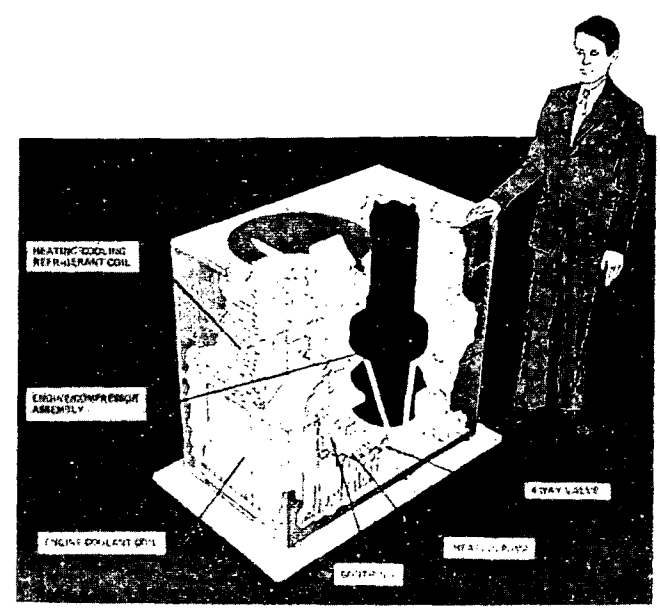


Figure 3. HAHP Outdoor Unit Design Concept

Recognizing the high potential of the heat activated heat pump as a product, the American Gas Association and the General Electric Company initiated a program to evaluate heat activated heat pumps in 1975. The result of this Phase I project was the selection of the Stirling/Rankine concept for further development. The current Phase II contract was then initiated in early 1976 by the AGA and General Electric. The Department of Energy joined with the AGA and GE in late 1976 to accelerate the development of this energy conserving product. In 1977, the Gas Research Institute assumed responsibility for the HAHP program for the American Gas Association. Today, the program is a joint GRI/DOE/GE development project, targeted for the development of a three-ton residential Stirling/Rankine HAHP. The residential HAHP product is projected to offer the consumer a three to five year payback on the premium initial price required by the product. With this level of rapid payback, it is projected that the Stirling/Rankine HAHP can attain a market growth history comparable to that of central air conditioning.

## SUMMARY OF ACCOMPLISHMENTS

The current Phase II program includes the development of an engineering prototype HAHP, followed by the development and field testing of a prototype residential unit. Development and testing of the engineering prototype was completed in early 1979. All of the components and subsystems met their performance goals except for the Stirling engine. The engine achieved an efficiency of 26%, versus a goal of 30%. Empirical data from the engineering prototype was then used to design the residential unit prototype, using a third order simulation model developed for the engine/compressor. As of the writing of this paper, development and testing of the system components and the combustor/engine/compressor subsystem has been completed, preparations for laboratory tests of the complete residential unit prototype system are underway, and installation preparations at the field test site are nearly completed. The prototype residential unit components met their goals, except again for the Stirling engine. The engine achieved indicated power and efficiency values of 2.8Kw and 27%, versus goals of 3.2Kw and 33%, respectively.

General Electric has conducted market and business assessment studies in parallel with the hardware development activities. Results of the studies indicate that the projected 3 to 5 year payback on the initial premium price of the HAHP can result in a market growth history comparable to that of central air conditioning. Industry shipments are projected at 100,000 units per year, 10 years after introduction.

## TECHNICAL ACCOMPLISHMENTS

- Developed and tested an engineering prototype of a complete Stirling/Rankine HAHP system.

The combustor/engine/compressor unit of the engineering prototype, designated Prototype 1, is illustrated in Figure 4. In the combustor, shown in Figure 5, a porous transpiration burner element, operating on natural gas, provides the heat input for the helium working fluid. The compact combustor operates with 15% excess air, yielding a CO emission level of less than 0.04% at the combustor exhaust. Testing verified the 82 to 83% efficiency goal for the combustor, operating with an 800°F air preheat level. The combustor is composed primarily of sheet metal components, lending itself to a low cost mass producible unit. Figure 6 illustrates the linear inertia compressor configuration used in the first prototype, and also depicts the hardware tested to demonstrate and characterize its performance. Testing yielded a 76% efficiency (isentropic) for this compressor configuration, exceeding its initial design goal. Figure 7 shows the first prototype unit engine, including the engine housing, regenerators, and heater head subassembly, and the dynamic components (displacer and power piston). The engine operated with a maximum helium temperature of approximately 1250°F and had a 3 kilowatt power output. Because of high thermal losses and a low operating frequency (26Hz), engine testing yielded only 20-26% operating efficiency, as opposed to the 30% efficiency goal.

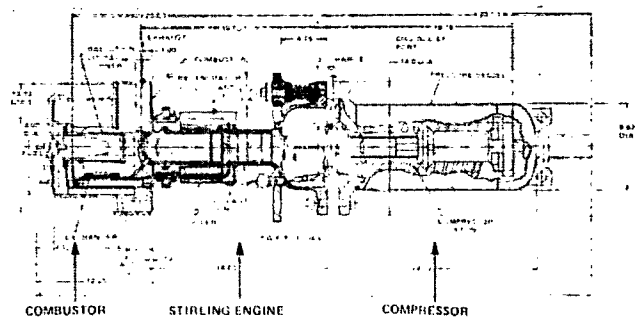



Figure 4. HAHP Combustor/Engine/Compressor

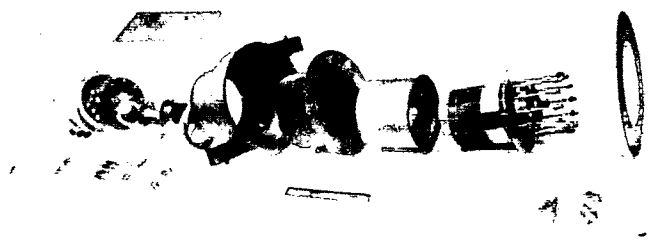


Figure 5. Transpiration Cooled Combustor

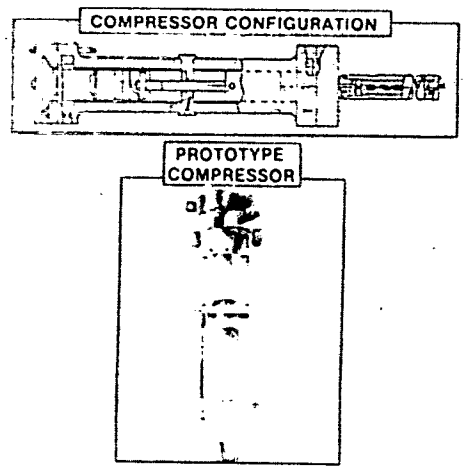



Figure 6. Linear Inertia Compressor

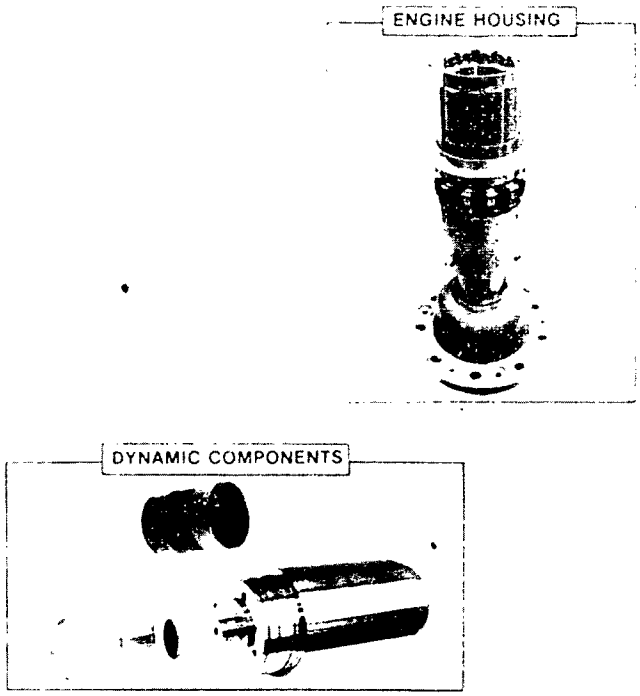



Figure 7. Free-Piston Stirling Engine

After the combustor/engine/compressor assembly testing was completed, the unit was combined with the previously tested outdoor fan/coil unit to form the complete outdoor unit shown in Figure 8. The outdoor unit was then tested in an outdoor environmental simulation chamber, with its matching indoor unit operating in conjunction in an indoor simulation loop.

Figure 8. Engineering Prototype Outdoor Unit

- Developed and tested the components and subsystems of a prototype residential unit designed for field testing.

Based on the experience gained from the engineering prototype, an improved prototype, designated Prototype 2, was designed and tested through the same process. The engine/compressor design was modified to reduce thermal losses and to simulate a production configuration to the extent practical. In addition, a gas spring was added to the engine power piston/compressor housing assembly to increase engine operating frequency. Testing of the combustor, compressor, indoor and outdoor fan/coil units, and the control system verified that component performance goals (which were similar to those of the first prototype) were all met. However, extensive testing of the combustor/engine/compressor unit, shown in Figure 9, yielded 2.2 to 2.8Kw engine indicated power, versus a 3.2Kw goal. Engine indicated efficiencies achieved in test were 22 to 27%, versus a 33% goal. And although the gas spring increased the operating frequency to the 30 Hz goal, the design must be revised to reduce venting and leakage losses.

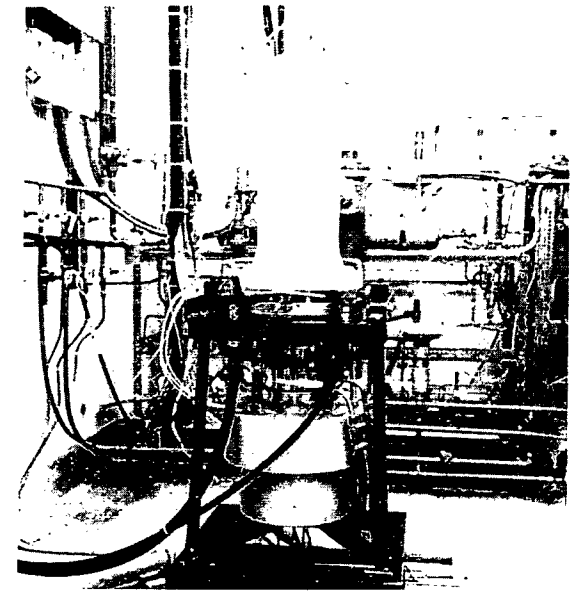


Figure 9. Combustor/Engine/Compressor In Test

- Developed a dynamic/thermodynamic simulation model of the engine/compressor.

Both the engineering prototype and the prototype residential unit engine/compressor were designed using a detailed computer simulation code that accounts for both the dynamic and thermodynamic behavior of the free piston Stirling engine and linear free piston compressor. The code, developed early in the Phase II program, has been periodically improved by expanding the level of component model details and through the use of test data to empirically refine design parameters. Results of early (1980) tests on the Prototype 2 engine (prototype residential unit) indicated the need to improve the correlation accuracy of the model. Revisions improving leakage and pressure loss models subsequently improved dynamic parameter predictions. Figure 10 shows the original simulation/test data correlation for the engine dynamic parameters (open symbols), and the improved correlation of the revised model (solid symbols). Additional improvements are required to achieve comparable correlation accuracy for the engine thermal parameters (heat input, power output, etc.).

- Selected and prepared a test site for field testing of the residential unit prototype.

Several test sites for field testing the prototype residential unit were offered by the Brooklyn Union Gas Company, Public Service Electric and Gas Company of New Jersey, Philadelphia Electric Company, Philadelphia Gas Works, and the Washington Gas Light Company, Washington, D.C. A test site proposed by Philadelphia Electric Co. (PECO) was selected and modified by PECO to accept the test unit. A sketch of the installation is shown in Figure 11. The indoor unit, a standard GE air handler, was installed

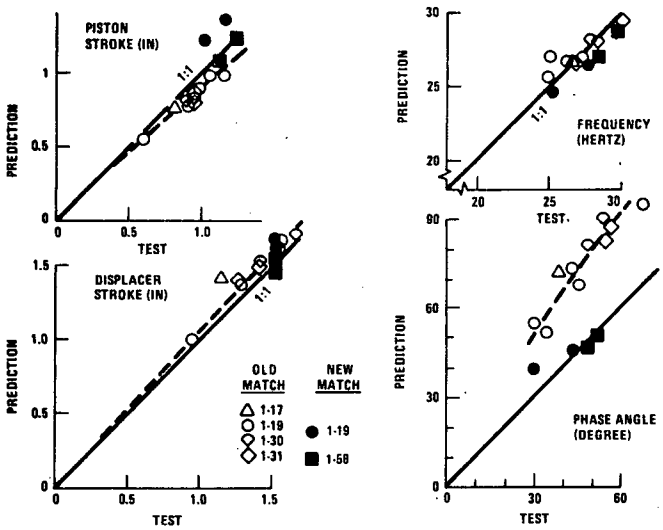



Figure 10. Engine Test Data Correlation

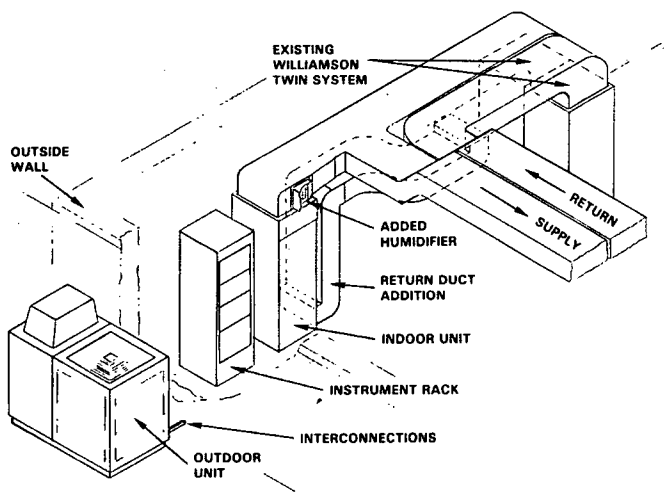



Figure 11. Field Test Installation Arrangement

with the modified ducting in parallel with the existing heating/cooling unit to permit side-by-side performance evaluations. The instrumentation rack houses all of the site related instrumentation (ambient temperature, interior temperature, on-off time periods, etc.) as well as indoor and outdoor unit measurements (temperatures, pressures, flow rates, etc.). Electricity and gas consumption are provided by a complete PECO submetering system.

A styled cabinet for the outdoor unit was designed and supplied by General Electric (Figure 12). Using advanced plastic cabinet materials and fabrication processes, the outdoor cabinet houses the combustor/engine/compressor unit with the fan/coil unit and system controls in an attractive, easy to install and service package.

- Conducted extensive market and business assessment analyses on the Stirling/Rankine HAHP product potential.

Throughout the Stirling/Rankine HAHP development program, the General Electric Company has been conducting market and business assessment studies in parallel with the hardware development contract activity. The studies have examined the potential market for all heat activated heat pump concepts (including absorption, Brayton/Rankine, Rankine/Rankine, etc.), the projected penetration of the Stirling/Rankine HAHP into the potential market, the economics of premium pricing the HAHP, the level of investment required by the manufacturer, and the expected return on investment, internal rate of return, cash flow, and other financial measures.

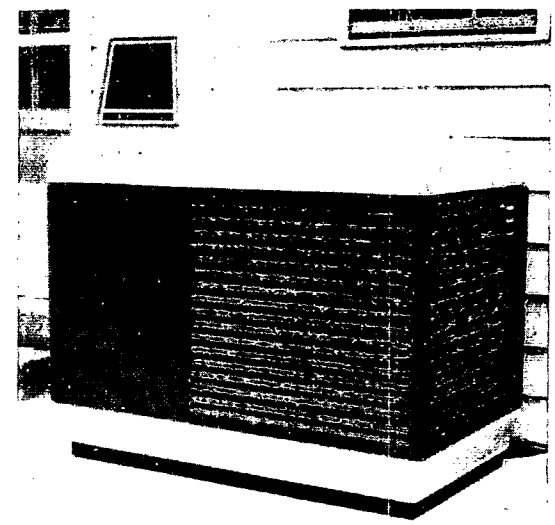


Figure 12. Prototype Residential Unit Outdoor Package

Performance projections for both the HAHP and a comparable gas furnace/electric air conditioner system (Nearest Equivalent Product or NEP) are shown in Table 1. These projections are for the products in the latter half of this decade, after gas heat pumps are an established product line in the HVAC industry.

Table 1. Performance of HAHP and NEP

| PERFORMANCE | NEP | GHP |
|---|---|---|
| HEATING COP @ 47°F | 0.95 | 2.31 |
| COOLING COP @ 95°F | 0.88 | 1.13 |
| PARASITICS, WATTS | 270/500 | 660 |
| HEATING SPF, W/O PARASITICS | 0.85 | 1.59 |
| COOLING SPF, W/O PARASITICS | 1.15 | 1.21 |
| HEATING SPF, WITH PARASITICS | 0.78 | 1.18 |
| COOLING SPF, WITH PARASITICS | 0.83 | 0.90 |
| ANNUAL SPF, WITH PARASITICS | 0.80 | 1.12 |

## 30% ELECTRIC POWER CONVERSION EFFICIENCY

Based on these performance projections and energy price forecasts, the estimated annual operating cost savings shown in Figure 13 would be realized. These savings would yield a 3-5 year payback to the consumer on the premium initial price required by the product. With this level of payback, Figure 14 shows an annual sales volume of 100,000 units/year predicted for heat activated heat pumps within 10 years after their introduction. If an aggressive campaign to introduce the product much more rapidly is undertaken, sales approaching 250,000 units/year may be realized.

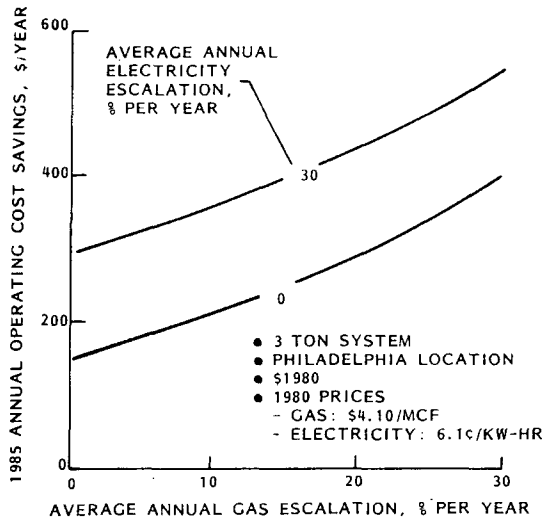



Figure 13. 1985 Savings of HAHP Over NEP

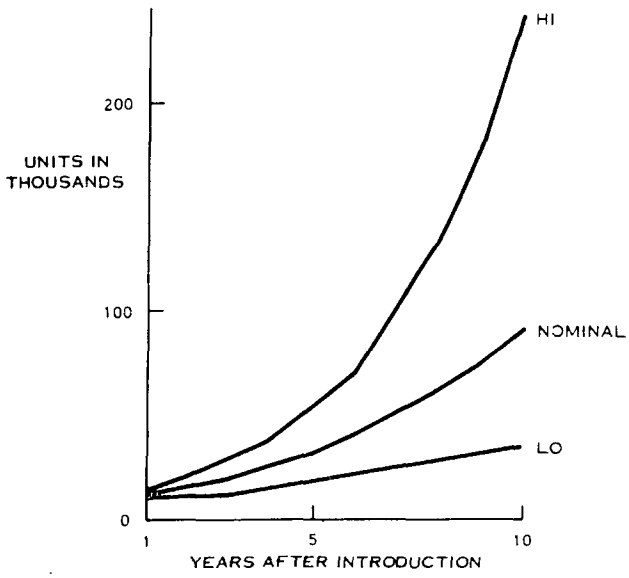



Figure 14. Sales Forecast for Gas Fired Heat Pumps

Recent changes in the relative escalation rates for gas and electricity have led to a reexamination of the performance goals established for the HAHP and its economic competitiveness. The results are shown in Figure 15, where the HAHP savings versus an electric heat pump are given as a function of the escalation rates for natural gas and electricity. These results have confirmed the importance of achieving the HAHP performance goals as natural gas escalates more rapidly than electricity.

## FUTURE ACTIVITIES

Contract activities. As the Phase II contract draws to a close, effort is being focused on completion of the development of the free-piston Stirling engine. As a result of the recent testing of the prototype residential unit engine, and the increased emphasis on achieving product performance goals, the field test is being deferred until product performance goals for the prototype are achieved. The remaining Phase II work will therefore consist of engine testing and final documentation of program results, scheduled for completion in June of 1981. The results of Phase II will then provide the basic hardware, test facilities, analytical tools and market/business assessments necessary to initiate the Phase III contract.

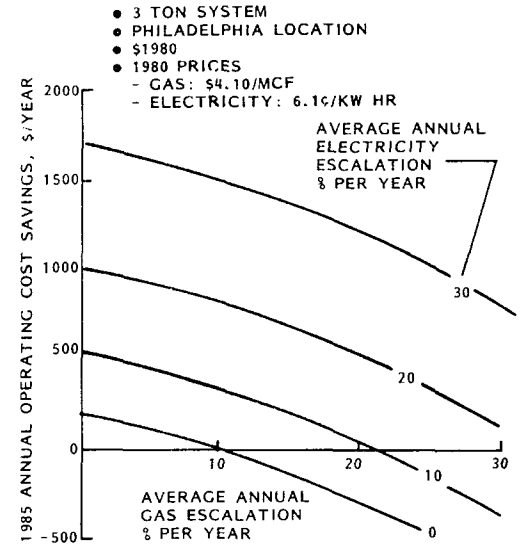



Figure 15. 1985 Savings of HAHP Over An Electric Heat Pump

Post-contract activities. As this paper is being written, the detail plan for the Phase III contract of the program is being established. The initial effort in Phase III will be to complete development of the Stirling engine to achieve the required prototype hardware performance (33% indicated efficiency), and to improve and refine the engine/compressor design tools to permit design of a preproduction prototype product.

Figure 16 shows the overall schedule for the HAHP development and commercialization project.

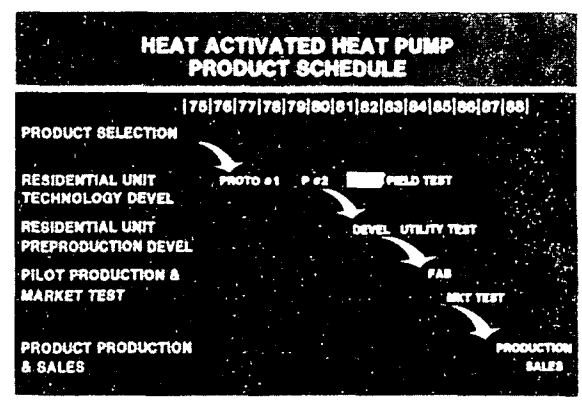



Figure 16. Heat Activated Heat Pump Product Schedule

After completion of the initial technology development of the Phase III program (Residential Unit Preproduction Development), 10 to 20 advanced units will be fabricated for extensive field testing by gas utilities in various climatic areas of the nation. The final phase of the project prior to production and market introduction will be a pilot production and market test program. In this final phase, 50 to 200 units will be produced and tested in a limited number of locales to conduct a market test and to prepare for product production, sales and service.

## PUBLICATIONS/REPORTS

Project Reports. Semi-annual reports of technical progress on the Phase II contract are available from NTIS.

- Development of a Gas Heat Pump, Project HC 115-1, Semi-Annual Report, January through June 1976

- Development of a Gas Heat Pump, Project HC 115-1, Semi-Annual Report, July through December 1976

- Development and Demonstration of a Stirling/Rankine Gas Activated Heat Pump, Contract No. EC-77-C-02-2911, Semi-Annual Report No. COO/2911-1, January through June 1977

- Development and Demonstration of a Stirling/Rankine Gas Activated Heat Pump, Contract No. EC-77-C-02-2911, Semi-Annual Report No. COO/2911-2, July through December 1977

- Development and Demonstration of a Stirling/Rankine Gas Activated Heat Pump, Contract No. EC-77-C-02-2911, Semi-Annual Report No. COO/2911-3, January through June 1978

- Development and Demonstration of a Stirling/Rankine Heat Activated Heat Pump, Contract No. EC-77-C-02-2911, Semi-Annual Report No. COO/2911-4, July through December 1978

- Development and Demonstration of a Stirling/Rankine Heat Activated Heat Pump, Contract No. EC-77-C-02-2911, Semi-Annual Report No. COO/2911-5, January through June 1979

- Development and Demonstration of a Stirling/Rankine Heat Activated Heat Pump, Contract No. EC-77-C-02-2911, Semi-Annual Report No. COO/2911-6, July through December 1979

- Development and Demonstration of a Stirling/Rankine heat Activated Heat Pump, Contract No. DE-AC02-77CS20043, Semi-Annual Report No. COO/2911-7, January through June 1980

Open Literature Papers. Several technical papers on project results have been published in the open literature, and are available from the publishing society/source.

- W. L. Auxer, "Development of a Stirling Engine Powered Heat Activated Heat Pump", Paper No. 779065, 12th Intersociety Energy Conversion Engineering Conference.

- T. J. Marusak, and W. S. Chiu, "The Matching of a Free Stirling Engine Coupled with a Free Piston Linear Compressor for a Heat Pump Application", Paper No. 789272, 13th Intersociety Energy Conversion Engineering Conference.

- W. D. C. Richards and W. L. Auxer, "Performance of a Stirling Engine Power Heat Activated Heat Pump", Paper No. 789453, 13th Intersociety Energy Conversion Engineering Conference.

- W. D. C. Richards and W. S. Chiu, "System Performance of a Stirling Engine Powered Heat Activated Heat Pump", Paper No. 799359, 14th Intersociety Energy Conversion Engineering Conference.

- "Natural Gas Heat Pump Implementations and Development", L. L. Dutram, Jr., and L. G. Sarkes, Conference on Research and Development of Heat Pumps in Europe and the United States, Haus der Technik, Essen, West Germany, September 1979.

- "Development of the General Electric Stirling Engine Gas Heat Pump", R. C. Meier, IGT Conference on Future Alternatives in Residential/Commercial Space Conditioning, June 1980.

DIAPHRAGM STIRLING ENGINE HEAT-ACTUATED HEAT PUMP DEVELOPMENT

MECHANICAL TECHNOLOGY INCORPORATED CONSOLIDATED NATURAL GAS RESEARCH CO.
CONTRACT NO. 86X-6161C
DR. ROBERT A. ACKERMANN PAUL SWENSON
PERIOD OF CONTRACT: BEGIN JUNE 1, 1980 END OCTOBER 31, 1982

## PROJECT OBJECTIVE

The objective of this program is to develop and demonstrate the performance of a diaphragm Stirling engine heat-actuated heat pump power module. The power module, consisting of a free displacer, resonant Stirling engine, hydraulic transmission, and resonant Rankine refrigerant (F-22) compressor, embodies several innovative concepts in free-piston Stirling engine heat pump design that will advance the state of the art of this technology.

## PROJECT DESCRIPTION AND STATUS

For the past two years Consolidated Natural Gas Research Company (CNG), together with Mechanical Technology Incorporated (MTI), has been developing a Stirling-engine-driven residential heat-actuated heat pump under the GASCON (gas conservation) program. In June 1980, DOE/ORNL became a third partner in this program, with the primary objective of developing a demonstrator engine/compressor power module based on some newly conceived and promising GASCON concepts. The key element in this new concept is the use of a hydraulic transmission that hermetically seals the engine working fluid from the compressor working fluid by a flexible metal diaphragm.

Progress has been made in three areas of the program. First, a compressor/engine matching analysis and a stability analysis have shown that the power module, which is representative of a two-degree-of-freedom resonant system, will operate stably over the full range of heat pump conditions. Second, a compressor design has evolved that has met our criteria for performance and cost; and third, tests employing a hydraulic simulator test rig has shown that the transmission losses are less than we had predicted, and that properly designed and fabricated diaphragms can attain long life.

## POWER MODULE DESIGN

The innovation in the power module design lies in separating the working fluids in the engine and compressor with a flexible metal diaphragm and transferring the power of the engine to the compressor through a hydraulic coupling.

As shown in Fig. 1, the power module will consist of three subassemblies: the engine, the hydraulic transmission, and the refrigerant compressor. The engine consists of the displacer, heat-transfer components, cooler, regenerator, heater head, and combustor. The hydraulic transmission consists of the engine power diaphragm, gas spring diaphragm, transverse vibration balancer, and hydraulic oil. The compressor consists of the piston, valves, and cylinder heads. These three subassemblies form a coupled resonant system that functions as a conventional Stirling engine; i.e., the displacer shuttles the engine working fluid (helium) between the hot and cold spaces generating the driving pressure wave for the system. The power piston (compressor) extracts work from the gas to power the load (refrigerant circuit). In the diaphragm system, power is extracted from the engine and delivered to the compressor through the hydraulic fluid which provides a force/displacement transfer path between these two components. This coupling also applies the proper dynamics to the engine to give it its resonant characteristics and volumetric phase relationships.

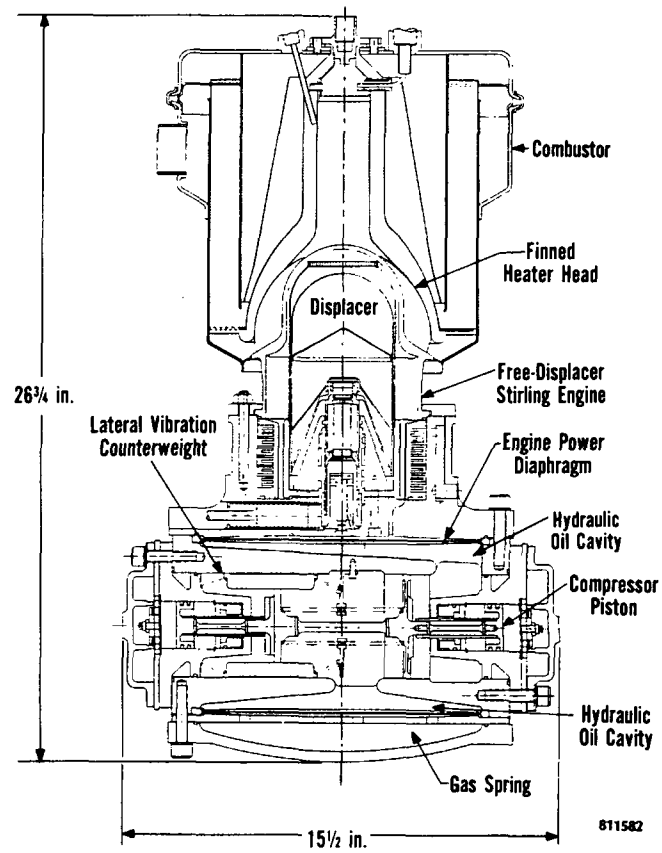



Figure 1 Diaphragm Stirling Engine Design for a Heat-Actuated Heat Pump Power Module

The benefits to be derived from this system are:

- The use of the power transfer diaphragm has reduced the system from a three-degree-of-freedom resonant system to a two-degree-of-freedom system. This will improve the operating control and stability of the system
- The hermetic separation of the working fluids thermally isolates the refrigerant suction flow into the compressor, minimizing the superheating of the suction vapor and the associated penalties in compressor performance
- The compressor closely resembles a conventional heat pump refrigerant compressor allowing for the use of many standard heat pump compressor parts.

## TECHNICAL ACCOMPLISHMENTS

- System analyses consisting of both matching and stability analyses were performed for the engine/compressor design. The matching analyses, performed to match the compressor and Stirling engine at four operating points (17°F, 47°F, 80°F, and 95°F), define the strokes, pressure, frequency, spring rate, and power that are required to match the engine and compressor over its full operating range. The stability analyses were performed to determine if the system will react in a stable manner to small perturbations

about the operating point, and also if the system will move with stability from one operating point to the next. The results of the matching analysis performed to mate an existing MTI Stirling engine to the hydraulically driven compressor are shown in Table I. The table shows that the system dynamics may be matched at each of the ARI test points by varying the mean engine charge pressure. However, as shown, the part-load performance of the engine drops 24% in moving from the design point (95°F) to the 47°F operating point. The conclusions drawn from this work were:

- The power module consisting of a two-degree-of-freedom system can be designed to operate stably at each of the ARI test points.
- Using an existing MTI engine for the breadboard system may not demonstrate a high overall system performance because of the poor part-load characteristics of the engine.

**TABLE I**
**SUMMARY OF ENGINE AND COMPRESSOR MATCHING FOR THE GASCON STIRLING ENGINE**

| Engine Operating Condition | | 95°F Day | 80°F Day | 47°F Day | 17°F Day |
|---|---|---|---|---|---|
| Heating/Cooling Load | (Btu/hr) | -36,000 | -18,000 | 14,000 | 40,000 |
| Charge Pressure | (bar) | 60.0 | 41.0 | 35.7 | 50.8 |
| Piston Amplitude | (cm) | 1.27 | 0.65 | 0.73 | 1.49 |
| Displacer Amplitude | (cm) | 1.02 | 0.39 | 0.34 | 1.02 |
| Phase Angle | (°) | 60.0 | 96.5 | 105.5 | 73.85 |
| Engine Power | (kW) | 2.75 | 0.50 | 0.40 | 2.40 |
| Cycle Efficiency | (%) | 31.0 | 27.0 | 23.5 | 26.0 |

The present status of the program and level of developments are as follows:

- Engine - Based on the matching analyses, a decision has been reached that a second generation of the existing engine developed under a CNG-MTI privately funded program will be evaluated for use on the breadboard system. Generically, this engine is similar to existing engines developed by MTI, but has a higher projected efficiency as the result of experience and test data collected on several free-piston Stirling engines.
- Compressor - The breadboard compressor design that has evolved from this program is shown in Fig. 2. Its operation is driven by a deflection of the engine power diaphragm which creates a force unbalance across the transverse vibration counterweight forcing it to move. Because of the incompressibility of the oil, the power piston is also acted upon but in the opposite direction to the counterweight. Compression spaces on either side of the piston, compress the refrigerant. The lower gas spring diaphragm and gas spring provide the restoring force for the system and give it its resonant characteristics. The advantages seen for the compressor design are:
  - The counterweight eliminates transverse vibrations internally, simplifying the mounting requirements
  - The horizontal orientation of the compressor leads to a smaller, more compact power module, in which standard heat pump valves and cylinder heads may be used
  - Performance calculations for the compressor, including valve and plenum pressure drop losses, indicate that this design will lead to adiabatic power efficiencies greater than 90% at each of the ARI test points.
- Hydraulic Transmission - To evaluate the uncertainties of the hydraulic losses, and diaphragm life associated with the hydraulic transmission, a hydraulic simulator test rig was developed at MTI. The rig, as shown in Fig. 3, consists of a linear motor mounted between two diaphragms. The space between the motor plunger and the diaphragms is filled with a hydraulic oil that transmits the motor plunger motion to the diaphragms. Gas springs on the outer side of each diaphragm produce the resonant dynamics for the system. The purpose of the simulator is to determine the practicability of hydraulically driving

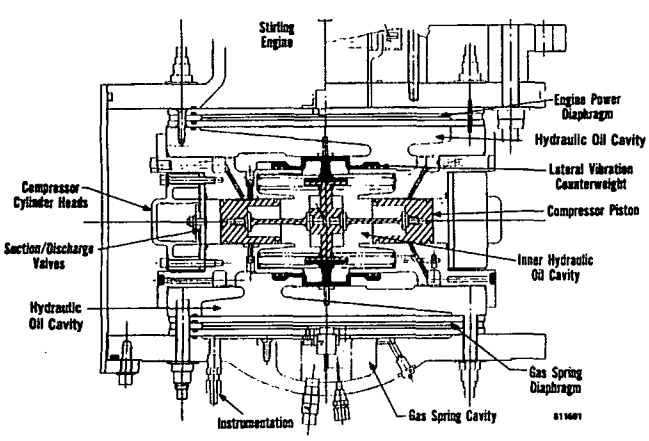



Figure 2 Hydraulically Actuated Resonant Refrigerant Compressor

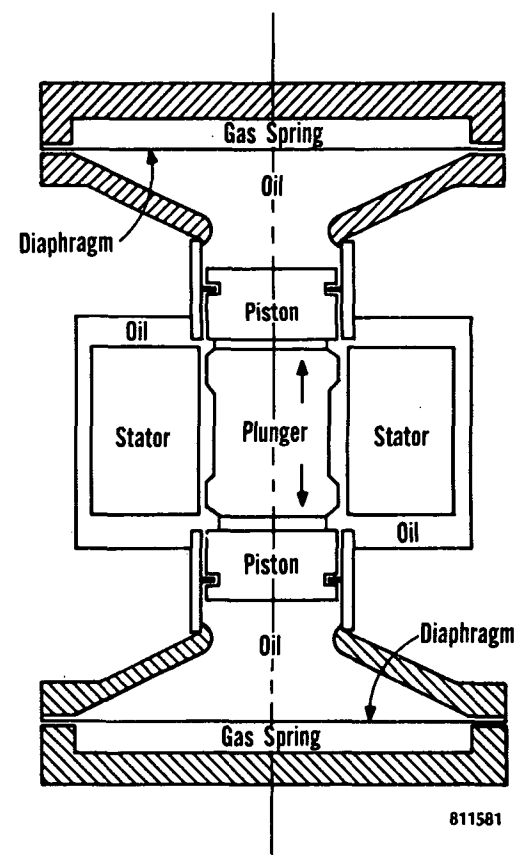



Figure 3 Hydraulic Simulator Test Rig

a compressor, and to evaluate the power losses associated with this type of transmission. The test objectives of the rig are:

- Demonstrate continuous long-term operation of the system and diaphragms
- Evaluate the mechanical losses
  - Oil viscous loss
  - Seal friction
  - Gas pring thermal hystersis

Testing to date has presented some very encouraging data. First, the total power loss measured is less than was analytically predicted. Fig. 4 shows the comparison between the predicted and measured loss. The significance of this is that the hydraulic transmission efficiency will be higher than the 90% originally predicted. Second, more than 10 million cycles have been put on a pair of contoured diaphragms. These diaphragms were developed by the Bendix Corporation to our specifications, and their operating for 10 million cycles at the design point (95°F) stress is an important step in demonstrating their potential for long life.

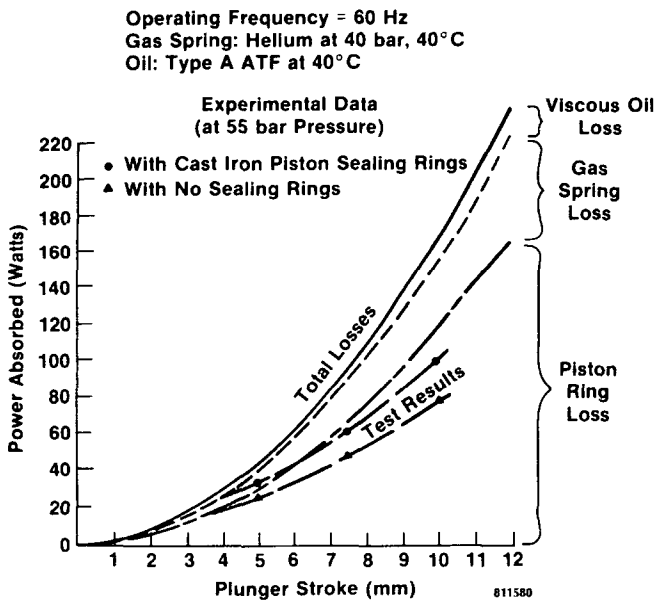



Figure 4 Comparison between Predicted and Measured Transmission Losses for the Hydraulic Simulator

• System Performance Calculations - Table II presents the COP for the power module at each of the rating points. The computed COPs are based on the results of the compressor design analyses, the measured hydraulic transmission efficiency, and the computed efficiencies for the privately funded second-generation engine. The system COPs at the cooling and heating design points are 1.27 and 1.63 respectively.

PROGRAM SCHEDULE

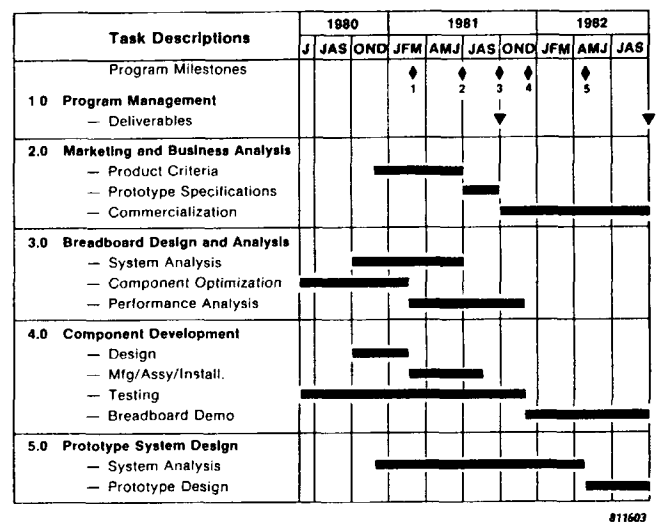
The original program schedule is shown in Fig. 5. The program, as defined, was designed to meet two objectives. The first was the design, fabrication, and testing of the breadboard power module, and the second was to design a prototype HAHP system for product evaluation. The scheduled plan was to start testing the breadboard unit by December 1981, and to complete both the testing and the prototype design by October 31, 1982. A recent work slow-down directive from DOE/ORNL will delay this program, however, and

**TABLE II**

**BREADBOARD HAHP PERFORMANCE SPECIFICATIONS**

| Engine Operating Condition | | 95°F Day | 80°F Day | 47°F Day | 17°F Day |
|---|---|---|---|---|---|
| Heating/Cooling Load | (Btu/hr) | -36,000 | -18,000 | 14,000 | 40,000 |
| Suction Pressure | (psia) | 90.7 | 111.3 | 83.4 | 42.2 |
| Discharge Pressure | (psia) | 277.4 | 193.6 | 188.8 | 264.4 |
| Evaporator Temperature | (°F) | 45.0 | 57.3 | 40.1 | 4.2 |
| Condenser Temperature | (°F) | 120.8 | 93.9 | 92.2 | 117.0 |
| Freon Flow Rate | (lb/hr) | 570.0 | 230.6 | 110.4 | 290.6 |
| Compressor Adiabatic Power | (kW) | 2.286 | 0.396 | 0.276 | 1.918 |
| Actual Compressor Power | (kW) | 2.741 | 0.440 | 0.306 | 2.062 |
| Operating Frequency | (Hz) | 60.0 | 55.8 | 53.8 | 55.9 |
| Compressor Efficiency | (%) | 92.5 | 90.2 | 90.2 | 93.0 |
| Hydraulic Transmission Efficiency | (%) | 94.0 | 90.0 | 80.0 | 91.0 |
| Engine Power | (kW) | 2.63 | 0.49 | 0.38 | 2.27 |
| Engine Efficienty | (%) | 38.0 | 32.0 | 30.0 | 38.0 |
| Thermal Input | (kW) | 6.92 | 1.53 | 1.27 | 5.97 |
| Combustor Efficiency | (%) | 83.0 | 80.0 | 80.0 | 83.0 |
| Combustor Thermal Power | (Btu/hr) | 28,435 | 6,527 | 5,399 | 24,542 |
| COP | | 1.27 | 2.76 | 2.59 | 1.63 |

reduce the work scope to include Tasks 1.0, 3.0, 4.0, and a portion of 5.0. A revised program schedule is being prepared, and it is anticipated that the breadboard demonstration testing will not start before April 1982.




**HAHP MASTER SCHEDULE**

**MILESTONES**

1. All Breadboard Hardware Committed to Fabrication
2. Stirling Engine Selection Decision
3. Prototype System Specifications Finalized
4. Start Breadboard System Testing
5. Submit Prototype Data Package to Potential OEM Candidates

811602

Figure 5 Original Program Schedule

DUPLEX STIRLING GAS FIRED
HEATING ONLY HEAT PUMP
SUNPOWER INC.
Contract No.: 5081-341-0464
William Beale
PERIOD OF CONTRACT: Begin May 1, 1981 End May 31, 1981

OBJECTIVE

The current program will complete the detail design of the duplex Stirling 10 kW heating only heat pump analyzed in the phase I study (Ref.1), construct the laboratory demonstration prototype, test the performance of the prototype over a range of source temperatures, and modify the hardware appropriately to meet the system performance requirements. Secondary objectives are an analytical assesment of the performance of a duplex Stirling System optimized for both heating and cooling, and a refinement of the phase I cost and payback time estimates.

BACKGROUND

The Duplex Stirling Concept. The duplex Stirling heat driven heat pump is essentially two free piston Stirling engines sharing a common piston and pressure enclosure. One working pair of displacer and piston act as a heat engine, driving the other pair in the same pressure envelop as a heat pump. In both halves the displacer receives its driving force from a differential area between expansion and compression spaces embodied in the displacer rod. The uniqueness of the design is that the two Stirling machines are served by a single piston common to the heat engine and heat pump. Both ends pump heat from the expansion space side to the compression space side, i.e. from the ends to the center. When high temperature heat is applied to its heater, the heat engine half produces mechanical energy which is used by the other half to pump heat from the cold source to the warmer sink. The resulting three element system is a potentially effective, simple and durable heat driven heat pump (Fig 1)

The concept is at least as old as the author's first Stirling engine work in 1964 (Ref 2) and has been independently proposed by others working on free piston Stirling engines, but has not yet been reduced to practice except in models of about 100 watt capacity. (Fig 2)

The duplex Stirling machine offers many advantages over competing heat driven heat pumps. This machine is completely hermetically sealed in a single pressure enclosure and uses a single common working fluid (helium or hydrogen) in both heat pump and heat engine. No sliding seals, flexing tubes, diaphrams or bellows are used to contain or isolate the working gas from a refrigerant or atmosphere. Aside from peripheral fans and pumps the basic machine contains only three moving parts which may be suspended by hydrodynamic gas bearings to insure long life. It has no overstroking problem, being balanced in energy produced and energy absorbed over the full range of outside temperatures. Not least, the machine is exceptionally quiet, and requires no acoustic enclosure or other noise abatement appurtanances.

The Duplex Stirling free piston heat pump is the latest embodiment of the free piston Stirling Concept originated at Ohio University in 1964, and carried on continuously since then by the same team, which moved out of the University in 1974 to form Sunpower Inc. The Sunpower team has designed and built many free piston Stirling engines and delivered them to major government and private research labs across the country. These engines have been fitted with linear alternators, inertia compressors, fluid pumps and Stirling heat pumps. In the course of their work the Sunpower team has solved problems in analysis, optimization, mechanical design, stability and control, burners and system configurations which has permitted it to commercialize free piston Stirling machines. The present heat pump program will have the advantage of their experience. (Ref 2,3,4)

SUMMARY

In phase I the Duplex Stirling Gas fired Heat Pump was analyzed with the aid of the Sunpower third order simulation and optimization code and found to show heating COP of 1.7 at the rating point of -8° C (17° F) with a heat delivery of 10 kW. Also it was found that a direct heat exchange between working fluid and source and sink air was consistent with high COP in the heat pump, thus eliminating the need for an intermediate heat transfer fluid and its associated pumps and controls. Further, it was demonstrated in the simulation that the heat engine power output could be matched to the heat pump load over the range of source temperatures without need of complex control systems.

Cost estimates, along with COP figures from the analysis indicated a payback time of 2 to 4 years in completion with existing combustion heating systems in northern climates.

TECHNICAL ACCOMPLISHMENTS

Phase I Analytical Study. The purpose of the Duplex Stirling Heating Only Heat Phase I Pump analytical Investigation was two fold: a) to analyze the concept in sufficient depth to prove or disprove its technical merit, and b) to make a preliminary estimate of the commercial prospects for such a machine, should its technical promise be realized in hardware.

A system capacity of 10 kW (34,000 BTU/hr) at the ARI low outdoor temperature heat pump standard, -8.3° C (17° F), was chosen as the design target. This number was based on a synthesis of information from manufacturers' current practices and preliminary estimates of SPF from an estimate COP vs outdoor temperature curve for the system.

The work was performed in three distinct phases. First, the engine and heat pump ends of the duplex Stirling machine were optimized for thermodynamic performance in isolation from each other. Second, the operation of the complete system over the expected range of outdoor temperatures was modelled and the matching problem was addressed. Third, the system was refined with emphasis placed on the principle indoor and outdoor air heat exchangers and associated air handling equipment.

Thermodynamic Optimization. Scaling of a successful 1 kW free piston Stirling engine and a previously developed computer model of a Stirling heat pump provided the starting point for the optimization work. The heat pump was scaled in such a way that its source and sink heat exchangers would be similar to air-freon heat exchangers common to the industry. The engine was scaled to match the power and stroke of the heat pump. A parametric search was then done for both engine and heat pump to optimize thermodyna-

mic performance. By this means, optimal strokes, phase angles, and heat exchanger geometry were arrived at. The concerns of fabrication difficulty and cost were also kept in mind.

The thermodynamic optimization was performed with a range of proprietary Sunpower computer techniques all of which have been proven by experience in practical designs already reduced to hardware.

Operation Over Range of Outdoor Temperature. A central issue of free-piston technology is matching a variable load without overstroking, understroking, or thermodynamic performance degradation. In the duplex Stirling technology the problem surfaces as a result of a seasonal and daily fluctuation in outdoor temperature. On a warm day the heat pump has an easier time pumping heat which the engine sees as a reduced load.

New techniques were developed to analyze the dynamic response of the duplex Stirling system over the anticipated range of outdoor temperatures. The previously optimized engine and heat pump halves were joined together and examined over the complete operating range. It was found that, as expected, there was a tendency to overstroke on warm days and understroke on cold days. Fortunately, simple means were available from the Sunpower proprietary storehouse of Stirling tricks to avoid this problem and still keep high thermodynamic perfomance over the range of outdoor temperatures without recourse to complex control gadgetry.

Description of System Resulting From Phase I Analysis. Figure 3 represents the final study concept in which the heat pump exchangers are placed directly in the air stream. Initially this approach seemed difficult due to the potential matching problem between the air side and heat pump side transfer requirements. However, computer simulation work showed that the matching would not be a problem and as a result the secondary fluid heat exchange loops could be eliminated. The incentive for removing these loops is twofold; the pumps are eliminated, thus reducing the auxiliary power requirements, and more importantly, the temperature drops between air and the heat pump working fluid are reduced.

Proof of Principle Model. In 1979 Sunpower undertook the construction of a demonstrator version of the duplex Stirling heat pump as a companion machine to its commercially successful 100 Watt linear alternator machine. Figure 2 is a photo of the system on test at Sunpower. It has been found to be nearly silent in operation and operates smoothly over the entire range of cold end temperatures from ambient to -70° C. The system is moderately efficient at its design point (heat pump COP .75) even though it is by no means an optimized engine, but was built solely to prove the feasibility of the duplex Stirling concept in operating hardware. (Ref 4 )

Commercialization Study. The COP values and the design layout allowed estimates of first cost and operating costs which were compared with those of direct combustion heaters of 60% and 90% efficiency. The results are shown below:

| CITY | PAYBACK PERIOD (YEARS) (60%) | (90%) |
|---|---|---|
| Minneapolis | 1.8 | 4 |
| Omaha | 2.5 | 4.4 |
| San Francisco | 6.5 | 8.2 |

FUTURE ACTIVITIES. At the time of writing of this paper, the second phase, construction of the prototype demonstrator, has just begun. It is expected that Phase II will take approximately a year in the following steps:

1) Detailed Design in which a thorough computer study will be done to assure proper system performance and stability, and shop drawing will be prepared.

2) Fabrication and Procurement of all test hardware.

3) Test and Modification of System to achieve and record desired performance over the range of outdoor temperature.

4) Report of Results.

In step 1, the initial decision to use a direct helium to air heat exchanger will be reviewed and compared with the alternatives of helium glycol-air, helium - helium - air (secondary loop of same gas at same average pressure as primary working fluid) and helium - heat pipe - air. This analysis requires the accurate handling of subtle interaction between cold and warm wall $\Delta T$, heat exchanger volume, pressure ratio, cyclic rates, displacer phase angle and amplitude, which can only be done with the use of a high order computer code including all thermodynamic and fluid dynamic effects, mechanical dynamics and parasitic losses and with a correctly interacting optimization routine. Without these analytical tools, the task would become far too complex to handle with any probability of a successful conclusion.

The detailed design will reflect the accumulation of 16 years of successful hardware programs on free piston engines at Ohio University and Sunpower, during which the problems of gas lubrication, sliding seals, piston centering, component dynamic interations, parasitic losses and the other practical concerns of mechanical design of these relatively new and unfamiliar machines will be addressed.

The most critical period of Phase II will be during the test and modification, when the real world intrudes upon the carefully organized structures of the engineers. However, a number of successful programs resulting in running hardware meeting customer specifications has given the Sunpower team confidence in the success of this important program.

Post Contract Activities. While the present study is intended to produce a heating only heat pump demonstration, preliminary studies already done indicate that there is liklihood of excellent thermodynamic performance from a duplex Stirling heat pump over the entire range of heating and cooling temperatures with little performance penalty compared to a heating only machine.

In fact the duplex Stirling gives promise of a very broad range of applications. The duplex Stirling can be made as a combined heat pump-air conditioner, using either air or a liquid as its heat source and sink. It can be made in larger sizes up to 100 kW per cylinder, or in much smaller sizes appropriate for heating or cooling single rooms or even refrigerators. It can be made to operate at much higher temperatures as a waste heat temperature booster, or at much lower temperature for food freezing or liquification of gases such as methane, and even for cryogenic uses.

In combination with a free piston - linear alternator, the duplex Stirling gives promise for use as a total energy package, which would use a single primary energy source - natural gas to heat and cool the house, supply electric power and hot water and possibly even liquid natural gas for use as a fuel for transportation.

## REFERENCES


1) Final Report to Gas Research Institute. Phase I Analytical Study Duplex Stirling Gas fired Heating Only Heat Pump. Sunpower Inc. 1980

2) Beale, William, Free Piston Stirling Engines - Some Model Tests and Simulations. SAE Paper 690230 1969

3) Gedeon, David, The Optimization of Stirling Engines. 13th IECEC Proceedings 1978

4) Senft, James, Advances in Stirling Engine Technology. 14th IECEC Proceedings 1979

5) Gedeon, David, Scaling for Stirling Engines. 16th IECEC 1981

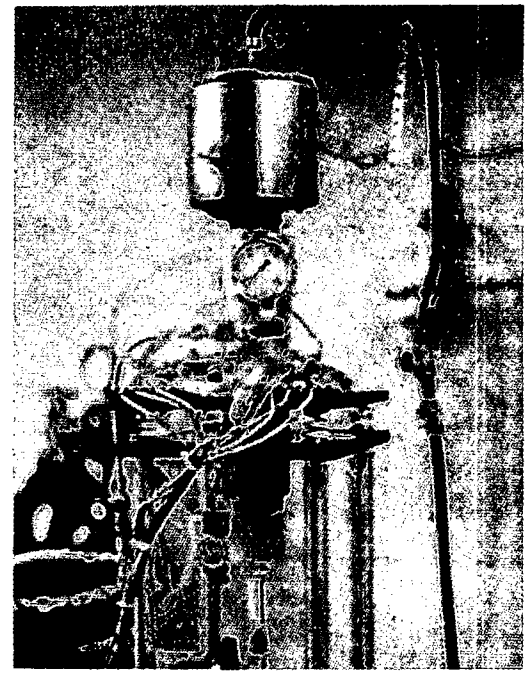



Fig. 2

Sunpower 100 watt Duplex Stirling Demonstrator on test

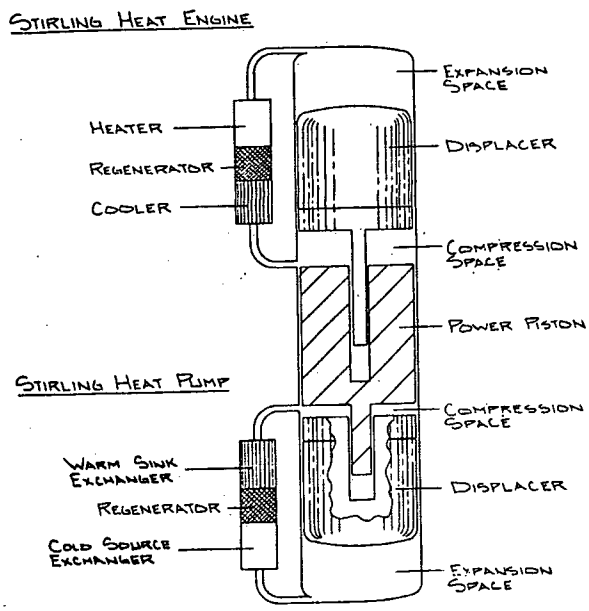
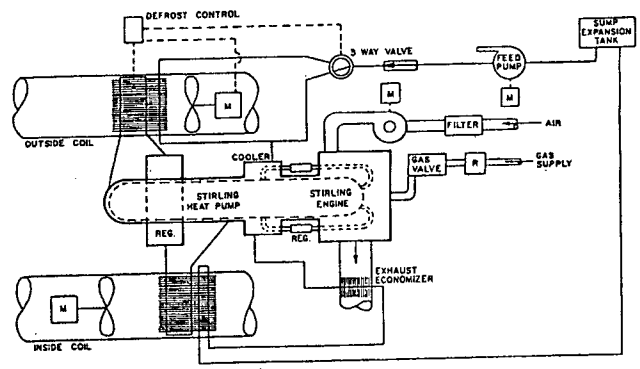



Fig. 1

Duplex Stirling heat driven heat pump




Schematic of the Sunpower Duplex Stirling Heating Only Direct Heat Exchange Concept

Fig. 3

FREE-PISTON STIRLING ENGINE TECHNOLOGY DEVELOPMENT
MECHANICAL TECHNOLOGY INCORPORATED
CONTRACT NO. 31-109-38-6420
GEORGE R. DOCHAT
Begin July 1, 1981 End December 31, 1981

## OBJECTIVE

The objective of this program is to gain an improved understanding of free-piston and Stirling engine technology by evaluating specific loss mechanisms within the engine during operation. The Technology Demonstrator Engine, developed by DOE and Mechanical Technology Incorporated (MTI), will be utilized as the test bed for this evaluation.

## BACKGROUND

The Technology Demonstrator Engine (TDE) is a 1-kWe free-piston Stirling engine/linear alternator system that has been designed and fabricated under Department of Energy Contract #EC-76-C-02-2764 to demonstrate the technology of this engine-power-conversion system. Complete description and previous test history is found in References 1 and 2. In summary, the TDE system consists of:

- natural-gas combustor;
- monolithic heater head;
- free-piston engine;
- linear alternator; and,
- necessary controls and sufficient instrumentation

The natural-gas combustor system has a recuperator to recover exhaust-gas thermal energy, and a turbulent-mixing combustor. The monolithic heater head is made of Inconel 617 and has both internal and external fins to augment thermal energy transfer from the combustor to the internal helium working fluid. The free-piston engine/alternator system has only two moving parts whose motion is determined by their respective masses and gas springs. The moving members (displacer and power piston) are "free;" that is, there is no physical connection between them. The displacer and power piston are supported by hydrostatic gas bearings with noncontacting clearance seals. The layout of the TDE system is shown in Figure 1.

As discussed in Ref. 2, the engine has achieved:

- demonstration of the 1-kWe steady state design power output;
- validation of analytical code predictions with test results; and,
- accumulation of over 350 hours of operational test data.

In addition, the TDE system is well-suited for use as a test bed to further understanding of loss mechanisms and to aid in increasing the performance of free-piston Stirling engines.

## PROGRAM DESCRIPTION

The program will utilize the TDE system as a test bed to evaluate specific loss mechanisms within the engine during operation. This will provide meaningful information and test data to increase understanding of free-piston operation, and will be of benefit to application development programs for heat-activated heat pumps, fluid pumps, engine generators, etc. The evaluation will include the current analytical procedures for calculating the specific losses, perform a series of tests designed to isolate each loss, comparison of actual changes in engine performance with predicted change, and to determine the adequacy of present analytical methods. Recommendations to improve overall engine/system performance and/or analytical methods will be made.

## EXPERIMENTAL EVALUATION

As discussed in Reference 2, the TDE has the necessary and sufficient instrumentation, flexibility of design dynamics, wide range of test conditions, ability to modify design and incorporate changes, and analytical codes to qualify for the intended purpose. The experimental assessment of the magnitude of four specific engine loss mechanisms will allow the validation of analytical techniques available for predicting same. The four loss mechanisms are:

1. compression and expansion space hysteresis loss;
2. displacer seal leakage loss;
3. displacer (dome-to-cylinder) gap loss; and,
4. regenerator hysteresis loss.

The experimental approach will involve back-to-back performance comparison of the engine with and without modifications to change the loss levels in each area. The magnitude of each loss will be assessed indirectly from measured and predicted changes in engine performance. The changes may, in fact, worsen overall performance. The approach will be to make changes that significantly affect performance that will then be attributed to that change.

Baseline Testing. The primary objective of this task is to measure and document the performance and operating characteristics of the TDE configuration in order to provide a baseline for engineering comparison and assessment. A further objective is to demonstrate the repeatability of the measured data, and the reliability of the selected engine hardware. Performance data and operating characteristics will be presented as maps of the following parameters:

- indicated power;
- indicated efficiency;
- mean heater temperature;
- displacer stroke ratio;
- displacer phase angle; and,
- engine frequency,

versus piston stroke. Two sets of maps will be provided. In the first set, the above parameters will be mapped against piston stroke for discrete values of control temperature, and in the second set, the parameter maps will be against piston stroke for discrete settings of the displacer gas spring control piston.

The number of data points taken for each map will be sufficient to properly characterize the engine performance. Each map point will be the result of a number of individual data points taken over a time period consistent with good sample statistics. In general, it is anticipated that the baseline performance maps will consist of at least five stroke points for each of three values of control temperatures and displacer spring

piston settings. Previous test data obtained from the TDE is shown in Figures 2 through 6, and provides a typical operating envelope over which the baseline data will be obtained (i.e., power, stroke, stroke ratio, pressure, volume control rod setting (VCR), displacer phase angle, etc.).

Compression Space Hysteresis Loss Test. The objective of the compression space hysteresis test is to validate the present analysis procedure by back-to-back performance tests of the engine with discrete changes in the compression space wetted surface. The change in wetted surface will be achieved by attaching surface area extensions to the face of the engine power piston. The surface area extensions will include two finned plugs and two conical piston face recessions. The wetted surface parameters for the two finned plug configurations are as follows:

| Case | C.S. Surface Area |
|---|---|
| Baseline | 157.3 $cm^2$ |
| Finned Plug A | 242.0 $cm^2$ |
| Finned Plug B | 334.5 $cm^2$ |

The baseline geometry in this case will be an engine test with a slightly recessed piston face to match the dead volume of the two finned plugs.

Displacer Seal Loss Test. The current TDE hardware uses a clearance seal between the displacer skirt and the cylinder wall to separate the compression and the expansion spaces. Flow through this seal results in parasitic losses and a potential thermal energy transfer from the hot expansion space working gas directly to the cooler. Typical seal clearances in this area are 3 to 4 mils — tighter clearances would be desirable. Sensitivity of engine performance to variations in seal clearance will be evaluated by back-to-back engine tests with different clearance values. The very small clearance performance will be evaluated by using a Rulon piston ring to close the seal gap. The following configurations will be evaluated:

- 3 mil clearance without ring
- 3 mil clearance with ring
- 6 mil clearance without ring
- 6 mil clearance with ring

A significant performance degradation is expected as the clearance with no ring is increased; however, the performance with the ring is expected to remain constant if the ring is effective. A comparison of different clearances with piston rings will, therefore, provide an indication of ring effectiveness.

Displacer Appendix Gap Loss Test. The appendix gap is the thin working gas annulus between the displacer dome and the engine cylinder. Previous analysis of the loss in this gap region, based on a model by Rios (Ref. 1), concentrated on the axial conduction losses generated by conductive transfers across the gap as the displacer reciprocates. This loss is descriptively referred to as the displacer "shuttle loss." The shuttle loss has been traditionally treated as a direct conduction-type loss; however, recent analysis indicates that it does, in fact, affect both power and efficiency.

In the present engine configuration, the appendix gap is 30 mils. Additional tests will be performed with appendix gaps of 80 mils and 120 mils to evaluate this effect.

Regenerator Matrix Loss Test. The regenerator matrix selection technique trades pumping, thermal, and void fraction loss effects to provide an optimal geometry and porosity for a specified matrix wire size. To aid in assessing the accuracy of the present regenerator model, four variations in regenerator parameters will be tested on the engine. The regenerator matrices for this test will be a woven and pressed wire material known by the tradename of Metex. This material is relatively inexpensive compared to the standard square-weave screens used in the present engine configuration. The regenerator parameter variation is as follows:

| Test | Matrix Type | Porosity | No. of Strands | Wire Size (mils) |
|---|---|---|---|---|
| Baseline | Screens | 0.90 | 1 | 1.0 |
| A | " | .70 | 3 | 2.0 |
| B | " | .75 | 3 | 2.0 |
| C | " | .80 | 3 | 2.0 |
| D | " | .85 | 3 | 2.0 |

The affect of porosity on engine performance will be of primary interest. High porosities have greater free flow area and, therefore, low pumping losses; however, the heat capacity and, therefore, effectiveness is low. Low-porosity regenerators have high heat capacity, but suffer high pumping losses. An optimum porosity, therefore, exists for a given engine geometry.

## PROGRAM ACTIVITY

The program to evaluate loss mechanisms within an optimum free-piston Stirling engine will begin as of July 1, 1981.

## REFERENCES

1. Dochat, G.R.; Moynihan, T.M.; Dhar, M., "1-kWe Free-Piston Stirling Engine/Linear Alternator Test Program," 15th IECEC, #809400.

2. Dochat, G.R.; Vitale, N.G.; Moynihan, T.M., "Development of Free-Piston Stirling Engine Test Bed Engine," 16th IECEC, #819784.

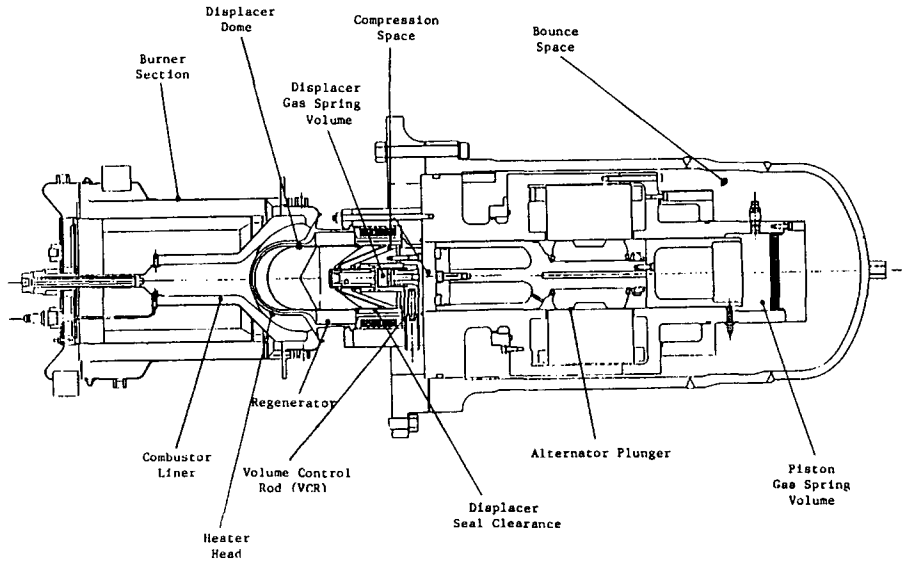



Figure 1 TDE Engine Layout

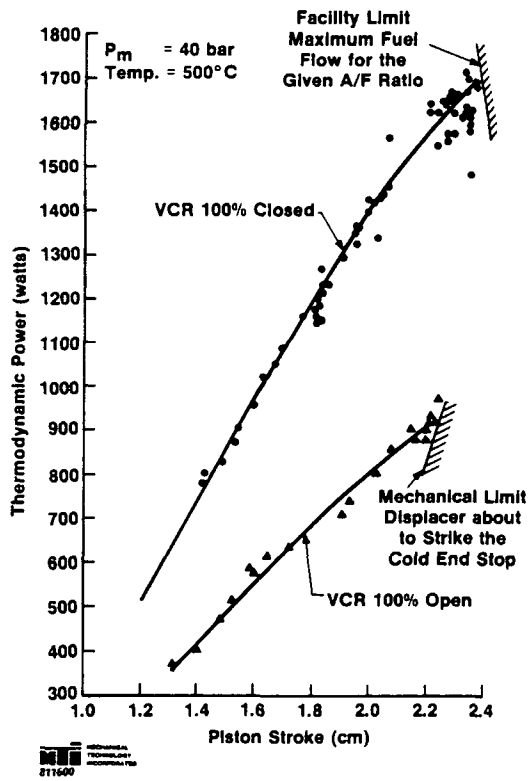



Figure 2 Thermodynamic Power Versus Piston Stroke

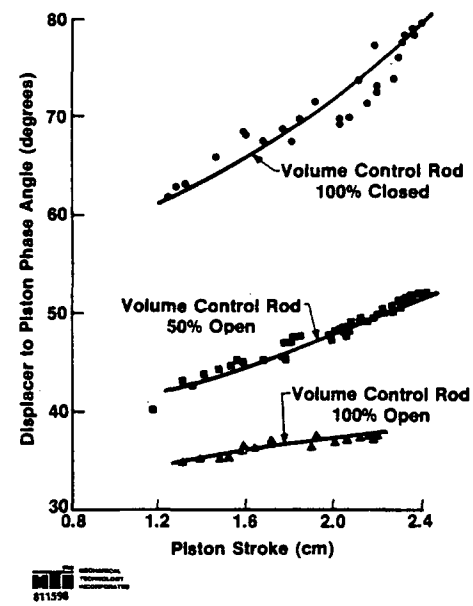



Figure 3 Displacer-to-Piston Phase Angle Versus Piston Stroke

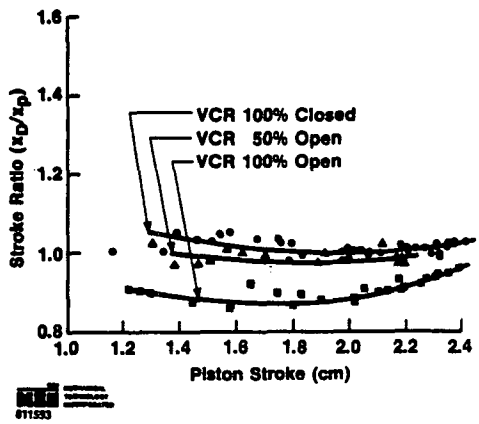



Figure 4 Stroke Ratio

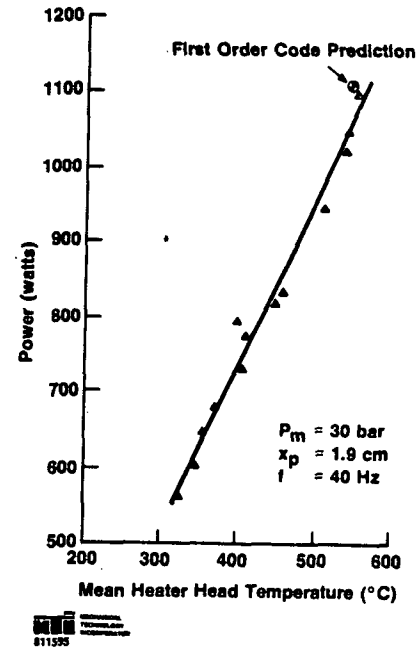



Figure 6 Total Cycle Power Versus Head Temperature

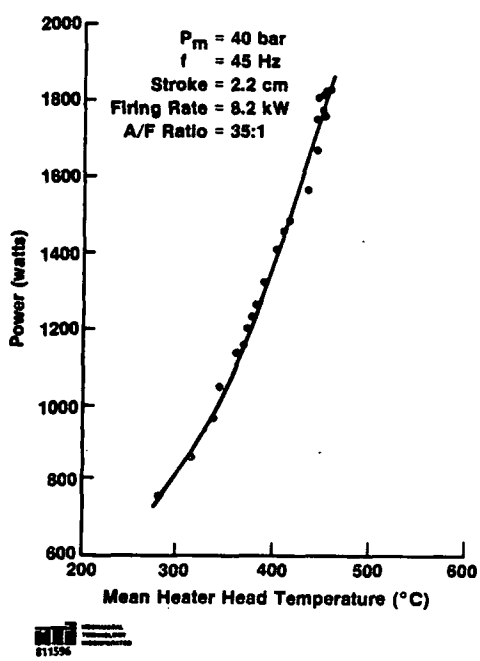



Figure 5 Total Engine Thermodynamic Power Versus Mean Head Temperature

KINEMATIC STIRLING ENGINE POWER SYSTEM DEVELOPMENT AND DEMONSTRATION
GENERAL ELECTRIC COMPANY
CONTRACT DE-AC01-76ET32002
PROGRAM MANAGER: J. A. BLEDSOE
PERIOD OF CONTRACT: JUNE 1976 TO DECEMBER 1980

## OBJECTIVE

The Kinematic Stirling Isotope Power System (SIPS) was designed to demonstrate the application of isotope powered dynamic systems for terrestrial and future space applications. The power system was required to provide more than 1 kW(e) at 100 Vdc with an overall system efficiency approaching 30%. It was required to operate continuous and unattended for periods of six months with only routine maintenance between operating periods.

## BACKGROUND

The SIPS is a Stirling cycle power conversion system which utilizes thermal energy received from a radioisotope heat source. The SIPS consists of the following major components and subsystems (see Figures 1 and 2):

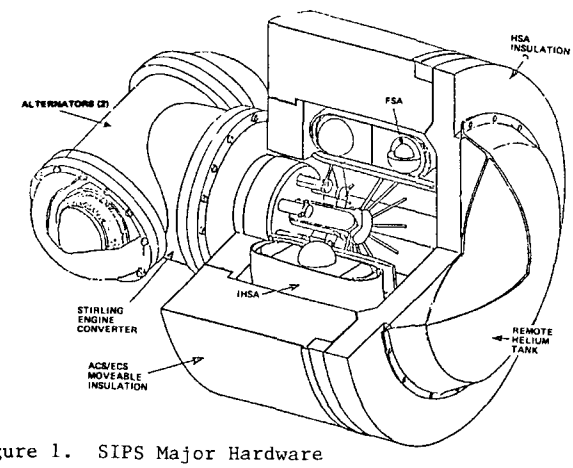



Figure 1. SIPS Major Hardware

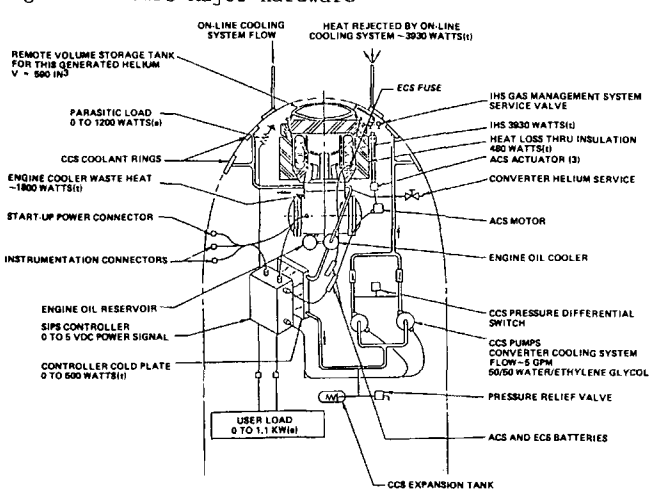



Figure 2. Stirling Isotope Power System

A Power Conversion System (PCS) which includes:

- Stirling engine converter
- Primary support structure
- Insulation assembly
- Auxiliary and Emergency Cooling Subsystem (ACS and ECS)
- Converter Cooling Subsystem (CCS)
- On-line Cooling Subsystem (OCS)
- Power regulation and engine control subsystems

An Isotope Heat Source Assembly (IHSA) which includes:

- Isotope Heat Source (IHS)
- Remote helium tank
- Gas management system
- HSA insulation pad
- Heat Source support structure

Figure 3 shows a system block diagram of SIPS. The movable insulation (ACS/ECS) is positioned in the open position and the heat source is installed into the power conversion system. The insulation is closed and the heat is radiated to the Stirling engine heat exchanger. A cooling pump from the converter cooling system circulates cooling fluid through the regenerators and oil cooler. When operating in water, only this internal system is required. However, for out of water operations, an external heat exchanger (on-line cooling system) is operated on conjunction with the internal system to effectively cool the SIPS.

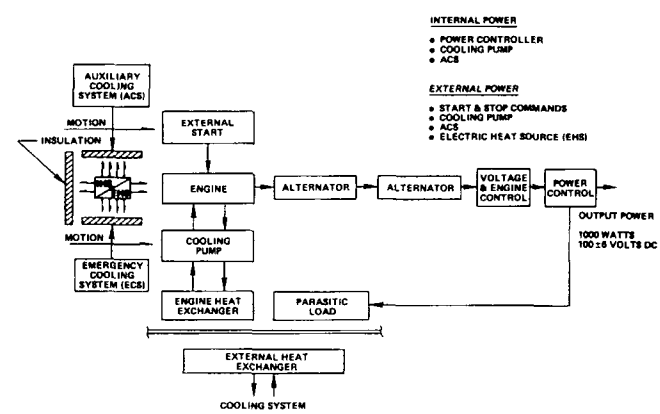



Figure 3. System Block Diagram of SIPS

When the prescribed starting temperatures are reached, the engine is started by an external starting system which utilizes the two alternators as starting motors. The Stirling engine then drives the two alternators (wired in series) to provide the electrical power for the demonstration unit. Their output is further conditioned and controlled to provide ≈ 1000 watts of power to the user at 100 ± 6 volts dc. The power controller also adjusts for load conditions, and during periods of low demand by the user the power is dumped to a parasitic load which dissipates the heat load through the demonstration unit.

As future mission power requirements increased, and size and weight continued to be system limitations, considerable interest in dynamic thermal energy converters was created, resulting in several study and hardware programs by the DOE. Add to the proceeding, the requirement for long life and high reliability, the

isotope heat source was introduced into the senario with a very positive performance history from the Multi-Hundred Watt Radioisotope Thermoelectric Generator (MHW-RTG) Programs. The SIPS Program was initiated to utilize the current proven technology of the MHW Programs coupled with the extensively developed Stirling engine technology.

Under U.S. Department of Energy contract EX-76-C-16-3084, the Space Systems Division of General Electric and its subcontractor Philips Laboratories fabricated a Stirling power conversion system capable of being powered by an isotope heat source. The isotope heat source and its' unique support systems have been designed and documented; however, as a result of program redirection, fabrication was not initiated.

The program redirection received in July 1978 transferred the management of the program to the office of Fossil Energy Utilization and emphasized engine testing and technology demonstration. In July of 1980, the management responsibility within DOE was changed to the office of Energy Conversion and Utilization Technology with the program emphasis remaining the same with additional testing being authorized up to 500 hours.

## SUMMARY OF ACCOMPLISHMENTS

Although the original program was redirected to a technology demonstration program in 1978, the emphasis on the engine development and fabrication continued as planned. All major components of the system completed the design and documentation phase. Major components and subsystems required for engine test all met their performance goals. The engine achieved an efficiency of ~28.6% with 3800 watts heat delivered to the engine as compared with the 30% predicted value at the design point conditions. Total engine operating time was over 500 hours and post test inspections revealed no major problems with the hardware. One minor problem was associated with the lubrication system which is easily corrected.

General Electric has completed a detail review of the engine performance, and prepared an outline of how the engine could be utilized to further enhance the Stirling engine technology database. Details of the engine performance and other system components are detailed in the final report which will be distributed after authorization by DOE.

## TECHNICAL ACCOMPLISHMENTS

Developed a Stirling Engine Design Compatible With An Isotope Heat Input. Figure 4 is an outline drawing of the SIPS engine. The SIPS engine design is of a single cylinder rhombic drive type, directly driving two internal alternators. Twelve heat receptor fins in the hot-side heat exchanger receive the radiant heat from the isotope heat source. The heat is transferred to the working fluid (helium) which undergoes compression and expansion according to the Stirling cycle. Thermodynamic waste heat, as well as other thermal and mechanical losses are rejected at the engine cooler and oil cooler to a stream of circulating coolant. The working spaces of the engine and crankcase are pressurized to the same mean pressure. A dry sump lubrication system is used to improve the mechanical efficiency of the drive system.

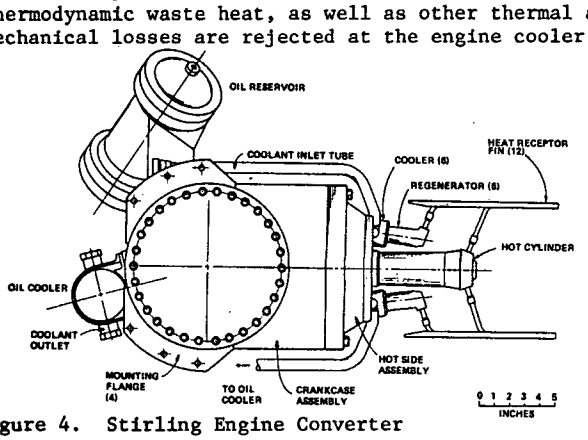



Figure 4. Stirling Engine Converter

A Philips rhombic drive configuration was chosen for the engine. This drive provides very quiet operation, and Philips has a great deal of experience with its design.

The engine speed was selected after consideration of a number of pertinent factors. It is known from experience that the flow losses, and to a certain extent the drive friction losses, decrease with decreasing speed. However, as a consequence, at constant power output the dimensions of the engine increases. However, the size of the engine is limited by available space in the SIPS envelope. Also, the alternator efficiency decreases if the speed is made too low. From these considerations, the conclusion was made that the engine speed should be between 1800 and 2200 rpm.

Helium was selected as the working fluid; hydrogen would have provided better performance, but containment would have been a difficult problem since it would diffuse rapidly through the hot heater tube walls, so it was ruled out.

The initial engine dimensions were arrived at by applying scaling rules which allow calculating dimensions for a new design based on the experience from an existing engine design. In this instance, the Philips 1-98 rhobic drive engine was chosen as the basis for this procedure.

The final dimensions of the engine were arrived at via an iterative process between computer optimization of the cycle efficiency and layout drawings of the engine. Layout drawing were necessary to permit a judgment to be made whether the outcome of the optimization was practicable, particularly with respect to size limitations. Table I list the essential dimensions of the SIPS engine.

Table I. SIPS Engine Dimensions

| | |
|---|---|
| Speed | 1950 rpm |
| Pressure | 95 atm |
| Piston $\phi$ | 49.2 mm |
| Displacer $\phi$ | 49.2 mm |
| Stroke | 22 mm |
| Heater Tube ID | 2.9 mm |
| Number Heater Tubes | 368 mm |
| Cooler Tube ID | 0.685 mm |
| Cooler Tube Length | 42.7 mm |
| Number Cooler Tubes | 117 |
| Number of Regenerators | 6 |
| Regenerator $\phi$ | 22.9 mm |
| Regenerator Length | 45 mm |
| Mess Wire $\phi$ | 41 $\mu$m |
| Regenerator Filling Factor | 32 % |

Utilizing these final dimensions, the power delivered at the output of the engine alternators was calculated for the nominal design case to be 1230 W @ 1950 rpm, 50°C coolant temperature, 750°C heater tube temperature, and 3500 W heat input to the engine.

Designed and Developed Key Engine Components for Isotope Powered Stirling Applications. In Stirling engine development, the seal between the buffer space and crankcase has always played an important role. Classically, the function of this seal is two-fold: to prevent loss of working fluid pressure to the crankcase, and to prevent leakage of lubrication oil into the buffer space.

In the SIPS design, the crankcase is pressurized; therefore, the only function of this seal is to exclude oil from the buffer space.

The seal selected for this application is reversed pumping ring, or "scraper seal." Figure 5 shows the configuration. A pumping ring has been turned upside down; it now scrapes oil from the rod when the rod moves upward and pumps excess oil back to the crankcase when the rod moves downward. This type of ring was used for many thousands of hours at Philips under the more severe operating requirements of a pumping ring in the Philips 1-98 engine and other designs.

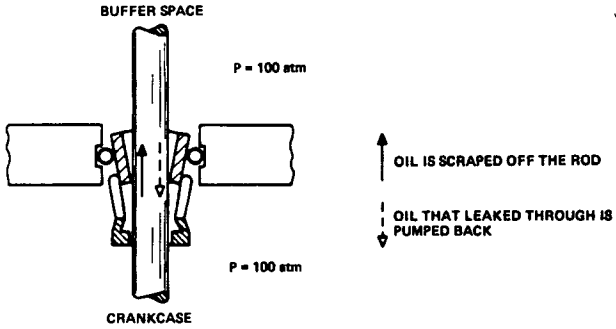



Figure 5. Scraper Seal Configuration

The hot-side assembly consists of a hot cylinder and six regenerator housings brazed into a flange, all of Udimet 700, and twelve heat receptor fins of Inconel 625 tube, sheet, and silver. The assembled brazement constitutes a unitized portion of the engine pressure vessel.

Two different nickel-based alloys are used in the construction of the hot-side of the SIPS engine. The first, Udimet 700, is a sigma-phase stablized, precipitation hardened alloy which requires a complete heat treatment schedule. The second, Inconel 625, is a solid solution hardened alloy and as such does not require any heat treatment other than a solution treatment of 1100-1200°C

The heat receptor fin/tube requires the use of a high thermal conductivity material (fin) in combination with a high creep strength material (tube). Silver and Inconel 625 were selected for this application. A silver casting process was developed which results in metallurgical bonding of the silver to the Inconel 625 and affords control of both rate and direction of solidification. Confirmatory testing involved investigations into the diffusion of silver into Inconel 625, the effect of silver diffusion on the creep-rupture strength of Inconel 625, and the effect of thermal cycling on the bond between the silver and the Inconel 625.

The structural design of the SIPS pressurized crankcase was patterned after the general form of a piping tee-fitting, and was forged from 316 stainless steel. This alloy was chosen because of its commercial availability, corrosion resistance, its reasonable strength and non-magnetic characteristics.

Two flange-mounted hemispherical end domes seal-off the "run" ends of the tee, while the hot-side assembly seals the "branch" end of the tee. The tee-forging starts as a solid piece of metal and is cored-out after forging. Thus, initially, the "T" has very thick wall sections, permitting a great deal of design freedom in establishing the internal geometry of the crankcase.

The structural design of the engine pressure vessel was undertaken in accord with criteria designed to maximize safety. All crankcase, oil reservoir, and oil color components are designed with a safety factor of 4 applied to the ultimate strength of the material, while a safety factor of 2.25 on the rupture stress (to 100,000 hours) was applied to the high-temperature components in creep-rupture service.

The alternators for the SIPS system were designed, developed, and fabricated by Inland Motor Division of Kollmorgen Corp. The rotors are mounted on the output shafts of the rhombic drive. The stators are supported by the crankcase of the Stirling engine. The rotor support bearing is therefore part of the engine and bearing losses are not charged against the alternator efficiency.

Conducted 500 Hours of Performance and Endurance Testing on the SIPS Engine. The development effort at Philips Labs culminated with the successful testing of the first (and only) SIPS engine, the test exceeding 238 hours in length. One previous test run was terminated when apparent lubrication problems caused anomalous engine behavior. These problems were isolated during an interim teardown period and steps taken to correct them. Typical performance of the engine resulted in 1065 watts of shaft output power resulting from a net heat input of 3620 watts, for an efficiency of 29.4%. Following completion of the test, the engine was disassembled and inspected. The major problem uncovered was the existence of approximately 700 ml of oil in the lower end bell cavity (200 ml in the upper cavity), and a significant deterioration of the alternator rotor. The oil leakage into the end bells was thought to have been caused by helium loss through crankcase electrical penetrations, which caused oil droplets in the helium to be transported from the crankcase to the upper end bell cavity and in turn to the lower cavity. Corrective measurements included a change in the engine's internal porting configuration. Disassembly of the oil pump showed some evidence of wear, and modifications were made to change the loading on the pump gears and shafts. The Bal-seal on the displacer was replaced with one having a lower force internal spring to attempt to reduce the wear rate on subsequent engine operations.

The engine was then reassembled and shipped to GE, where additional testing was conducted. The goal was to perform off-design point testing while increasing the endurance test time on the engine. Due to a series of minor problems with engine components, a total of sixteen start-ups were performed over a period of three months and a single test of greater than 167 hours duration was accomplished. The engine was purposely stalled after fifteen data points had been obtained at off-design point conditions. The problems encountered on earlier attempts included oil pump drive gear wear, oil pump shaft to gear lock problems, oil pump internal friction and binding, oil filter clogging, and displacer seal leakage. The oil pump and lubrication system problems all seemed to be related to pump friction. Materials for the drive gears were changed and the pump reworked to overcome those problems. The displacer seal leaks were traced to a thin teflon spacer that inadvertently acts as a flapper valve and sealed the Bal-seal at the low point of the engine's pressure cycle; thus not permitting it to expand to effect a seal at higher pressures. During operation, the engine also experienced oil leakage from the crankcase into the buffer space and end bell cavities. Oil drain valves were added to these components to permit oil removal, and make-up oil was then added to the crankcase through a valve added above the oil reservoir. Over two liters of oil were removed from each of the two spaces during the final test. A shift in the pressure relationship between the buffer space and crankcase was also encountered during the test.

The test results gained are comparable in both power and efficiency to the test values observed at Philips Laboratories. Post test disassembly and inspection showed only slight discoloration of metal surfaces and some oil residue in the lower end of the hot side. Figures 6, 7 and 8 illustrate both the system test schematics and the engine at its test support equipment. Figures 9 and 10 depict the predicted performance and the actual engine data profile during the testing during the first 238 hours of testing.

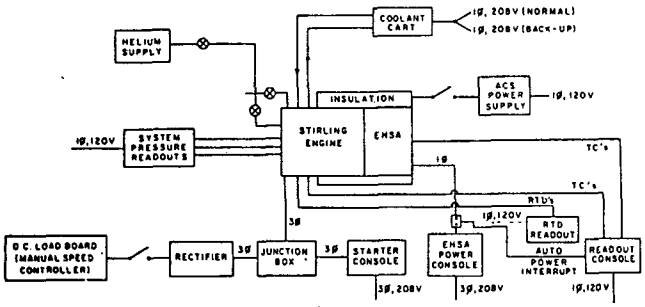



Figure 6. SIPS Test Schematic

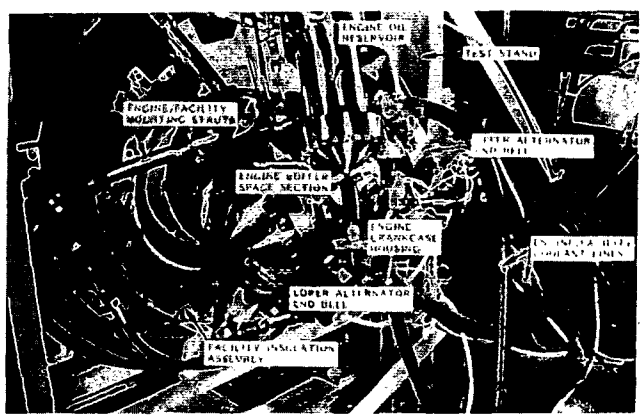



Figure 7. SIPS Engine Mounted to Stand

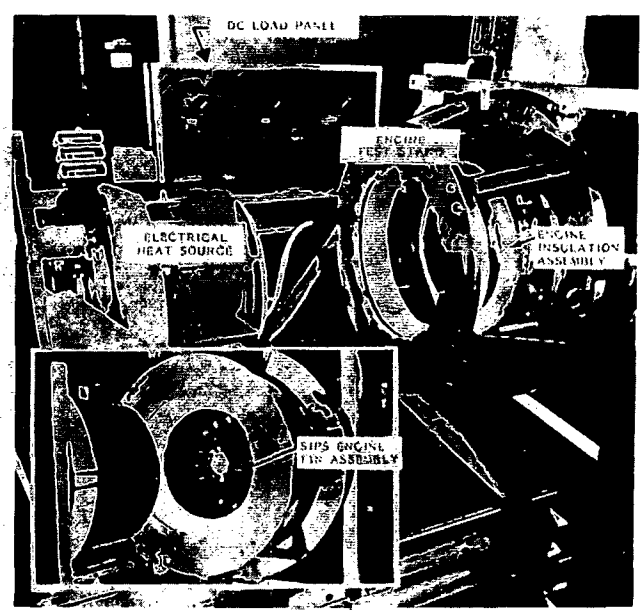



Figure 8. SIPS Engine Test Support Equipment

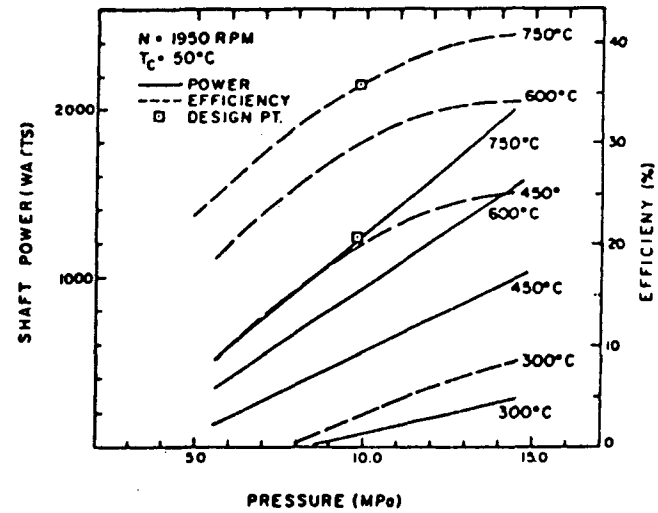



Figure 9. Predicted Shaft Power and Efficiency vs Pressure

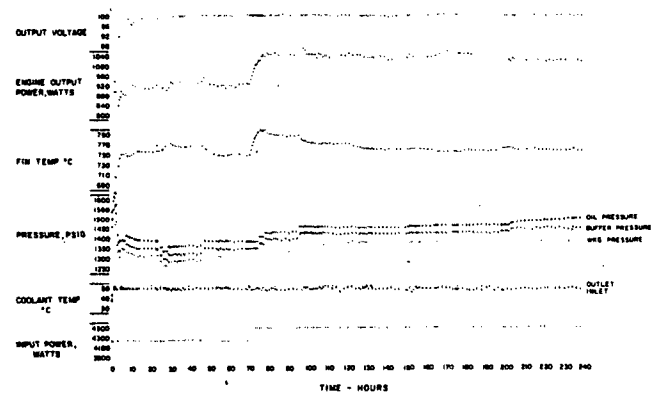



Figure 10. Engine Data Profiles

## FUTURE ACTIVITIES

Contract Activities. The contract activities have essentially been completed with the technical project manager's approval of the final report. Evaluation of proposed additional test activities are underway.

Post-Contract Activities. At this time, no future activities are under contract. The following is a summary of General Electric's preliminary plan for future engine development.

The SIPS engine has proven itself to be capable of laboratory operation for extended periods of time, and will certainly lend itself well to technology enhancing test programs aimed at furthering the development and understanding of the Stirling engine. The engine employs a reliable, mature rhombic drive to convert the linear motion to rotary motion and the heat source is simple, reliable and well suited for laboratory use. The performance of the baseline engine has been well characterized, and the possibilities for engine modifications are excellent. Of particular value is the engine's pressurized crankcase design, which eliminates the large pressure differential across the rod seals. This provides a great deal of flexibility when testing dynamic seals since the crankcase pressure can be altered, a capability not available on many Stirling engines. These seal tests could be designed to evaluate the positive type (roll-sock) seal as well as the scraper type seals presently in the engine, since the original design was capable of utilizing the positive type in the event a development problem precluded the use of the scraper seals. These tests could prove the effectiveness of the seals as well as aid in material selection and life predictions.

The SIPS engine also would be an ideal test bed for evaluation displacer and piston seals, providing the option of changing the liner against which the piston rings mate to allowing material pairs compatibility to be studied. Studies of oscillating flow characteristics could be performed to increase understanding of periodic flow heat transfer by adding additional instrumentation to the engine's hot side, and further evaluation would be possible if the regenerator, cooler and/or heater head geometry were modified. Evaluation of the effects of lubricant fouling of the hot side is possible by metering fixed quantities of the fluid of interest into the cooling space while monitoring performance. Another area of possible study is the evaluation of alternate forms of heat sources, allowing materials and fuels to be varied. The design of the SIPS engine was based on radiative heat exchanger from the external heat source, and resulted in relatively long hot side tubes to gather the heat from the fins, consequently changing the system to a forced convection or conduction type of exchange would be quite simple. The development of advanced materials for the high temperature areas of a Stirling engine held great promise for the cycle's future. Again the SIPS engine provides an excellent tool to allow techniques to be developed which permit materials such as ceramics to be implemented.

In summary, the SIPS engine is an available tool which lends itself extremely well to laboratory use for enhancing the development of the Stirling cycle. Minor refurbishment requirements requiring only a short schedule time and minimal cost would allow expedient use without the expense of developing and fabricating an engine designed specifically for these tests.

PUBLICATIONS/REFERENCES


- Wiley, R., Lehrfeld, D., "Development of a 1 Kw(e) Isotope Fueled Stirling Cycle Power System," Proceedings of the 13th IECEC, San Diego, California, August 1978

- Bledsoe, J. A., Lehrfeld, D., "Stirling Power System Development and Test Results," Proceedings of the 25th American Nuclear Society, Atlanta, Georgia 1979

- Lehrfeld, D., Sereny, A., Bledsoe, J., "Predicted Performance and Testing of a Preprototype, Small, Stirling Engine Generator," Proceedings of the 14th IECEC, Boston, Massachusetts, August 1979

- Richards, W. D. C., Lehrfeld, D., "Stirling Engine Power System Development and Test Results," Proceddings of the 15 IECEC, Seattle, Washington, August 1980

LIGHT COMMERCIAL BRAYTON/RANKINE
SPACE CONDITIONING SYSTEM

AIRESEARCH MANUFACTURING COMPANY
2525 W. 190TH STREET, TORRANCE, CALIFORNIA 90509

CONTRACT NUMBER 86X-24706C

DAVID FRIEDMAN, KENNETH TRESTER, WILEY COWELL

PERIOD OF CONTRACT:
BEGIN MARCH 1979 END SEPTEMBER 1982

SUMMARY

A heat-actuated space conditioning system that provides more efficient use of natural gas has been developed for 26.4- to 87.8-kW (7.5- to 25-ton) commercial applications. The system consists of a subatmospheric, natural-gas-fired, Brayton-cycle engine that drives a Rankine-cycle heat pump. A centrifugal Freon compressor is driven directly from the Brayton engine rotating group through a permanent-magnet coupling. Unique features that offer high life-cycle performance include a hermetically sealed magnetic coupling, air foil bearings, an atmospheric in-line combuster, and a high-temperature recuperator. Predicted overall engine efficiency is 27 percent and predicted overall coefficient of performance (COP) at the energy source is 1.0 in cooling and 1.2 in heating.

INTRODUCTION

The increasing energy shortage has resulted in new developments in energy-saving devices. In the space conditioning field, heat pumps driven by off-site electric motors have been successfully marketed as energy-saving systems. However, on-site, gas-powered generation systems can significantly improve upon the performance ratings of these existing systems. As shown in Figure 1, with a gas-powered engine directly driving the heat pump, the heat loss from the power cycle is not wasted. The heat can be recovered and put directly into the building. With the electric-motor-driven system, however, 71 percent of the energy required never reaches the building because it is lost at the power plant and during transmission.

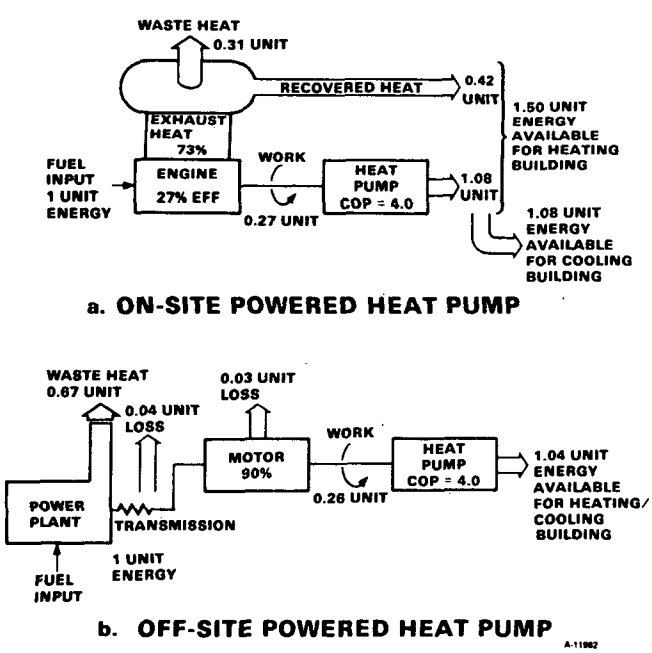



Figure 1. Off-Site vs On-Site Heat Pump Cycles

AiResearch Manufacturing Company, a division of The Garrett Corporation, and Dunham-Bush have been investigating the critical technology for a gas-fired Brayton/ Rankine 35-kW (10-ton) space conditioning system (Garrett and Dunham-Bush are members of The Signal Companies). This program, first sponsored by the American Gas Association, is now sponsored by the Gas Research Institute and the Department of Energy. The system offers high performance potential by utilizing the most advantageous working fluid in each cycle: air is used in the heat engine where high turbine inlet temperatures provide high thermal efficiency, and Freon 12 is used in the heat pump because of its desirable thermodynamic characteristics. The total system is slightly larger than a 35-kW (10-ton) electric rooftop heat pump and measures 1.83 m high by 1.68 m wide by 2.29 m long (6 ft by 5.5 ft by 7.5 ft). Figure 2 shows a scale model of the space conditioning system.

The principal technical achievements during this program have been the development of (1) the Brayton/ Rankine rotating group, including a magnetic coupling that provides efficient torque transmission at the high speeds associated with the gas turbine engine, and (2) an engine cycle that provides high thermal efficiency while burning natural gas at low pressure.

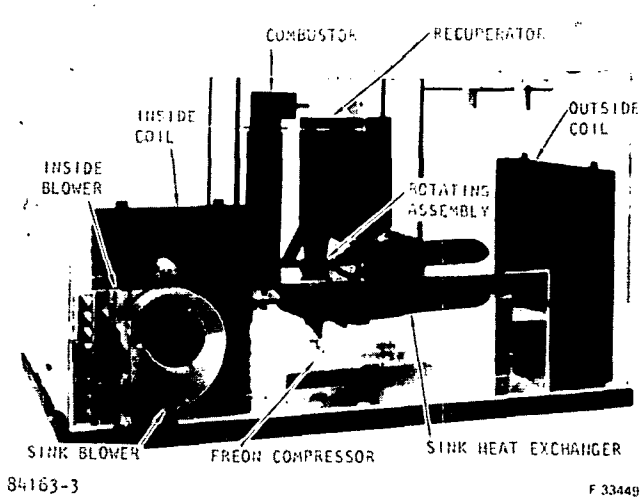



Figure 2. Model of Brayton/Rankine Space Conditioning System

## CYCLE DESCRIPTION

The basic system, shown in Figure 3, consists of two thermodynamic loops. The first is a conventional vapor compression heat pump system that uses a high-speed centrifugal vapor compressor instead of the usual positive displacement type. The second loop is the Brayton-cycle engine, which runs the centrifugal compressor and supplies waste heat energy to the building during the heating mode of operation.

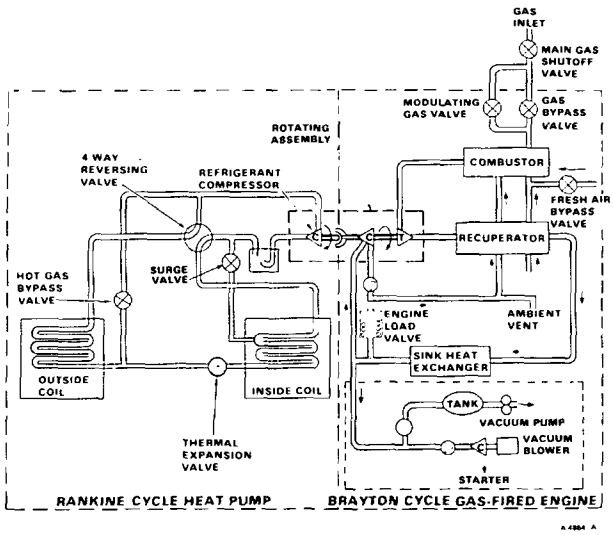



Figure 3. Basic Brayton/Rankine System Concept

The actual Brayton cycle is a semi-open subatmospheric-pressure cycle consisting of a centrifugal compressor, a radial inflow turbine, a recuperator, a sink heat exchanger, and an in-line atmospheric combustor. Ambient air is drawn through the recuperator, where it is preheated before being introduced into the atmospheric-pressure, natural-gas-fired combustor in stoichiometric proportion. Compressor discharge gas is also cycled through the recuperator and used as diluent to provide added flow and the desired turbine inlet temperature. Expansion takes place through the turbine component, from which sufficient power is extracted to drive both the Brayton and Freon compressors. The turbine discharge, which is at subatmospheric pressure, is processed through the recuperator, where it preheats combustor inlet air. The temperature of the low-pressure gas is further reduced by using the sink heat exchanger to reduce compressor power consumption. The compressor pumps the gas back to atmospheric pressure and a small portion is exhausted; the remainder is recycled as diluent. The sink heat exchanger cooling air, which now has a temperature greater than 37.8°C (100°F) is combined with building recirculation air in the heating mode, supplementing the vapor cycle unit heating capacity.

## SYSTEM PERFORMANCE PREDICTIONS

Figure 4 presents engine performance characteristics for a 35°C (95°F) ambient day. Cycle efficiency and output shaft power are plotted as functions of speed for varying turbine inlet temperatures. At the design point of 75,000 rpm and turbine inlet temperature of 843°C (1550°F), the predicted cycle efficiency is nearly 30 percent and the delivered power is 7.65 kW (10.2 hp), which is sufficient to drive a 35-kW (10-ton) refrigeration system that provides 35 kW of cooling for the 35°C (95°F) ambient day.

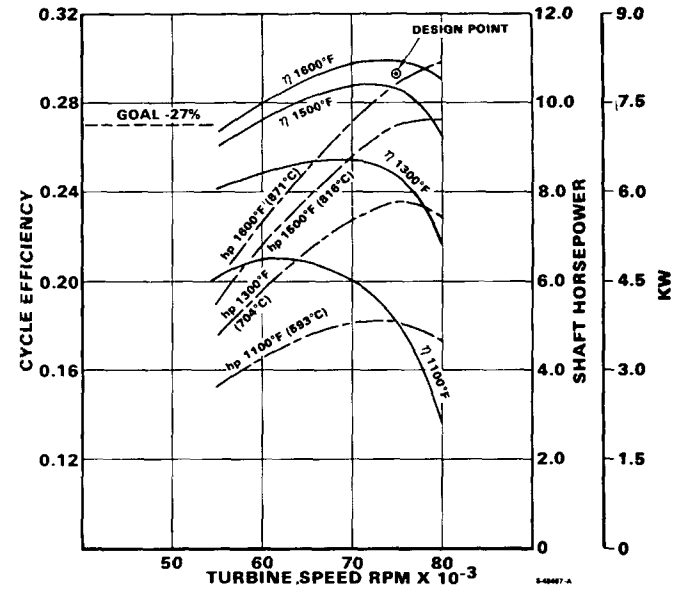



Figure 4. Brayton Engine Performance

The combined performance characteristics for the heat pump system for a 35°C ambient day cooling mode of operation show a system COP of 1.3. (With all parasitic losses included, the overall COP is reduced to 1.0.) This is equivalent to an electrically driven heat pump energy efficiency ratio (EER) of 15.3 (11.8 after inclusion of parasitic losses). A similar performance curve was generated for an 8.3°C (47°F) ambient heating day. With the system operating to match predicted heat load--turbine inlet temperature at 621°C (1150°F)--the system COP is 1.5 (1.2 with all parasitic losses included). This is equivalent to a heating COP of 5.17 for an electrically driven heat pump system (4.14 after inclusion of parasitic losses). In Figure 5 the Brayton/Rankine system COP is plotted as a function of ambient temperature and compared against existing equipment. As shown, Brayton/Rankine system performance surpasses that of existing conventional systems at all conditions.

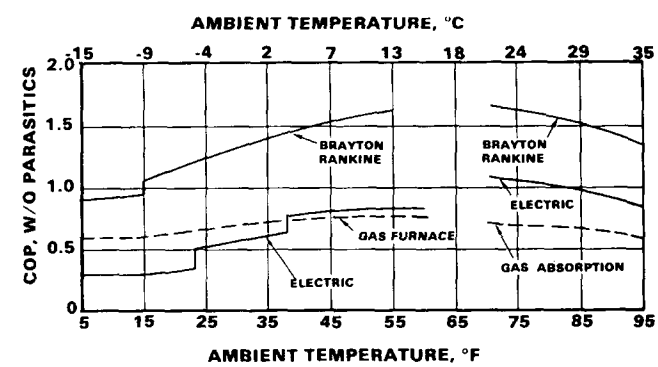



Figure 5. Comparative Performance

## ENERGY SAVINGS AND PAYBACK COST ANALYSIS

An energy consumption and energy cost analysis was made to compare the Brayton/Rankine heat pump system with a variety of competitive year-round space conditioning systems. The following competitive systems were used in the analysis:

(a) Present conventional electric heat pump (HP) with EER of approximately 8.0 at 35°C (95°F)

(b) Present conventional year-round air conditioning (YAC) with gas heating furnace efficiency of 60 percent overall and electric cooling EER of 8.0 at 95 percent

(c) Future all-electric heat pump with predicted EER of 11.5 at 35°C (95°F)

(d) Future year-round air conditioner incorporating pulse-combustion gas furnace with overall efficiency of 90 percent, and electric cooling EER of 11.5 at 35°C (95°F)

(e) Future two-speed heat pump with high efficiency at maximum design condition, while providing lower cyclic losses at off-design conditions

Three cities were chosen to represent climatic conditions for three different areas of the United States: Canton, Ohio; Birmingham, Alabama; and Phoenix, Arizona. Air Force weather data were used for each city to establish the time for each 2.9°C (5°F) temperature bin. A representative load line for a commercial 35-kW (10-ton) system was generated for each city, and a computer program was designed to determine the energy consumption for each system. Included within the analysis were degradation factors for on-off cycling losses, where appropriate, and for defrost. The degradation factors for cyclic losses were utilized for heat pumps, air conditioners, and the Brayton/Rankine system.[1] The pulse-combustion furnace and standard furnace were assumed to have 90-percent and 60-percent efficiency, respectively, at all conditions, with no degradation factors.

Figure 6 shows the results of the computer energy consumption analysis for the three cities and each of the five systems analyzed. The figure shows the annual seasonal energy efficiency ratios (SEER's) for each system. To compare gas systems with electrical systems, the gas source energy numbers were divided by a conversion factor of 0.29 to arrive at a comparable SEER. The data include all parasitic losses for each system, including electrical fans for indoor and outdoor coils, furnaces, and sink heat exchangers. As shown, the Brayton/Rankine system performance exceeds that of an advanced year-round conditioning system or a heat pump, even with those systems showing a performance improvement of approximately 40 percent over their current counterparts in the marketplace.

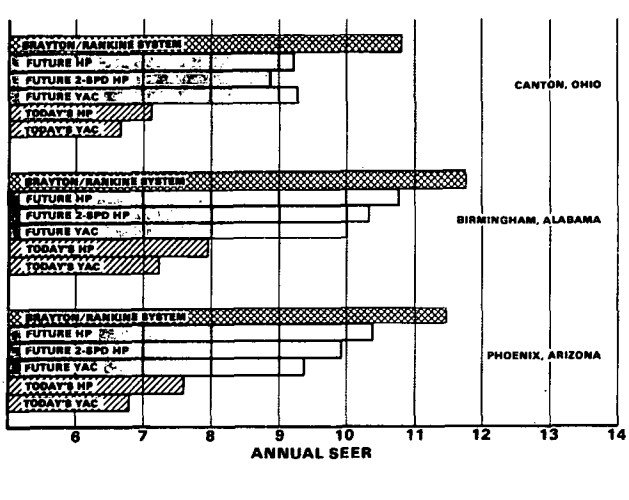



Figure 6. Annual Seasonal Energy Efficiency Ratios

Figure 7 shows the energy savings converted to dollar savings, with each system being compared with the Brayton/Rankine space conditioning system. The figure reflects three different energy rates for both gas and electricity for each of the three cities analyzed. As shown by the figure, annual savings over future YAC's or HP's are greater than $600, with nominal energy costs of $4 per million Btu for gas and 7 cents per kW-hr for electricity. With high energy costs, the savings against today's systems could be as great as $2400 per year.

Figure 8 shows the total cost savings to the user over the expected life of the equipment. The Brayton/Rankine system is compared with present and future YAC's for the Canton, Ohio, test case. Energy rates of $4 per million Btu for gas and 7 cents per kW-hr for electricity were assumed. The equipment cost

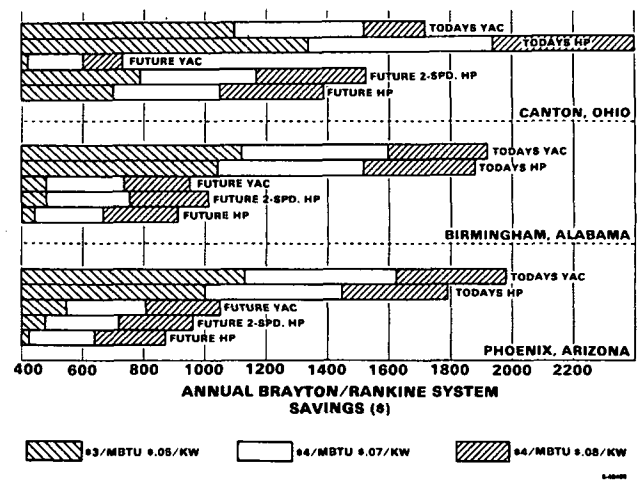



Figure 7. Comparative Annual Savings

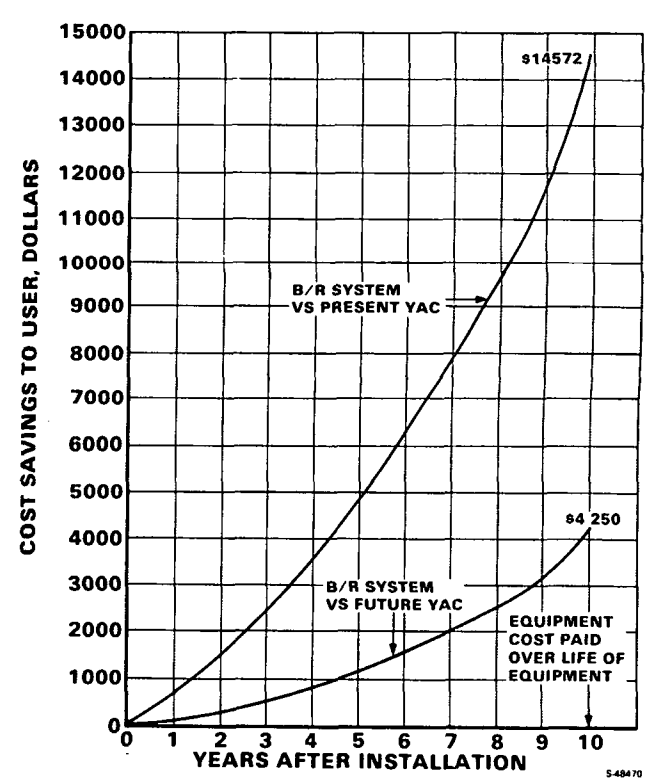



Figure 8. Cost Savings to User

was included in the analysis and was assumed to be financed for 10 years at a 10-percent interest rate. Energy costs were compounded annually at 8 percent. Installation costs and maintenance costs were assumed to be the same for all systems. The equipment was assumed to have no salvage value at the end of 10 years, and depreciation was ignored in the analysis.

Estimated equipment costs were based on expected 1984 prices. The future YAC cost was assumed to be 14 percent greater than that of the baseline conventional YAC, while the Brayton/Rankine system was assumed to have a 42.5 percent greater selling price than the baseline conventional YAC.

As shown, even with the higher first cost, the Brayton/Rankine system shows positive cash flow in the first year when compared against either YAC under this type of financing arrangement. The savings to the user over the life of the equipment is shown to be $14,572 when compared with the conventional system, and $4,250 when compared with the future YAC.

It should be noted that the savings would be even greater if the Brayton/ Rankine system were compared with an electric heat pump in the northern climate of Canton, Ohio.

The results of this analysis should encourage prospective purchasers if those purchasers are the actual users of the equipment, rather than contract installers who do not pay the monthly energy bills.

## HARDWARE DEVELOPMENT

Freon compressor and magnetic coupling. The first development effort was the design, fabrication, and test of the Freon-12 compressor and magnetic coupling. The Freon compressor unit shown in Figure 9 is basically very simple; its one moving part comprises a single-stage, shrouded impeller at one end of the shaft, and the driven magnet cartridge at the other end. The shaft is supported on two standard, precision ball bearings. The compressor wheel is made of aluminum alloy. A stationary cover seal encloses the driven magnet and forms a part of the Freon compressor unit. The magnets are samarium-cobalt. The primary requirement of the magnetic coupling is to run at rated speed without slipping.

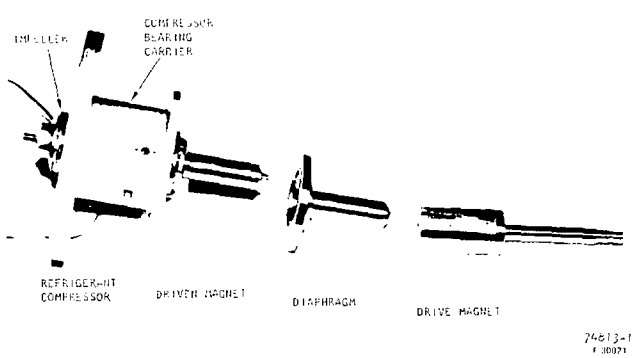



Figure 9. Refrigerant Compressor and Magnetic Coupling

Brayton rotating group. The last phase in the program was the development and test of the Brayton rotating group shown in Figures 10 and 11. The Brayton rotating assembly is connected to the Freon compressor via the magnetic coupling. The assembly consists of a cast, single-stage, radial turbine and a single-stage, radial compressor; these are bolted back-to-back to the shaft to form an integrated rotating unit. The shaft is supported on long-life,

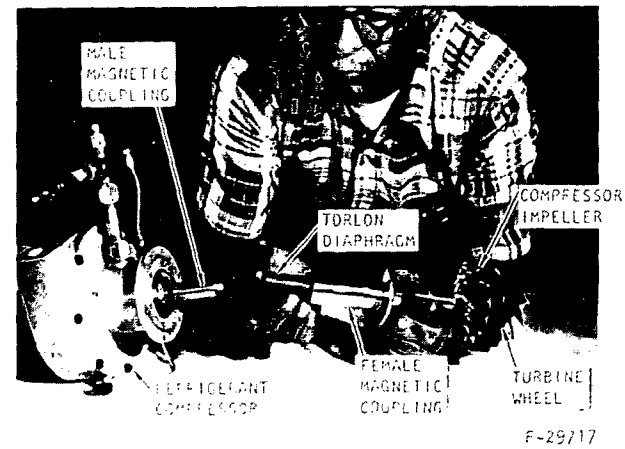



Figure 10. Brayton/Rankine Rotating Group Details

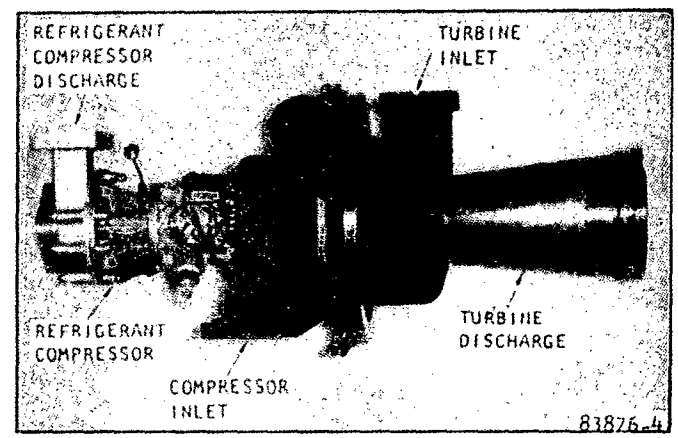



Figure 11. Assembled Brayton/Rankine Rotating Group

maintenance-free, foil air bearings. The foil bearings use the loop air as both coolant and lubricant and are entirely self-acting.

The turbine wheel is of high-temperature cast alloy, and the compressor wheel is cast steel. Cast turbine and compressor scrolls, a turbine nozzle, a compressor diffuser, a thermal shield, and a compressor inlet housing complete the drive design.

The design arrangement draws cooling air over the bearings into the compressor inlet housing. A labyrinth seal keeps the bearings essentially at ambient atmospheric pressure. A thermal shield between the turbine and compressor wheels, plus other thermal design features, minimize thermal leakage.

The unit shown in Figures 10 and 11 is compact, having overall dimensions of 0.74 m (29 in.) in length, including the attached Freon compressor, and 0.36 m (14 in.) in diameter at its widest point.

Recuperator. The high-temperature recuperator is constructed from a formed tube sheet of nitride-dispersion-strengthened, 400-series, stainless steel. Its counterflow configuration provides an effectiveness of 0.90.

The core consists of alternate layers of gas and air fins. The development unit is shown in Figure 12. Air enters a plenum and is distributed by turning fins into the central counterflow section of the core, where most of the recuperation takes place. After leaving the counterflow section, air flows through

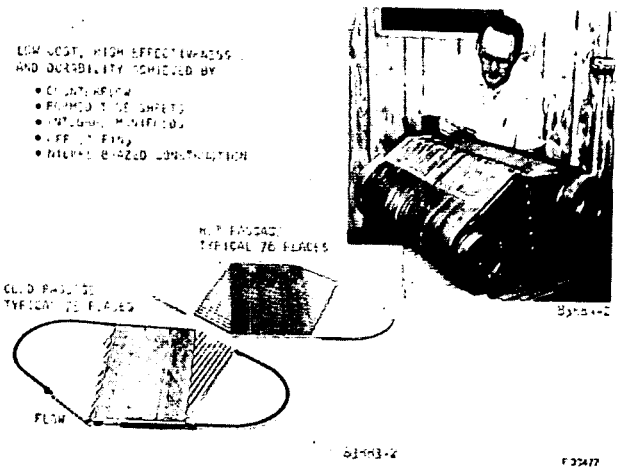


Figure 12. Recuperator Core Construction

additional turning fins and out the return air plenums. Gas enters at the return air end and passes through the core in a counterflow direction. High-performance rectangular strip fins are used in the counterflow section in both the air and gas passages.

Both the air and gas fins are brazed to the tube sheets with a high-temperature, nickel-base braze alloy. Brazing maximizes heat transfer and makes the entire core an integral structural assembly capable of withstanding high pressures and temperatures without external support. This type of design is used by AiResearch in its large industrial regenerators.

Combustor. The use of a subatmospheric gas turbine leads to a distinct advantage in design of the combustor, which is a low-cost, in-line, atmospheric design that utilizes natural gas at only a few inches of water pressure. The in-line design, in which all combustion pro ducts are introduced into the flow stream, results in a very efficient combustor that provides 100-percent combustion efficiency with only 10-percent excess air above stoichiometric proportion. The diluent and quenching gasses are recirculated combustion products that provide a reasonably low combustion chamber temperature. The unit is a cylinder having a diameter of approximately 0.18 m (7 in.) and a length of approximately 0.75 m (30 in.).

## PROGRAM STATUS

Development. The development program schedule shown in Figure 13 has proceeded basically as depicted, with most of the major technical challenges already met. It is expected that the development effort will be completed in early 1981, and a full demonstration prototype system will be fabricated and assembled, ready for test, by mid-1981.

The introduction of this system into the marketplace is still a substantial effort. A demonstration unit is yet to be fabricated; the field test program, scheduled for 1982, is still to be accomplished; and the economics of the approach has yet to proceed past the preliminary analysis stage. Assuming these tasks are completed successfully, full production is anticipated in the 1984 time frame.

Program Expansion. The program is being expanded to allow a second heat pump manufacturer to participate in the development effort, and also to survey the international market. In this regard, assistance in development efforts is being negotiated with foreign gas utilities.

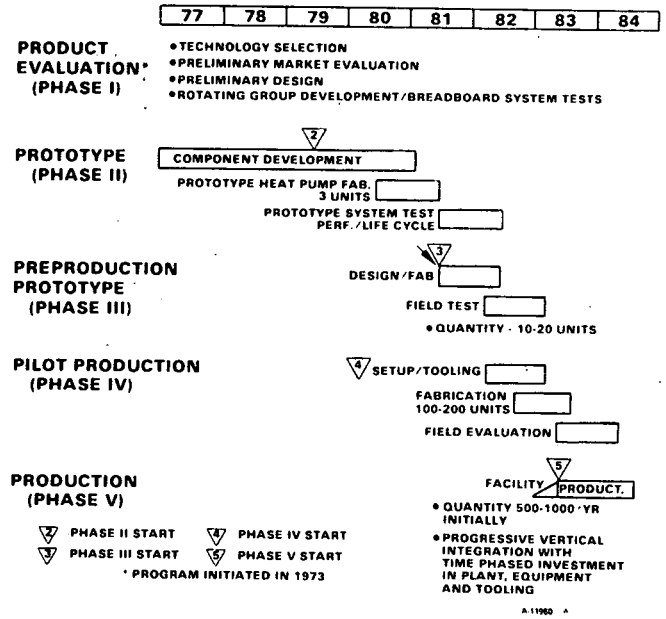



Figure 13. Brayton/Rankine Conditioning System Master Program Schedule

AiResearch is also investigating the application of common hardware for use in power generator sets, including solar power generation. To this end, a study contract has been awarded to AiResearch by GRI to investigate the application of this hardware for use in a solar-assisted, gas-fired heat pump. It is planned to utilize support hardware similar to that needed for a solar-dispersed power generating system.

Figure 14 shows the overall arrangement of the solar-assisted, gas-fired heat pump. Basically, the system would incorporate the subatmospheric Brayton engine and the 10-ton Rankine heat pump under development. The system will require repackaging for integration with a solar parabolic concentrator and a thermal receiver. The Brayton engine will be mounted at the focal point of the concentrator behind the receiver, and the heat pump will be on the ground. A simplified system schematic is shown in Figure 15.

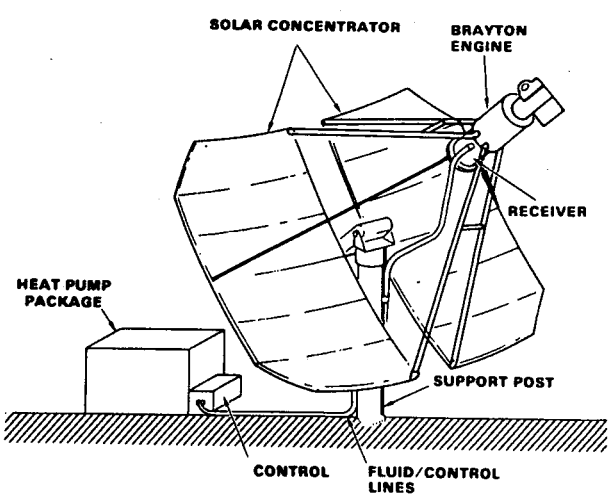



Figure 14. Overall Arrangement of Solar-Assisted, Gas-Fired Heat Pump System

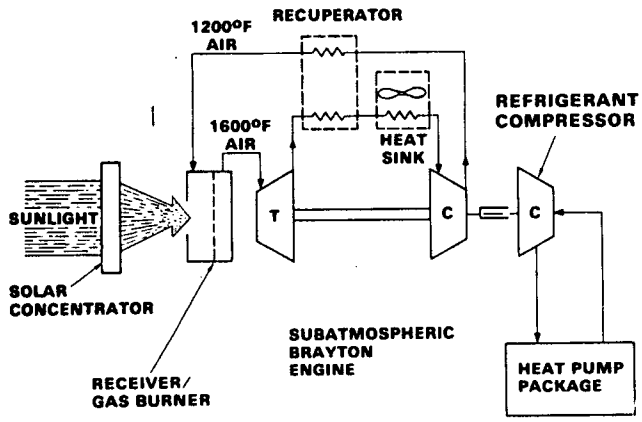



Figure 15. Schematic Diagram of Solar-Assisted, Gas-Fired Heat Pump System

Solar equipment used in the system has the highest efficiency of any such hardware now under development. The high temperatures attainable from the concentrator/ receiver correspond to high potential power system efficiencies, which will result in minimum collector area requirements and minimum installation cost.

Reference:

[1]Didion, D.A., and G.E. Kelly, "New Testing and Rating Procedures for Seasonal Performance of Heat Pumps," ASHRAE Journal, September 1979, Vol. 21, No. 9, pp. 40-44.

# Development and Demonstration of a High COP Rankine Heat Pump for Generation of Process Steam Combined Cycle Steam Compression Industrial Heat Pump

William Koebbeman
Mechanical Technology, Inc.

PAPER NOT AVAILABLE

SOLAR HEATING AND COOLING SYSTEMS

HONEYWELL, INC.
ROBERT BUCHHOLZ AND GEORGE MARSCHALL

PERIOD OF CONTRACT: JULY 1976 TO SEPTEMBER 1981

## ABSTRACT

Solar-assisted, 3-ton residential and 25-ton multifamily residential and commercial heating and cooling systems have been installed at 10 different sites in 9 states. Eight sites were instrumented to acquire data for system performance analysis. The systems provide solar-assisted heating, cooling, DHW preheat, and electrical power generation utilizing a Rankine cycle turbine, powered by liquid-cooled, flat-plate collectors to drive a motor/generator inline with the Freon compressor. This paper gives a brief description of the solar system, site installation, and site performance.

FULL TEXT NOT AVAILABLE

DEVELOPMENT OF A SOLAR-POWERED RANKINE CYCLE HEAT PUMP

RESEARCH CENTER AND HAMILTON STANDARD DIVISION OF UNITED TECHNOLOGIES

UTRC: F. R. BIANCARDI, G. MELIKIAN; HSD: J. W. SITLER

DE-AC03-77CS34510

## OBJECTIVE

United Technologies is currently designing, building and conducting laboratory and field tests of unique solar-powered heat pumps and solar cooling systems sized for multifamily residential and light commercial applications. The heat pumps and solar chillers are specifically designed to operate at moderate peak temperatures and to permit efficient air cooling. The basic design data has been developed under prior NSF/ERDA, and DOE-sponsored programs in which UTRC demonstrated the operational feasibility and performance advantages of Rankine-cycle solar heating and cooling systems.

## TURBOCOMPRESSOR CONCEPT

The UTC heat pump (HP) system, shown in Fig. 1, incorporates a Rankine-cycle power loop in which a centrifugal turbine is used to drive a centrifugal compressor in a vapor compression refrigeration loop. The thermal energy to the power loop is provided at temperatures up to 300 F (149 C) by a medium-concentration solar collector array. Auxiliary energy for cooling is provided by a fossil-fuel-fired furnace. Heating can be provided by direct solar, direct furnace, or furnace-driven heat pumping (Fig. 1) of low-temperature solar energy in combination with power loop heat recovery. The HP is rated at 18 tons for cooling and approximately 500,000 Btu/hr for heating. It utilizes low-maintenance air-cooled condensers for heat rejection and conventional HVAC design heat exchangers and controls. Refrigerant 11, a common fluorocarbon is used as the working fluid in both the power and cooling loops.

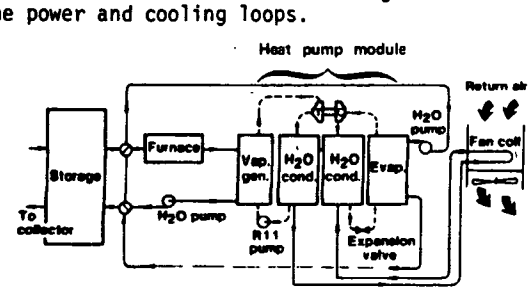



**Fig. 1 Heat pump system in heat pump mode.**

## DESCRIPTION OF WORK AND SUMMARY

The first prototype 18-ton solar-powered turbocompressor heat pump module has been successfully designed, built and tested for more than 250 hr. in a specially-designed laboratory facility at UTRC (see Fig. 2). Operation in both the cooling and heat pump mode was demonstrated over a wide range of building, climatic, and collector/storage conditions. The design point performance of the heat pump in both the cooling and heat pump modes has been confirmed, and performance mapping of the module completed. The heat pump demonstrated the wide operating range possible (using 200 to 300 F hot water) and high heat pump mode performance levels, such as a COP of 1.4 to 2.5 and 500,000 Btu/hr capacity. In cooling, a COP of 0.5 to 0.75 and up to 20 tons was demonstrated. In a simulation of operation in an actual building, the heat pump smoothly and accurately followed the building load for a full day.

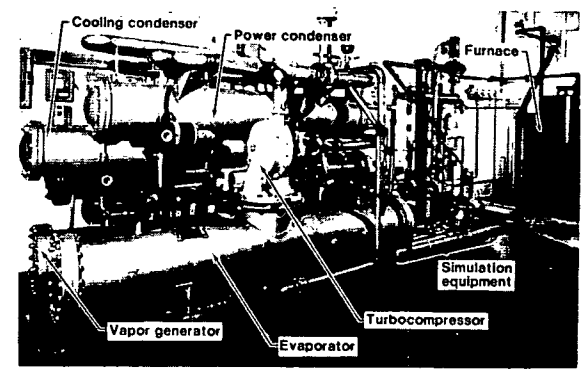



**Fig. 2 Heat pump module in UTRC simulation laboratory.**

A detailed assessment of the individual module components was completed and performance, cost and reliability improvements were identified. No evidence of R11 decomposition or component wear or corrosion was found.

Detailed seasonal performance and economic analyses were conducted for the UTC solar HP in various geographical areas and indicate significant potential energy and cost savings with the concept. A brief summary of the results are presented herein and summarized in published UTRC reports.

The first UTRC heat pump module was modified and installed in a complete UTRC-designed (SOLERAS) solar cooling system for field tests in a Phoenix, AZ office building. The system installation and initial test results are briefly described herein.

A second, advanced, higher performance heat pump module has been designed and is currently being fabricated at UTRC. This heat pump incorporates numerous operating and reliability improvements identified during the laboratory and field tests. Still higher performance configurations have been designed and are under evaluation.

## TECHNICAL ACCOMPLISHMENTS

- Performance Mapping. A primary objective of the 1980 efforts was to conduct extended testing of the unit to establish performance over the entire operating range in both the cooling and heat pump modes. Utilizing the automatic data acquisition facility in the UTRC heat pump laboratory, data was recorded for over 2000 data slices taken at 400 sets of water inlet conditions, simulating various collector/storage tank output, ambient air and building load conditions. The mapping

was conducted over vapor generator water inlet temperatures from 200 F to 300 F, evaporator temperature from 55 F to 125 F, and condenser inlet temperatures from 80 F to 125 F.

Figure 3 shows the measured heat pump mode performance and the extremely wide range of output capacities provided by the heat pump. Good agreement between test and predicted values of performance and capacity was found for both the cooling and heat pump operating modes.

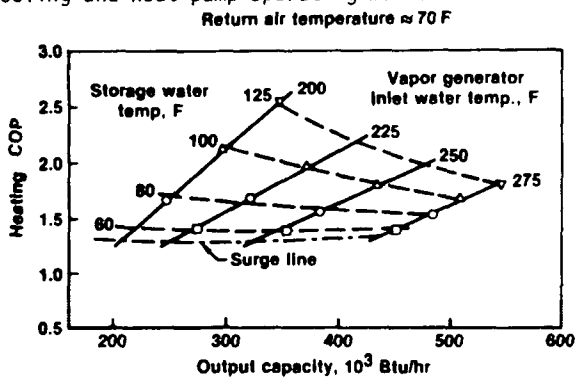



*Fig. 3 Measured heat pump mode performance.*

● FULL-DAY SIMULATIONS. The heat pump was also run over a range of conditions to simulate operation in an actual building. The boundary conditions for operating the system in the solar cooling mode for an office building located in New York were provided by the UTRC-modified TRNSYS computer program. The hot water storage tank temperature was varied from 275 F to 220 F representing the continual extraction of energy from hot storage. Ambient temperature for the day followed a typical pattern with a night low of approximately 75 F and a mid-day peak of 89 F. Typical results of the full-day simulation tests are shown in Fig. 4 where the solar cooling system output profile is compared with the TRNSYS predicted building loads. Cooling produced ranged from 4 to 17 tons and a peak COP of of 0.75 was attained. During morning operation at lower ambient temperatures, the module output exceeded the instantaneous demand. Therefore, solar cooling was turned off as room temperature was allowed to increase into the allowable range. The integrated water-side cooling output was 96 percent of that required by the building. The pertinent results are summarized below.

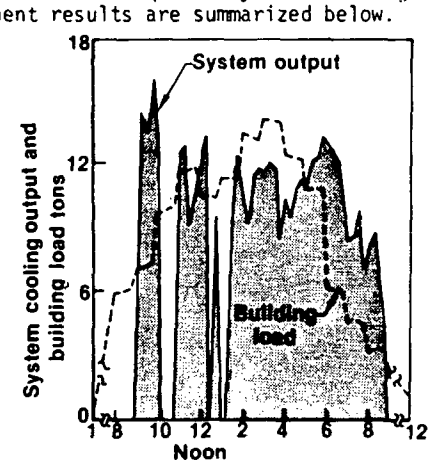



**Fig. 4 Typical full day system operation.**

● ENDURANCE TESTING AND COMPONENT ASSESSMENTS. Sufficient instrumentation was incorporated in the heat pump components so that detailed assessments of component performance, wear, and durability could be made continuously. In addition, the major module components, i.e., heat exchangers, turbocompressor and valving were disassembled and inspected after the endurance testing was completed. After 250 hrs of testing, the turbocompressor condition was essentially as "new" (Fig. 5). All of the major components were found to be in extremely good condition, except for the water-cooled condensers which experienced water-side fouling due to the use of untreated city-water. Chemical tests and visual inspection of the R11 working fluid showed no sign of decomposition even though much of the testing was at elevated temperatures (280-290 F). This can be attributed to careful material selection and elimination of oil from the power loop. The data gathered during this testing was used as input for the design and fabrication of the advanced heat pump module.

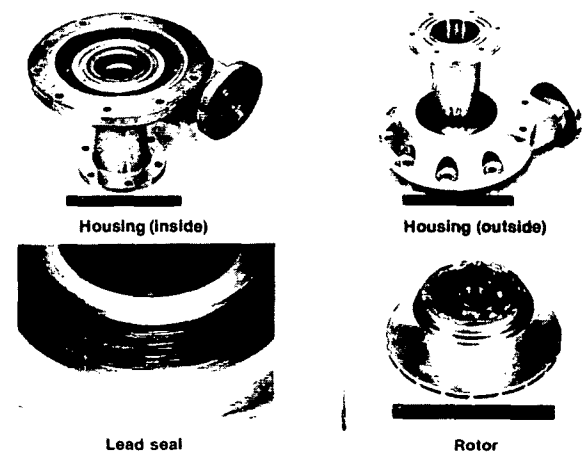



**Fig. 5 Turbine elements after testing.**

● SEASONAL PERFORMANCE ESTIMATES. The annual performance of the UTC solar heating and cooling system installations has been estimated by means of an analytical simulation procedure which includes performance models for each of the major components and subsystems, the thermal load characteristics of typical buildings such as multi-family residences or light commercial office buildings and meteorological data for specific U.S. locations.

The UTC heat pump installation performance analyses were conducted with hourly weather data. The National Bureau of Standards, under DOE sponsorship, has generated consistent weather and insolation data (SOLMET) for numerous U.S. locations for use in evaluating solar heating and cooling systems. SOLMET data tapes have been selected for typical years at locations which represent a wide variety of heating/cooling loads, geographic areas and solar insolation levels.

Building heat loss/gain factors depend on the size, construction, and use of the building as well as its geographical location and orientation. For the studies, building models were established to simulate the heating and cooling loads of energy efficient (as per ASHRAE Standard 90-75) multi-family residential buildings.

The design point heating load was designed on the basis of coincident wind velocity and the 97.5 percent dry bulb temperature for a heating season. The design point cooling load was designed on the basis of a 2.5 percent temperature level for a cooling season. Based on these design point procedures, the modeled buildings require a nominal 15- and 20-ton capacity cooling system if located in a region with climate typical of northeastern U.S.

The UTC-modified TRNSYS simulation computer program was used to determine the effects of collector characteristics (type, configuration, surface area, tilt angle, etc.), storage tank characteristics (capacity, segments, etc.) and system operating strategies on seasonal performance of the UTC solar-powered heat pump installations in various geographical locations. These results include solar contribution (energy supplied by solar collector/total thermal energy supplied to HP system), parasitic power consumption and time in each operating mode as well as time profiles of all system and building conditions (temperatures, heating/cooling rates, humidity, etc.). A sampling of these results are presented herein. A typical annual energy distribution profile for a New York location is shown in Fig. 6. These results show a substantial portion of the winter heating load is delivered by solar-assisted heating pumping and essentially all of the summer cooling load is delivered by the HP in the solar-powered cooling mode. These computer generated plots are for building simulation calculations made every 15 minutes for an entire year; however, such detailed results are usually unnecessary for most performance and economic evaluation studies.

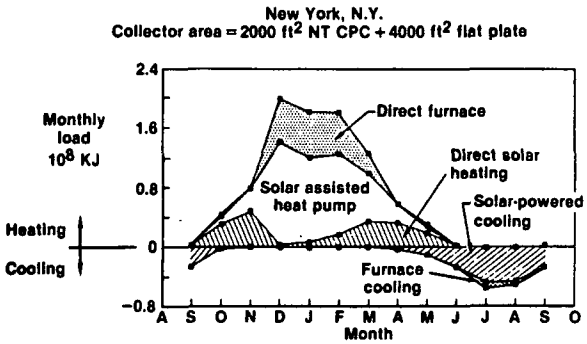



Fig. 6 Distribution of energy delivered.

Detailed annualized performance characteristics were calculated for alternative UTC heat pump installation configurations covering a range of collector and storage tank types and sizes. Although their average thermal COP's were similar, the optimum installations had significantly higher solar contributions and lower parasitic power consumption and operating time. A summary of the performance characteristics for an optimum installation in various cities is given in Table 1. These performance simulations indicate a significant portion of the selected buildings heating and cooling energy needs can be supplied by solar energy, thereby reducing its annual fuel and electricity requirements. For example, the UTC solar-powered HP installation can provide a seasonal solar contribution (SSC) of 50 to 60 percent with 2 to 4000 $ft^2$ of collector. In addition, the heat pump average thermal COP in heating is generally 1.90 or above.

When used in cooling dominated areas such as Arizona and Texas, the UTC system can provide a substantial fraction of the building cooling energy needs with a minimum amount of collector. For example, in Phoenix, a system utilizing 1000 sq ft of advanced collector would result in an estimated 46 percent seasonal solar contribution, while 2000 sq ft of collector would increase the solar contribution to more than 80 percent.

Detailed building indoor conditions for all six (6) regions evaluated, indicate that standard comfort conditions can be met with the UTC solar installation in addition to providing significant energy savings.

● ECONOMIC ANALYSIS. A detailed evaluation of the economic potential of the UTC system has been conducted for a UTC solar HP installation in New York and other cities.

Life-cycle costing was used to evaluate the economic performance of the UTC solar-powered systems relative to conventional HVAC systems. The conventional HVAC system includes an electric air conditioner with an overall seasonal COP of 2.3 and a fossil-fueled furnace with a seasonal efficiency of 80 percent. The baseline general economic assumptions used in these evaluations are presented in various UTC reports. Solar systems have a 20 percent investment tax credit and a conventional HVAC 10 percent. Both systems were assumed to have a 20-yr life and 10-yr depreciation period.

After-tax discounted rate-of-return, and payback period were calculated for the systems. After-tax discounted rate-of-return values greater than 10 to 12 percent are necessary for cost effective building HVAC investments. Payback period, which represents the years required to pay back an initial investment difference with future cost-saving differences (not discounted) between two alternative investments, should be less than 10 years for a cost effective investment. These economic indicators were evaluated with respect to changes in the cost of fuel, collector cost and area, and heat pump module cost. The results of these evaluations identified the hybrid collector configuration with a total of 4000 $ft^2$ as the most cost effective installation.

Selected results for this configuration are shown in Fig. 7. Rate-of-return is shown as a function of the heat pump module cost for a range of collector costs and a fuel cost of $10.50/10^6$ of oil. The Fig. 7 results indicate that the UTC system is cost effective at HP

| Location | Collector Type | Collector Area | Solar Energy Collected $10^6$ Btu/yr | Average Thermal COP: HP Heating | Solar Cooling | Furnace Cooling | Solar Contribution, %: Heating | Cooling | Total |
|---|---|---|---|---|---|---|---|---|---|
| New York, NY | NTCPC/FP | 2000/4000 | 610 | 1.94 | 0.85 | 0.66 | 48.6 | 92.4 | 56.3 |
| New York, NY | NTCPC | 4000 | 594 | 1.92 | 0.76 | 0.65 | 40.1 | 97.1 | 52.1 |
| Phoenix, AZ | ACPC | 2000 | 729 | - | 0.82 | 0.62 | 100 | 76.7 | 81.0 |
| Fort Worth, TX | NTCPC | 2000 | 471 | 2.02 | 0.77 | 0.61 | 49.1 | 56.3 | 53.3 |
| Santa Maria, CA | NTCPC | 2000 | 626 | 2.15 | 0.71 | 0.62 | 70.2 | 73.2 | 70.6 |
| Madison, WI | NTCPC | 4000 | 719 | 1.93 | 0.82 | - | 30.5 | 100.0 | 39.7 |
| Apalachicola, FL | NTCPC | 2000 | 420 | 2.10 | 0.79 | 0.62 | 76.9 | 50.5 | 57.2 |

Table 1. Alternative UTC heat pump installation performance

module costs of up to $80,000 at a collector installed cost of $20/ft$^2$, values which should be obtainable with added research and development.

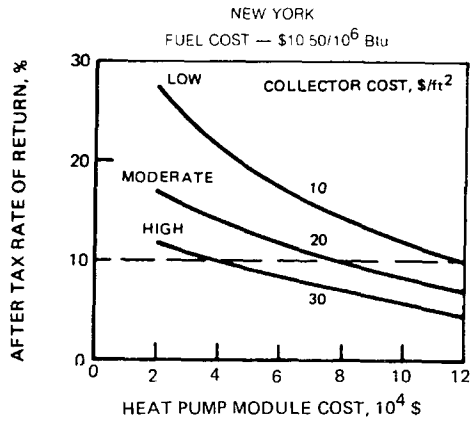



**Fig. 7 Rate-of-return of UTC solar systems compared to a conventional system.**

● SOLERAS SYSTEM. The first UTRC heat pump module was modified for the joint DOE/Saudi Arabian SOLERAS solar cooling field test program, in which SERI is the operating agent. These modifications included replacement of the water-cooled condensers with R11-to-air condensers, addition of a purge unit and unique micropressor control and data acquisition systems.

In the SOLERAS program, the UTRC solar turbocompressor chiller module provides approximately 75 percent of the cooling in the headquarters office building of the Hamilton Test Systems, located in Phoenix, Arizona.

● Design and Analysis. Extensive modeling of the building load, ambient conditions, and system components such as the chiller, collectors, storage tanks and building fan-coils was used in the design of the SOLERAS solar cooling system. Tradeoff analyses (Fig. 7) were used to optimize the collector array size (1316 ft$^2$ net area or 75 ft$^2$/ton of cooling). Similar analyses were used to optimize the cold tank (2000 gal) and hot tank (1500 gal) capacities.

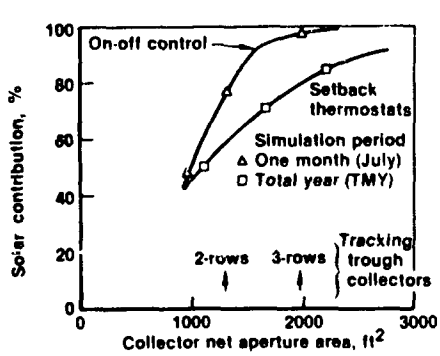



**Fig. 8 Collector area optimization.**

● Fabrication & Installation. The installation of the parabolic trough collectors on a carport at the rear of the HTS headquarters is shown in Fig. 9. The chiller is located in a separate equipment room in the carport and hot and cold tanks are buried. Chilled water is delivered to three ceiling-mounted fan-coils. Installation was initiated in December 1980 and the entire system installation was completed by the end of February, 1981.

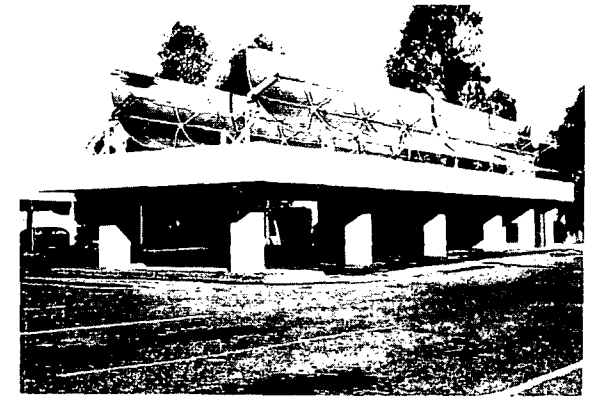


**Fig. 9 UTRC solar cooling installation.**

● Checkout and Acceptance Tests. Highly successful component and system checkout tests were begun in March and the system was delivering up to 20 tons of cooling in April 1981. Typical results obtained during the acceptance tests are shown in Fig. 10 over the course of a day. Early in the day when tank water levels are about 300 F, the UTC system provides about 18 to 21 tons of cooling, about half of which is stored in the cold tank and the remainder delivered to the building fan coils. Although the collector continuously collects about 5 tons (200,000 Btu/hr) of energy, the hot water temperature and thus, chiller output falls off during the day. The system COP has varied from 0.55 to about 0.75.

● ADVANCED HEAT PUMP DESIGN COMPLETED. UTRC and HSD are currently fabricating and assembling an advanced solar-powered heat pump module with higher cooling mode performance capability and lower cost features which were identified as a result of laboratory and field testing. Another unitary heat pump configuration with a cooling mode design point COP of 0.8 is also under evaluation and optimization.

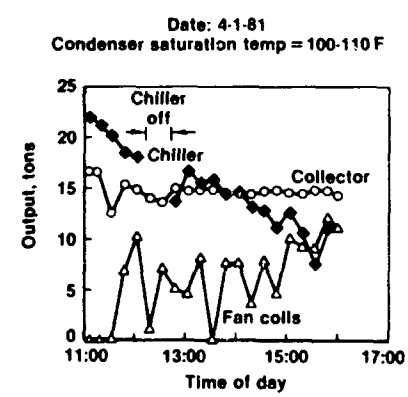



**Fig. 10 Solar cooling system test performance.**

## FUTURE ACTIVITIES

● CONTRACT ACTIVITIES. Module design and assembly of the advanced heat pump is continuing under the current program prior to subsequent laboratory testing in the UTRC facility. Field testing of the SOLERAS system will continue for two and, possibly, three complete cooling seasons.

● POST CONTRACT ACTIVITIES. Additional field test sites and adaptations of the system concept to other sizes and applications are being explored.

## CONCLUSIONS

● The Rankine cycle-powered heat pump and cooling only concept has been conclusively proven as an efficient, reliable approach for thermally driven systems for building applications.

● The predicted performance levels and operating characteristics of the heat pump and components have been verified through laboratory and field tests. Very high performance levels have been demonstrated in both heat pump and cooling modes and even further significant improvements are feasible.

● The UTRC Rankine turbocompressor concept incorporates proven technology and reliability and, therefore, must be considered for numerous thermal drive applications using solar waste heat and/or fossil-fuel fired energy sources.

## ACKNOWLEDGMENTS

Technical effort under this program was provided by Messrs. M. D. Meader, C. E. Kepler, A. M. Landerman, B. W. Rhodes, T. N. Obee, M. D. Krosney, H. E. Khalifa and T. J. Anderson of UTRC. Their assistance is greatly appreciated.

## PUBLICATIONS/REPORTS/REFERENCES

1. Development of Solar Powered 18-ton Rankine Cycle Heat Pump, Proceedings of Annual DOE Active Solar Heating and Cooling Contractors' Review Meeting, March 1980.

Various reports and papers are available. See IECEC, DOE and prior contractors' meetings.

# Heat Sources

# AN INVESTIGATION OF DIRECT EXPANSION IN GROUND SOURCE HEAT PUMPS

E-TECH, INC.
SUBCONTRACT 86X-61612C
MARK D. KALMAN

PERIOD OF CONTRACT: BEGIN SEPT, 1980 END AUG, 1982

## OBJECTIVE

The objective of this project is to perform an experimental investigation and feasibility assessment of direct expansion of refrigerant in an earth heat exchanger as a means for utilizing the ground as a heat source/sink for heat pumps. Particularly we want to demonstrate that the proposed direct expansion concept is technically feasible as a ground source evaporator and a ground sink condenser. The economic viability of the system including the practicality of its installation is being considered. Sufficient thermal performance and design data is being collected to permit evaluation of the energy conservation potential of this technology.

## BACKGROUND

This project is concerned with a new concept for utilizing the ground as a heat source for space heating heat pumps. It is different from other ground source systems currently being studied because it allows direct expansion of the refrigerant in an earth heat exchanger. Most existing ground source heat pumps rely on an intermediate heat transfer fluid such as water or an anti-freeze solution.

The direct expansion concept could offer several advantages over both air source heat pumps and ground source heat pumps using intermediate fluids. Depending on the soil conditions and ambient air temperatures, the direct expansion concept has the following potential advantages compared to the air source heat pump:

- a much more stable evaporator temperature, resulting in higher capacities and efficiencies especially at very low ambient air temperatures.
- elimination of the defrost cycle, thereby offering a higher operating efficiency and a significant increase in reliability.

Compared to a ground source system based on a water or anti-freeze circuit in the ground, the direct expansion concept has the following advantages:

- elimination of a heat exchanger and pump thereby increasing both the efficiency of the system and its reliability.
- a much lower limit on evaporator temperature since freezing of the heat transfer fluid is not an issue.

If the proposed investigation is successful in showing that the direct expansion concept is both technically and economically feasible, then because of the above advantages, heat pumps will become feasible for locations where they currently are not cost effective. It is likely that in most cases, the need for supplemental resistance heating can be eliminated or substantially reduced. This would result in a significant reduction in both energy consumption and required peak power generating capacity.

## SUMMARY

A fully instrumented subscale ground coupled heat pump system has been developed and built. This experimental apparatus has been used to test and obtain data on three different earth heat exchanger configurations under heating conditions (ground cooling). Various refrigerant flow control and compressor protection devices have been tested for their applicability to the direct expansion system. Undisturbed earth temperature data has been acquired at various depths. The problem of oil return at low evaporator temperatures and low refrigerant velocities has been addressed. An analysis has been performed to theoretically determine what evaporator temperature can be expected with an isolated ground pipe configuration with given length, pipe size, soil conditions and constant heat load.

## TECHNICAL ACCOMPLISHMENTS

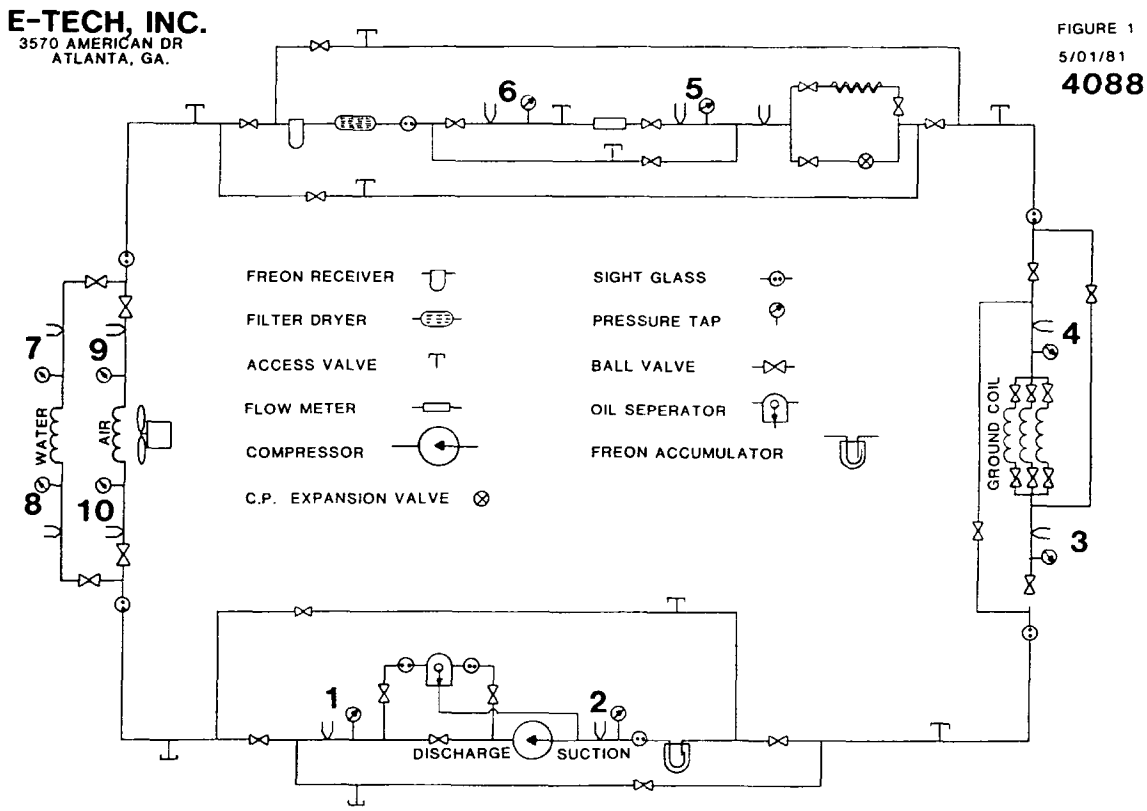
Figure 1 is a schematic of the subscale system. At the points 1-10 temperature and pressure are measured. A turbine flowmeter is used to measure refrigerant flowrate at the saturated or subcooled liquid point. The system is manually reversible so that ground cooling or heating can be performed. An oil separator is used on the discharge to help prevent oil circulation in the system. A suction accumulator is used to prevent liquid refrigerant floodback. Both water and air coils can be employed on the opposite side of the system from the ground coils so that space conditioning and water heating operations can be studied.

The first two ground coil iterations consist of 16 ten foot long 5/8" OD copper tubes in a star configuration from a single header. In the first iteration, freon flows down the 5/8" tubes and up through 3/16" OD tubes concentrically in the center of the 5/8" tubes. Under quasisteady state conditions, the ground evaporator heat input was 4500 BTU/HR with a $0^{o}$ evaporator and $100^{o}$ condenser using R22. The heat input as 3200 BTU/HR with a $0^{o}$ evaporator and $150^{o}$ condenser. This condensing temperature would be required for water heating. The second iteration ground coil is designed to solve the low refrigerant velocity problem experienced in the first iteration. 3/8" instead of 3/16" tubes are used. A copper wire helix is inserted between the 3/8" and 5/8" tubes to increase velocity. With this configuration, the ground evaporator heat input was 8000 BTU/HR with a $10^{o}$ evaporator and $100^{o}$ condenser.

The third and fourth iteration ground coils consist of single 20 foot deep 1-1/8" and 7/8" tubes with 3/4" and 5/8" center tubes respectively. The refrigerant flows down the center tube and up through the annulus between the two tubes. These were installed using a water drill and were much easier to install than the first two iterations. With the 1-1/8" OD tube, the evaporator heat input was approximately 4000 BTU/HR with a $0^{o}$ (flooded) evaporator and $100^{o}$ condenser.

Three methods are being tested to insure oil return and proper lubrication of the compressor. These are:

1. Use of refrigerant mixtures. A 15% R-12, 85% R-22 mixture can be used to insure better oil solubility at low temperatures in an R-22 system without deterring performance.




**DIRECT EXPANSION GROUND COIL SUB-SCALE HEAT PUMP SYSTEM**

2. Use of an oil separator. An oil separator on the compressor discharge will help keep oil from circulating in the system.

3. Flooded evaporator. Since the oil tends to separate from the vapor and not the liquid, a flooded evaporator will help insure oil return. Liquid floodback (slugging) is prevented by using a suction accumulator. A capillary tube is used as an expansion device since it is the only practical way to have control with a flooded evaporator. The capillary tube is sized to give minimal liquid return to the compressor at the lowest expected evaporator temperatures.

An analysis was performed to determine approximately what evaporator temperature can be obtained with an isolated ground pipe configuration with given length, pipe size, soil conditions and constant heat transfer theory and considers the buried pipe as a cylindrical heat source or sink. First, the undisturbed earth temperature at the ground pipe depth is determined. Then a T due to heat transfer to or from the pipe is subtracted or added to the undisturbed earth temperature to obtain the evaporator temperature. This assumes soil - good pipe contact, negligible pipe wall and inside pipe boundary layer resistance, and constant soil properties.

## FUTURE ACTIVITIES

Contract activities. One or more 2-1/2 ton prototype systems are to be developed, built and installed in residences. These will employ ground coil configuration(s) and refrigerant flow control and compressor protection devices that are applicable to the system as determined with the small scale prototype. The cost effectiveness of the concept on a geographic basis will also be studied.

Post contract activities. Assuming success of the full scale system study, and that the concept is deemed cost effective, we will enter into a demonstration phase to establish the performance and cost effectiveness of the direct expansion system.

## REFERENCES

1. Heat Conduction with Engineering and Geological Applications. L.R. Ingersoll, O.J. Zobel, A.C. Ingersoll (McGraw Hill Book Company, Inc., New York, 1948).

2. Earth Heat Exchangers for Ground Source Heat Pumps. M.D. Kalman, Masters Dissertation, Georgia Tech., 1980.

3. ASHRAE Handbook & Product Directory, 1980 Systems. American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., 1980.

# CONDUCT A STATE-OF-THE-ART SURVEY OF EXISTING KNOWLEDGE FOR THE DESIGN OF GROUND-SOURCE HEAT PUMPS

BATTELLE, COLUMBUS LABORATORIES

SUBCONTRACT NO. 7800, LETTER RELEASE NO. X06

DAVID A. BALL, PRINCIPAL INVESTIGATOR

SEPTEMBER 1980 THROUGH MAY 1981

## INTRODUCTION

A ground-coil heat-pump system is defined as a heat pump which either places the outdoor or evaporator coil directly in the ground or where the evaporator exchanges heat with the ground through a secondary fluid (usually either water or antifreeze solution) which is circulated through the ground in a closed loop. This differs from an air-source system which uses outdoor air as a heat source for the evaporator. It also differs from a well-water system which usually involves bringing well water to the evaporator and then either reinjecting the well water or discharging it on the surface; thus, well-water systems can generally be classed as open loop whereas ground-coil systems can be classed as closed loop. Hybrid systems using combination ground coil/air source or ground coil/well water are possible, sometimes desirable, and are currently used in some instances in both Europe and the U.S. today. However, this study concentrated on closed-loop ground-coil heat-pump technology.

There are some apparent advantages to ground-coil heat-pump systems. For instance, the ground exhibits less temperature variation over the year than the ambient air and thus provides for potentially higher heat-pump efficiency than with air-source systems. Also, a ground-coil system does not require defrosting as do air-source systems. Well-water systems offer many of the same advantages but require a good source of well water and either a separate reinjection well or a surface drainage system which is accessible without legal or code restrictions. Thus, ground-coil systems are believed to have wider applicability than well-water systems although there are also some locational restrictions on ground coils because adequate ground moisture must be present to insure good heat transfer between the ground and the buried coil.

Ground-coil systems present a more challenging design problem than either air-source or well-water systems. For instance, the performance of the outdoor coil on an air-source system can be readily predicted once the outdoor air temperature and humidity are known. Well-water system performance is directly related to the performance of the well which can generally be determined at the time of drilling with established well testing techniques. However, the performance of ground-coil systems is highly dependent on ground heat transfer characteristics. Soil type (clay, silt, sand, etc.) and moisture content can cause ground thermal conductivities to vary by orders of magnitude from one part of the country to another, from one lot to another within the same town, or from one point in time to another for the same lot. Thus, there are many challenges and uncertainties in designing and installing ground-coil heat pumps.

This project was undertaken with the objective of reviewing historic and current methods for designing ground-coil heat pumps. The efforts concentrated on European and North American experience. The following discussion presents the historic background followed by a description of some specific design methods which have been developed or are currently under development.

## HISTORIC BACKGROUND

The first known record of the concept of using the ground as a heat source for a heat pump is in a Swiss patent issued in 1912 to Heinrich Zoelly[1]. Thus the idea itself is about 70 years old.

Despite Zoelly's 1912 patent, it was not until after World War II that serious ground-coil development efforts began in both North America and Europe. This surge in interest lasted until the early 1950's when gas and oil became widely used as heating fuels.

Starting about 1946 in the U.S., 12 major ground-coil research projects were undertaken. These projects were all monitored by a joint AIEC-EEI committee, and the results of these projects were recorded in a series of articles in the EEI journal[2,3,4]. These projects systematically examined many ground-coil parameters including effects of pipe size, pipe spacing, depth of burial, and the effects of extended surfaces on the soil side of the pipe. Thermocouples were buried in the soil to record soil temperature as a function of time and heat extraction. Experiments included both vertical and horizontal coil configurations. They also included direct expansion of refrigerant into the ground coil and the use of secondary fluids including water and aqueous ethylene glycol antifreeze solutions. The ground coil itself generally consisted of copper tubing for direct expansion or cast iron pipe for secondary fluids.

The AIEC-EEI heat pump committee concluded in 1953[4] that "even with this considerable amount of information (from the 12 research projects begun in 1946-1948) there is still no apparently workable design equation which can be used by the individual contractor". The head of the AIEC-EEI heat pump committee, E. R. Ambrose, summarized experience on ground-coil heat pumps in 1966[5]. Ambrose indicated that an average safe selection of horizontal ground-coil size is 200 to 400 feet of coil per ton of heat pump capacity buried 4 ft deep by 4 ft apart. However, Ambrose also states that "the earth has not been found to be a practical and dependable heat source and sink". He further states that after a few years, it was found that the space between the earth and the pipe increased until heat transfer was no longer satisfactory, this in spite of the use of sand as backfill.

It was also during this time period that Ingersoll and Plass[6] developed the classic theory for ground pipe heat conduction. This theory is still used today as the basis for some complex computer simulation programs both in the U.S. and Europe which are discussed later in this paper.

In Europe during the early '50's, workers in Britain and Germany were active in ground-coil heat pump research. Notable among these were Griffiths[7]

and Sumner[8] in the U.K. and von Cube[9] in Germany. In Europe, Sumner and von Cube installed systems for residential heating, while Griffiths also commenced work on measurements into the properties of and heat transfer within soil.

The von Cube installation (at his own house) is over 25 years old and is today still in operation recording a seasonal performance factor, SPF (an annual average coefficient of performance), of about 3. The system used direct expansion of the refrigerant in horizontal steel pipes placed 0.5 to 0.8 meters deep. The experience accumulated in this installation has been of great use in subsequent work on this type of system.

The next period of intense acitivty on ground-coil heat pumps started in Europe in 1974, following the first OPEC oil price rise, and continues to the present day. In Europe, over 30 R&D projects have been identified concerning the development of design methods, installation techniques or accumulation of operating experience. In addition, over 4,000 horizontal ground-coil installations have been made in Sweden, with an unknown but significant number of similar installations in Germany. All the installations are for heating only with all but a few having hydronic heat distribution systems. Horizontal coils represent the majority of installations in Europe to present, though R&D on vertical systems has in recent years been given emphasis because of the need for systems which use less land.

In the U.S., the resurgence of activity on ground-coil systems did not occur until about 1977 to 1978 when a number of experimental and/or analytical investigations were begun. These programs involve testing of a variety of ground-coil variables and, in many cases, solar collectors have been added creating hybrid systems. In many ways, the current programs are very similar to the experimental efforts of the 1940's and 1950's. The most notable difference is the use of plastic pipe today instead of metal.

An unknown number of working ground-coil installations exist in the U.S., mostly installed by local contractors. Based on a response to a news release on this project, it is estimated that possibly as many as 1,000 such systems are active today. These installations are widely varied in basic design and, in most cases, were designed by intuition, many times using "novel or patented ideas".

As a result of the recent resurgence of interest in the U.S. and Europe, a number of ground-coil heat-pump design methods have been developed, most employing computer techniques. The following section reviews some of these design methods in more detail.

## DESIGN METHODOLOGIES

In this project, approximately 27 individual design and performance evaluation methods have been studied. In addition, numerous systems have been installed in various locations of the U.S. and Europe, based on rules of thumb (experience) evolved from experimental prototype installations.

Figures 1 and 2 represent methodologies developed in U.S./Canada and in Europe, respectively, in categories of increasing complexity from left to right in the diagrams. Modeling categories include steady-state analytical and transient analytical; lumped parameter; finite difference; and finite element in 1, 2, and 3 dimensions. A discussion of each of these modeling categories follows.

Rules of thumb have served well for specific localities where soil conditions, rainfall, and surface drainage are fairly uniform, since design specs are based on experimental results. For example, a typical rule of thumb for horizontal coils is 350 to 400 ft of 1-1/4-inch diameter polythylene tubing per ton of heat pump capacity with coils 4 ft deep and 4 ft apart[10]. Design information of this type should be used in other localities with careful comparison of soil conditions and weather-related factors. Design optimization can only be conducted slowly over a period of years with this approach based on changes to experimental systems. Even then, weather vagaries can easily mask real differences due to design changes.

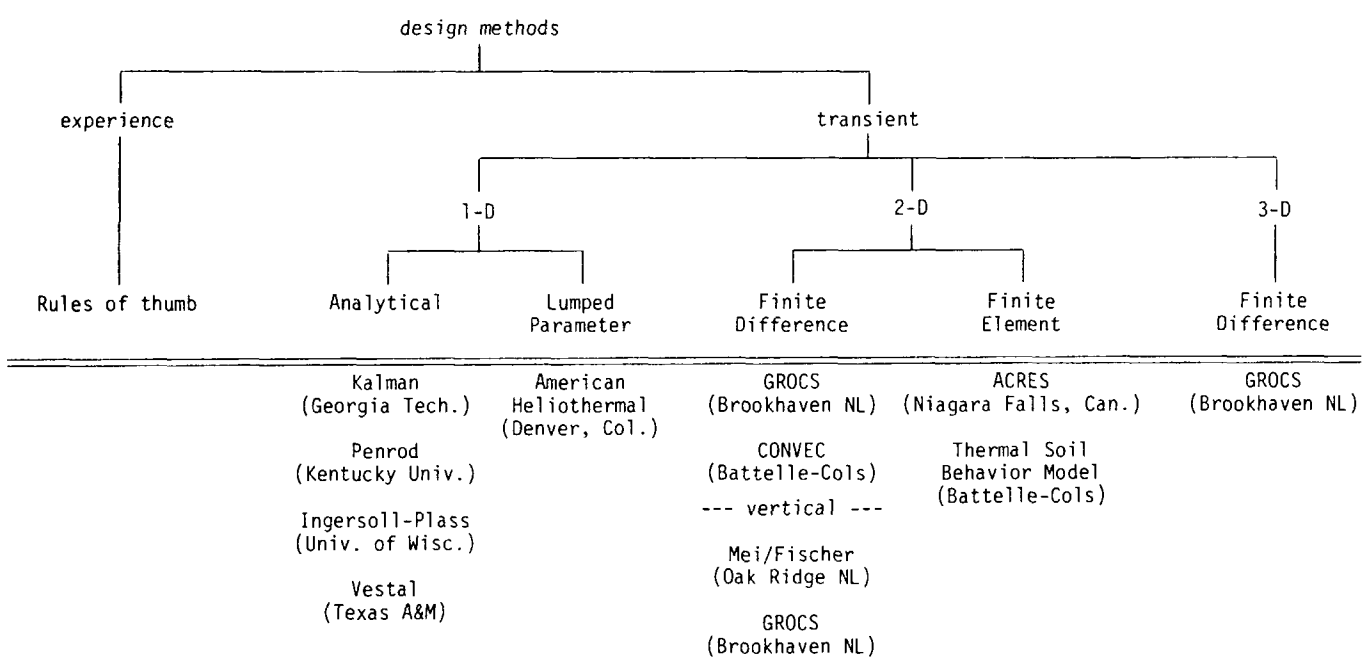



Figure 1. USA/Canada Design Methodologies

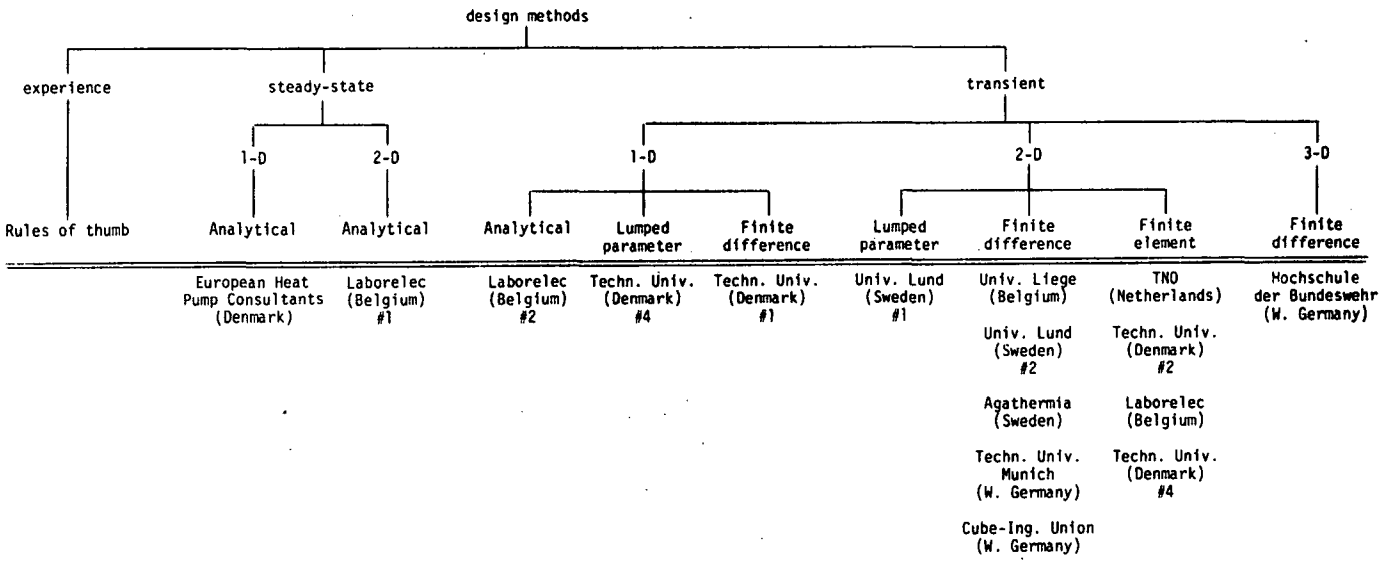



Figure 2. European Design Methodologies

For effective design optimization, a modeling approach is needed in which performance can be predicted in response to controlled weather conditions. The steady-state models, indicated on Fig. 2, are the next level of sophistication. In these models, heat extraction from the ground is modeled in monthly steps from monthly values for weather and heating-load parameters. Constants in the heat-transfer equations have to be obtained from experimental measurements or from more complex models.

Transient models provide time-dependent fluid and ground temperatures and other values which are useful for performance evaluation and design optimization. Much of the design methodologies of the late '40's to early '60's in the U.S. utilized the Kelvin line-source theory applied to ground coupling by Ingersoll. This theory is the basis of the Kalman, Penrod, and Ingersoll-Plass models in the U.S. and the Laborelec #2 model in Belgium. This theory relies on an integral solution to the 1-dimensional conduction of heat from (or to) a line source (or sink) to the far field which is assumed to vary in temperature as a function of burial depth and weather conditions.

The next level in sophistication of transient models is the lumped-parameter models in which the earth surrounding the coil is layered into concentric isothermal slabs. Weather influences are incorporated in this model by varying the temperature of the far field adjacent to the outer slab. This type of model can be used to study short-term performance in the vicinity of the pipe. Extension to long-term usage will require corrections due to the non-symmetry of the temperature field around such a buried coil.

*One-dimensional finite-difference models* can be effectively utilized to compute ground temperature at any depth in response to weather conditions as in a subprogram of the Battelle CONVEC model. In Model #1 of the Technical University of Denmark, the coil is simulated as a thin infinite slab, and conduction in the depth plane only is considered.

Two-dimensional transient models are the next level of sophistication in model categories. Some of these models are very detailed and require extensive computing time. Yet, with these complex models, the output results are only as good as the model structure. For example, the nodal geometry on many models is set up for modeling transient heat flow within a slab of earth bounded on either side by adiabatic surfaces through the center of a coil and through the midpoint between coils, at the top by the ground surface, and at the bottom by a constant temperature or constant heat flux. Other models assume far-field conditions on either side of the pipe. Ground surface temperature is specified as a mean annual value, single annual sinusoidal value, or a value computed from local terrain and weather parameters. Some models explicitly solve for local conditions in the vicinity of the pipe while others use a steady-state analytical solution for heat transfer from the pipe wall to nearest boundary nodes. Some models allow the user to set up the internal nodal geometry while the nodal geometry is internally generated in other models. Only a few models account for freezing, and none of the models account for moisture diffusion due to thermal effects.

The 2-dimensional transient models include finite-difference and finite-element models. The finite-difference models are most familiar to thermal engineers in that nodal elements represent specific volume elements. Thermal properties can readily be specified or computed as a function of temperature and moisture for each element. Temperatures at the center of each nodal element are computed from an energy balance at each time step. The time step must be selected to a low enough value to maintain stability. With very small nodal volumes in the vicinity of the pipe or elsewhere, a small time step is required, and the computing can become a major factor.

Finite-element models inherently provide a means to extend the time step while maintaining accuracy and stability. However, the physical significance of nodal elements are lost since temperatures are computed at the junctions of nodal boundaries. Model stability is related to thermal properties rather than time step; some mathematical adjustment of thermal properties may at times be required to achieve an energy balance in the integral solution at each node.

Not all of the 2-dimensional models listed in Figs. 1 and 2 are publically available. Some have never been fully developed and documented because student developers at universities graduated or funding was lost. Descriptive comments on a few of the

more significant models are provided below beginning with the U.S./Canada models.

Significant U.S. Models (Fig. 1). The GROCS model was developed by Brookhaven National Laboratory as a user-oriented, self-contained ground-coupling subprogram for use with TRNSYS to model either a buried tank or serpentine coil. The buried-tank simulation has been used to simulate the performance of an isolated vertical coil. The user controls nodal geometry and specifies the density and specific heat of each node. Neither freezing nor moisture migration are modeled. Far-field temperatures are obtained by linear interpolation from user-supplied data on ground temperatures as a function of depth and time of year. Multiple layered serpentine coils can be simulated. Nodal volumes can be provided along the length of the coil to give 3-dimensional results. An earlier version of GROCS accommodated weekly heat extraction to specified nodes, whereas the TRNSYS-compatible version is set up to receive hourly heat-extraction values.

The CONVEC model is a large program developed by Battelle to model the earth coupling adjacent to a buried gas pipeline in Arctic soil with underground water movement. The model has a 1-dimensional subprogram for computing ground surface temperature as a function of terrain, snow cover, and weather conditions. Moisture movement due to hydraulic (but not thermal) effects is modeled. Predictions of the time-dependent frost line in the ground have been experimentally verified. Internally generated nodal geometry is probably too complex for heat-pump earth-coupling studies. Thermal properties of the earth are computed as a function of moisture content and temperature. The unique feature of this model is the use of an internally computed time step to maintain stability.

The Mei/Fischer model developed by the Oak Ridge National Laboratory simulates the thermal performance of a tube-within-a-tube single vertical coil. Fluid temperature as a function of depth and earth temperature as a function of depth and radius from the pipe are provided in response to time-dependent heat extraction or addition. Constant piping and ground properties are assumed.

The Kalman model uses the method of Ingersoll and Plass to predict monthly values in response to monthly average values for a two-year period. Values for the second year are used to provide operating cost data. Kalman's major contribution to the Ingersoll and Plass approach includes accounting for temperature drop across the wall of the buried coil and formalizing the procedure in a computer program.

The ACRES finite-element model includes separate models for horizontal and vertical coils. The nodal structure of each model is very detailed, freezing can be accommodated, and ground properties are computed as a function of moisture and temperature. At the ground surface, temperature and moisture are specified as annually varying sinusoidal values. Moisture at several depths is specified and varies linearly from that at the upper depth level to the surface value. Snow cover is accommodated by an adjustment to the surface temperature. Soil-properties relationships used are for soil local to the Niagara Falls area. Time-dependent values of heat extraction or addition are used. Considerable effort was required in development of the models to achieve modeling stability.

Significant European Models (Fig. 2). As is the case with the U.S. models, the most significant European models fall into the category of 2-dimensional finite difference or finite element. of the five finite-difference models developed in Europe, one, for a single vertical pipe, has not been verified experimentally and work has been discontinued on it (Univ. Liege). The remaining four are all for horizontal pipes, with the University of Lund model being the basis of the more developed AGA-Thermia model. The Cube-Ingenieur Union model shows great similarity to the Technical University of Munich model, but is more sophisticated and has been verified. Thus, basically two finite-difference models are available, the AGA-Thermia and Cube-Ingenieur Union models. Both use constant sized rectangular elements; in the neighborhood of the pipe, an analytical solution is fitted to the adjoining elements. In many other respects the two models are very similar. For instance, they ignore moisture diffusion and allow for freezing by adjusting the thermal properties of the soil from those of frozen to unfrozen ground at 0 C. The AGA-Thermia model assumes 100 percent freezing at 0 C while the Cube-Ingenieur Union model assumes a gradual increase in the fraction of the ground which is frozen and uses experimental data for the latent heat of fusion. Both permit variable soil properties, e.g., in the Cube-Ingenieur Union model in vertical layers, and require meteorological data as input (air temperature, precipitation, radiation, etc.). Both have been tested against experimental data and have been used for sizing installations. However, they both require lengthy computing time for a complete year calculation (one to three hours), which results from adjusting the time step to avoid stability problems near the pipe. The time step is usually two hours, though results are printed for bi-weekly or monthly intervals.

There are four 2-dimensional transient finite-element models. Of these, two were developed by the Technical University of Denmark, and these vary only in that one does not account for freezing (Tech. Univ. of Denmark #4). The other calculates the enthalpy over a field of triangular elements and then derives the temperature using the appropriate values of thermal and physical properties. The TNO model uses a TNO proprietary rectangular element program, while that of Laborelec uses a Laborelec proprietary program with triangular elements. Both solve for temperature and adjust some soil thermal and physical properties between -0.5 and +0.5 C to allow for freezing. The TNO model adjusts only the thermal conductivity by a factor of up to 1.2 while the Laborelec model adjusts only the thermal capacity of the soil, using the same data as Cube. Both models use a soil surface temperature, expressed as a Fourier series (almost a simple sinusoidal function) of the air temperature, as the upper boundary condition. Both use a simple heat pump model (Carnot efficiency), ignore moisture diffusion and have sinusoidal heat demand functions. The TNO model has been tested against experimental data (albeit with preset brine temperatures) and computes rapidly (about 15 minutes for a season) while the Laborelec model is not yet completely tested (results are due in 1981) and requires two to three hours computing time per season.

The most complete model is that of Schlosser at the Technical University of Denmark. This solves for enthalpy with triangular elements, has an upper boundary condition which uses convection and radiation data (as for AGA-Thermia and Cube-Ingenieur Union models) and can simulate both horizontal and vertical coils. The ground thermal properties can

therefore be adjusted both spacially and temporally. The major weaknesses of this model are the simple heat pump model (Carnot efficiency), its unproven accuracy, and its slow computing speed.

Discussion. Of the models which predict the ground temperature in the neighborhood of the coil, i.e., the 2-dimensional and 3-dimensional finite-difference and finite-element models, only five are complete and to some extent tested (AGA-Thermia, Cube-Ingenieur Union, TNO, Laborelec, Technical University of Denmark). Of these, the last three are not generally available at present, while the first two could be made available and are extensively tested. These models, though they use very simple heat pump models and do not allow for moisture diffusion, would form the basis for a computer model for use in the U.S. for horizontal coil simulation, prediction or design. Neither are at present suitable for vertical coils. None of the European models have a detailed heat-pump model, and only the Kalman and American Heliothermal models in the U.S. have reasonably detailed heat-pump models. The GROCS and Battelle-revised American Heliothermal models can be used with the University of Wisconsin TRNSYS model which has a heat-pump component model.

Use of the Models. Detailed 2-dimensional models such as the Laborelec, Cube, AGA-Thermia, or CONVEC models, can be used to predict time-dependent temperature variations and frost-line progression or regression. Cost optimum maximum heat-extraction rates and coil diameters can be computed as shown in Figures 3 and 4 from results computed with the Cube program.

## CONCLUSIONS

Horizontal serpentine coils have been and are at present the most common coil configuration. Best design data exist for horizontal coils in heating-only applications with moist soil. Applications in dry soil or where significant summer cooling is required are not as well understood at this time.

A seasonal performance factor (SPF) of about 3.0 can be expected for a properly designed and installed residential ground-coupled heat-pump system.

Long-term (∿20 yrs) durability of buried steel and cooper tubing has been demonstrated. Life expectancy of thin-walled polyethylene tubing in the heating-only application is expected to be equally as good; however, present experience is limited to less than five years. In the cooling application with heat-rejection temperatures exceeding 100 F, some cracking has been experienced upon subsequent cool-down for heating operation due to localized stresses induced by conformity of the tubing to bedding material (stones) when hot[11].

Receding of the soil from the pipe after a period of several years was experienced in the late '40's. An understanding of this phenomenon may be crucial to the long-term operating success of these systems.

A wide variety of design methodologies are or have been utilized by system designers. Rules of thumb related to coil length and burial depth, based on performance of prototype installations, have served well for a particular locality.

The finite-difference and finite-element design methods predict fluid and earth temperatures for a specific model geometry. The accuracy of predictions are dependent on the level of detail in the model

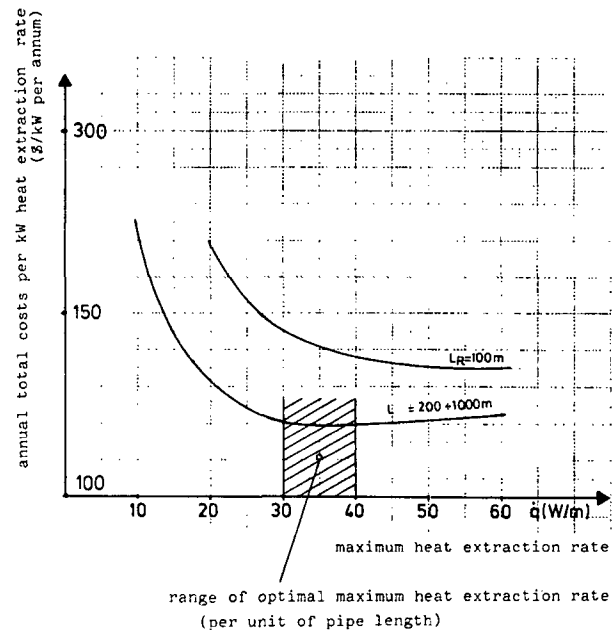



Figure 3. Specific Costs per kW Heat Extraction as a Function of Maximum Extraction Rate per Meter of Pipe

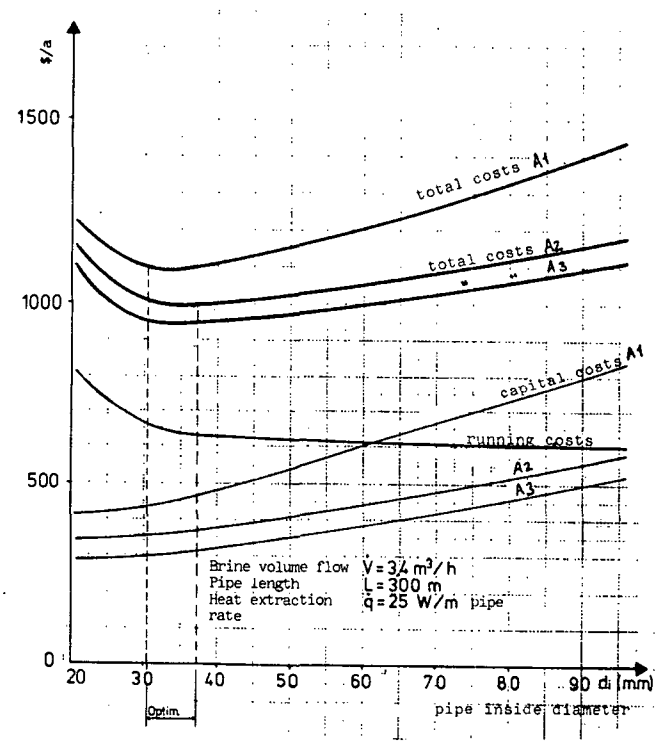



Figure 4. The Running and Investment Costs for a Horizontal Ground Coil from Hard Polyethylene (to DIN 8074) as a Function of Pipe Internal Diameter. Three alternative installation techniques are included. The heat extraction rate and pipe length are fixed.

geometry, use of constant or variable thermal properties, and modeling of freezing. No model, at present, models moisture diffusion due to thermal effects.

Brookhaven National Laboratory and Oklahoma State University have found that cycling operation enhances heat extraction from the ground because the soil adjacent to the pipe has a chance to recover to far-field conditions. This transient phenomenon is not well understood and should be further explored. Any model used for such effort should be able to accommodate variable thermal conductivity, moisture migration, and freezing.

European experience over the past 30 years can be summarized by a lack of suitable design methods, an insufficient knowledge of the effect of different soil properties on system capacity, but mainly a lack of use of these systems because of their total economic uncompetitiveness in the era of cheap fossil fuels. Design methods have been developed in the last 10 years, though these are generally mathematical models of heat transfer within the ground which use finite-difference or finite-element techniques and are therefore complex and expensive to use. Simpler models, more suitable for on-site use, have been developed, but these are inaccurate and therefore no better than empirical methods. Thus, no one in Europe has reached the stage yet of being able to develop a design manual, based on theory and experiment, which enables the optimum design to be selected for each installation. In fact, in Europe at present, the economic optimum is often used (i.e., a compact system to minimize investment cost) by many installers, rather than conservatively designed systems based on best experience, without regard to the long-term consequences for the earth.

In many ways, the conclusions regarding U.S. experience are similar to those expressed above for Europe. Despite an intense effort of experimental work in the '40's and '50's, no suitable, general design guidelines could be developed. Modern-day research and installation efforts have benefited over the earlier work by availability of plastic tubing and by use of computerized analytical techniques. However, there are still, today, major uncertainties regarding heat transfer in dry soils (especially where heat is being rejected to the soil in the summer), the long-term effects on heat transfer and materials of cyclic heat extraction and addition to the soil, and the effects of moisture migration away from the coil during heat rejection to the soil.

REFERENCES


1. Wirth, E., Aus der Entwicklungsgeschichte der Wärmepumpe Schweizerische Bauzeitung, 73, Nr. 42, pp. 647-651 (1955).

2. "Initial Survey of Heat Pump Research", EEI Journal (March 1948).

3. "Progress Report on Heat Pump Research", EEI Journal (August 1949).

4. "Research Results Concerning Earth as a Heat Source or Sink", EEI Journal (December 1953).

5. Ambrose, "Heat Pumps and Electric Heating", Wiley and Sons, Inc. (1966).

6. Ingersoll and Plass, "Theory of the Ground Pipe Heat Source for the Heat Pump", Heating, Piping, and Air Conditioning, Vol. 20, No. 7 (July 1948).

7. Griffiths, M. V., "Heat Pump Sources. Heat Transfer from Soil to Buried Pipes", ERA report Y/T 18, Electrical Research Association, Leatherhead, U.K. (1955).

8. Sumner, J. A., "Domestic Heating by the Heat Pump", J. Inst. Heating and Ventilating Engineers, 23, pp. 129-151 (July 1955).

9. von Cube, H. L. and Stiemle, F., Wärmepumpen, Grundlagen und Praxis, VDI-Verlag VDI-Verlag GmbH, Düsseldorf (1978).

10. Bose, J. E., "Design and Testing of Solar Assisted Earth Coils", Final Report to DOE, Contract EM-78-S-01-4257 (January 1981).

11. Based on experience at Brookhaven National Laboratory as per Phillip Metz; meeting of February 4, 1981, between Metz, Fischer (Battelle) and Devault (ORNL).

GROUND ENERGY COUPLING*
BROOKHAVEN NATIONAL LABORATORY
CONTRACT NUMBER DE-AC02-76CH00016
PHILIP D. METZ
PERIOD OF CONTRACT: BEGIN 10-1-80 END 9-30-81

## OBJECTIVE

The goal of this research program is to determine the feasibility of ground coupling for various heat pump systems, and if feasibility is confirmed, to create comprehensive and reliable design guidelines for optimal ground coupled heat pump systems.

## CONCEPT

The earth is a moderate temperature thermal medium of virtually limitless extent. Ground coupling is the use of the earth as a heat source/sink or storage element. Thus, for example, ground coupling is an attractive option to provide additional storage, auxiliary heating, and space cooling for solar heat pump systems. Stand-alone ground coupled space conditioning systems are also possible. They have the advantages over conventional air-source heat pumps of higher efficiency, no defrost cycle, and operation without resistance heating - even in very cold locations. It is also possible to use the ground as a heat source or sink for industrial processes.

## SUMMARY

The Brookhaven National Laboratory (BNL) Ground Coupling Research Program began with a review of the ground coupling, ground behavior and heat flow modeling literature. Analytical heat flow models were developed to approximately design ground coupling devices for use in solar heat pump space conditioning systems. A digital computer program called GROCS (GRound Coupled Systems) was written to model 3-dimensional underground heat flow in order to simulate the behavior of ground coupling experiments [1,7] and to provide performance predictions which have been compared to experimental results. [2] GROCS also has been integrated [3] with TRNSYS, [4] the University of Wisconsin solar system transient simulation computer program. This permits the hour-by-hour simulation of complete ground coupled heating systems as reported previously. [2]

Both soil thermal property experiments and 12 ground coupling device experiments are underway at BNL. The latter include 4 buried tanks, 7 serpentine earth coils in various configurations, lengths and depths, and 1 group of six sealed vertical wells. One earth coil has been used to heat a house on the BNL site with no use of resistance heating.

A key step in the BNL Ground Coupling Research Program is the development of an experimentally validated model of ground coupling. This is accomplished by comparing model predictions under various soil thermal property assumptions to experimental results to assess model validity and to provide guidance for model refinement. Work also has been conducted on simplified design procedures. Initial model validation will be followed by prototype system design and further model validation and refinement, followed by engineering field tests and comprehensive, reliable design guidelines for ground coupled heat pump systems.

*Work performed under the auspices of the Office of Buildings and Community Systems, U.S. Department of Energy.

## TECHNICAL ACCOMPLISHMENTS

• Heat Flow Model Development. Initially, analytical models were used to study both steady-state and time-dependent ground heat flow in simple geometries. Next, finite difference equation models were used to solve somewhat more realistic problems. Eventually a FORTRAN computer program called GROCS [5] was developed to solve complex 3 dimensional underground heat flow problems. Some features of GROCS include the use of 20 finite elements or "free" blocks of earth whose temperatures are determined by finite difference heat flow equations and by heat inputs used to simulate the effect of a ground coupling device, and 10 "rigged blocks" which provide realistic far-field depth and time-dependent temperature boundary conditions. The major approximations used by GROCS at present are:

(1) Twenty finite size free blocks of earth,
(2) A finite time step interval,
(3) One constant thermal conductivity (k) for every block,
(4) One constant volume heat capacity ($C\rho$) for each block,
(5) Horizontal boundary conditions a finite distance from the device modeled,
(6) No consideration of variations in ground moisture content, or moisture flow, or of freezing,
(7) Weekly heat inputs (for the most recent version).

Copies of the stand-alone program GROCS [5] as well as the TRNSYS compatible subroutine version [3] are available with documentation at nominal cost.

Work also has been conducted on the development of simplified analysis procedures and design tools. This work is divided into three parts:

(1) Determination of ground coupling device behavior given soil thermal properties.
(2) Determination of minimal soil thermal property knowledge for accurate device design.
(3) Identification of simplified methods to acquire minimal soil thermal property identification.

• Soil Thermal Property Experiments. Since 1978, experiments have been conducted at BNL to determine soil thermal properties including density ($\rho$), moisture content ($\chi_w$), heat capacity (C), thermal conductivity (k) and thermal diffusivity ($\alpha = k/C\rho$). Both undisturbed and ground coupling device influenced properties have been measured.

Undisturbed soil density, heat capacity and moisture content have been measured periodically by weighing, drying and calorimetry of soil samples. Temperature sensors buried far from any heat flow experiments at depths of from 0.6 m to 3.7 m have been read weekly. These provide a basis for comparison of the temperatures and performances of all the in-ground heat flow experiments. Additionally, this data has been used to compute undisturbed soil thermal diffusivity as described in detail in other reports. [5,6]

Soil thermal conductivity has been determined by analysis of the behavior of the in-ground heat flow experiments (as discussed in more detail below) and by the use of buried conductivity probes. Each probe consists of a constantan wire to which is soldered a copper-constantan thermocouple. The wire

is heated by passing an AC current through it and the temperature of the wire is measured as it is heated and cooled. This data is channelled through a datalogger to a microcomputer which computes conductivity on-line. Six of these probes have been installed near the Long Straight Pipe experiment described below.

- In-Ground Heat Flow Experiments. Presently twelve ground coupling device experiments are in operation at BNL. These consist of 4 buried tanks, 7 buried earth coils, and 1 group of six sealed vertical wells. The first 9 in-ground heat flow experiments, constructed in 1978, are shown in Figure 1 (and also described in detail in Ref. 1).

Tank Experiments. Each tank is constructed from two cylindrical precast reinforced concrete rings, a precast slab lid (all commonly available in local concrete yards), and a placed concrete floor. They were waterproofed using a cementitious product used to waterproof swimming pools. Each tank is about 2.4 m high with its base 3.7 m deep. The tanks are described further in Table 1. They were built at a cost of approximately \$90/m$^3$ (\$0.34/gallon) in 1978 dollars. [1] Temperature sensors are located at the inlet and outlet of each tank, inside them, and in the ground at various locations around them.

Table 1
Tank Experiment Design

| Tank | Outer Diameter (m) | Outer Height (m) | Storage Volume (m$^3$)** |
|---|---|---|---|
| A* | 2.4 | 2.4 | 8.4 |
| B | 3.0 | 2.4 | 13.6 |
| C | 2.4 | 2.4 | 8.4 |
| E | 2.4 | 2.4 | 8.4 |

*Tank A has 5 cm polystrene foam covering its top half.
**1 m$^3$ $\simeq$ 264 gal

Field Experiments. All 5 field experiments shown in Figure 1 are made from nominal size 1-1/2 in. medium density polyethylene pipe (actual O.D. 4.633 cm). Table 2 describes the field designs (see Ref. 1 for more detail). All field are filled with an antifreeze solution. Temperature sensors are located at the inlet and outlet of each field, every 30 m along the pipe, and at various locations in the ground near each field.

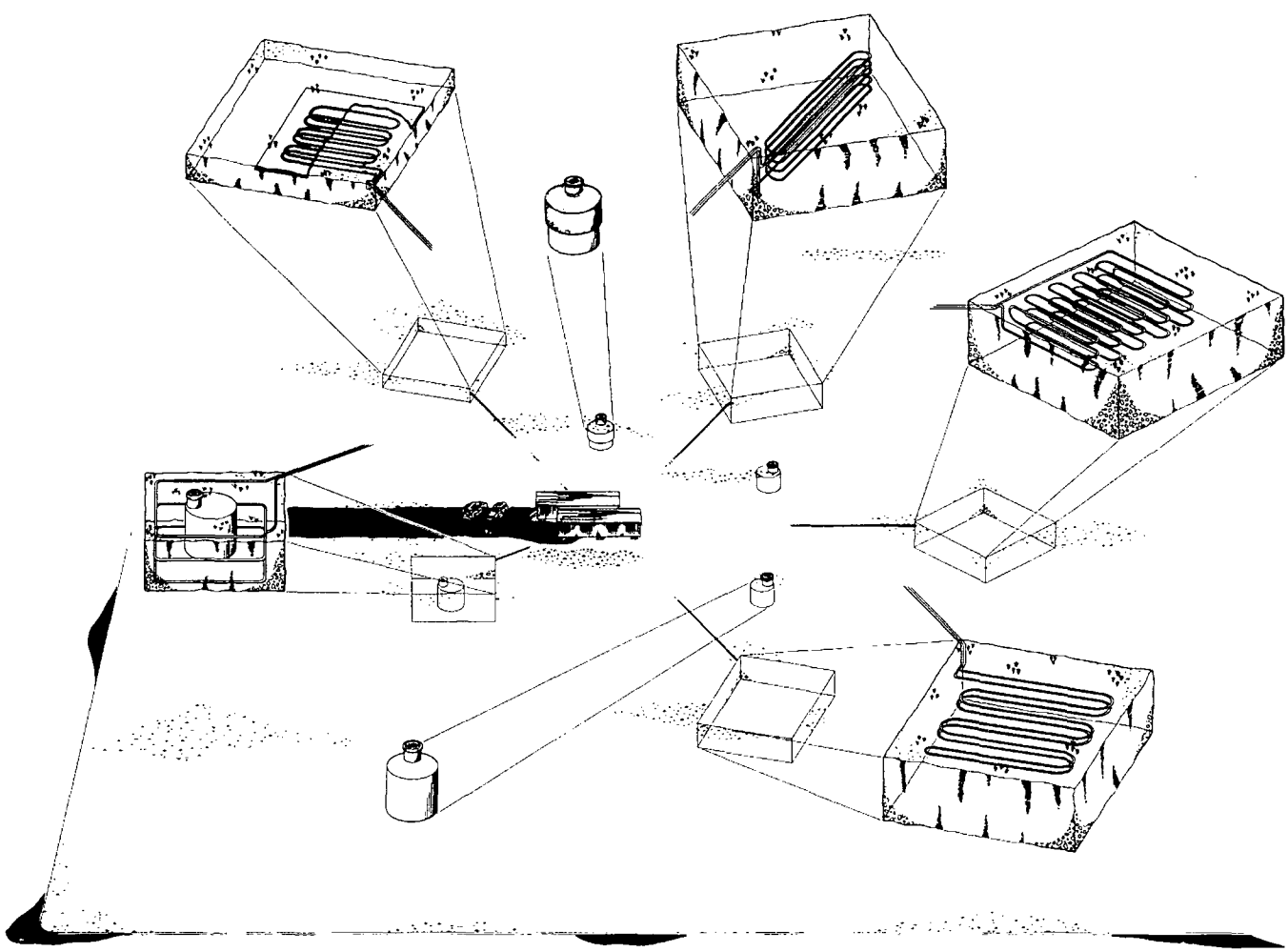


Figure 1. Perspective Drawing of the First Generation Experiments

Table 2
Field Experiment Design

| Field | Location in Figure 1 (o'clock) | Length (m) | Depth (m) |
|---|---|---|---|
| A | 1 | 170 | 1.2 to 3.7 |
| B | 3 | 270 | 1.2 to 3.7 |
| C | 5 | 160 | 1.2 |
| Coil E* | 8 | 80 | 1.2 to 3.7 |
| F** | 11 | 100 | 0.6 |

*Coil E surrounds Tank E and is part of an "advanced configuration".
** Field F is covered with 5 cm of polystyrene foam.

Vertical Wells. Two experiments were constructed in 1980. The first consists of six 18 m long sealed vertical wells located at the corners of a hexagon 6 m on a side. Each 6.4 cm inner diameter steel casing contains a dip tube of nominal size 1-1/4 in. PVC pipe down its center. An antifreeze solution flows through the wells in series. The advantages of sealed vertical wells over horizontal earth coils similar to those discussed above are that the wells require practically no land area and are deeper, and hence in contact with moister, high conductivity soil of almost constant moderate temperature. However, the cost of installed steel well casing in this size is approximately $20/m while nominal 1-1/2 in. shallow earth coils cost approximately $5/m installed.

Long Straight Pipe. The other experiment constructed in 1980, the "Long Straight Pipe" (LSP), is a single, horizontal, 18 m long nominal 1-1/2 in. polyethylene pipe buried 1.2 m deep. The pipe is surrounded with temperature, thermal conductivity and moisture sensors which are all read weekly. The LSP is connected to a 1.5 $m^3$ thermal buffer storage tank which enables us to control its temperature to within 0.5 C of any selected value. The LSP is used to provide a more detailed look at the thermal behavior of the ground near a buried pipe which extracts heat from and freezes the ground in winter or which heats the ground in summer.

Ice Tank. In March, 1981 Tank C was converted to freezable storage for a heat pump. As shown in schematic form in Figure 2, a double coil made from about 140 m of nominal 1-1/2 in. polyethylene pipe was installed in the tank. An economical and safe method to freeze almost the entire tank was identified. In operation an antifreeze solution is circulated through the coil to freeze or melt the solution in the tank. The advantage of this approach is that the latent heat thermal mass of the 8.4 $m^3$ tank, neglecting heat from the ground, is about 2.7 X $10^9$ J. This is enough heat to supply a 19 X $10^6$ J/C-day building, through a heat pump with a coefficient of performance (COP) of 3.0 for more than one week with the outdoor temperature at -12 C. Since the ice tank, built for about $900, is 2.4 m in diameter, it occupies very little land. Given the addition of heat from the surrounding ground, waste grey water, the ambient, the sun, or off-peak electricity, a very efficient, economical and compact ground coupled heat pump system for heating and cooling may result.

- System Experiment - The Arkansas House. All of the above experiments are operated with weekly heat inputs and outputs based on GROCS-TRNSYS computer simulations of ground coupled heat pump systems in a 19 X $10^6$ J/C-day building in the local (3400 heating C-day/year and 900 cooling C-day/year) climate. This strategy facilitated modeling the sensitivity of device performance to soil properties, device design

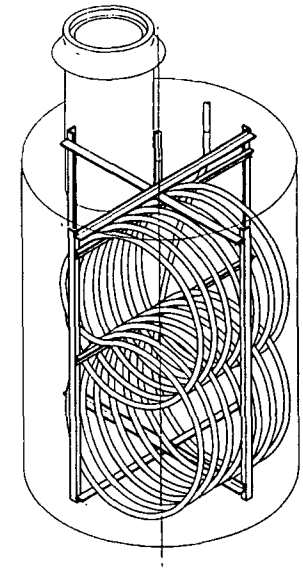


Figure 2. Ice Tank Design

and operating scenario. However, little was learned about the cycling performance or practical problems associated with an actual ground coupled heat pump installation. In order to obtain this information, beginning in February 1980, an earth coil based ground coupled heat pump system was designed, for a small (104 $m^2$) 3 bedroom house on the BNL site. The system was installed in September 1980, and has been operated and monitored since then. The goals of this project are to:

1. Experimentally test the validity of GROCS model-based earth coil design methods.
2. Determine the cost and feasibility of a residential ground coupled heat pump space conditioning system requiring no resistance heating in the local climate (3400 heating C-day).
3. Identify and solve any design, construction or operational problems which might arise.
4. Measure earth coil and system performance under realistic conditions.
5. Identify ways to improve system performance and better optimize the system.

The heating system heat requirement for the house was measured to be about 9.5 X $10^6$ J/C-day by using resistance heating of the house between February and May 1980. This leads to a design heat rate of about 3.4 kW at the local design temperature of -12 C. Based upon the underground heat flow computer program GROCS developed at BNL, an antifreeze filled serpentine earth coil was designed to just meet the building's design heating needs with an average brine temperature never less than -7 C. The 155 m long, 1.2 m deep serpentine earth coil made from nominal size 1-1/2 in. (actual O.D. 4.633 cm) medium density polyethylene pipe was installed by a local plumbing contractor using two different chain driven trenchers for $500 ($\sim$$3/m) not including materials. The earth coil layout is shown in Figure 3. A commercially available water-to-air heat pump is used with minor modifications. The indoor hardware layout is shown in Figure 4. System performance has been monitored using killowatt-hour meters, a "Btu-meter" and a datalogger-microcomputer data acquisition system. The house temperature has been kept between 21 and 23 C all winter despite outdoor temperature as low as -24 C. During the period November 20, 1980 to March 2, 1981 while the outdoor

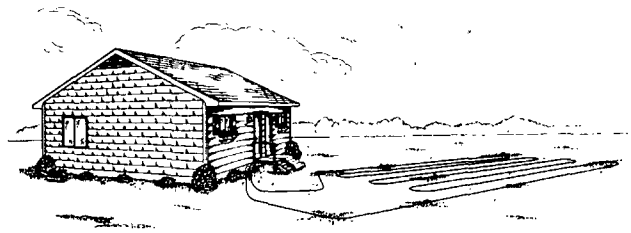


Figure 3. Installed Earth Coil Layout

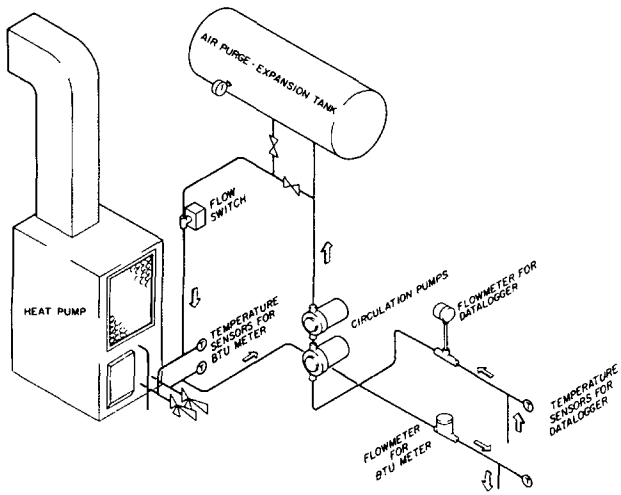



Figure 4. Indoor Hardware Layout

temperature average -2 C, the system extracted approximately 10.2 X $10^9$ J from the ground with an almost constant heat pump COP averaging about 2.3 and a system COP of 2.2. No resistance heating was used. Detailed performance results have been reported previously. [7]

- Ground Coupling Model Validation. The goal of this research program is the development of easy to use, reliable and comprehensive design guidelines for ground coupled heat pump systems. A key step toward this goal is the development of an experimentally validated model of ground coupling which will facilitate the reliable design of optimal ground coupling devices on paper.

Model validation is accomplished as follows. A physical model of each experiment suitable for GROCS is created. This, together with weekly experimental heat addition/withdrawal data for the experiment provide the input for GROCS along with values of k and $C\rho$ deduced from the underground thermal property experiments. The GROCS output is the underground temperature of each free block at selected regular time intervals. These temperatures, particularly those of blocks which represent a ground coupling device, are compared to those experimentally observed, and the goodness of the fit of the GROCS predictions to the experimental results indicates the validity of the model. A number of reports have been made on the results of these comparisons. [2,5,7,8] Work also is underway on the development of simplified analysis procedures and design tools whose accuracy is checked using the experimentally validated detailed computer model.

## FUTURE ACTIVITIES

### Contract Activities.

- Current Research. Experiments and analysis for the development of an experimentally validated model of ground coupling will continue.

- Prototype System Development. Given an experimentally validated model of ground coupling, prototype ground coupled heat pump systems can be developed. The Arkansas House described above is an example of one such system for which an accurate design procedure has already been established [7] (at least locally). Based on model validation to date, other prototype systems can be designed similarly. The performances of these systems will provide information leading to further model validation and refinement, and better design guidelines.

- Engineering Field Tests. Based on the model of ground coupling validated as described above, engineering field tests of optimized ground coupled heat pump systems will be designed and conducted.

- Comprehensive and Reliable Design Guidelines. Completion of the engineering field tests will permit the creation of comprehensive and reliable design guidelines for optimal ground coupled heat pump systems.

- Other Experiments. Ground coupling system design and behavior are strongly dependent on climate, soil type and application. Therefore fundamental experiments, prototype system development and engineering field tests must be carried out in a number of locations.

- Supporting Developments. Research is required to develop thermally optimized heat pumps with the appropriate safety devices and controls for ground coupling applications. Specifications and standards for such heat pumps do not exist and must be promulgated (e.g. via ARI). Similarly, specifications and standards for materials used in ground coupling such as plastic pipes, and pipe-antifreeze combinations do not exist and must be developed (e.g. via ASTM).

### Post-contract Activities.

The use of the earth as a heat source/sink or storage element has been attempted before. Nevertheless ground coupled heat pump systems did not become widespread, despite thermal and economic advantages, because comprehensive and reliable design guidelines were never developed. Once the thermal behavior of ground coupling devices is reliably and comprehensively understood, the commercial sector will have the knowledge necessary to sell products or synthesize advanced systems as it sees fit.

## REFERENCES

1. Metz, P.D., Design, Construction and Operation of the Solar Assisted Heat Pump Ground Coupled Storage Experiments at Brookhaven National Laboratory, Proc. Fourth Ann. Heat Pump Tech. Conf., Oklahoma State Univ., Stillwater, OK, April 1979, BNL 25908.

2. Metz, P.D., Development of a Validated Model of Ground Coupling, Proc. 1980 Ann. Mtg. Am. Sect. Int. Solar Energy Soc., Phoenix, AZ, June 1980, pp. 344-348, BNL 27661.

3. Andrews, J.W., A TRNSYS-Compatible Model of Ground-Coupled Thermal Energy Storage, submitted to ASME Winter Ann. Mtg., November 15-20, 1981.

4. Klein, S.A., et al., TRNSYS, a Transient Simulation Program, Univ. of Wisconsin-Madison, 1976.

5. Metz, P.D., A Simple Computer Program to Model Three-Dimensional Underground Heat Flow with Realistic Boundary Conditions, submitted to ASME Winter Ann. Mtg., November 15-20, 1981.

6. Metz, P.D., A 4-Parameter Least Squares Fit Method to Determine Soil Thermal Diffusivity, in preparation.

7. Metz, P.D., Design, Operation and Performance of a Ground Coupled Heat Pump System in a Cold Climate, invited paper submitted to 1981 16th IECEC Conf., Atlanta, GA, August 9-14, 1981.

8. Metz, P.D., Experimental Results From the First Year of Operation of the Solar Ground Coupling Research Facility at Brookhaven National Laboratory, Proc. Second Int. Conf. on Alternative Energy Sources, Miami Beach, FL, December 1979, p. 527-540, BNL 27137.

OTHER PUBLICATIONS

1. Andrews, J.A., Kush, E.A., and Metz, P.D., Heat Pump Development for a Solar Assisted Heat Pump Space Conditioning System, Proc. Third Ann. Heat Pump Tech. Conf., Oklahoma State Univ. Stillwater, OK, April 1978, BNL 24353.

2. Metz, P.D., The Potential for Ground Coupled Storage Within the Series Solar Assisted Heat Pump System, Proc. 1978 Ann. Mtg. Am Sect. Int. Solar Energy Soc., Denver, CO, August 1978, BNL 24579.

3. Metz, P.D., The State of the Art of Sensible Heat Storage for Solar Source Heat Pump Systems, Proc. Solar Energy Storage Options Workshop, San Antonio, TX, March 1979, BNL 25909.

4. Andrews, J.W. and Metz, P.D., Computer Simulation of Ground Coupled Storage in a Series Solar Assisted Heat Pump System, Proc. 1979 Int. Solar Energy Soc. Congr., Atlanta, GA, May 1979, BNL 26216.

5. Metz, P.D., Experimental Results from the Solar Ground Coupling Research Facility at Brookhaven National Laboratory, Proc. 1979 Int. Solar Energy Soc. Congr., Atlanta, GA, May 1979, BNL 26219.

6. Metz, P.D. The Ground Coupled Solar Heat Pump Research Program in the United States, Proc. 5th Ann. Heat Pump Tech. Conf., Oklahoma State Univ., Stillwater, OK, April 1980, BNL 27383.

7. Metz, P.D., Kush, E.A., and Andrews, J.W., The Use of Earth Coupling in Solar Assisted Heat Pumps, Proc. Nordic Symposium on Earth Heat Pump Systems, Chalmers Univ. of Tech., Sweden, October 1979, BNL 26748.

8. Metz, P.D., The Use of Ground Coupled Tanks in Solar Assisted Heat Pump Systems, in preparation.

PERFORMANCE MONITORING OF A GROUND-COUPLED SOLAR ASSISTED HEAT PUMP SYSTEM
OKLAHOMA STATE UNIVERSITY - OKLAHOMA GAS AND ELECTRIC COMPANY
EPRI 1191-6
DR. JERALD D. PARKER, PROFESSOR OF MAE
PERIOD OF CONTRACT: Begin 9-1-79 End 12-31-81

## OBJECTIVE

This study is sub-contracted to Oklahoma State University through Oklahoma Gas and Electric Company. The objective is to compare the performance of a solar- and ground-assisted heat pump system, a ground-assisted heat pump system, and a conventional air-to-air heat pump system. Performance will be measured in terms of:

1. Overall life cycle costs,
2. Peak demand on the electrical system,
3. Reliability and maintenance factors,
4. kWh consumption to consumer.

## CONCEPT OR BACKGROUND

Excessive peak demands for electricity force utility companies to build more power plants than might otherwise be needed. The Oklahoma Gas and Electric Company and the Electric Power Research Institute sponsored this project to determine the relative effectiveness of two heat pump systems in reducing and/or shifting the demand at peak times. The effort attempts to compare an air-to-air heat pump with a ground-source heat pump, and a solar-assisted ground-source heat pump.

The heat pump systems being compared in this project are installed in three houses in Perkins, Oklahoma. Perkins is a small town eleven miles south of Stillwater, in North Central Oklahoma. The north facing houses were built on adjacent lots in a new subdivision which includes primarily small, two- and three-bedroom, single-family houses. The houses have been designated the east, the west, and the middle house.

Description of the Houses. The houses chosen were designed and built with energy efficiency as a major consideration. The floor area excluding the garage is approximately 1100 sq ft. The houses are built on 10 cm (4 in) concrete slabs with 5 cm (2 in) styrofoam insulation 30 cm (12 in) deep around the perimeter. The walls are insulated with 9 cm (3.5 in) fiberglass batting and 1.27 cm (0.5 in) styrofoam (4-19), the ceiling is insulated with 30 cm (12 in) (R-36) of rock wool, and the floors are padded and carpeted. All plumbing is located on one wall to reduce heat loss in piping and the air ducts are located within the heated space of the house to minimize duct losses. Also, the houses are equipped with steel insulated doors with magnetic seals and storm windows. The exterior of the houses are bricked except for the rear area of one house which is behind collector panels.

Description of the Systems. A 1 1/2 ton Carrier air-to-air heat pump is installed in the east house as the base line of comparison. The west house has a Commandaire ground source heat pump with a 76m (250 ft) well (cased and sealed) of 13 cm (5 in) PVC pipe. The well is filled with tap water at 103 kPa (15 psig). The water is pressurized to prevent air build-up in the system. Water is taken from the top of the well through the heat pump and is returned to the bottom of the well in a 1 1/4 inch PVC pipe. As the water filters to the top, heat is transferred to or from the surrounding earth. See Figure 1. This system is similar to several installed in Oklahoma and described in reference 1.

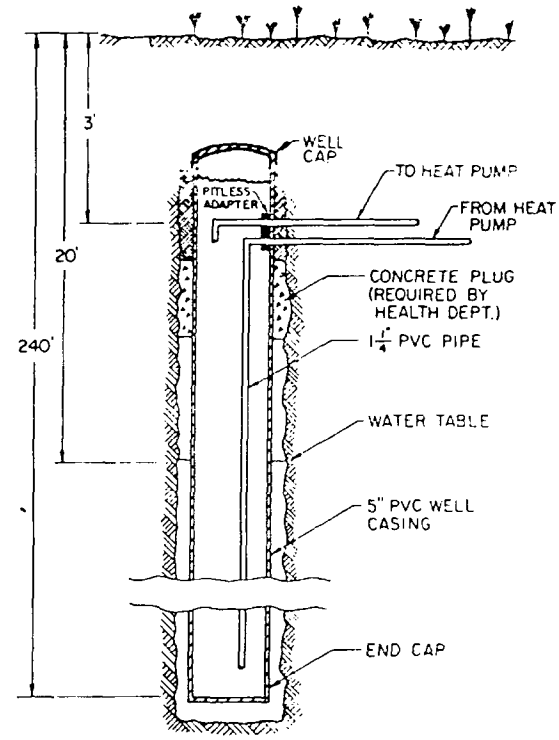



Figure 1 Well detail

The solar-assisted ground-source heat pump in the middle house has an array of five collector panels in addition to the heat pump and well. A schematic of this system is given in Figure 2. The panels are bare steel plates 2.1 m X 1.2 m (7 ft X 4 ft) with 1.27 cm (0.5 in) copper tubing spaced 10 cm (4 in) center-to-center. This design was chosen for durability since children will be playing in the backyard. The array is mounted against the rear (south) wall of the house to add heat to the well on sunny winter days. The system which is shaded on summer days is also designed to dissipate heat from the well when desirable. The solar loop controller starts the solar loop pump when the temperature of the collector panels is 11 C (20 F) hotter in winter or cooler in summer than the well water temperature. The controller also starts the well water pump if it is not operating at the time. When the temperature difference decreases to 3 C (5 F), the pumps are switched off. (The well water pump will continue to run if the compressor is on.)

Common to all three houses is the domestic water preheater. When the compressor is running, the Carrier Hot Shot water preheater circulates water from the bottom of the hot water tank through a coiled tubing heat exchanger with the hot gas exiting the compressor and then returns the water to the top of the tank. This device operates when the compressor is operating in either the heating or cooling mode.

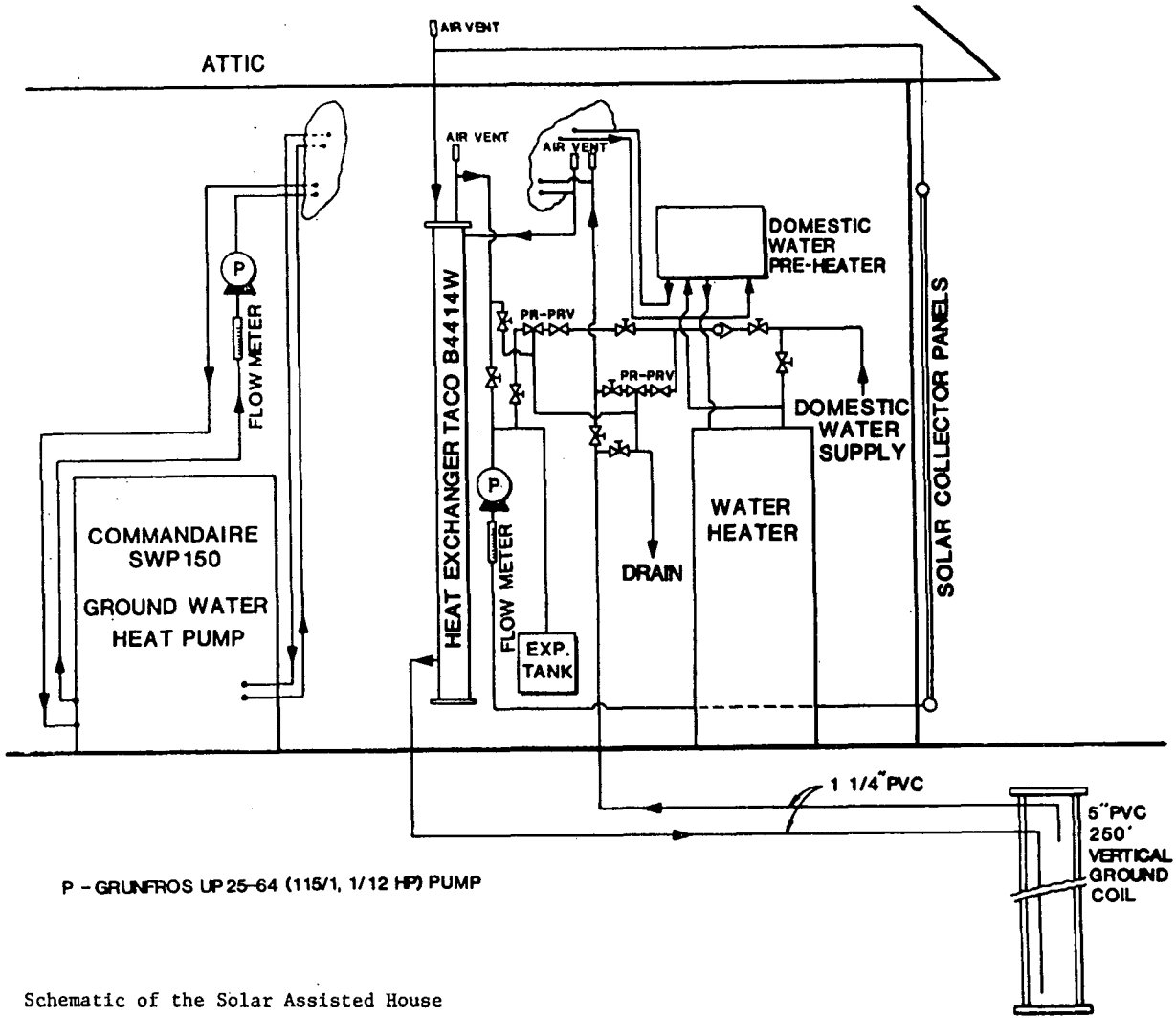



Figure 2 Schematic of the Solar Assisted House

Data Acquisition System. Table lists all the parameters monitored for the comparison of the three systems. The data acquisition system is composed of three parts: the transducers, the data logger, and the data sotrage and transmission devices. A schematic of the on-site system is given in Figure 3. The "brain" of the system is a Campbell Scientific CR5 data logger which receives analog or pulse signals from various sensors, conditions the signals and outputs the data in meaningful units to recording devices. This system and some of the transducers were chosen due to their successful use in a similar EPRI project conducted by Berkeley Solar Group.[2] The data logger is located in a thermostatically controlled shed attached to the side of the middle house. Also in the shed are the output and recording devices. Most of the sensors are located in the air handler closet and the utility closet in each house. The weather monitoring sensors are located on a 25 ft pole behind the middle house.

Data Handling and Sample Data. As shown in Figure 3 data may be recorded on audio tape, transmitted by phone to OSU, or printed on paper tape. The major mode of collection is the audio tape. The raw data from the tape are transmitted to the computer by the A235 computer tape interface manufactured by Campbell Scientific. The data are edited for inaccurate or missing values, formatted, and then stored as raw voltage or pulse information on a magnetic tape for permanent storage. The ten-inch audio tape is then reused. From the magnetic tape, the data can be manipulated mathematically or plotted.

## SUMMARY

Approximately five months of good data have been collected and studied. These data are for November 1980 through April 1981. Some of these data have been processed and analyzed. They show reductions in both energy consumption and in peak demand for the ground and solar assisted ground systems. A simulation program has been written to help analyze the data in a more meaningful way. This program, illustrated in Figure 4, simulates the building, the heat pump system, the ground source exchanger and when needed, the solar loop.

## TECHNICAL ACCOMPLISHMENTS

- A working data collection, logging and processing system has been developed and installed in the three test houses and five months of data collected.

- A simulation program has been developed and is being used to help make comparison between the three heat pump systems.

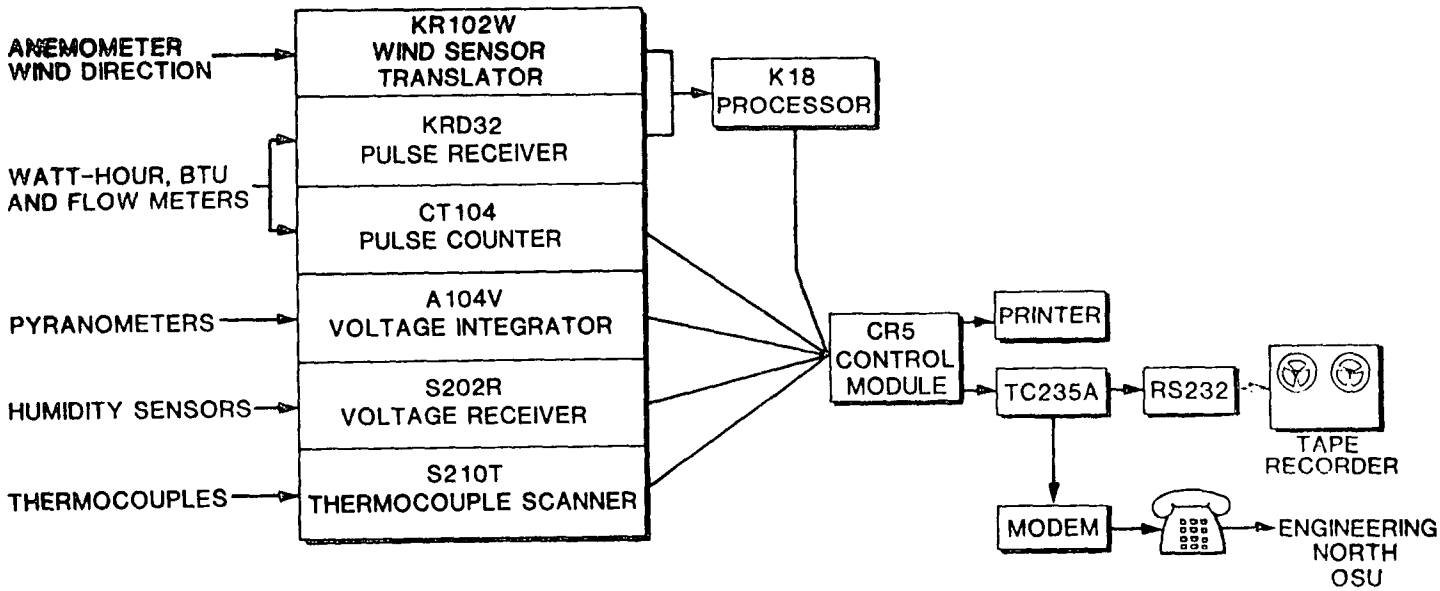



Figure 3 Basic Data Collection System

Table 1

PARAMETERS MEASURED

| | | | |
|---|---|---|---|
| | | Weather Data | Solar Insolation<br>Dry Bulb Temperature<br>Humidity<br>Wind Speed and Direction |
| | East House | Middle House | West House |
| Environmental Measurements | Dry Bulb Temperature<br>Humidity | Dry Bulb Temperature<br>Humidity<br>Water Temp from Well<br>Water Temp to Well<br>EGS Temp from Collector | Dry Bulb Temperature<br>Humidity<br>Water Temp from Well<br>Water Temp to Well |
| Heat Transfer Measurements | Hot Water Usage<br>Hot Water Btu | Hot Water Usage<br>Hot Water Btu Rate<br>Well Water Flow Rate<br>Well Water Btu Rate<br>Solar Loop Flow Rate<br>Solar Loop Btu Rate | Hot Water Usage<br>Hot Water Btu<br>Well Water Flow Rate<br>Well Water Btu Rate |
| | Total<br>Resistance Heat<br>Hot Water<br>Hot Shot<br>Compressor and Crankcase Heater<br>Inside Blower<br>Outside Blower | Total<br>Resistance Heat<br>Hot Water<br>Hot Shot<br>Compressor<br>Inside Blower<br>Well Pump<br>Solar Loop Pump | Total<br>Resistance Heat<br>Hot Water<br>Hot Shot<br>Compressor<br>Inside Blower<br>Well Pump |

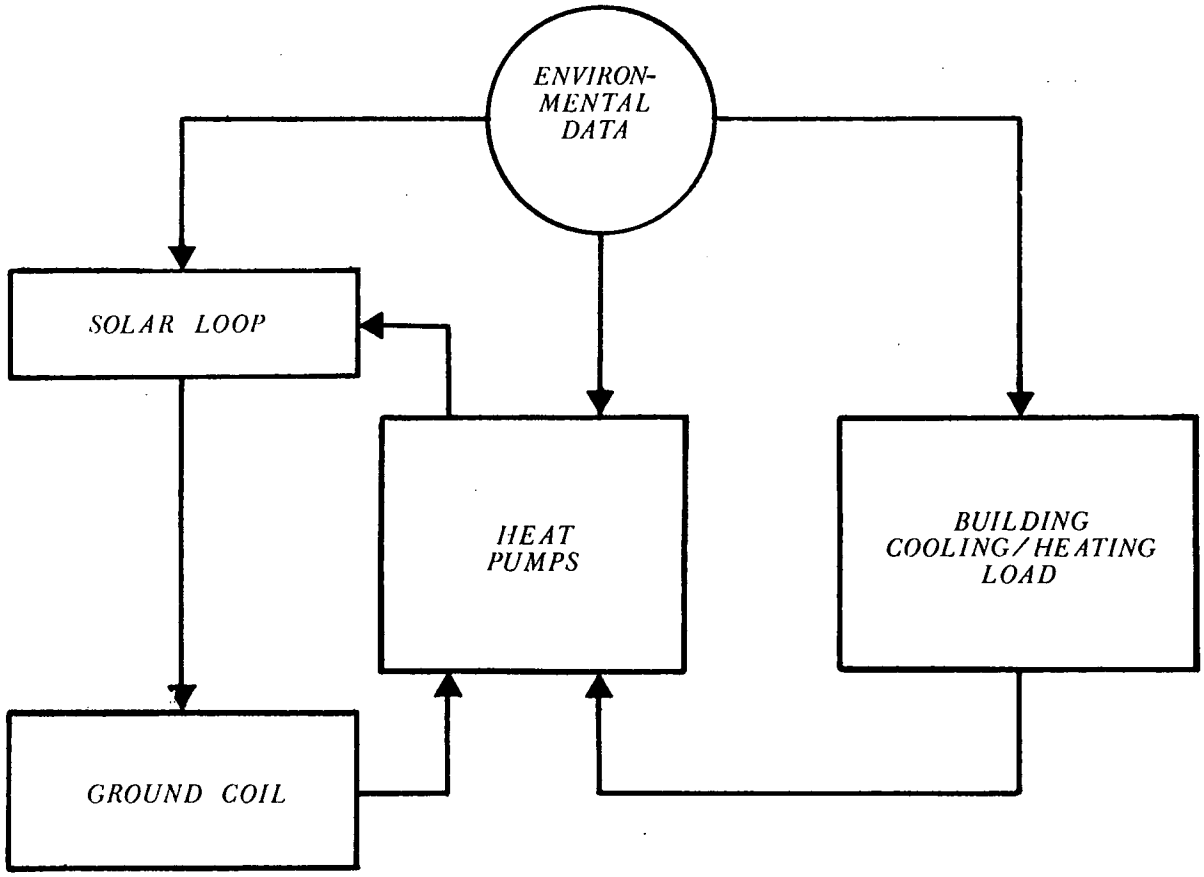



Figure 4 Flow of Logic for the General Simulation Program

FUTURE ACTIVITIES

Contract Activities. Data will continue to be collected through December 1982. An automatic dialup system is being installed to assure that satisfactory data collection is taking place.

Post-contract activities. The results of this study may cause an increased use of the ground source heat pumps. If installed in large numbers there could be a significant effect on the peak requirements of the utility.

REFERENCES

1. Bose, J. E., C. W. Ledbetter and J. R. Partin. "Experimental Results of a Low-Cost Solar-Assisted Heat Pump System Using Earth Coil and Geo-Thermal Well Storage," Proceedings of the Fifth Annual Heat Pump Technology Conference, Oklahoma State University, April, 1980.

2. Wexler, Anthony, Performance Monitoring of a Passive Solar Heated House, Stockton, California. Electric Power Research Institute Research Report 1191-3, Interim Report, September, 1979.

3. CR5 Digital Recorder Operator's Manual. Logan, Utah: Campbell Scientific, Inc., 1978.

SOLAR HEAT PUMP SIMULATOR*
BROOKHAVEN NATIONAL LABORATORY
CONTRACT NUMBER DE-AC02-76CH00016
MARK A. CATAN
PERIOD OF CONTRACT: BEGIN 10-1-80 END 9-30-81

## OBJECTIVE

The solar assisted heat pump (SAHP) system's potential for energy conservation depends on the heat pump's ability to achieve the high coefficient of performance (COP) that is possible with the high source temperatures which can be supplied by solar energy. A nominal goal of a COP equal to 6 at 32 C (90 F) source water temperature was established for the SAHP program.

The objectives of the solar heat pump hardware simulator project have been threefold:

1. To demonstrate the possibility of high heat pump COP's to an initially skeptical industry;
2. To explore practical means to realize and enhance this high performance potential;
3. To test prototype heat pumps produced by contractors under the SAHP program.

The identification of cost-competitive systems capable of utilizing the high COP potential of the heat pump has to date largely been a systems analysis activity which has been carried out at Brookhaven National Laboratory (BNL) and at other laboratories. Examples of such systems may be found in some (but not all) types of ground-coupled solar heat pump systems; bivalent SAHP's; gas-fired SAHP's; swimming pool-based solar-assisted heat pump systems; heat-pump-based solar ponds; and heat-pump-based integrated community energy systems. As such systems are identified, the simulator can provide a useful facility for performing tests of heat pump operating modes and modified heat pump components consistent with the needs of such systems.

## BACKGROUND

The SAHP program in the United States began in 1976 at the time of the formation of the United States Energy Research and Development Administration, forerunner of the present U.S. Department of Energy (DOE). The fundamental idea of the SAHP system is to collect solar energy at temperatures too low for direct use in space heating, and to use this heat as a thermal source to the evaporator of a heat pump, which processes the heat to the load at a higher temperature. The intent of the concept is to improve the efficiency of both the solar collector and the heat pump by operating in the 5 to 40 C temperature range, which is low for a solar collector (relative to collectors used for direct space heating) but high for a heat pump (relative to air-source heat pumps). The efficiency of a solar collector increases as its operating temperature is lowered, while for a heat pump the potential exists for increasing efficiency by raising the source temperature. The SAHP concept required the redesign of the heat pump to take advantage of these source temperatures, since currently available water-source heat pumps did not achieve the continuously rising COP with rising source temperature that one would expect from the Second Law of Thermodynamics.

The problems arise when the refrigerant evaporating temperature, and therefore the saturation vapor pressure is high. The suction density and mass flow increase quickly with increasing evaporating pressure. Most prominent among the symptoms caused by excessively high refrigerant mass flowrates and compressor back-pressure are: excessive strain on the compressor (particularly the valves) and compressor motor, causing poor reliability; poor efficiency caused by high condensing temperatures; "pegging" of the thermal expansion device causing the evaporator to become starved, leading to excessive superheat; and wide mismatch between capacity and load resulting from the wide range of source temperatures expected in a solar source system.

Several devices may be employed in a heat pump to thwart the deleterious effects of high evaporating temperatures and allow the potential benefit of high source temperatures to be realized. The most important are, the use of a capacity controllable compressor, a condenser which is large relative to the compressor, and an expansion device capable of controlling superheat over a wide range of mass flowrates.[1] Other devices such as a very effective evaporator may be used to improve performance and are more justifiable in terms of cost because of the energy savings possible with solar input.

The BNL solar heat pump simulator was designed to demonstrate that high COP's could indeed be obtained in vapor compression heat pumps; to explore, in concert with industry, the means to performance improvement, and to test heat pump prototypes resulting from the SAHP hardware development program.

## SUMMARY OF TECHNICAL ACCOMPLISHMENTS

A SAHP hardware simulator was constructed at BNL to demonstrate the potential of the vapor compression heat pump to obtain high COP's at high source temperatures, to explore the means to obtain such high efficiencies, and to test prototype hardware resulting from the SAHP development program. The original water coolant system which simulated heating loads was upgraded to accommodate liquid to air heat pumps. A further refinement to the simulator was the addition of an on-line data acquisition and reduction facility. Testing of an experimental mockup heat pump designed to operate efficiently under SAHP system conditions demonstrated that very high COP's can be achieved with conventional components. One prototype marketable SAHP constructed by Northrup has been tested under steady state conditions using the simulator.

## TECHNICAL ACCOMPLISHMENTS

- Constructed and Operated the Heat Pump Hardware Simulator. At the same time contracts for the development of SAHP's were awarded to Lennox Industries and Northrup Incorporated, construction of a SAHP simulator was undertaken at BNL. The purpose of the hardware simulator is to create a realistic environment for the fast and accurate testing of liquid source heat pumps. The simulator is capable of:
  - testing heat pumps of up to 8 ton capacity
  - testing liquid-to-water or liquid-to-air heat pumps
  - on line data reduction by digital computer

*Work performed under the auspices of the Active Solar Heating and Cooling Division, U.S. Department of Energy, Contract No. DE-AC02-76CH00016.

- simulating source temperatures from -10 to 45 C (14 to 113 F) and load temperatures from 10 to 70 C (50 to 158 F)
- automatic operation via computer control system

The three subsystems of the SAHP simulator are depicted semischematically in Figure 1 as they are arranged for the testing of water to air heat pumps. The Water Source Subsystem and Water Load Subsystem alone comprise the simulator as it was used for testing an experimental water to water laboratory heat pump test bed. The Air Load Subsystem, which was added to the simulator later, is discussed separately.

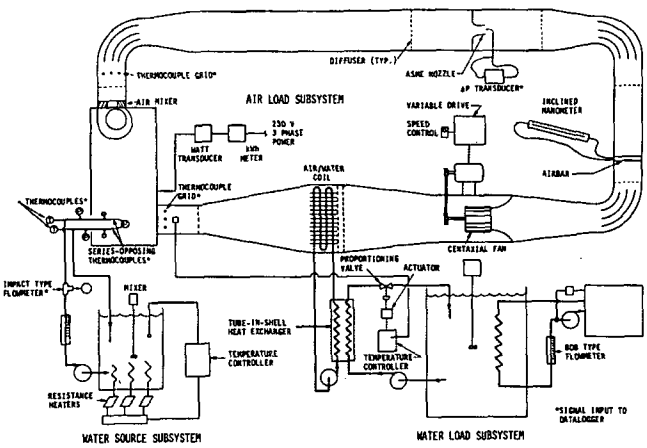



Figure 1. Solar Assisted Heat Pump Hardware Simulator

The Water Source Subsystem supplies constant temperature water to the heat pump simulating the short term behavior of the solar system storage. The temperature in the tank is maintained by replacing the heat withdrawn by the heat pump via electric resistance heaters. The heaters are governed by a servo temperature controller which can be adjusted to create any desired water temperature condition.

By circulating water at controlled temperatures through it, the Water Load Subsystem withdraws heat from the heat pump condenser simulating the heating load. The heat picked up from the condenser is transferred via a heat exchanger to a second water circuit which contains a large mass of cold water maintained at low temperatures by a chiller. The flow rate in the second circuit is governed by an adjustable temperature controller, to keep the water which withdraws heat from the condenser at controlled temperatures, allowing the condensing temperature of the heat pump to be regulated.

• Constructed and Tested the Laboratory Model Heat Pump. At the same time construction of the simulator was begun, construction of a residential size experimental solar heat pump was undertaken at BNL to be tested with the simulator to explore the details of the vapor compression cycle at high source temperatures and establish performance goals, and, to test the contractor-supplied prototypes. The Laboratory Model Heat Pump (see Figure 2) was of water-to-water type. It used a variable speed compressor (first an open type and then a Lennox two-speed hermetic), heat exchangers that were large relative to the compressor, specially selected thermal expansion valves, and a receiver. The tests of this device demonstrated that very high COP's are attainable, exceeding 9 at entering water temperature (EWT) of 43 C (110 F), with motor unloading obtained at low speed.[2,3] All components behaved well and no

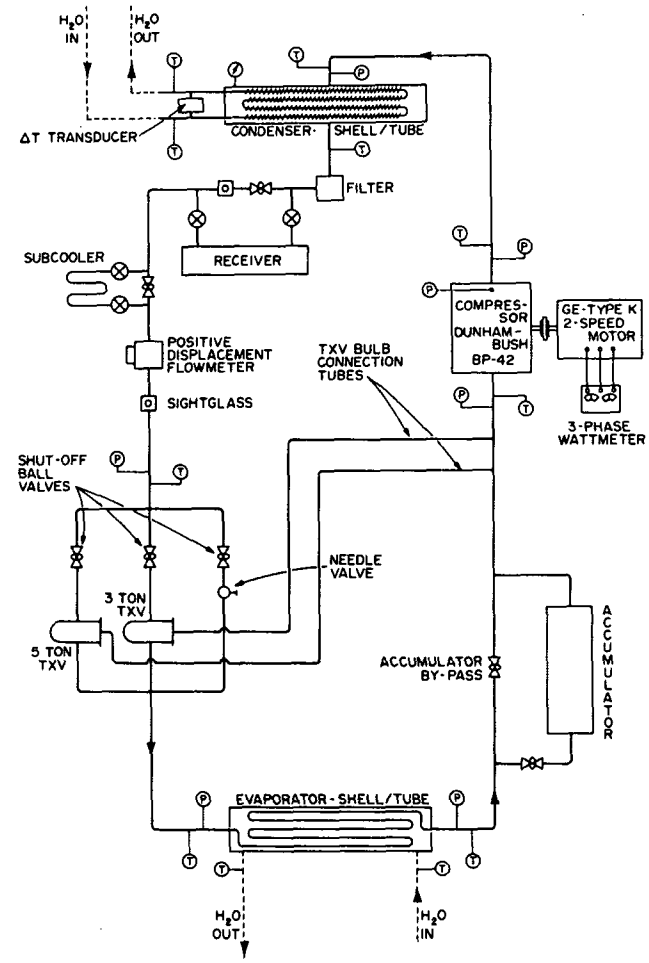



Figure 2. Laboratory Model Heat Pump

detrimental effects to the compressors were noted. This heat pump was not of a practically packaged and cost-competitive design, but it was composed of components available "off-the-shelf", including the Lennox two-speed hermetic compressor.

• Constructed an Air Load Subsystem to Test Water-to-Air Heat Pumps. The capability for testing liquid-to-air heat pumps was added to the previously liquid based heat pump hardware simulator at BNL at the construction of an instrumented duct system called the Air Load Subsystem (see Figure 3).

The Air Load Subsystem withdraws heat from the heat pump's supply air via an air/water coil connected to the water load subsystem and returns it to the heat pump to be reheated. The heat pump employs a small centrifugal blower capable of supplying 566 L/s (1200 cfm) of air against 0.25 kPa (1 in. $H_2O$) of static pressure. Additional fan power is required to maintain nominal flow through the flow measuring devices. This is supplied by a centaxial booster fan to which electrical power is supplied by a variable frequency 3-phase drive which allows the fan speed to be adjusted precisely.

Procedures and instrumentation configuration for testing follow closely the recommendations of ASHRAE Standards.[4,5] A long radius ASME air flow nozzle is mounted in the long straight section of duct in the air loop. A pitot tube, mounted at the nozzle outlet, provides for determination of velocity head of the airstream leaving the nozzle. An Annubar airbar is positioned in the narrow vertical section

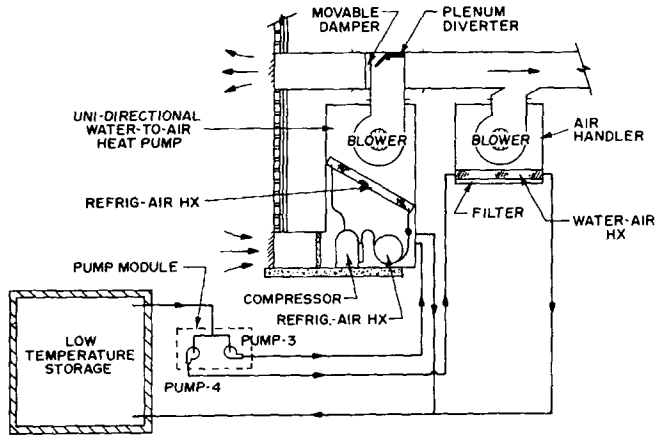



Figure 3. Northrup SAHP System

of duct downstream of the nozzle for redundant flow measurement. Diffusing plates (41% open area) with 4 mm diameter perforations were placed upstream and downstream of the nozzle and airbar to promote uniform flow. A differential pressure transducer is connected to the pitot tube and an incline manometer connected to the airbar. Both means of airflow determination were compared with a 36 point rectangular traverse of the nozzle outlet, at conditions typical of those expected during testing, and found to agree within 2%.

Nine thermocouples are mounted in the supply air duct and twelve in the return at the centers of equal areas of the local duct cross sections. Flow mixers and diffusers are used where appropriate to ensure uniform velocity and temperature distributions at the site of temperature measurements.

A strain-gauge impact type flowmeter and a full-view rotameter flowmeter are installed in the water source subsystem for flow measurement. Liquid temperatures are measured via thermocouples with integral leads of high accuracy Claude S. Gordon thermocouple wire. Two additional high accuracy calibrated thermocouples are configured to read electrical potential difference so that a sensitive measure of the change in water temperature can be obtained.

Power is supplied to the heat pump through a single phase watt transducer. An additional wattmeter and kilowatt meter are installed so that either can be switched in to verify the signal of the watt transducer. A switch is installed so that power to the heat pump fan can be transferred to a supply which is monitored separately allowing compressor power consumption to be isolated.

The datalogger is used to read all voltage signals from the SAHP simulator instrumentation (including thermocouples). The datalogger scans up to 60 channels in under 30 seconds. A Northstar microcomputer coupled with the datalogger comprise an automatic data acquisition and reduction system for the entire simulator which supplies printed records of raw and reduced data at one minute intervals during testing.

- Tested the Northrup Prototype Solar-Source Heat Pump.

Test Procedure. A prototype SAHP has been delivered by Northrup Incorporated to BNL as part of a DOE sponsored SAHP development contract. The prototype has been tested on the BNL SAHP simulator with the recently installed Air Load Subsystem.

Features of the final Northrup design are:

- Dual compressor capacity is achieved via 1-1/2 ton and 2-1/2 ton rotary compressors with common suction and discharge lines and separate accumulators
- The evaporator is a coaxial coil with refrigerant in the inner line
- The condenser is a three row slanted air-cooled condenser coil.
- The expansion device is an electric expansion valve
- Cooling is achieved via a separate water/air heat exchanger which receives chilled water from the heat pump, the condenser being cooled by outside air.

The configuration of the Northrup system is depicted in Figure 3.

The conditions under which the heat pump was tested were staggered to minimize the appearance of extraneous trends in the data. Steady state was ensured by allowing the unit to run for thirty minutes after target temperatures and flow rates were established.

Air inlet temperature was held at a constant 21.1 C (70 F) during all tests. Compressor power was read via the watt transducer signal which was input to the computer. The humidity of the air stream was measured regularly by sampling the airstream and passing the sample through a wet bulb psychrometer.

A sequence of tests was performed at constant actual air volume flow rate of 520 L/s (1100 $ft^3$/min) at a range of entering water temperatures. Air flow rate was determined in accord with accepted ASHRAE procedures[5] via the pitot tube and nozzle apparatus and via the airbar. The air flow rate was adjusted precisely via the variable frequency drive. A second sequence of tests was performed at constant entering water temperatures at a range of air flow rates. A third sequence of tests was performed at constant condensing temperatures of 43 and 47 C (109 and 117 F) by adjusting the air flow to keep condensing temperature at a constant value. Condensing temperature was taken to be the average of the saturation temperature corresponding to the compressor discharge pressure and the temperature as read from the thermocouple located at the midpoint of the condenser refrigerant line. The two values obtained never differed by more than 1 C. Power consumption as given by the kilowatt-hour meter and wattmeter, and airflow as calculated from the signal of the airbar were compared regularly with the sensors which were read by the datalogger.

Data Reduction. Heating capacity was calculated from the change in air enthalpy and from the compressor power and change in water enthalpy in accord with ASHRAE Standard 37[4] (Air Enthalpy Method and Outdoor Water Coil Method). Both measures gave excellent agreement during all tests (within 2%).

Supply and return air temperatures were taken to be the averages of the respective thermocouple readings Compressor power was computed from the watt transducer signal which was regularly verified by comparison with the kilowatthour meter and wattmeter during testing. Air mass flow was computed from the pitot tube signal in accord with ASHRAE procedure. The difference between entering and leaving water temperatures was computed from the potential difference between the series opposed type T thermocouples immersed in the entering and leaving water streams using standard type T thermocouple data tables.

Experimental Performance. The evaporating temperature, (taken to be the saturation temperature corresponding to the compressor suction pressure), and the superheated refrigerant vapor temperature leaving the evaporator (as measured by the thermocouple soldered to the evaporator outlet refrigerant line), never differed by more than 2 C (3.5 F). This indicates uniform control of the evaporator superheat over a range of entering water temperatures by the electric expansion valve. No apparent signs of liquid carryover were present during tests. One hundred twenty hours of run time were clocked on the unit before some early tests were duplicated. No significant change in performance was detected.

The compressor capacity control switch installed by Northrup operated consistently as the entering water temperature passed through the temperature range between 14 and 16 C (57 and 61 F). Transfer of power from the small compressor to the large one upon fall of the entering water temperature could occur while the heat pump was running, but switching from the large compressor to the small could not occur until the heat pump was turned off and reactivated. The time delay relay installed by BNL proved to be a reliable device for protection of the large compressor during capacity transfer.

The measured COP versus entering water temperature at constant airflow is given in Figure 4.

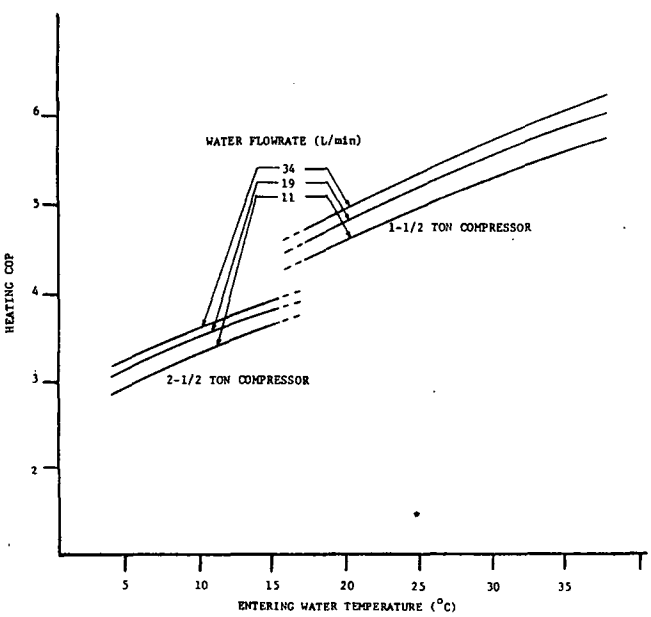



Figure 4. COP vs. EWT at Constant Airflow

Noticeable in Figure 4 is the inclining COP versus entering water temperature trend which is one basis for the SAHP's potential. The abrupt increase in COP at the point of compressor capacity transfer demonstrates the advantage of compressor capacity control (the increase in COP is partly due to having the heat exchangers large relative to the compressor after transfer is made to the small compressor). The temperature at which the capacity is transferred may not be ideal for al conditions encountered. For example, extrapolation of the 1-1/2 ton compressor curve downward past the temperature at which capacity is transferred suggests that higher COP's might be achieved at low temperatures if the temperature at which capacity is transferred is lowered. This would be desirable if the heating load were lower than the heating capacity under these conditions and a "smart" capacity transfer switch, capable of transferring compressor capacity based on outdoor temperature as well as entering water temperature might enhance the seasonal performance that is possible with this machine. Also noteworthy is the relative insensitivity of the COP to lowering of the water flowrate compared to the capacity which is affected more strongly by the water flow. This suggests that control of water flow rate to reduce the cycling losses by decreasing capacity might be worthwhile. Figure 5 gives compressor power input versus entering water temperature at constant condensing temperatures. Evident in this figure is the decline in power requirement of the compressor with increasing

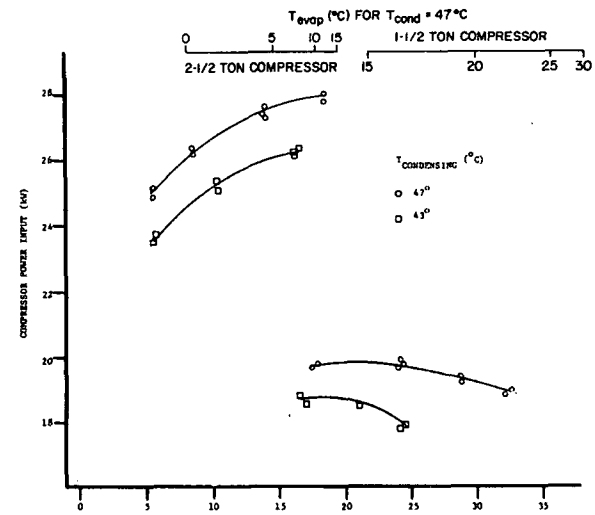



Figure 5. Compressor Power vs. EWT at Constant Condensing Temperature

evaporating temperature which is the unloading feature of the rolling piston rotary compressor documented by Northrup. This feature probably contributes substantially to the inclining COP versus entering water temperature trend apparent in Figure 4.

Figures 6 and 7 show the Northrup test results and the BNL test results (data points) in superposition. The greatest discrepancy between the two heat capacity results amounts to less than 10%. The COP data are in excellent agreement.

The contribution of the heat pump fan was not included in calculation of heating capacity and COP. Fan power requirement was approximately 300 watts, water pumping power requirement was approximately 150 watts at 30 L/m (8 gpm) (the latter from measured pressure drop and assumed 50% efficiency).

A nominal goal of a COP of 6 at 32 C (90 F) entering water temperature was set for the SAHP program contractors. Northrup claimed to have reached this goal. The results presented here shows that this goal has been approximately attained with the prototype. Since the prototype is of a design which may be readily manufactured, these results are encouraging to the DOE program.

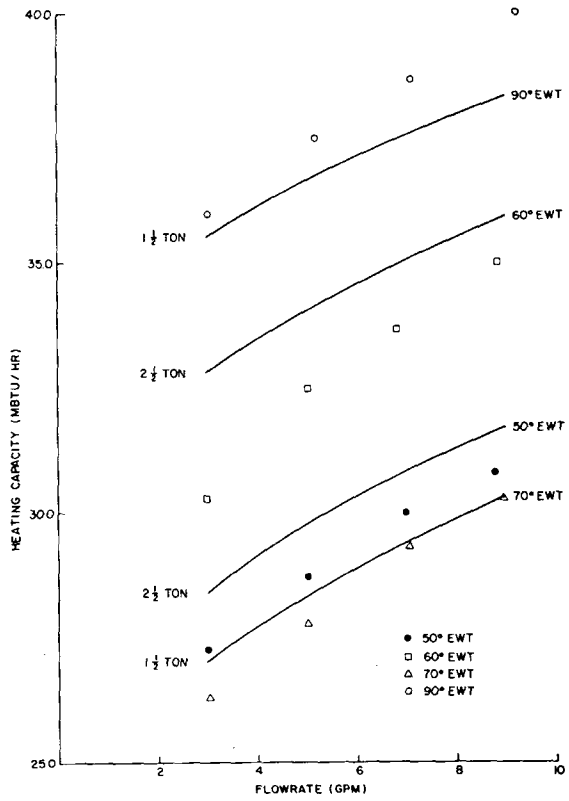



Figure 6. Heating Capacity vs. EWT - Northrup and BNL Test Results

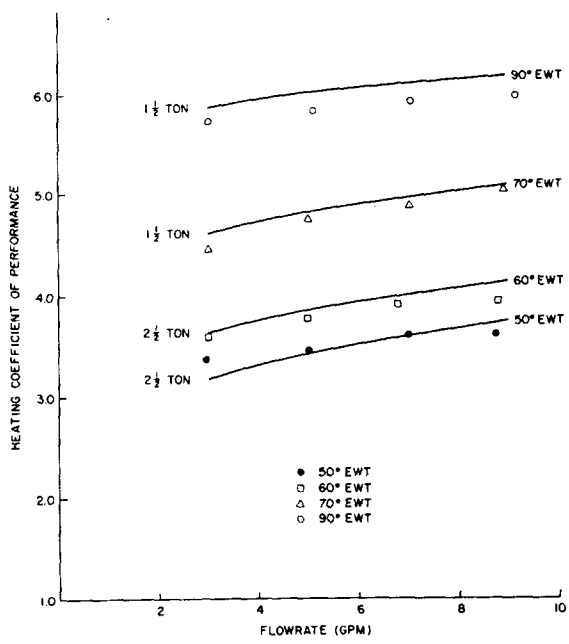



Figure 7. COP vs. EWT - Northrup and BNL Test Results

## FUTURE ACTIVITIES

In order to predict the seasonal performance of the Northrup prototype SAHP the unsteady state behavior of the unit will be determined via further tests on the simulator. The simulator will be modified so that room air will be drawn into the unit, supply air being cooled and vented into the test room. This will allow constant temperature air to be drawn into the unit during the start up period of the heat pump. The data acquisition system scanning rate will be increased to obtain higher temporal resolution in the data. Performance results will be obtained at various heat pump cycle rates with various entering water temperatures. The benefit of allowing the heat pump fan to run longer than the compressor to distribute residual heat in the condenser will be assessed. The unsteady state test data will then be used in computer simulations to determine the performance potential of the Northrup SAHP.

The Lennox prototype SAHP will arrive in mid-1981. It will be tested in a fashion similar to that in which the Northrup unit was tested. The air load subsystem will not be a closed loop because unlike the Northrup unit cooling is done in the refrigerant/air coil of the unitary machine requiring air humidity pretreatment and measurement.

Several manufacturers have been invited to participate in a heat pump component optimization study whose object is to identify heat pump components, particularly heat exchangers, which promise to make further performance improvements possible in practical (i.e., cheap, reliable, compact, etc.) water-to-water and water-to-air heat pumps. There will be two parallel paths to this effort. One will emanate from experimental and computer analytical studies and be directed toward the detailed characterization of total solar assisted heat pump heating system performance. The other path will be a program of experimental heat pump component studies. The two programs will share salient results, eventually leading to a clearer view of what SAHP systems are the most capable of exploiting the potential of the vapor compression cycle, and the heat pump configurations that are most suited to those systems. Eight manufacturers have demonstrated interest in this program. They represent a wide range of heat exchanger products and several novel approaches to linking the heat pump with the solar heat storage or source have been suggested.

## REFERENCES

1. Catan, M. A. and Kush, E. A., Results of Tests of a Prototype Solar Assisted Heat Pump. Proc. of 16th Ann. IECEC, Atlanta, GA, August 1981.

2. Kush, E., Experimental Study of a Series of Solar Heat Pump. Proc. 4th Ann. Heat Pump Conf., Oklahoma State Univ., Stillwater, OK, April 1979.

3. Kush, E., Performance of Heat Pumps at Elevated Evaporating Temperatures with Application to Solar Input, Proc. of ASME 1979 Winter Ann. Mtg., Paper No. 79-WA/Sol-19, New York, NY, December 1979; also ASME J. of Solar Energy Engr., August 1980, pp. 203-210.

4. ASHRAE STANDARD 37-78: Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment.

5. ASHRAE STANDARDS 41.1-74, 41.5-75, 41.7-78, 41.8-78.

CRAWL SPACE-ASSISTED HEAT PUMPS
OAK RIDGE NATIONAL LABORATORY
CONTRACT NUMBER: IN-HOUSE*
PRINCIPAL INVESTIGATOR: RALPH N. MCGILL
PERIOD OF CONTRACT: CONTINUING

## OBJECTIVE

The objective of this project is to determine the extent to which the performance of an air-to-air heat pump can be improved and made more efficient by utilizing the thermal energy that is stored naturally in the earth of the crawl space of a house to preheat (or precool) the source air of the heat pump. Both experimental and analytical studies are being used to achieve the objective.

## BACKGROUND

It has been suggested for some time that the efficiency of a residential heat pump could be improved by using the earth of a crawl space under the house as a thermal energy storage reservoir. Figure 1 shows a schematic representation of the concept. In the winter, the thermal energy stored naturally in the crawl space earth could be used to preheat the source air for the heat pump. This could bring about efficiency improvements by (1) increasing the effective capacity of the heat pump for a given outdoor air temperature, and (2) reducing the demand for auxiliary resistance heating. Additionally, it is likely that by preheating the outdoor air with the crawl space, the need for defrost cycles could be reduced. However, whether or not the number of defrost cycles would, indeed, be reduced depends greatly on the heat pump design. On the other hand, a factor that may diminish the efficiency improvements could be increased losses from cycling since the heat pump would cycle more often at higher source air temperatures. We may reasonably expect, though, a net improvement in the seasonal coefficient of performance (COP) of the heat pump and reduced energy consumption by using the crawl space earth storage; but, at this point we do not know how much improvement may be realized.

In the summer, the crawl space earth could be used to precool the source air for the heat pump. Such precooling would result in an increased effective capacity of the heat pump for a given outdoor air temperature. However, the improvement in seasonal COP would probably be considerably less than in the winter because the only gain would be in increased capacity for a given outdoor air temperature.

To the best of our knowledge, the first study of the crawl space-assisted heat pump concept with a resultant publication was the masters degree work by Lee Smith (1). He studied the potential enhancement of heat pump performance both by the crawl space of a house and by the attic air space. He found that using the attic air space for heat pump source air preheating was not advisable. His prediction was that there would be a small net increase in energy usage for the winter months by using the attic. On the other hand with regard to using the crawl space earth for preheating the source air, he found some very promising results. Smith's work had been supported by the Oak Ridge National Laboratory (ORNL). Consequently, after his encouraging report, we have continued working on the concept under the Thermal Energy Storage Program.

Two status reports on our work have been given by Ternes (2), (3) at the Thermal Energy Storage Annual Contractors' Review Meetings sponsored by the U.S. Department of Energy. Some of the material presented here was included in the report by Ternes at the Review Meeting of October, 1980.

The work by Ternes centered on two objectives: (1) Determine, by field tests in real houses, the amount of source air preheating that can be obtained by using the crawl spaces in two different modes. (2) Verify, through experiment, that the preheating, indeed, comes from the

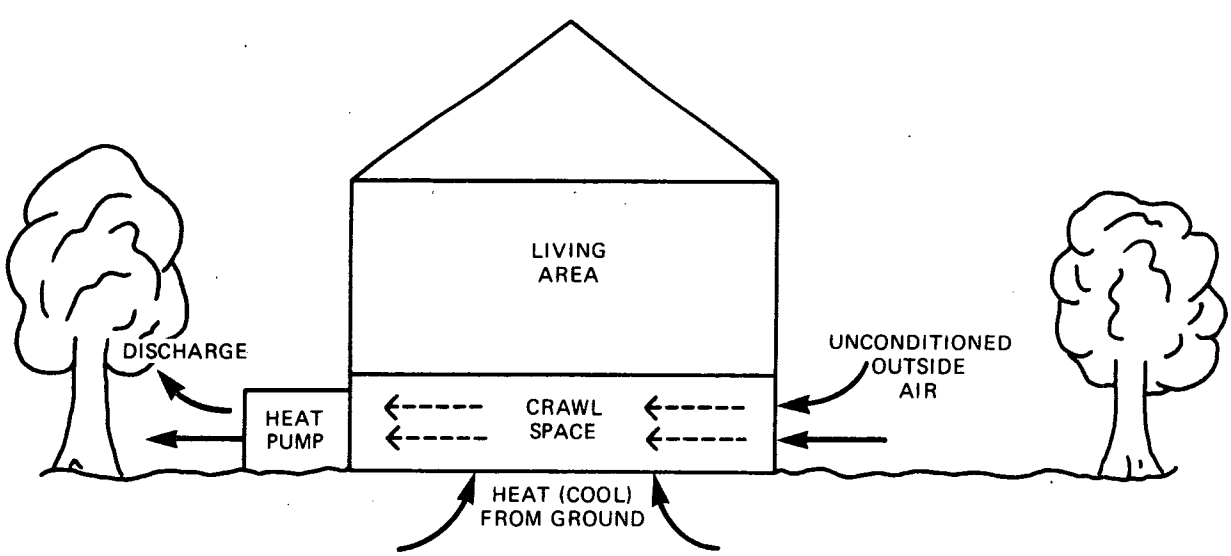



Figure 1. Schematic diagram of the crawl space-assisted heat pump concept.

*Research sponsored by the Division of Thermal and Mechanical Energy Storage Systems, U.S. Department of Energy under contract W-7405-eng-26 with the Union Carbide Corporation-Nuclear Division.

earth in the crawl space and not simply by means of heat leaks from the house above.

For the first objective above, Ternes conducted experiments with both the "single pass mode" and the "total recycle mode" of using the crawl space to preheat the source air. In the single pass mode, which is depicted schematically in Figure 2, the outdoor air is drawn through the crawl space vents and into the crawl space, where it is warmed by the relatively warmer earth, then is directed to the outdoor heat exchanger of the heat pump. Thus, the heat pump uses preheated source air rather than the ambient air. For the total recycle mode, shown in two different conceptual representations in Figure 3, the crawl space is sealed from the outdoors, and the heat pump continually recycles the same air within the crawl space. In all of the crawl space arrangements, there is usually required the placement of polyethylene "walls" or barriers to force the air to take a circuitous route through the crawl space to gain as much heat as possible. These walls are seen in both Figure 2 and Figure 3.

In his field tests of the two modes of operation, Ternes monitored many temperatures, including that of the outdoor air, the heat pump source air, the crawl space air, the ground surface of the crawl space, and that of the earth in the crawl space at depths of 0.15 m and 0.3 m. He presented preliminary results from these tests in his second report (3). The present report includes more complete results.

To accomplish his second objective, Ternes constructed a "tunnel" or duct arrangement in the crawl space of the control house at the ACES complex in Knoxville. See Figure 4 for a plan view of the experiment. Three sides of the tunnel were formed with 0.3 m thick

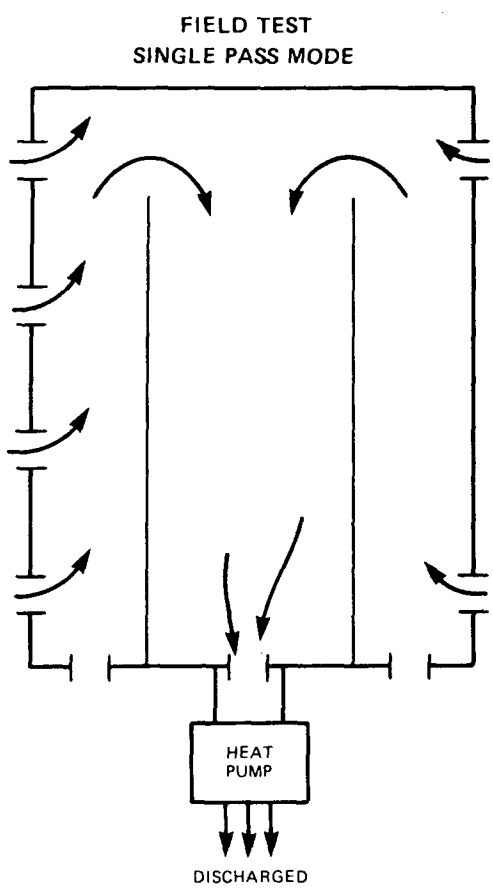



Figure 2. Schematic diagram of the single pass mode.

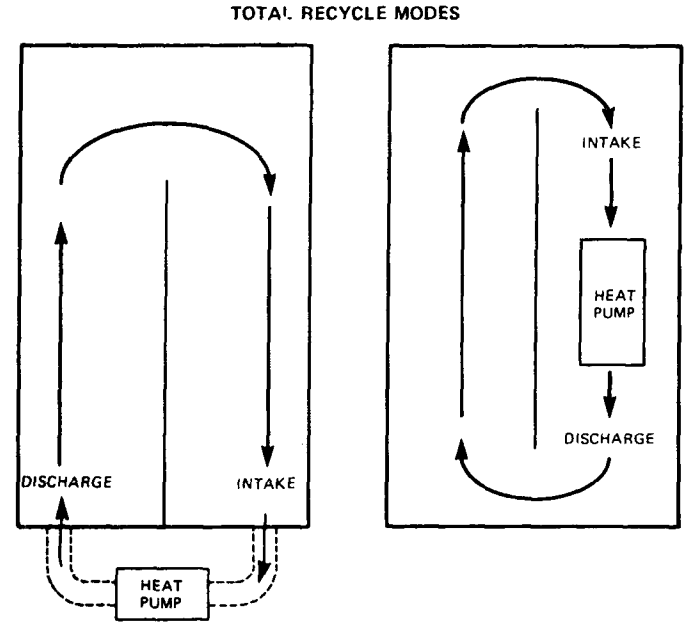



Figure 3. Schematic diagram of two different concepts of the total recycle mode.

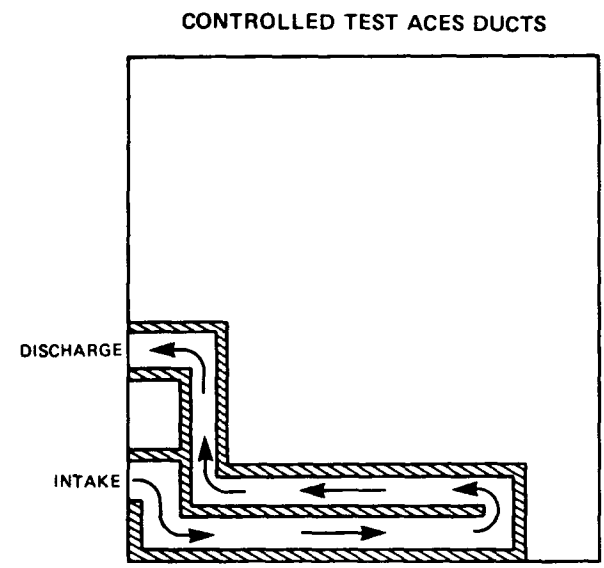



Figure 4. Plan view of the tunnel experiment.

polystyrene sheeting, while the ground itself formed the fourth side. All joints were sealed tightly, and no part of the tunnel was in contact with the house above. Therefore, the tunnel was isolated from the house to a very great degree. A fan was used to draw outdoor air through the tunnel and exhaust it back to the atmosphere. Again, numerous temperatures were recorded during the experiment. Since heat exchange with the house above and air infiltration into the tunnel were eliminated in this test, then any temperature gain by the air flowing through the tunnel must have been by heat transfer from the earth. Results from this experiment are also included in this report.

In all of the field tests, vapor barriers of polyethylene sheeting were placed over the earth of the crawl space. This is common practice in houses with

crawl spaces because it prevents excessive moisture buildup in the crawl space.

## SUMMARY

The work of this project, thus far, has shown that substantial preheating of the heat pump source air can be accomplished by using the crawl space in either the single pass or the total recycle mode. The total recycle mode showed slightly more temperature gain than the single pass mode over most of the temperature range. The tunnel experiment showed preheating of the air very similar to that of the single pass mode, thus, proving that the heat comes from the earth and not from the house above. A heat transfer analysis of the tunnel data confirmed the fact that the heat comes from the earth, and, in fact, showed that the heat transfer from the upper 0.3 m of the earth can account for all of the heat gained by the air.

## TECHNICAL ACCOMPLISHMENTS

- The amount of preheating of the heat pump source air for both the single pass and total recycle modes are shown in Figure 5. The conventionally installed heat pump source air temperature would be as depicted by the straight line shown in Figure 5. The curves for the single pass and total recycle modes are the best curve-fits for the considerable number of data points, which are not shown for the sake of clairty. Both the single pass mode and the total recycle mode show substantial temperature gains over the conventional installation up to an ambient temperature of about 10°C. As one can see, the total recycle mode shows greater temperature gains over most of the range, compared to the single pass mode. Also, it is notable that the greatest temperature gains for both modes are at the lower ambient temperatures, with the gains tapering off as the ambient temperature rises, which is as we would expect. Although one can conclude from Figure 5 that the total recycle mode is the more desirable mode for preheating, the same conclusion is not true for precooling in the summer months. Our data on precooling is extremely limited and is not included here for that reason. Nevertheless, our preliminary results indicate that the single pass mode would be preferred in the cooling season. Consequently, a total recycle installation would probably have to be arranged so that it could be switched to the single pass mode of operation when cooling is needed.

- The third curve on Figure 5 is from the ACES tunnel experiment. In this case, the ordinate of the graph is not the temperature of the air entering the heat pump, but, simply, is the tunnel outlet temperature. Again, the curve is the best fit of numerous data points, all taken while the fan that draws the outdoor air through the tunnel was running every other hour at 1.3 $m^3$/min. The important conclusion from this curve is the fact that warming of the outdoor air did occur even though the tunnel system was isolated and insulated, as much as possible, from other possible sources of heat. This means that the earth of a crawl space is, indeed, capable of warming the outdoor air that is drawn over it and is capable of sustaining that effect for long periods of time. We may also conclude that in the field tests of the real houses, represented by the single pass and total recycle curves, the preheating of the source air came, for the most part, from the crawl space earth and not from heat drawn from the house above. When comparing the results from the tunnel experiment to those from the single pass mode, one should not make absolute comparisons because crawl space dimensions, flow paths, air flow rates, etc. are not necessarily scaled properly to permit absolute comparisons.

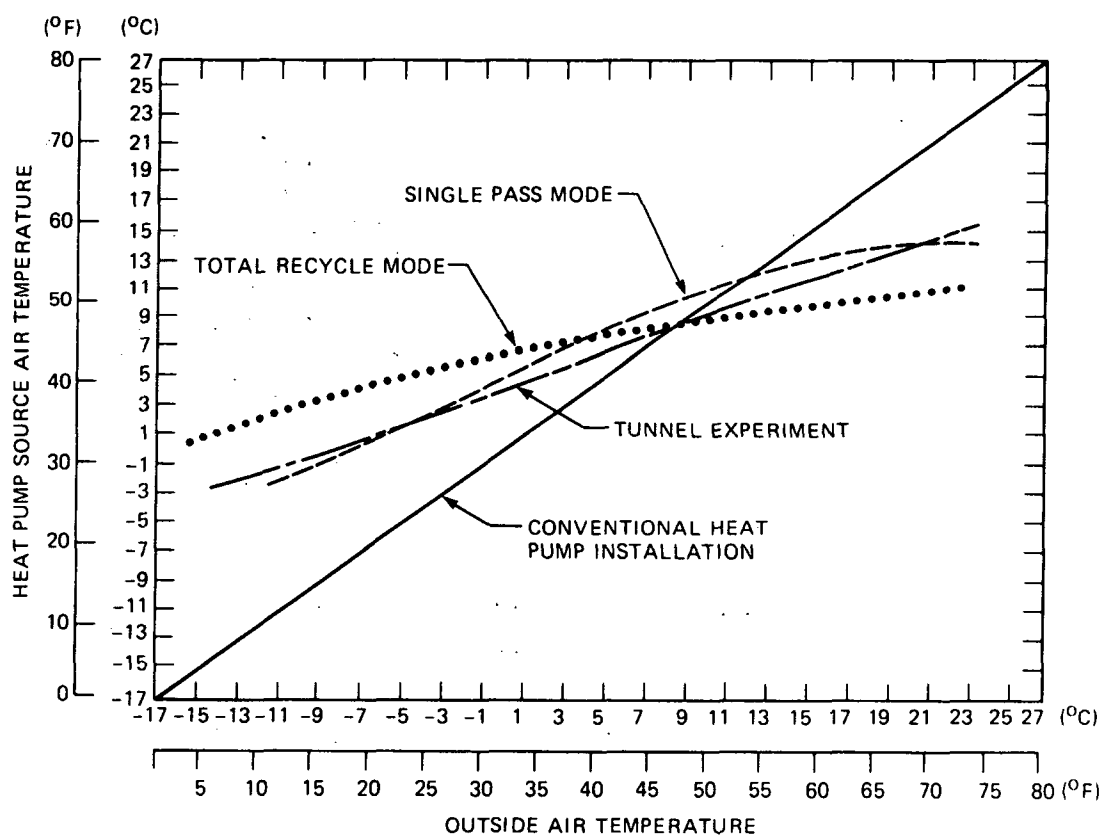



Figure 5. Composite graph of the field test results.

• We concluded above that the preheating of the outdoor air by the crawl space is by heat transfer from the earth in the crawl space. We have postulated that the heat transfer mechanism that occurs within the earth of the crawl space is simple heat conduction. This postulate is supported by an analysis of some ACES tunnel data by Alan Solomon of our Computer Sciences Division. The thermal conductivity of earth is given as approximately 0.87 to 1.3 W/mK (4). If we use our data to calculate the thermal conductivity of the earth in the tunnel experiment, and if that thermal conductivity is within the range given above, then the postulate that the heat transfer within the earth is by simple conduction is consistent with the data. Solomon's analysis was much more detailed than that which is presented here; but, briefly, the approach was as follows: The heat gained by the air can be equated to that lost by the earth (assuming heat transfer only by conduction). The temperature of the earth at three different depths for three different locations along the length of the tunnel was known, as well as the air temperature at those locations. Knowing all of this, one can calculate a thermal conductivity of the earth needed to yield the amount of heat transferred. Solomon had to average a number of variables to reach his conclusions. Nevertheless, he found the following: (1) The calculated thermal conductivity of the earth in the tunnel did not change with different air flow rates. (2) Most importantly, the overall average calculated thermal conductivity was found to be 1.04 W/mK, which is precisely within the range quoted above. Although this work has not been extensive, our preliminary conclusion is that the heat transfer mechanism that occurs within the crawl space earth is simple conduction.

## FUTURE ACTIVITIES

Thus far, in this project we have shown that heat pump source air can be preheated (precooled) by using the crawl space of a house. However, we have not yet proven that such preconditioning of the source air will be beneficial to the seasonal efficiency of a heat pump. Any predictions that we could make at this point would require many assumptions and much speculation. Therefore, we do not wish to predict at this point. However, we can say that we suspect that there will be a net benefit to be derived by using the concept.

The major thrust of this project in the future will be aimed at actual comparative testing of the concept. Presently, three houses are under construction which we will lease starting in August. The houses, which will remain unoccupied, are identical in layout and orientation and are situated on roughly the same terrain on contiguous lots. The heating and cooling loads of the houses should be very nearly the same; but, we will check them anyway. All three houses will have the same heat pump system, except one house will be set up to use the single pass mode and one set up for the total recycle mode. The third house will have the conventional installation and, thus, will serve as the control for the experiment. Each house will be heavily instrumented to measure numerous temperatures and all relevant data with regard to the heat pump performance. We believe that these tests will be very definitive and will preclude the need for speculation and assumptions about the benefits of using the crawl space thermal storage.

At the same time, computer modeling efforts are continuing. Solomon (ORNL) is developing a heat transfer model of crawl spaces which can be refined with more data from the three houses. Ultimately, his model will be integrated with a heat pump model so that crawl space-assisted heat pump performance can be predicted for various parts of the country with different climates. Again, it is anticipated that a heat pump model can be refined with more extensive data from the three houses.

Finally, we will be trying to determine answers to other questions that relate to such a use of crawl spaces. For example, could there be expected any long term detrimental effects on the construction materials in the crawl space? And, can the use of the preconditioned air be a benefit to the longevity of heat pump components?

## REFERENCES


(1) Lee O. Smith, "Solar/Geothermal-Assisted Air-to-Air Heat pump," Masters Thesis, University of Tennessee, Knoxville, 1978.

(2) Mark P. Ternes, "Crawl Space-Assisted Heat Pump," Thermal Energy Storage - Fourth Annual Review Meeting, December 3-4, 1979, DOE Publication CONF-791232.

(3) Mark P. Ternes, "Crawl Space-Assisted Heat Pump," Proceedings of the DOE Thermal and Chemical Storage Annual Contractors' Review Meeting, October 14-16, 1980.

(4) W. McAdams, Heat Transmission, McGraw-Hill, N.Y., 1954, p. 460.

WELL-WATER-SOURCE HEAT PUMP
FIELD PERFORMANCE STUDY
UNIVERSITY OF PITTSBURGH
CONTRACT ORNL/sub-7985
RICHARD S. DOUGALL
PERIOD OF CONTRACT: Begin 10/80; End 10/82

## OBJECTIVE

The objective of this study is to obtain and analyze data pertaining to the field performance characteristics of two well-water-source heat pumps (WWSHPs) which are installed in two residences located in rural south-central Pennsylvania. These were existing single-family residences which were retrofitted with the WWSHPs. The heat pumps will supply heating in the winter and cooling during the summer. The heat pump installations are being instrumented with an automated data gathering system which will allow the performance of the systems to be evaluated. The first major aim of this project is to evaluate the data obtained so that seasonal performance factors and energy consumption characteristics of the WWSHPs may be evaluated. The second major aim of this project is to develop a mathematical model that simulates the actual performance satisfactorily and then to use this model to compare WWSHPs with other heating and cooling systems.

## CONCEPT OR BACKGROUND

A large number of rural and suburban residences in the northeastern section of the United States depend on forced-air furnaces fueled by oil and bottled gas as their primary means of space heating. The availability and cost of these particular fuels have become a serious concern in recent years. Replacement by electric resistance heating is generally not economical. Because of the normal climate of this region, air-to-air heat pumps and solar heating systems are usually operating far from optimum conditions when heating is in greatest demand.

The types of residences of interest in this study represent great opportunities for energy conservation. In general, the existing homes are large and "loose" thermally. Usually the requirements for space heating are high because of high rates of infiltration as well as high heat transmission through the building envelope. Therefore, the strategy for reduction in energy use and cost would be to reduce the load and evaluate the best heating system.

As far as the type of heating system, there is one geographic feature that offers possibilities in an energy conservation strategy for the northeast. This is the widespread availability of ground water in wells, ponds, creeks, rivers, and springs. This ground water can serve as a heat source for heat pump systems since its minimum temperature is at least 0 deg. C (32 deg. F) and significantly higher for wells and springs. This overcomes the primary disadvantage of air-to-air heat pumps in severe climates.

Another advantage of ground-water heat pumps is the ease with which they can be retrofitted into existing hot-air ducting. However, there is probably some penalty in this retrofit application since the original ducting was probably designed for lower air flow rates at higher temperatures than those that would be found from a heat-pump system.

Based on these concepts, EPRI contracted with Allegheny Electric Cooperative of Harrisburg, PA to run field tests of WWSHPs at two test sites located in south-central Pennsylvania. One site is located in Bedford, PA and was fitted with a TETCO water-to-air heat pump for heating and a well-water cooling coil for air conditioning. The second site is in Huntingdon, PA and has a Century water-to-air heat pump with reverse-cycle air conditioning. The aim of the field tests is to obtain the major performance parameters of WWSHPs such as seasonal performance factors, cycling losses, consumption of water, energy use, etc. The most critical item in this list is the seasonal performance factor which is defined as:

SPF = energy to air/elec. energy to system

To determine the energy supplied to the air, air flow rates through the system and air temperature rises across the system need to be determined over long periods of time when the system is operating. Many other parameters such as temperatures, humidity, and electric power consumption must also be determined. It is desirable to have a data acquisition system that is very flexible in operation so that it can monitor all phases of WWSHP performance. Conventional data loggers can perform this type of monitoring, but they generate an enormous amount of data, most of it being of limited use. Therefore, it was decided to design a data acquisition system around a minicomputer that would have complete control of the data gathering process. This system would only monitor the WWSHP when it was in operation. In addition, it would do preliminary data analysis and thus further reduce the amount of data which must be stored.

The Department of Mechanical Engineering, University of Pittsburgh was informally a part of this project up to this point. However, the sophistication of the sensors and data gathering system required additional technical manpower. Furthermore, it was felt desirable to do some mathematical modeling of WWSHP systems so that information generated by these field tests could be evaluated for other residences and other geographic locations. Therefore, the University of Pittsburgh obtained a contract with DOE (at ORNL) to perform the data analysis and modeling portion of this project.

## SUMMARY

Since this project is in its early stages, the results obtained have been minimal up to this point. In fact, it was only in the early spring of 1981 that the location of the second test site was finally resolved. The main results so far have been the designing of the instrumentation and data acquisition system and obtaining a preliminary design of the control logic and data reduction software for the minicomputer to be located at the sites.

## TECHNICAL ACCOMPLISHMENTS

- A data acquisition system centered around a minicomputer has been selected for this project. A schematic of the basic configuration is shown in Figure 1. An HP-85 computer is the minicomputer that controls the system. An HP-3597A data logger functions as the multiplexer and A/D converter. The instrumentation listed in Table 1 will all be connected to the data logger. The instrument readings

will be sampled at preselected time intervals under control of the HP-85. The raw data is to be temporarily stored in the memory of the HP-85 where these values are converted to appropriate engineering units of temperature, flow, etc. The HP-85 also outputs the results to either the built-in minicassette tape unit or the built-in paper printer in the form of summaries or complete sets of performance data.

o The minicassette tapes which have the capability of storing the data gathered for one or two week periods have been selected as the temporary data storage media. At the end of this time, the cassette with the data will be removed and a new cassette will be inserted into the HP-85. This new cassette will not be a completely blank tape. It will have a copy of the data control program on it and another file to automatically reload the control program when a power cutoff has occurred. The data cassettes will be removed from the houses by engineers working for the local electric utilities and mailed to the University of Pittsburgh where further analysis of the data will take place.

o The instrumentation monitoring the houses has been selected and can be divided into groups which have similar scanner sequences. These are:

o Slow Scanning Rates (every 5 min.): Indoor space temperatures, outdoor temperature, outdoor humidity, and house energy usage.

o Fast Scanning Rates (every 10 sec. when systems are on): Compressor, fan, water pump, and resistance heating energy; supply air and water temperatures; air and water flow rates; supply air humidity; and air and water temperature differences across system.

o Instruments falling into the slow scanning rate category are (tentatively) set to be read every five minutes and the results will then be averaged over each hour. The (possible) exception to this approach is the wattmeter measuring total electric usage (excluding heat pump, fan, water pump and resistance heating energy) which is monitored every minute.

o The fast scanning rate instruments will be monitored only when the corresponding system is in operation. (There are two systems operating separately: the heat pump and the water pump.) The scanning rate will be as fast as the data acquisition system can handle which is probably about every 10 seconds. Information will then be averaged over each cycle of operation, i.e., cycle water consumption, cycle energy delivered to the supply air, and cycle electrical energy for the fan, compressor, and resistance heating (if operating).

o A simplified flow chart showing the present status of the monitoring procedure is shown in Figure 2. This shows the fast scans when the heat pump and/or water pump are operating and the slow scans for the other instrumentation. Also shown is an averaging process at the end of each hour and each day. This averaging and data reduction will reduce the data down to a smaller package of information. (An option to record all information will also be built into the program, but will not be used for long periods of time, i.e., more than one day at the most.)

## FUTURE ACTIVITIES

Contract Activities. The next major task to be completed is to install the instrumentation and data acquisition system in the residences. Then, this equipment must be debugged and the data gathering operation started. Once data has been obtained, analysis of WWSHP performance can be started.

A second major area is the modeling of WWSHPs. One of the most critical areas in this type of modeling is to obtain an accurate prediction of the heating and/or cooling load placed on the system. A version of NBSLD (1) contained a heat pump simulation (2) is available at the University of Pittsburgh. This computer program will form the basis for the modeling effort.

Post-Contract Activities. No specific plans have yet been made for post-contract activities. However, it is clear that the instrumentation and data acquisition systems could be used for other field tests of heat pump systems. Two particular systems that could be investigated are ground-coupled heat pumps and solar-assisted heat pumps.

It is clear that any models generated for the WWSHPs could be useful for comparisons of other hypothetical and actual situations.

## PUBLICATIONS/REPORTS/REFERENCES

At the present time, this project has not generated any reports.


1. T. Kusuda, NBSLD, The Computer Program for Heating and Cooling Loads in Buildings, Nat. Bureau of Standards Publication No. NBS-BSS-69, Washington, DC, July, 1976.

2. R.D. Ellison and F.A. Creswick, "A Computer Simulation of Steady-State Performance of Air-to-Air Heat Pumps", Report ORNL/CON-16, March, 1978.

TABLE 1
Quantities Being Measured and Type of Sensor Used

| | |
|---|---|
| **Electrical Energy** | |
| Compressor | Wattmeter |
| Fan | Wattmeter |
| Water Pump | Wattmeter |
| Resistance Heat | Wattmeter |
| House Total | Wattmeter |
| **Temperatures** | |
| Indoor Space Temperatures | Thermistors |
| Water Inlet Temperature | Platinum RTD |
| Supply Air Temperature | Platinum RTD |
| Outdoor Air Temperature | Platinum RTD |
| **Temperature Differences** | |
| Water | Thermopile |
| Supply Air | Thermopile |
| **Flowrates** | |
| Supply Air (Calibration Values) | Pitot Tube Array |
| Supply Air (Data Values) | Hot Film Anemometer |
| Water | Target Flowmeter |
| **Humidity** | |
| Supply Air | Humidity Sensor |
| Outdoor Air | Humidity Sensor |

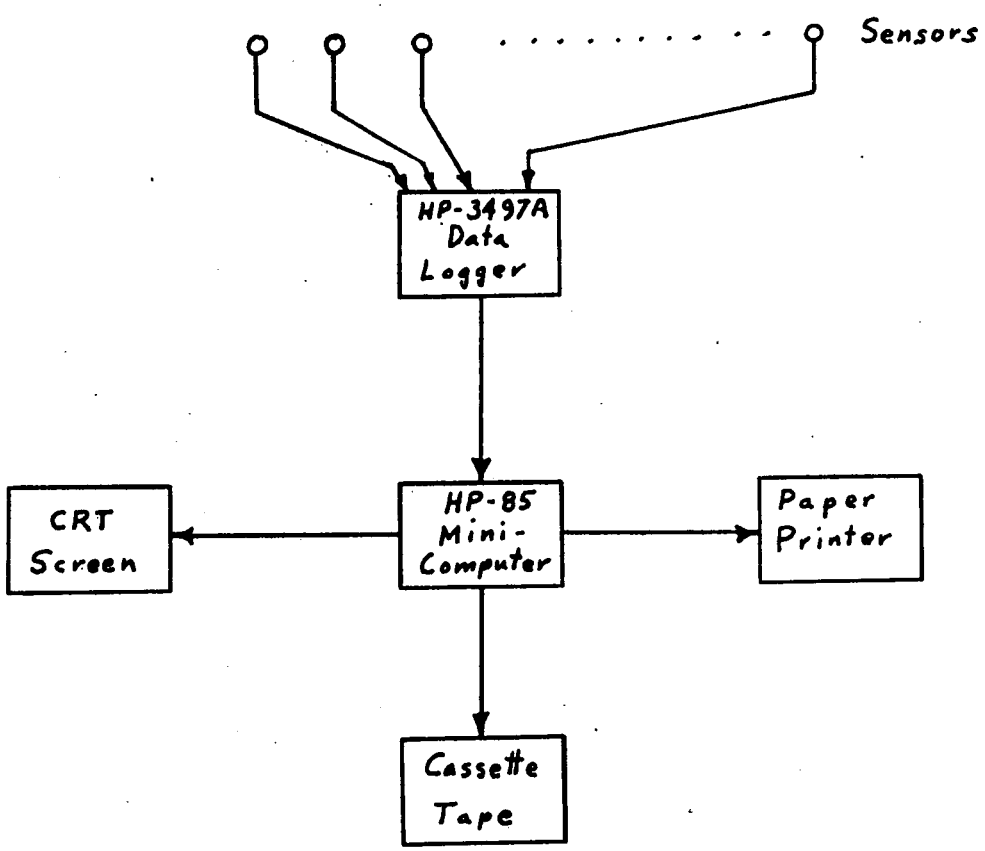



FIGURE 1 - SCHEMATIC OF THE DATA ACQUISITION SYSTEM

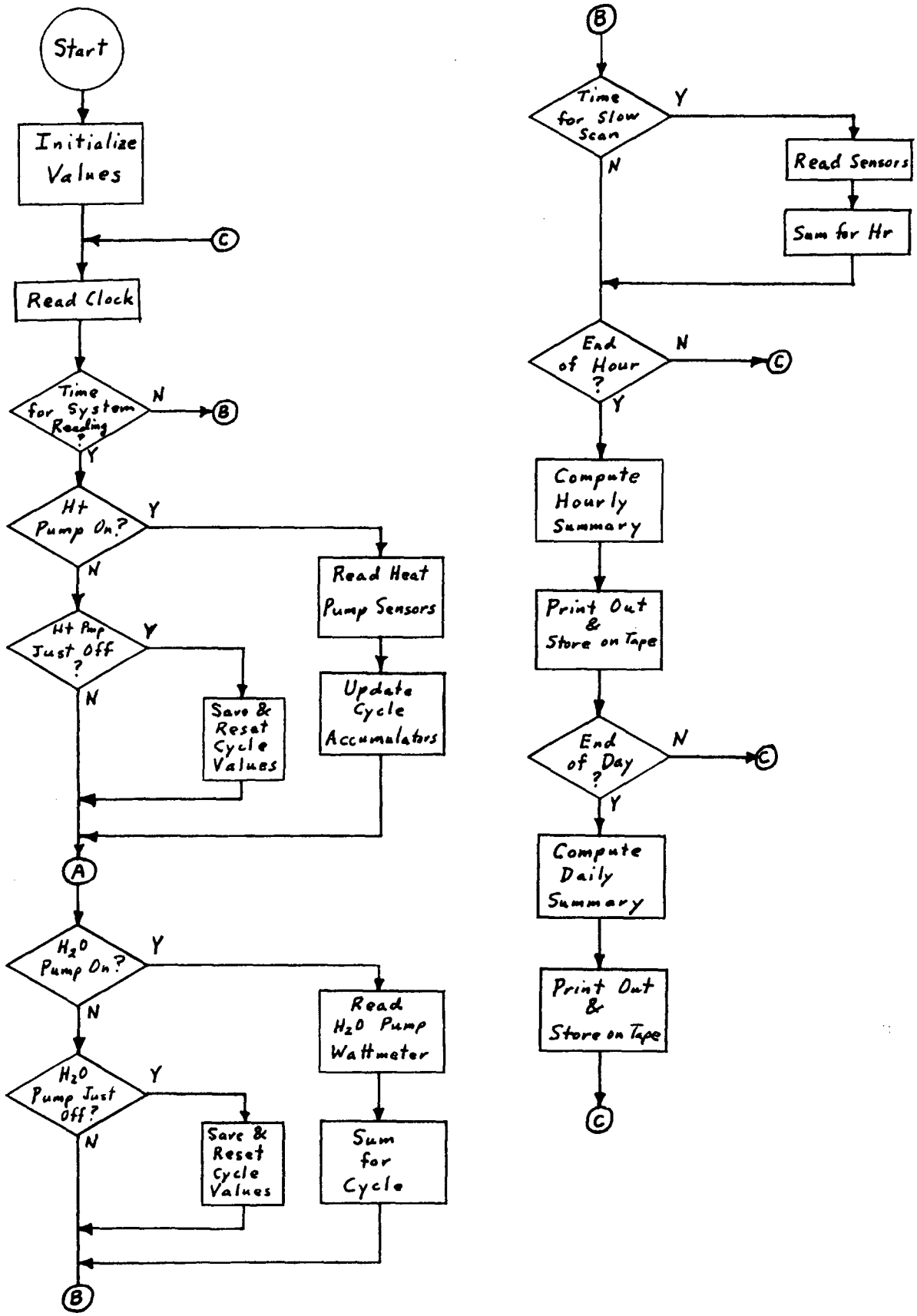



FIGURE 2 - FLOW CHART OF MONITORING PROCEDURE

# Advanced Heat Pump Technology

# ALTERNATIVE HEAT PUMP CYCLE STUDIES*

OAK RIDGE NATIONAL LABORATORY
CONTRACT NUMBER W-7405-eng-26
F. A. CRESWICK

## OBJECTIVE

The objective of this study is to conceive and evaluate alternative heat pump cycles for space conditioning applications that offer improved performance or the option to use a non-chlorofluorocarbon (CFC) working fluid.

## CONCEPT OR BACKGROUND

Heat pump cycles that have been or will be examined in this study include various air-cycle systems, the Stirling cycle, and an ejector cycle. Thus far, no specific mechanical configurations have been developed.

Air cycles were examined initially to evaluate a system that employed water injection.[1] Because air cycles without water injection have the potential for low sensitivity to heat-source temperature, low cycling losses, and for some configurations, elimination of frosting losses, the study was scoped to include several variations of dry cycles as well. Air cycles have had very little use to date because of their inherent low efficiency. Block diagrams of two air-cycle configurations are shown in Figure 1.

The Stirling Cycle heat pump (not a Stirling-engine-driven heat pump) offers a number of potential advantages: high efficiency, low sensitivity to heat source temperature, exceptional durability and reliability, simple cycle reversal and defrost, and capacity modulation by pressure control. It is conventional wisdom that the Stirling cycle is inferior to the vapor-compression cycle for space conditioning applications but there appears to be no supporting documentation.

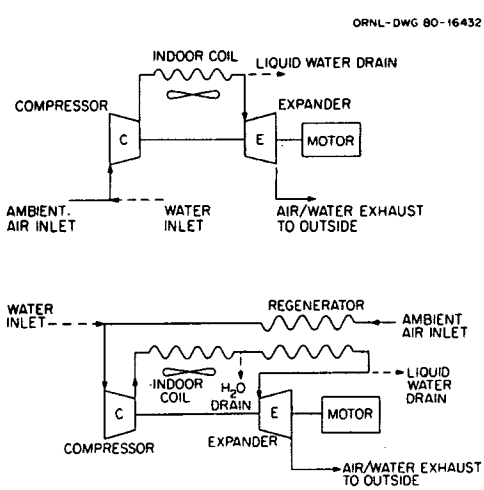



Figure 1. Air cycles with water injection. Top: nonregenerative cycle. Bottom: regenerative cycle.

The fact that both air-and Stirling-cycle systems could operate without the use of CFC refrigerants provides an additional incentive for looking at these alternatives.

The ejector cycle is of interest as a heat-actuated heat pump because of its simplicity. With its low efficiency, it is not attractive for space cooling use, but it could be an approach to a low-cost heat-actuated heat pump.

## SUMMARY

The analytical studies of various air cycles have shown that it is unlikely that a machine in this category could be developed that would match the efficiency of conventional vapor-compression systems. If there were other compelling reasons to develop such a machine, a regenerative cycle and a gas-centrifuge configuration would be promising approaches.

Preliminary studies of a Stirling heat pump indicate that efficiency and specific output in an acceptable range can be obtained. Studies have not progressed to the point where efficiency limits or an optimum configuration have been defined.

Ejector heat pump studies are at a very preliminary stage and no conclusions have been made.

## TECHNICAL ACCOMPLISHMENTS

### Air-cycle systems.

- Thermodynamic cycle calculations were performed for five dry air-cycle system variations in which component efficiency and pressure drop were used as parameters:
  - Open simple cycle,
  - closed simple cycle,
  - open regenerative cycle,
  - open multi-stage cycle,
  - a cycle in which the working-fluid air ventilates indoor space.
- It was concluded that the regenerative cycle offered the best cycle efficiency for a given compressor and expander efficiency, and the least sensitivity to low heat-source temperature. Calculated achievable compressor-only efficiencies were the following.

| Compressor & expander efficiency | Compressor-only COP at 47°F (8.3°C) | Pressure ratio for maximum COP |
|---|---|---|
| 0.95 | 2.40 | 1.9 |
| 0.90 | 2.03 | 2.1 |
| 0.85 | 1.78 | 2.3 |
| 0.80 | 1.61 | 2.4 |
| 0.75 | 1.48 | 2.5 |

*Research sponsored by the Office of Buildings and Community Systems, U.S. Department of Energy under contract W-7405-eng-26 with the Union Carbide Corporation.

- Assuming a maximum achievable component efficiency of 0.90, the peak achievable COP is about 2.0. This is less than half of the achievable efficiency of an advanced vapor-compression system.

- It was concluded that the water injection scheme did not provide any usable efficiency improvement in an air-cycle system.

- It was concluded that a gas-centrifuge configuration of an air-cycle system offered interesting possibilities for achieving high compression and expansion efficiency. In this concept, working fluid flowing radially outward within the centrifuge undergoes compression; inward flow experiences expansion. Heat exchangers are located inside the centrifuge near the hub and the periphery.

Stirling-cycle systems.

- A computer program developed by NASA-Lewis Research Center[2] to simulate the performance of a GPU-3, 10-hp, Stirling engine was borrowed and adapted to operation at ORNL. This engine has a displacement of 7.3 in.$^3$ (120 cm$^3$).

- It was found that, by loading in appropriate temperature values, heat pump operation could be simulated by the program.

- Using a 35°F (2°C) "heater tube" temperature and a 120°F (49°C) "coolant" inlet temperature the following performance was calculated.

| Motor speed (rpm) | Internal pressure, psia (kPa) 400 (2760) | 800 (5520) | 1200 (8270) |
|---|---|---|---|
| | *Compressor-Only COP* | | |
| 3500 | 1.86 | 1.73 | 1.40 |
| 1750 | 2.34 | 2.24 | 2.13 |
| 1150 | 2.47 | 2.44 | 2.35 |
| | *Heat Output, kW* | | |
| 3500 | 8.6 | 16.5 | 22.1 |
| 1750 | 4.0 | 7.9 | 11.5 |
| 1150 | 2.6 | 5.1 | 7.5 |

- These efficiency and output values were judged to be in an acceptable range for a residential heat pump.

- Because the engine (heat pump) geometry is fixed in this computer program, no attempt was made to devise an improved heat exchanger configuration for the heat pump application. It is assumed that an optimized geometry would produce high efficiency values.

## FUTURE ACTIVITIES

Contract activities. Air-cycle system studies have been concluded and no future work is presently planned.

For the Stirling heat pump, further computer simulation work is planned in which the efficiency potential will be explored and a "best" configuration will be devised. If the performance potential is sufficiently attractive, it is likely that a laboratory system will be built and tested. If possible, this machine will use an existing open refrigeration compressor as the driving mechanism.

For the ejector heat pump, a feasibility study will be conducted which will assess heating efficiency potential with existing ejector technology, required ejector improvements, and candidate working fluids.

Postcontract activities. Not applicable.

## PUBLICATIONS/REFERENCES


1. F.A. Creswick, The Theoretical Thermal Performance of Air-Cycle Heat Pumps with Water Injection, ASHRAE Preprint, January 1981.

2. Roy Tew, Kent Jeffries, and David Miao, A Stirling Engine Computer Model for Performance Calculations, DOE/NASA/1011-78/24, NASA TM-78884, 1978.


## REPORTS


F.A. Creswick, Analytical Evaluation of the Potential of Air-Cycle Machinery for Improving the Efficiency of Heat Pumps for Space Conditioning, ORNL/CON report (Draft), Oak Ridge National Laboratory.

# THE DOE HEAT-PUMP-CENTERED INTEGRATED COMMUNITY ENERGY SYSTEMS PROJECT

ARGONNE NATIONAL LABORATORY
DOE CONTRACT NUMBER W-31-109-ENG-38
PRINCIPAL INVESTIGATOR: JAMES M. CALM, PE
PERIOD OF CONTRACT: OCTOBER 1978 - MARCH 1981

## ABSTRACT

The Heat-Pump-Centered Integrated Community Energy Systems (HP-ICES) Project seeks to conserve energy by developing heat pump systems for district heating and cooling. Seven contractor teams were involved in concept development and subsequent application feasibility studies. A technical and economic assessment of the systems developed was performed based on the results of these and two related studies. The assessment concludes that district heating and cooling with heat pumps can conserve energy resources, and particularly nonrenewable fuels, in an environmentally and economically attractive way. The application potential is believed to be broad, and the energy savings of widespread implementation would be substantial. No one system is universally applicable, but many options exist. Market forces are already promoting many of the required technologies, but further research, development, and demonstration could accelerate implementation.

## OBJECTIVE

The primary objective of district heating and cooling with heat pumps is to conserve energy. Flexibility in substituting nonscarce resources for oil and gas, benign or favorable environmental impacts, and improved utility-load management are also expected.

The HP-ICES Project is part of the Department of Energy's (DOE's) Community Systems Program. The project supports development and promotes implementation of heat pump systems for district heating and cooling. The actual development of heat pumps, as compared to systems using heat pumps, is not included in this project.

## BACKGROUND

Energy Use in the United States: Approximately 26% of the energy annually consumed in the United States is used to heat and air condition buildings and to heat water for domestic and other service uses. Of this amount, about 83%, or 17 quadrillion Btu (quads), is supplied by direct combustion of natural gas and petroleum distillates. This amount includes an appropriate share of the fuels used to generate electricity used for these services.

District Heating and Cooling: District heating replaces independent building heating systems with more efficient central production of hot water or steam that is distributed to connected buildings by a thermal utility. This utility delivers energy in thermal form to subscribers, or customers, just as conventional utilities deliver energy in electrical or chemical (fuel) form. Steam and hot water for these systems are normally supplied either by boilers or as spent steam from an electrical-generation topping cycle in a cogeneration plant. In either case, the energy involved is derived from combustion of fuels or refuse or from a nuclear reaction.

Commercial district heating was first applied in the United States in 1877. Many large northern cities have, or had, small systems, usually with steam distribution. Single-building heating and cooling systems have become the norm in the United States, largely due to historically low energy prices. This remains true today even in urban areas with high thermal-demand densities. Large, single-owner building complexes such as universities, military installations, industrial complexes, and medical centers are notable exceptions. Central hot-water or steam heating for multiple buildings is not uncommon for these facilities.

District cooling provides air conditioning in a similar manner, using chilled water. The chilled water is generally produced by chillers driven by electric motors or steam turbines (or, less commonly, by engines or gas turbines). In limited applications, heat recovered by chillers is beneficially used for water and space heating, usually by chillers equipped with double-bundle condensers.

Because less use is made of air conditioning than of heating throughout the world, district cooling systems are not common. However, portions of several cities in the United States have such systems. Central cooling is also popular in single-owner, multiple-building facilities.

Heat-Pump-Centered Integrated Community Energy Systems: HP-ICES are thermal utilities that use heat pumps to provide heating, or heating and cooling, to buildings. These district heating, or district heating and cooling, systems may also provide refrigeration, water heating, or other energy services. Since heat pumps primarily transfer energy from existing and otherwise probably unused sources rather than convert it from electrical or chemical to thermal form, HP-ICES offer significant potential for energy savings. By powering these heat pumps with nonscarce fuels, which would be impractical in most conventional systems, less abundant fuels such as oil and natural gas can be conserved. Secondary benefits of HP-ICES in comparison with conventional systems include reduction of adverse environmental effects, more reliable provision of services (in contrast to increasingly frequent utility curtailments and interruptions), and delivery of services to consumers at lower costs (including acquisition, operation, and maintenance costs). HP-ICES can be classified as centralized, distributed, or cascaded, depending on the physical location of the heat pumps. A more detailed introduction to HP-ICES is presented in References 1 and 2.

## PROJECT DESCRIPTION

The HP-ICES Project was originally expected to progress from an initial system development phase through more detailed site-specific analyses, ultimately leading to physical demonstrations and evaluation. It was expected that several system concepts would be examined in parallel, but that only those with the highest potential for energy conservation and commercial viability would actually be constructed and operated. There are no current plans to continue the HP-ICES Project beyond the initial system development phase, although its resumption is recommended based on the findings of this project phase.

Table 1 identifies the major tasks completed for the HP-ICES Project. The concept innovation, system

Table 1 Outline of System Development Tasks

| Objectives | Tasks |
|---|---|
| Concept innovation | * propose and evaluate concepts |
| System development | * define systems<br>* evaluate potential applications<br>* analyze performance<br>* analyze economics<br>* analyze environmental impacts<br>* evaluate potential for expansion<br>* identify variations to afford site-specific adaptations<br>* identify component testing requirements |
| Application-feasibility studies | * describe communities<br>* describe systems<br>* estimate performance<br>* estimate economics<br>* evaluate institutional considerations |
| System assessment | * normalize analyses and data<br>* compare performance and economics of systems on uniform basis<br>* perform parametric studies of selected factors<br>* evaluate potential of individual systems |

development, and application-feasibility study tasks were primarily performed by competitive procurements, managed by the Energy and Environmental Systems Division of Argonne National Laboratory (ANL). The results of this work have been published in seven interim and seven final reports. The system assessment tasks were performed by ANL staff and consultants. A summary,[1] evaluation software description,[3] and an assessment[2] document the results of these efforts. A list of these reports as well as conference and journal publications resulting from the HP-ICES Project is included at the end of this project report.

Two additional studies were identified that addressed systems which could be classed as HP-ICES. These related studies were included in the summary and assessment reports, although they were not originally performed for the HP-ICES Project. The seven HP-ICES, two related studies, and primary subcontractors that were involved are identified in Figure 1.

## SUMMARY

District heating and cooling systems have been found to use energy efficiently and to substitute abundant resources for nonrenewable premium fuels. Employment of heat pumps in such community energy systems would increase their conservation potential. Adverse environmental impacts would be concurrently decreased by reducing both the combustion requirements and dissipation of low-temperature waste heat. In fifteen application feasiblity studies, using diverse sources and sinks, HP-ICES showed promise to:

- Substantially reduce on-site energy use for space heating and cooling and water heating,
- Substitute nonscarce resources for premium fuels,
- Significantly reduce resultant source-energy use,
- Offer load-management advantages,
- Reduce adverse environmental impacts, and
- Result in favorable life-cycle costs and, for some systems, payback periods that are shorter than five years.

Because the sources and sinks used by the HP-ICES are both varied and abundant, broad application potential is believed to exist for these systems.

The systems examined in the application feasibility studies and the resultant source (primary) energy savings are summarized in Tables 2 and 3.

## TECHNICAL ACCOMPLISHMENTS

Further data collection, system studies, and monitored physical demonstrations are required. Market forces are already promoting the required technologies, but further research, development, and demonstration (RD&D) could accelerate implementation. On the basis of currently available information, several tentative conclusions can be drawn that may be of value in guiding RD&D efforts, including prototype development:

- District heating with heat pumps is generally more capital-intensive than conventional systems for building heating and cooling. However, it can reduce costs by up to a third on a life-cycle basis for a 30-year period at expected energy prices. Higher escalation of energy prices and longer analysis periods result in much greater cost reductions.
- Centralized systems are best suited for contiguous areas of high thermal demand density, such as urban cores and major building complexes.
- Distributed systems are best suited for medium demand density areas, such as single-family housing and townhouse developments. Distributed systems are also attractive in high-density areas, particularly if individual buildings employ central-building systems.
- Cascaded systems tend to be very capital-intensive and are less attractive than the

HEAT-PUMP-CENTERED INTEGRATED COMMUNITY ENERGY SYSTEMS (HP-ICES) PROJECT

SYSTEM DEVELOPMENT STUDIES

| Source ↓ / Heat Pump System → | Centralized | Distributed | Cascaded |
|---|---|---|---|
| Annual Storage | | | |
| Ice (latent) | D | H | J |
| Water (sensible) | | A | J |
| Natural Sources | | | |
| Geothermal | | | |
| Ground Water | | A | |
| Solar | | | |
| Air | C | | F |
| Surface Water | U | H | |
| Recovered Heat | | | |
| Building | C | | |
| Electrical Generation | R, U | | |
| Industrial | R | F, R | F |
| Sewage | | G | G |

HP-ICES STUDIES

A: University of Alabama

C: Consolidated Natural Gas Service Company

D: Dubin-Bloome Associates PC

F: Franklin Research Center

G: Georgia Tech Research Institute

H: Honeywell Energy Resources Center

R: Rocket Research Company

RELATED STUDIES

U: Urban Development Corporation

J: Johns Hopkins University

Figure 1 HP-ICES Project System Development Studies

centralized or distributed systems. Cascaded systems may be viable if a favorable heat source is separate from the area of heat use and the central plant can be located close to the heat source.

- No HP-ICES are attractive for very low thermal demand densities due to parasitic pumping burdens and distribution pipe costs.

Regarding the heat sources and sinks examined, the following tentative conclusions (pending further investigation) are indicated:

- Natural sources appear very attractive. Groundwater and surface water are probably the most practical sources and sinks.* Air is a universally available source and sink, but may be less practical for district heating than for individual building systems.
- Recovered heat from buildings is very attractive, but is generally insufficient by itself.
- Recovered low-temperature heat from industry is unattractive due to the high energy and investment costs to utilize it. High-temperature heat should be used directly and low-temperature heat should be recovered for internal use within the industrial complex.
- Sewage heat and condensing heat from power plants may be attractive sources depending on local factors.
- Annual-storage cycles using aquifers may be very attractive, but require further study.
- Annual cycles employing ice-making and storage are probably more practical at the district or central heating scale than at the individual building scale, but also require more study.

These findings are suggested as guidelines and not as rigid application rules. Many local factors can result in exceptions to them for specific sites.

* Groundwater is a carrier of geothermal energy. At shallow depths, generally less than 60 ft, groundwater is also an indirect source of solar energy. Surface water is a natural collector of solar energy.

Table 2 Application Systems Description

| Study Code[a] and Community | System | Type of System | Services[b] | Heat Pump Source(s) or System for Heating | Heat Pump Sink(s) or System for Cooling | Backup[c] | Topping[c] | Storage | Pipe[d] |
|---|---|---|---|---|---|---|---|---|---|
| A Louisville | HP-ICES | Distributed | H,C | Groundwater storage | Groundwater storage | -- | -- | Aquifer | 2U |
| | Reference | Conventional | H,C | Furnaces, boilers[e] | Chillers | -- | -- | -- | -- |
| A Fort Rucker | HP-ICES | Distributed | H,C | Groundwater storage | Groundwater storage | -- | -- | Aquifer | 2U |
| | Reference | Conventional | H,C | Furnaces | Air conditioners | -- | -- | -- | -- |
| C Monroeville | HP-ICES | Centralized | [e] | Office cooling, air | Community heating, air | BLR | BLR | -- | 2-6I[e] |
| | Reference | Conventional | [e] | Conventional[e] | Conventional[e] | -- | -- | -- | -- |
| D Market Sq. | HP-ICES | Centralized | H,C,W | Freezing ice in storage | Melting ice in storage | -- | -- | Ice bin | 4I |
| | Reference | Central plant | H,C,W | Boilers | Chillers | -- | -- | -- | 4I |
| D Park Plaza | HP-ICES | Centralized | H,C,W | Freezing ice in storage | Melting ice in storage | -- | -- | Ice bin | 4I |
| | Reference | Central plant | H,C,W | Boilers | Chillers | -- | -- | -- | 4I |
| F Heckman | HP-ICES | Distributed | H | Industrial waste heat | -- | -- | -- | -- | 2I |
| | Reference | Central plant | H | Boilers | -- | -- | -- | -- | 2I |
| F Walden | HP-ICES | Cascaded | H,C | Air, boilers | Air | BLR | -- | -- | 2U |
| | Reference | Conventional | H,C | Conventional[e] | Air cond., chillers[e] | -- | -- | -- | -- |
| G Conyers | HP-ICES | Cascaded | H,C | Wastewater | Air | BLR | -- | -- | 2U |
| | Reference | Conventional | H,C | Furnaces | Air conditioners | -- | -- | -- | -- |
| G Cornelia | HP-ICES | Distributed | H,C | Wastewater | Air | BLR | -- | -- | 2U |
| | Reference | Conventional | H,C | Furnaces | Air conditioners | -- | -- | -- | -- |
| H Bloomingdale | HP-ICES | Distributed | H,C,W | Freezing ice in storage | Melting ice in storage | -- | -- | Ice pond | 1U |
| | Reference | Conventional | H,C | Furnaces, resistance[e] | Air cond., chillers[e] | -- | -- | -- | -- |
| R Everett | HP-ICES | Centralized | H,W | Industrial waste heat | -- | BLR | BLR | Tank | 2I |
| | Reference | Conventional | H,W | Furnaces | -- | -- | -- | -- | -- |
| R Seattle | HP-ICES | Centralized | H,W | Transformer, industrial | -- | BLR | BLR | Tank | 2I |
| | Reference | Conventional | H,W | Furnaces | -- | -- | -- | -- | -- |
| R San Antonio | HP-ICES | Distributed | H,C,W | Industrial waste heat | Air | BLR | BLR | Tank | 2U |
| | Reference | Conventional | H,C,W | Furnace | Window air conditioners | -- | -- | -- | -- |
| U Roosevelt Island | Reference | Conventional | H,C,W | Resistance | Chillers[f] | -- | -- | -- | 2+2I[g] |
| | HP-ICES 1 | Centralized | H,C,W | River water | River water[f] | -- | EBLR | Tank | 2+2I[g] |
| | HP-ICES 2 | Centralized | H,C,W | Utility condenser water | River water[f] | -- | EBLR | Tank | 2+2I[g] |
| | HP-ICES 3 | Centralized | H,C,W | River water | River water[f] | -- | EBLR | Ice bin | 2+2I[g] |
| | HP-ICES 4 | Centralized | H,C,W | Utility condenser water | River water[f] | -- | EBLR | Ice bin | 2+2I[g] |
| | Total en. | Central plant | H,C,W | Engine heat recovery | Absorption chillers[f] | -- | -- | -- | 2+2I[g] |
| | Central 1 | Central plant | H,C,W | Boilers | Absorption chillers[f] | -- | -- | -- | 2+2I[g] |
| | Central 2 | Central plant | H,C,W | Boilers | Compression chillers[f] | -- | -- | -- | 2+2I[g] |
| J Hypothetical | HP-ICES | Cascaded | H,C | Freezing ice in storage | Melting ice in storage | ER | -- | Ice pond | 2U |
| | Reference | Conventional | H,C | Air source heat pumps | Air, chillers | -- | -- | -- | -- |

[a]See Figure 1 for study code legend.
[b]H = space heating, C = space cooling, W = service water heating.
[c]BLR = boiler(s), EBLR = electric boiler(s), ER = electrical resistance heating.
[d]Number of distribution pipes, I = insulated, U = uninsulated.
[e]Varies by building.
[f]Some low-income, subsidized housing sections utilize through-the-wall air conditioners.
[g]Insulated two-pipe system for space heating and cooling with a separate, insulated two-pipe system for service hot water.

Table 3 Annual Source Energy Consumption[a]

| Study Code[a] and Community | System | Type of System | Region[c] | Coal (s.tons) | Gas (Mcf) | Oil (bbl) | U-235 (grams) | Hydro ($10^3$ kWh) | Other |
|---|---|---|---|---|---|---|---|---|---|
| A Louisville | HP-ICES | Distributed | East South Central | 10,400 | 8,870 | 1,850 | 67,900 | 4,220 | -- |
| | Reference | Conventional | East South Central | 9,130 | 7,800 | 30,100 | 51,600 | 2,860 | -- |
| A Fort Rucker | HP-ICES | Distributed | East South Central | 2,980 | 2,250 | 556 | 17,500 | 1,100 | -- |
| | Reference | Conventional | East South Central | 3,070 | 25,800 | 713 | 16,300 | 850 | -- |
| C Monroeville | HP-ICES | Centralized | Middle Atlantic | 1,200 | 58,900 | 4,590 | 28,200 | 720 | -- |
| | Reference | Conventional | Middle Atlantic | 1,490 | 63,000 | 5,660 | 35,000 | 893 | -- |
| D Market Sq. | HP-ICES | Centralized | South Atlantic | 2,600 | 7,990 | 5,360 | 37,700 | 531 | -- |
| | Reference | Central plant | South Atlantic | 2,170 | 67,800 | 4,860 | 31,800 | 387 | -- |
| D Park Plaza | HP-ICES | Centralized | New England | 494 | 161,000 | 22,400 | 169,000 | 896 | -- |
| | Reference | Central plant | New England | 760 | 423,000 | 33,000 | 235,000 | 1,440 | -- |
| F Heckman | HP-ICES | Distributed | Middle Atlantic | 366 | 242 | 1,350 | 10,200 | 238 | -- |
| | Reference | Central plant | Middle Atlantic | 65 | 20,300 | 292 | 1,790 | 42 | -- |
| F Walden | HP-ICES | Cascaded | East North Central | 409 | 482 | 3,200 | 2,690 | -- | -- |
| | Reference | Conventional | East North Central | 1,520 | 38,600 | 622 | 9,920 | 33 | -- |
| G Conyers | HP-ICES | Cascaded | South Atlantic | 816 | 2,330 | 1,660 | 11,800 | 178 | -- |
| | Reference | Conventional | South Atlantic | 450 | 24,600 | 1,060 | 6,600 | 78 | -- |
| G Cornelia | HP-ICES | Distributed | South Atlantic | 1,270 | 4,230 | 2,610 | 18,400 | 72 | -- |
| | Reference | Conventional | South Atlantic | 1,710 | 19,600 | 3,530 | 24,600 | 387 | -- |
| H Bloomingdale | HP-ICES | Distributed | East North Central | 17,900 | 20,100 | 6,060 | 117,000 | 392 | -- |
| | Reference | Conventional | East North Central | 12,700 | 325,000 | 4,880 | 84,200 | 282 | -- |
| R Everett | HP-ICES | Centralized | Pacific | 6,870 | 40,100 | 18,900 | 247,000 | 104,000 | -- |
| | Reference | Conventional | Pacific | 15,700 | 44,800 | 249,000 | 234,000 | 93,900 | -- |
| R Seattle | HP-ICES | Centralized | Pacific | 253 | 1,270 | 628 | 6,950 | 3,200 | -- |
| | Reference | Conventional | Pacific | 465 | 1,330 | 7,380 | 7,640 | 2,790 | -- |
| R San Antonio | HP-ICES | Distributed | West South Central | 252 | 103,000 | 1,320 | 3,470 | 351 | -- |
| | Reference | Conventional | West South Central | 169 | 69,700 | 2,840 | 2,280 | 89 | -- |
| U Roosevelt Island | Reference | Conventional | Middle Atlantic | 1,720 | 22,100 | 6,380 | 44,300 | 1,090 | -- |
| | HP-ICES 1 | Centralized | Middle Atlantic | 945 | 21,800 | 3,510 | 23,600 | 592 | -- |
| | Total en. | Central plant | Middle Atlantic | -- | -- | 21,400 | -- | -- | -- |

[a]Source fuel and energy classifications match Form 12.2 of ASHRAE Standard 90C-1977.

[b]See Figure 1 for study code legend.

[c]Geographical regions match Attachment B to ASHRAE Standard 90C-1977.

## FUTURE ACTIVITIES

No future activities are presently planned for this project. Based on the final reports of the individual studies performed on district heating with heat pumps and the system development summary[1] and assessment,[2] the following RD&D efforts are recommended:

Further studies:

- Conduct detailed, site-specific feasibility studies leading to technology demonstrations (the resumption of the HP-ICES Project as originally planned[1] is suggested).
- Develop component and system data bases on the performance and costs of district heating with heat pumps.
- Document experience with existing and new systems.
- Conduct further generic studies to quantify criteria for selecting centralized, distributed, or cascaded systems.
- Investigate requirements and costs for in-building distribution of low-temperature heat (e.g., low-cost panel heat coils and air systems).
- Periodically reexamine all concepts, including those considered unattractive at present, with updated performance and cost data and escalation projections.

Software:

- Expand existing simulation programs to accommodate centralized, distributed, and cascaded HP-ICES, since existing software is limited to in-building systems (e.g., DOE-2 and BLAST) and centralized plants (e.g., OASIS), and lacks capability for practical simulation of complete systems.
- Develop a component simulation library for heat pumps and low temperature distribution (inter- and intra-building).

Hardware:

- Develop optimal, water-source heat pumps for the temperatures encountered.
- Develop low-cost heat meters for single-mode (heating) and dual-mode (heating and cooling) use.
- Continue studies of impacts of temperature modifications and thermal storage in aquifers.
- A prototype demonstration program would accelerate implementation by proving concept viability, enabling acquisition of actual data, and promoting concept awareness (i.e., serving as a showcase). Private sector demonstration is likely, but could be accelerated through indirect or direct subsidies at the federal, state, or local level. The indirect approach would offer risk abatement while the direct approach would offer sharing of expected costs. The assessment report outlines specific demonstration recommendations.[2]

Substantial information on central and district heating systems using heat pumps is available in other countries. Much of this information, which could be of benefit to U.S. conservation efforts, is unpublished, and that which is published is neither readily available in the U.S. nor readily understood because of language barriers. Documentation and evaluation of European experiences with large-building and community heat pump systems are recommended. To do so would require a concerted effort, but would be far less expensive and quicker than duplicating the European efforts. Foreign interest in U.S. advances with heat pump systems is strong. An exchange of findings would be essential to insuring cooperation and may provide markets for export of American products. Therefore, a DOE-sponsored program is recommended to:

- Collect and evaluate data on foreign, particularly European, experiences with heat pump systems for central and district heating.
- Disseminate such information to research, engineering, and manufacturing groups in the U.S.
- Provide a contact for foreign inquiries into related American efforts and developments.
- Establish a continuing information exchange with foreign groups.
- Provide technology on such systems to developing countries, and facilitate their access to U.S. research efforts and suppliers when sought.

## ACKNOWLEDGEMENT

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## REFERENCES

1. Calm, J.M., Heat-Pump-Centered Integrated Community Energy Systems: System Development Summary, Argonne National Laboratory Report ANL/CNSV-7 (February 1980).

2. Calm, J.M., G.R. Sapienza, and N.P. Biederman, Heat-Pump-Centered Integrated Community Energy Systems: System Development Assessment, Argonne National Laboratory Report ANL/CNSV-18 (March 1981).

3. Sapienza, G.R., and J.M. Calm, HPASS, A Computer Program for Evaluation of District Heating with Heat Pumps: Users Manual, Argonne National Laboratory Report ANL/CNSV-TM-63 (March 1981).

## HP-ICES PROJECT PUBLICATIONS

* Baker, N.R., T.D. Donakowski, R.B. Foster, D.L. Sala, R.R. Tison, T.P. Whaley, and B.D. Yudow,

* Documents noted with an asterisk are recommended as primary references; the remaining citations are for background information. Documents indicated as "Available from NTIS" may be ordered from the National Technical Information Service, U.S. Department of Commerce, 5285 Port Royal Road, Springfield, Virginia 22161.

Heat-Pump-Centered Integrated Community Energy Systems, System Development, Consolidated Natural Gas Service Company Final Report, Argonne National Laboratory Report ANL/CNSV-TM-29 (January 1980). Available from NTIS.

* Calm, J.M., G.R. Sapienza, and N.B. Biederman, Heat-Pump-Centered Integrated Community Energy Systems: System Development Assessment, Argonne National Laboratory Report ANL/CNSV-18 (March 1981). Available from NTIS.

Calm, J.M., and G.R. Sapienza, Suitability of the CASES Software for Evaluation of District Heating Systems Based on Annual Storage Cycles, Argonne National Laboratory Report ANL/CNSV-TM-40 (July 1980). Available from NTIS.

Calm, J.M., and G.R. Sapienza, Site and Source Energy Savings for District Heating and Cooling with Heat Pumps, Proceedings of the 5th Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, pages II.1-II.10 (April 1980).

* Calm, J.M., Heat-Pump-Centered Integrated Community Energy Systems: System Development Summary, Argonne National Laboratory Report ANL/CNSV-7 (February 1980). Available from NTIS.

Calm, J.M., Conservation Potential of Heat-Pump-Centered District Heating and Cooling, Proceedings of the 2nd International Conference on Energy Use Management, Los Angeles, California, Paper #F.1-3, Volume III, pages 953-960 (October 1979).

Calm, J.M., Heat Pump Systems for District Heating and Cooling, ASHRAE Journal, 21(9):54-58 (September 1979); reprinted in Ground Water Heat Pump Journal, 1(4):16-21 (Winter 1980).

Calm, J.M., and P.T. Bauer, District Heating and Cooling with Heat Pump Systems, Proceedings of the 14th Intersociety Energy Conversion Engineering Conference, Boston, Massachusetts, Paper #799357, Volume II, pages 1681-1686 (August 1979).

Calm, J.M., and G.R. Sapienza, Integrated Community Energy Systems Engineering Analysis and Design Bibliography, Argonne National Laboratory Report ANL/CNSV-2 (May 1979). Available from NTIS.

Calm, J.M., Community Heating and Cooling Systems, Proceedings of the 4th Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, pages III.1-III.8 (April 1979). Calm, J.M., and J.J. Roberts, Heat-Pump-Centered Integrated Community Energy Systems, Proceedings of the 3rd National Conference on Technology for Energy Conservation, Tucson, Arizona, Paper #XB-2, pages 399-404 (January 1979); reprinted in Sun Up Energy News Digest, 4(2):10,15 (February 1980) and 4(3):19 (March 1980).

Calm, J.M., Recovery of Wasted Heat with Centralized and Distributed Heat Pump Systems, American Society of Mechanical Engineers Paper #78-WA/HT-63 (August 1978); presented at the ASME Winter Annual Meeting, San Francisco, California (December 1978).

Crane, R.E., H.G. Lorsch, and R.G. Werden, Two Applications for Heat Pump Centered Integrated Community Energy Systems (HP-ICES), Proceedings of the 7th Energy Technology Conference, Washington, District of Columbia, pages 522-534 (June 1980).

Crane, R.E., H.G. Lorsch, and R.G. Werden, Integrated Community Energy System Based on Existing Heat Pump Technology, Proceedings of the 5th Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, pages I.1-I.6 (April 1980).

Crane, R.E., H.G. Lorsch, and R.G. Werden, A Diesel-Driven Heat Pump Centered Integrated Community Energy System, Transactions of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers 1980 Semiannual Meeting, Los Angeles, California, Paper #LA-80-12-5 (February 1980).

* Crane, R.E., and R.G. Werden, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Franklin Research Center Final Report, Argonne National Laboratory Report ANL/CNSV-TM-24 (December 1979). Available from NTIS.

Crane, R.E., H.G. Lorsch, and R.G. Werden, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Franklin Research Center Interim Report, Argonne National Laboratory Report ANL/ICES-TM-26 (February 1979). Available from NTIS.

Dubin, F.S., A. Halfon, and P. Herzog, The Heat Pump Centered Integrated Community Energy System (HP-ICES), Transactions of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers 1980 Semiannual Meeting, Los Angeles, California, Paper #LA-80-12-4 (February 1980).

* Dubin, F.S., A. Halfon, and P. Herzog, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Dubin-Bloome Associates Final Report, Argonne National Laboratory Report ANL/CNSV-TM-8 (August 1979). Available from NTIS.

Dubin, F.S., A. Halfon, and P. Herzog, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Dubin-Bloome Associates Interim Report, Argonne National Laboratory Report ANL/ICES-TM-29 (February 1979). Available from NTIS.

* Honeywell Incorporated, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Honeywell Energy Resources Center Final Report, Argonne National Laboratory Report ANL/CNSV-TM-18 (November 1979). Available from NTIS.

Honeywell Incorporated, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Honeywell Energy Resources Center Interim Report, Argonne National Laboratory Report ANL/ICES-TM-31 (March 1979). Available from NTIS.

Katter, L.B., A Community Energy System Supplied from Waste Industrial Energy Through Heat Pumps, Transactions of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers 1980 Semiannual Meeting, Los Angeles, California, Paper #LA-80-12-1 (February 1980).

* Rocket Research Company, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Rocket Research Company Final Report, Argonne National Laboratory Report ANL/CNSV-TM-20 (November 1979). Available from NTIS.

Rocket Research Company, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Rocket Research Company Interim Report, Argonne National Laboratory Report ANL/ICES-TM-27 (February 1979). Available from NTIS.

Sapienza, G.R., and J.M. Calm, HPASS, A Computer Program for Evaluation of District Heating with Heat Pumps: Users Manual, Argonne National Laboratory Report ANL/CNSV-TM-63 (March 1981). Available from NTIS.

Schaetzle, W.J., C.E. Brett, and D. Grubbs, A Heat Pump

Integrated Community Energy System Using Annual Aquifer Energy Storage, Transactions of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers 1980 Semiannual Meeting, Los Angeles, California, Paper #LA-80-12-2 (February 1980).

* Schaetzle, W.J., C.E. Brett, and M. Seppanen, Heat-Pump-Centered Integrated Community Energy Systems, System Development, University of Alabama Final Report, Argonne National Laboratory Report ANL/CNSV-TM-25 (December 1979). Available from NTIS.

Schaetzle, W.J., C.E. Brett, and D.M. Grubbs, Community Integrated Cooling and Heating Systems, Proceedings of 2nd International Conference on Energy Use Management, Los Angeles, California, Volume II, pages 985-992 (October 1979).

Schaetzle, W.J., C.E. Brett, and D.M. Grubbs, Annual Energy Storage in Aquifers for Community Energy Systems, Proceedings of the International Association of Science and Technology for Development Symposium, ACTA Press, Zurich, Switzerland (June 1979).

Schaetzle, W.J., and C.E. Brett, Heat-Pump-Centered Integrated Community Energy Systems, System Development, University of Alabama Interim Report, Argonne National Laboratory Report ANL/ICES-TM-30 (March 1979). Available from NTIS.

Tison, R.R., N.R. Baker, B.D. Yudow, D.L. Sala, T.D. Donakowski, and P.F. Swenson, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Consolidated Natural Gas Service Company Interim Report, Argonne National Laboratory Report ANL/CNSV-TM-7 (August 1979). Available from NTIS.

Trammell, B.C., Wastewater Heat Recovery for Heat Pump Centered Integrated Community Energy Systems, Presented at the 2nd International Conference on Energy Use Management, Los Angeles, California, Volume II, pages 1009-1016 (October 1979).

Wade, D.W., Heat Pump Centered Heating and Cooling Utilizing Wastewater Heat Recovery, Transactions of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers 1980 Semiannual Meeting, Los Angeles, California, Paper #LA-80-12-3 (February 1980).

* Wade, D.W., B.C. Trammell, and B.S. Dixit, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Georgia Institute of Technology Final Report, Argonne National Laboratory Report ANL/CNSV-TM-23 (December 1979). Available from NTIS.

Wade, D.W., Wastewater Heat Recovery Schemes for Heat-Pump-Centered Integrated Community Energy Systems, Proceedings of the 4th Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, pages VI.1-VI.8 (April 1979).

Wade, D.W., B.C. Trammel, B.S. Dixit, D.C. McCurry, and B.A. Rindt, Heat-Pump-Centered Integrated Community Energy Systems, System Development, Georgia Institute of Technology Interim Report, Argonne National Laboratory Report ANL/ICES-TM-28 (February 1979). Available from NTIS.

Yudow, B.D., N.R. Baker, R.R. Tison, and P.F. Swenson, Feasibility of a Heat-Actuated Heat-Pump-Centered Integrated Community Energy System, Proceedings of the 14th Intersociety Energy Conversion Engineering Conference, Boston, Massachusetts, Paper #799358, Volume II, pages 1687-1692 (August 1979).

# PERFORMANCE RATING OF HEAT PUMPS

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PAPER NOT AVAILABLE

STEADY-STATE PERFORMANCE CHARACTERISTICS OF
LATENT HEAT TES/HEAT PUMP SYSTEMS

RESEARCH TRIANGLE INSTITUTE

CONTRACT NUMBER: 62B:13835C
DE-AC-01-79ET-26707

TONY W. SIGMON

PERIOD OF CONTRACT: Begin 7/79 End 12/81

## OBJECTIVE

The objective of the contracted efforts is to determine the relative viability of various coupled latent heat thermal energy storage (TES) heat pump systems for residential space heating and cooling applications and to describe the sensitivity of their relative viability to selected geographic, economic, and system performance parameters.

## CONCEPT OF BACKGROUND

The combination of heat pump systems and TES have the advantage of: (1) providing a means of substituting lower cost thermal energy during heating operation for that which is normally supplied by resistance heaters, (2) improving the efficiency of heat pump systems during both heating and cooling operation, and (3) decreasing heat pump operating cost by allowing the heat pump to operate primarily during periods in which low cost electricity is provided under a time-of-day (TOD) electricity rate structure. Although the focus of a significant amount of previous work has been directed towards the consideration of heat pumps combined with sensible heat storage subsystems, the advantages associated with latent heat TES have precipitated research and development efforts which consider phase change systems as well. Latent heat TES subsystems, in general, store more energy per unit volume than do sensible heat systems, allow for storage at a constant temperature or over a small temperature interval, and do not require the additional internal heat exchangers needed by many sensible heat TES subsystems. Because of these advantages, numerous analyses have been completed in which the technical and economic advantages of latent heat TES were explored. With the increased interest that is being shown by both public and private sectors in the application of TES, various manufacturing concerns are beginning serious marketing efforts aimed at pushing TES systems into the marketplace.

The work presented here is part of a program of work that has been sponsored by the Department of Energy through the Division of Energy Storage. The ultimate objective of this work is to provide an evaluation of the operational performance of alternative TES/heat pump designs under a range of application constraints. This will allow for the identification of those systems that are projected to be most cost effective under a specified application scenario.

## SUMMARY

Two current projects are currently being completed that wholly or in part address various technical issues involved in the implementation of heat pump systems combined with TES.

The first of these involves the determination of steady state performance characteristics for six generic TES/heat pump configurations and the comparison of the operational performance of these systems with other space heating and cooling TES technologies. Of these latter systems four are commercial or near commercial air conditioner or heat pump coupled TES systems. Steady state performance has been established for all systems. Operational performance and system life cycle cost has been determined for the six generic designs for a limited set of application conditions. Operational performance is being established through use of a computer simulation which utilizes a control algorithm to identify appropriate modes of operation under a time-of-day (TOD) electric utility rate structure. This software package is presently being used to compute operational performance for the other systems considered.

The intent of the second project is to establish a reliable method of estimating seasonal energy use by TES/heat pump systems, to utilize this methodology to evaluate a large number of possible system designs, identify a small number of systems that merit more detailed analysis, and, to the extent possible, conduct these detailed studies. The approach developed to evaluate performance is a modification of the bin analysis approach used to estimate energy use by conventional heat pump systems. The software to be used in establishing energy use using the bin technique has been developed and is presently being applied to a number of system designs.

## TECHNICAL ACCOMPLISHMENTS

• TES Subsystem Model: The TES subsystem considered is of a rectangular design and consists of alternate layers of storage material and refrigerant flow passages as shown in Fig. 1. The TES material is contained within individual storage modules which are in contact with the refrigerant circuit. Refrigerant enters the single circuit refrigerant line at the top refrigerant layer, runs the length and width of this layer then enters the next lowest layer and continues until the refrigerant eventually exits at the bottom refrigerant layer.

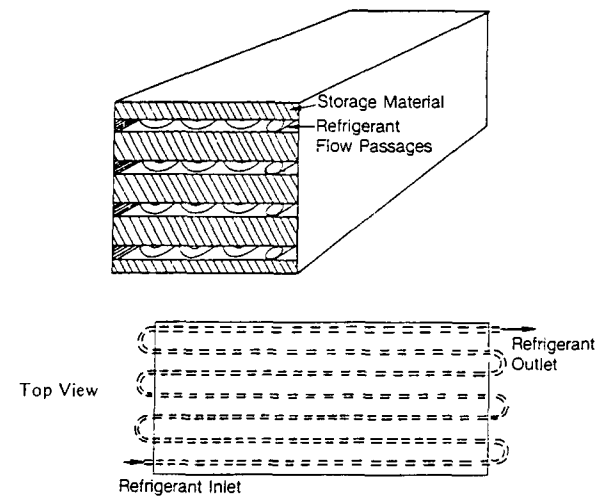


Figure 1. TES Subsystem Design

The TES subsystem is modeled by dividing the subsystem into a subelement as shown in Fig. 2. The width of the subelement is the center-to-center spacing of the refrigerant line, while the length is the total length of the refrigerant circuit. In order to achieve symmetry, the subelement is assumed to consist of two half thicknesses of a storage material layer surrounding an equivalent refrigerant passage. The height of this equivalent passage has been found by determining the dimensions of a square that has the same cross-sectional area as the circular refrigerant passage.

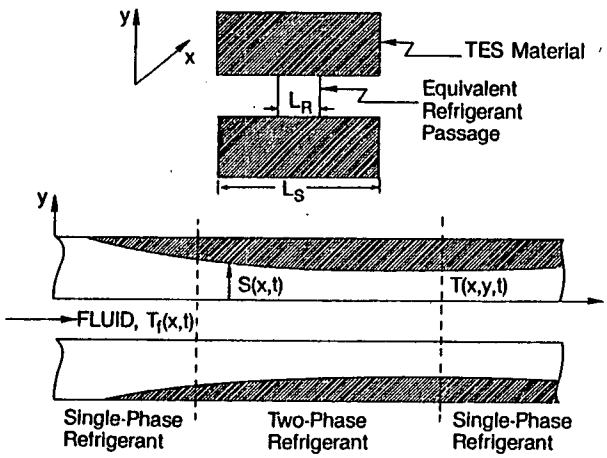


Figure 2. TES Subelement Geometry

The one dimensional heat conduction equation was solved for the melt/freeze process taking place within the TES material as a result of convective heat transfer at the refrigerant/TES material boundary.

These equations have been solved numerically for the case when the wall temperature $T_w$ is a function of only the streamwise variable x. The resulting solutions were then used to determine a total heat transfer rate given inlet conditions. The length of the subelement is divided into three distinct sections. In the entrance region the refrigerant is either a single phase liquid or vapor; a two-phase region follows in which the refrigerant is either condensing or evaporating, followed by another single phase region in which the refrigerant is either subcooled for the condensing case or superheated when the subsystem acts as an evaporator. Heat transfer characteristics have also been determined for the ease of air circulation through the subsystem.

- System Steady State Performance: Steady state performance characteristics, i.e., capacity and input power required to drive the system during each mode of operation, have been computed for six generic TES/heat pump system designs and of a conventional air-to-air heat pump system.

Table 1 provides information relative to those configuration types considered in this previous study. Each of the six designs is charged indirectly within the refrigeration cycle of the heat pump and is discharged either indirectly within the refrigeration cycle or directly by circulating return air through the subsystem. Each conceptual design was studied for only one storage material. Steady state performance characteristics of each system were determined for nominal 3-, 4-, and 5-ton air-to-air heat pumps coupled with a latent heat TES subsystem with total design storage capacities of 100, 200, 300, 400, and 500 K Btu. The conventional heat pump consisted of four primary components: two air-to-air heat exchangers, compressor, and capillary tube. The TES subsystem was of a rectangular design and had the capability for both refrigerant/storage material and air/storage material heat transfer as discussed previously.

Table 1. Baseline and TES/Heat Pump Systems

| System | Heating Mode Charging/Discharging | Cooling Mode Charging/Discharging |
|---|---|---|
| Base | Heat Pump[a] | Heat Pump |
| L1 | Indirect[b]/Direct[c] | Heat Pump |
| L2 | Indirect/Indirect | Heat Pump |
| L3 | Heat Pump | Indirect/Direct |
| L4 | Heat Pump | Indirect/Indirect |
| L5 | Indirect/Indirect | Indirect/Direct |
| L6 | Indirect/Indirect | Indirect/Indirect |

[a]Nonstorage mode

[b]Refrigerant storage material heat transfer

[c]Air/storage material heat transfer

A computer simulation of the performance of conventional air-to-air heat pump systems has served as the framework for the computation of steady state performance characteristics. The initial version of this simulation was developed at the Massachusetts Institute of Technology for the primary purpose of determining the change in heat pump performance as a result of modifications in compressor design and operation. This work has since been modified by individuals at Oak Ridge National Laboratory (Ellison, 1978) and has been used as a starting point for this analysis.

The simulation was used to compute system thermal capacity and power requirements by determining heat transfer and refrigerant mass flow balance conditions within the refrigeration cycle for each mode of operation. The computational scheme used to determine these equilibrium values is based upon iterating over refrigerant states until a heat transfer/mass flow balance point is reached for a specific set of source/sink conditions. Each system is described at the component level using manufacturer's performance data for the four primary heat pump components and a detailed heat transfer model of TES subsystem performance developed by the authors.

For conventional air-to-air heat pump performance, the refrigeration cycle has been simulated using standard components from which heat pump heating or cooling capacity and system input power are determined for an appropriate range of ambient temperature and a specified set of indoor conditions. Refrigerant (R-22) properties are allowed to vary with temperature and pressure and are computed at various locations within the refrigeration cycle.

Performance characteristics of each system during charging and discharging operation are found in a similar manner to that used for conventional heat pump operation. In these cases, however, the TES subsystem model takes the place of the indoor heat exchanger during charging and replaces the outdoor coil during indirect discharge. Direct discharge performance is determined using an air/storage material heat transfer model developed for

the TES subsystem and, therefore, does not involve the refrigeration cycle of the heat pump.

· Operational Analysis: Annual system operational performance for these six generic designs was determined for six simulation sites by interfacing building thermal loads computed using TRNSYS with steady state performance data. A system control strategy subroutine defined the operating modes for the TES/heat pump systems as a function of various independent parameters for each system. The TES systems were controlled in a manner so as to maximize the shift in energy use and demand from utility defined on-peak to off-peak hours while the basecase heat pump was allowed to operate in the conventional manner. The electric utility rate structure was input to the control algorithm to identify the time periods appropriate for each mode of operation.

Each TES/heat pump system considered was allowed to charge only during the off-peak period and only by utilizing the TES subsystem as an energy source or sink for the heat pump cycle. Therefore, charging could only take place when the heat pump was not in operation to satisfy the building thermal load. The systems could be discharged depending upon whether the system discharged directly or indirectly. For indirect systems the compressor is utilized to discharge; therefore, space heating and cooling could be provided either by discharging the subsystem or by conventional heat pump operation. Direct systems, on the other hand, were allowed to discharge in parallel with conventional heat pump operation since compressor operation was not required during discharge. The output of the control simulation consisted of the following for each hourly interval:

- Day of the year,
- Designation of heating or cooling season,
- Hour of the day,
- Designation of off- or on-peak period,
- Building thermal load, and
- System thermal output electrical demand and utilization factor for each mode of operation.

Each system is designated as having the capability to provide for either storage heating and/or cooling as an input to the control algorithm. In those cases when the system configuration does not allow for storage for space heating or cooling the heat pump is expected to operate in the conventional manner to supply the needed capacity.

· System Installed Costs: Conceptual cost estimates for the various TES and baseline HVAC equipment have been generated to be used along with results of the system simulations of technical performance to identify cost-effective storage alternatives. An equipment list has been specified for each TES/HVAC system in order that a total system cost can be estimated. Using these data, it has been possible to estimate the cost of TES/HVAC systems as a function of size of storage capacity parameters.

Cost estimates of any energy technology are seldom unique and can vary over a range of values according to labor rates, geographic location, installation factors, and other parameters. Two levels of cost detail were generally used: (1) conventional HVAC and TES equipment were costed as packaged units, as available from commercial manufacturing firms and (2) TES equipment was costed in terms of its components when necessary, especially for prototype systems.

A consistent methodology was adopted for cost estimating so that a comparative cost analysis could be conducted. Cost estimates have been presented at the "retail" level, which includes: (1) a markup on material of 15 to 25 percent (average 20 percent) for the manufacturer's margin, (2) a 10 percent markup on equipment for the distribution of products to the vendor, and (3) an estimate of 50 percent of total equipment cost for contractor installation of TES equipment. These factors were applied uniformly across all cost data even though it was obtained, and a revised estimate was made to be consistent with the above general assumptions.

· Economic Analysis: The economic performance of each system has been based upon the energy costs associated with the operation of the system and a total annualized life cycle cost associated with the purchase and operation of the system. Each of these criteria is a significant function of the electric utility rate structure under which the system must operate. The demand, energy, and customer charge components each play an important part in the projected cost effectiveness of each system. Through this analysis, as in the development of energy use characteristics, the primary scope of activity involved comparing systems operating at the same climatic location.

· Bin Analysis: The bin method of estimating energy requirements of HVAC systems as outlined in the ASHRAE Systems Handbook is a relatively accurate method of computing annual and seasonal energy consumption of HVAC equipment, such as heat pumps whose performance is extremely dependent on outdoor air temperature, without resorting to the more sophisticated procedure of computing energy use patterns on a sequential hour by hour basis.

The bin method was modified to estimate and compare energy consumption of the heat pump and TES systems for both the load leveling and load shifting operational strategies. Changes were made to consider differences in seasonal as well as on- and off-peak operation of the systems including charging and discharging of the storage system.

To account for the differences in seasonal on- and off-peak operation of the TES systems, the temperature bins were considered in 5° F intervals as with the conventional bin method but were additionally divided into summer and winter on- and off-peak periods as defined by the rate structure considered. This further division was necessary both to compute energy costs based on a TOD rate schedule and to differentiate between the operation of the TES systems during the different periods.

Steady state performance data necessary to the bin analysis computations for both the TES/heat pump and conventional heat pump systems have been determined using previously computed results and newly computed characteristics of those systems and modes of operation that had not been considered prior to this analysis.

## FUTURE ACTIVITIES

Near term activites to be completed include utilization of the bin analysis software to analyze operational performance of a large number of TES/heat pump conceptual designs and the determination of the annual cost associated with the purchase and operation of these systems. These results will then be used to identify those system types that justify more detailed analysis.

These detailed studies will then be conducted utilizing an improved TES subsystem heat transfer model that accounts for the natural convection heat transfer that takes place within the phase change material during charging or discharging and more detailed representations of the other major system components.

PUBLICATIONS/REPORTS/REFERENCES


Davidson, J. H., et al. "Thermal Energy Storage Systems Descriptions and Steady State Performance Characteristics." Draft Report submitted to Department of Energy. Conducted under Contract Number DE-AC-01-79ET-26707, March 1981.

Ellison, R. D. and F. A. Creswick. "A Computer Simulation of Steady-State Performance of Air-to-Air Heat Pumps." Oak Ridge National Laboratory Report No. ORNL/CON-16, March 1978.

Ferrell, G. C., et al. "Capital Cost Data for Thermal Energy Storage and HVAC Systems." Draft Report submitted to Department of Energy. Conducted under Contract Number DE-AC-01-79ET-26707, March 1981.

Sigmon, T. W., J. H. Davidson and D. H. Abbott. "Modified Bin Analysis for TES/Heat Pump Systems." In preparation. Conducted under Contract Number 62B:13835C.

Sigmon, T. W., J. H. Davidson and J. M. Doster. "The Effect of Storage Material Melt Temperature on Coupled TES/Heat Pump System Performance for Space Heating and Cooling Applications." Presented at the 3rd Miami International Conference on Alternative Energy Sources, Miami, Florida, December 1980.

Sigmon, T. W., J. H. Davidson, J. A. Edwards, et al. "Simulation and Evaluation of Latent Heat Thermal Energy Storage Heat Pump Systems." Presented at the 15th Intersociety Energy Conversion Engineering Conference, Seattle, Washington, August 1980.

Sigmon, T. W., et al. "Simulation and Evaluation of Latent Heat Thermal Energy Storage Heat Pump Systems." Draft Task Report submitted to Department of Energy. Conducted under Contract DE-AC-01-79ET-26707, March 1981.

# COMPUTER AIDED HEAT PUMP DESIGN AND OPTIMIZATION*

OAK RIDGE NATIONAL LABORATORY
CONTRACT NUMBER W-7405-eng-26
S. K. FISCHER AND C. K. RICE

## OBJECTIVE

The objectives of this project are to develop and demonstrate analytical methods to predict the performance of existing and proposed heat pumps; to develop and demonstrate computer programs to serve as tools for the design and optimization of high efficiency heat pumps.

## BACKGROUND

The MIT heat pump model (1) served as a starting point in the development of analytical methods for heat pumps at ORNL. The MIT program was modified for our use and documented (2, 3) as a preliminary version of the ORNL heat pump model. An improved version of the program was made available to other users in 1979. Copies of the computer program have since been sent to about 30 requestors for use on their computers. Our own use of the program in our optimization studies as well as the experience of other users has revealed the need for additional capabilities in the model. New capabilities were added to the program during the past year, using correlations from the literature, when they were available, or correlations developed at ORNL. These additions have improved the model significantly and extended its applicability. The result is a second generation of the computerized model, one that can be used by engineers who are not necessarily computer experts.

At ORNL, the heat pump model was used in an attempt to predict the increased heat pump efficiency that could be expected from the use of improved components, and to explore the practical limits of efficiency. It soon became obvious that a systematic approach is required to obtain consistent and credible results. The heat pump model was connected to an optimizing program; the combined programs were used to vary, simultaneously and independently, selected design parameters in such a way as to maximize the predicted heat pump efficiency, and thus assure maximum benefit from improved components.

## SUMMARY

A computer model of electric-motor-driven heat pumps was developed to provide detailed analyses of the performance of existing or proposed heat pumps. To the extent possible, the model is based on the physical principles underlying the processes and on basic correlations, with minimal use of empirical correlations. This approach was adopted so as to give the model maximum flexibility; to allow its use for analyzing design concepts that depart from current practice. The computer program is organized to facilitate its use by engineers who want to concentrate on a heat pump design, not a computer program.

The heat pump model was connected to an optimization program in order to maximize the predicted heating efficiency by simultaneously varying ten design parameters to find the combination giving the highest COP at a specified heating capacity. The program we used is based on the steepest ascent algorithm with refinements which allowed us to specify a minimally acceptable supply air temperature and an upper limit on the total heat exchanger area, as well as holding the heating capacity constant. We used this combined code to find the designs with the highest COPs for a series of system improvements: increasing compressor efficiencies, fan efficiencies, and total heat exchanger area.

We used the specifications for a heat pump tested in our laboratory (i.e., heat exchanger descriptions, air flow rates, compressor displacement, etc.) as input data for a base case system for validation and comparison purposes. When ten design variables were optimized for this system, there was a 21% improvement in predicted heating efficiency. Assumed increases of 17% and 34% in overall compressor efficiency (from 47% to 55% and 63%) led to 11% and 22% additional improvements in maximum predicted COP, respectively. It is noteworthy that the optimum system with the 63% compressor efficiency had a total displacement 20% smaller than our base case with a required shaft power of 2.24 kw (3.0 hp) rather than 3.36 (4.5 hp).

This analysis showed us the optimal designs and some trends from one level of improvement to the next. It did not, however, give us any indication of the sensitivity of efficiency to variations from the optimal design. We used the heat pump model to do parametric studies to show the flexibility or latitude that exists about the optimum for selecting system components. This analysis is displayed graphically as plots of contours of COP for wide ranges of design variables, two at a time, about their optimal values.

## TECHNICAL ACCOMPLISHMENTS

Developed a practical, validated design tool for heat pumps and air conditioners. The heat pump computer model was significantly improved from the program reported in 1978 (2). New capabilities were added, engineering methods and correlations were improved, and the program was made significantly easier to use.

It requires as inputs:

- geometric description of the heat exchangers and interconnecting pipes;
- a specification of the flow control device (or as an option, the desired condenser subcooling);
- the desired suction line superheat;
- the air temperatures at the entries to the heat exchangers;
- estimates of the condensing and evaporating temperatures and refrigerant mass flow rate (to get the calculations started); and
- data for either the efficiency and loss or map-based compressor model.

The model can be used to calculate:

- heating and cooling capacity, latent and sensible, and COP;

*This research was sponsored by the Office of Buildings and Community Systems, U.S. Department of Energy, under contract W-7405-eng-26 with the Union Carbide Corporation.

- compressor and fan motor power consumptions;
- refrigerant temperatures and pressures throughout the circuit and the mass flow rate;
- air-side temperature, pressure, and humidity changes; and
- compressor efficiency and heat exchanger effectiveness at the conditions specified.

The user has the option to specify:

- a map-based or an efficiency and loss compressor submodel;
- draw through or blow through fans;
- motor efficiency curves;
- any of eleven refrigerants;
- correlations for smooth or wavy fins; and
- thermostatic expansion valve (TXV), capillary tube, or fixed condenser subcooling to control refrigerant flow (in the latter case the TXV and capillary tube parameters necessary to achieve the desired level of subcooling are computed and printed).

The model is being validated against data collected in our laboratory and is being documented. It will be available for distribution this summer when these two tasks are completed.

Developed and demonstrated a technique for heat pump design optimization. We demonstrated the use of heat pump design techniques to find the system configuration with the highest predicted efficiency while satisfying constraints due to engineering considerations, and those chosen to limit component costs or sizes. This method allows engineers to design systems which obtain the maximum possible benefit from each improvement in individual components. We have plotted steady-state heating COP (at the ARI high temperature rating point) in Figure 1 against a normalized scale for total heat exchanger area. A unit which was studied in our laboratory (4) has been used as a base case and arbitrarily assigned a relative heat exchanger area of 1.0. Several points and three curves representing improved systems have been plotted with respect to the base case and labelled with their overall compressor efficiencies. Two of the individual points are commercially available, state-of-the-art heat pump systems and one is the predicted maximum COP attainable by our base case heat pump, if the system were optimized. The nine points that were used to draw the three curves were obtained by substituting high efficiency fans for those in our base case unit and optimizing the configurations for three levels of heat exchanger area and three compressor efficiencies. Figure 1 shows an increase in predicted COP for our base case from 2.4 (labeled with a 1) to a maximum of 2.9 (labeled with a 2) by finding the optimum combination of the design variables, but without increasing any component efficiencies or total heat exchanger area or decreasing heating capacity. This figure also shows the fairly uniform increases in predicted COP resulting from 17% to 34% improvements in overall compressor efficiency (from 47% to 55% and 63%, respectively). Another conclusion that can be drawn from the figure is the decreasing improvement in system operation resulting from increasing levels of total heat exchanger area. Doubling the total heat exchanger area yields an increase of 15% in predicted COP over the optimum configurations with smaller condensers and evaporators for all three levels of compressor efficiency. The relative heat exchanger area, however, must be quadrupled to give a 30% increase in COP.

Devised a technique for evaluating design tradeoffs. We demonstrated a means of displaying design flexibility by using the heat pump model to plot lines of constant COP and capacity over wide ranges of design variables. Figure 2 shows how the predicted COP and capacity change as the condenser and evaporator air flow rates vary about the values found by the optimization program. As may be seen from the figure, condenser air flow rate may be varied from about 1300 to 1900 cfm, providing the evaporator air flow rate is correspondingly adjusted, without changing the capacity more than 2% or reducing the COP more than 1.5%. Plots of this nature

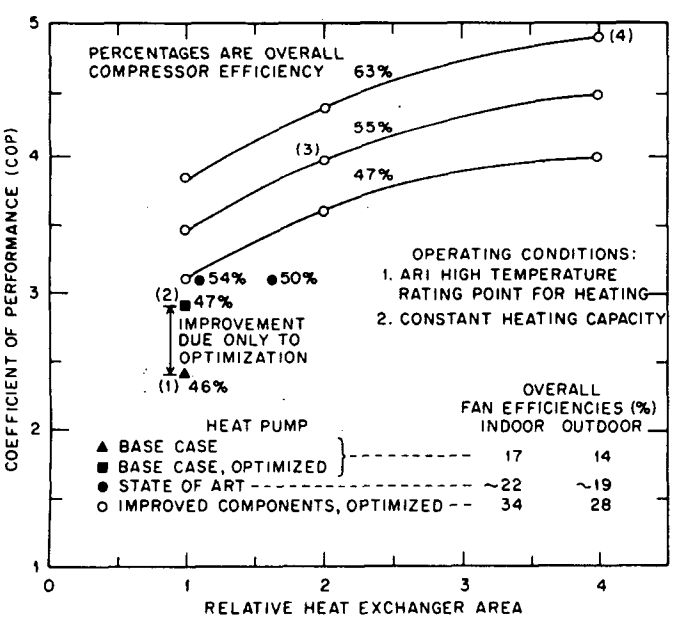


Fig. 1 Effects of compressor efficiency, heat exchanger area, and design optimization on heat pump COP.

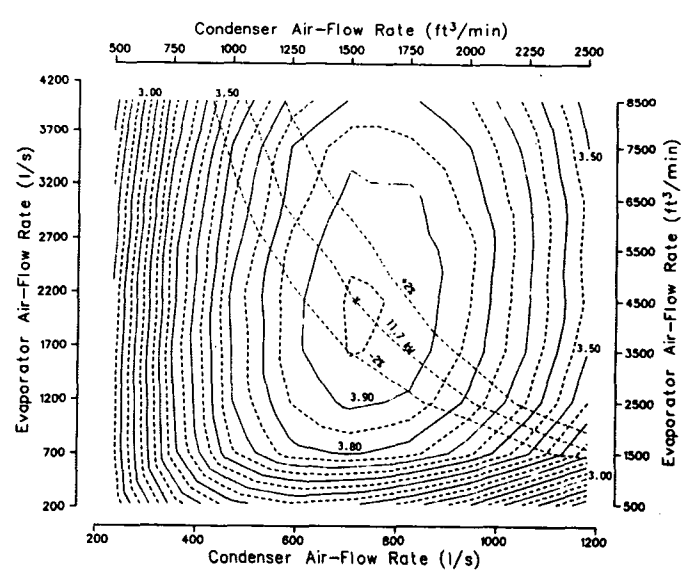


Fig. 2 Sensitivity of COP and heating capacity to air-flow rates at 47 F.

can be used to analyze the sensitivity of COP to variations about the optimum design and how design parameters can be adjusted simultaneously to maintain high efficiency and desired capacity.

Showed that optimum design is more important in high performance heat pumps. We found that good system design becomes much more critical as system efficiency increases. Figure 3 shows how the maximum predicted COP varies as the total compressor displacement changes for several levels of system improvements. The lowest curve was produced by

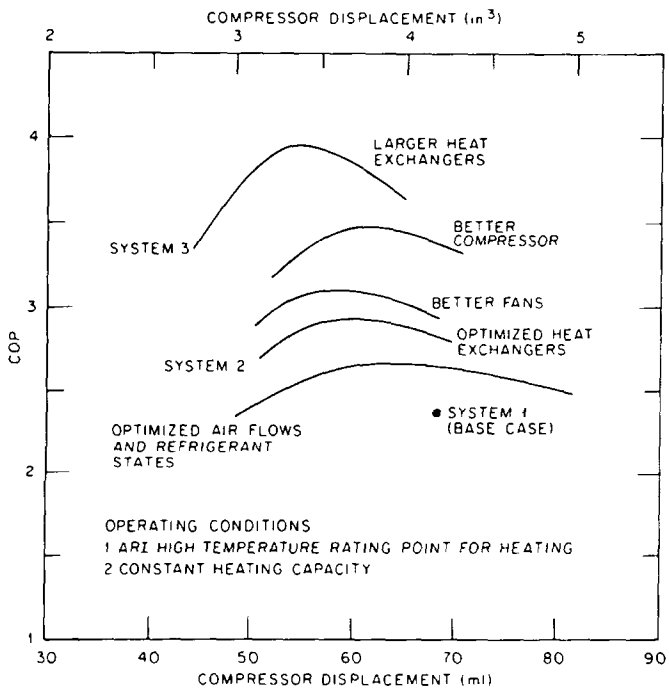


Fig. 3 COP vs compressor displacement for a series of successive system improvements.

holding the geometry of the heat exchangers fixed at our base case design while the COP was maximized for each of several compressor displacement values by finding the optimum air flow rates and condenser subcooling (the heating capacity was held constant). The other curves are all the results of finding the maximum COPs after optimizing the heat exchangers and adding additional levels of system improvement. It is noteworthy that the predicted COP is substantially increased from the base case and the compressor size reduced by 20% while the heat pump capacity remains constant. (Though it is not shown on these curves, the required motor shaft power is also reduced — from 3.36 (4.5 hp) to 2.24 kw (3.0 hp) — and thus, a smaller motor is acceptable). The most important point, however, is that as each successive component improvement is made, the system efficiency curve has a sharper peak and the proper sizing of the compressor is increasingly important. Similar analysis could be done with any of the design parameters on the horizontal axis.

## FUTURE ACTIVITIES

Contract activities. The ORNL Heat Pump Design Model will be maintained. Minor changes to the program will be considered if the demand for new capabilities is strong and as new engineering methods and correlations become available.

It is anticipated that the optimizing technique will be applied to single-speed heat pumps for both heating and cooling performance and to capacity modulating systems. Methods will be developed to optimize performance for best seasonal performance instead of steady-state efficiency. Heat pump designs that produce maximum efficiency and are cost-competitive will be sought.

Post-contract activities. Not applicable.

## REFERENCES


1. C. C. Hiller and L. R. Glicksman, Improving Heat Pump Performance via Compressor Capacity Control-Analysis and Test, Report No. 24525.96, Heat Transfer Laboratory, Massachusetts Institute of Technology (1976).

2. R. D. Ellison and F. A. Creswick, A Computer Simulation of Steady-State Performance of Air-to-Air Heat Pumps, ORNL/CON-16 (March 1978).

3. R. D. Ellison, et al., "Heat Pump Modeling: A Progress Report," Proceedings of the Fourth Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, April 9-10, 1979.

4. A. A. Domingorena, Performance Evaluation of a Low-First-Cost, Three-Ton, Air-to-Air Heat Pump in the Heating Mode, ORNL/CON-17 (October 1978).


## PUBLICATIONS


1. C. K. Rice, S. K. Fischer, et al., "Design Optimization of Conventional Heat Pumps: Application to Steady-State Heating Efficiency," ASHRAE Transactions 1981, Vol. 87, Pt. 1.

2. C. K. Rice, S. K. Fischer, et al., Design Optimization and the Limits of Steady-State Heating Efficiency for Conventional Single-Speed Air-Source Heat Pumps, ORNL/CON-63, in preparation.

3. S. K. Fischer and C. K. Rice, Heat Pump Design and Simulation Model (working title), in preparation.

LOW TEMPERATURE WASTE HEAT UTILIZATION CHILLER SYSTEM
FOSTER-MILLER ASSOCIATES, INC.
41X-28906C
DR. SCOTT HYNEK
8/20/79 - 6/30/82

OBJECTIVE

To design, develop, and demonstrate an industrial refrigeration system to be powered primarily by the 140F (70C) waste hot water that is available at many industrial sites as well as government owned gaseous diffusion plants. This system should maximize the ratio of refrigeration capacity to capital cost.

To investigate commercial applications of this system and to commercialize it by teaming with a manufacturer.

BACKGROUND

As the cost of energy continues to rise faster than the cost of machinery, the minimum temperature for economically recoverable waste heat continues to decrease. The Department of Energy, anticipating the day when 140°F water constitutes an exploitable resource, is funding, through the Oak Ridge National Laboratory, the development of several systems to convert 140°F water, which exists in abundance at government-operated gaseous diffusion plants, into process steam or industrial refrigeration. Foster-Miller Associates, Inc. (FMA) has designed and is developing one such system, using an organic Rankine cycle to drive a reverse Rankine refrigeration cycle.

Figure 1 shows how these two cycles are integrated schematically. Both cycles reject heat through the same evaporative condenser, because two smaller condensers with the same cumulative capacity would cost more. Both cycles necessarily use the same working fluid; R-22 was chosen over R-12 because it affords greater capacity per unit displacement in both the expander and the compressor; R-717 was rejected because it is incompatible with the copper windings of the hermetic motor/generator.

Figure 2 shows how these two cycles are integrated mechanically. The expander, compressor, boiler feed pump, oil pump, and motor/generator are all contained in a semi-hermetic enclosure. This eliminates exposed shafts and any associated leakage. It also minimizes the number, required effectiveness, and friction of internal shaft seals. The expander and compressor share a crankcase, crankshaft, and lubrication system; this is accomplished by transforming two cylinders of a four-cylinder refrigeration compressor into expander cylinders.

Figure 3 shows how these same two cycles can be integrated schematically as a heat pump. The concept of using the same fluid for both cycles is less appropriate for this configuration, however. The numerical ratio of expander cylinders to compressor cylinders is approximately doubled. The higher temperatures favor the use of R-114 or R-600; the latter, though thermodynamically superior, is quite flammable and could present problems in siting.

SUMMARY

The preliminary system design is complete; standard components (including condenser and compressor) have been selected and special components (such as rotary expander valve and boiler feed pump) have been designed.

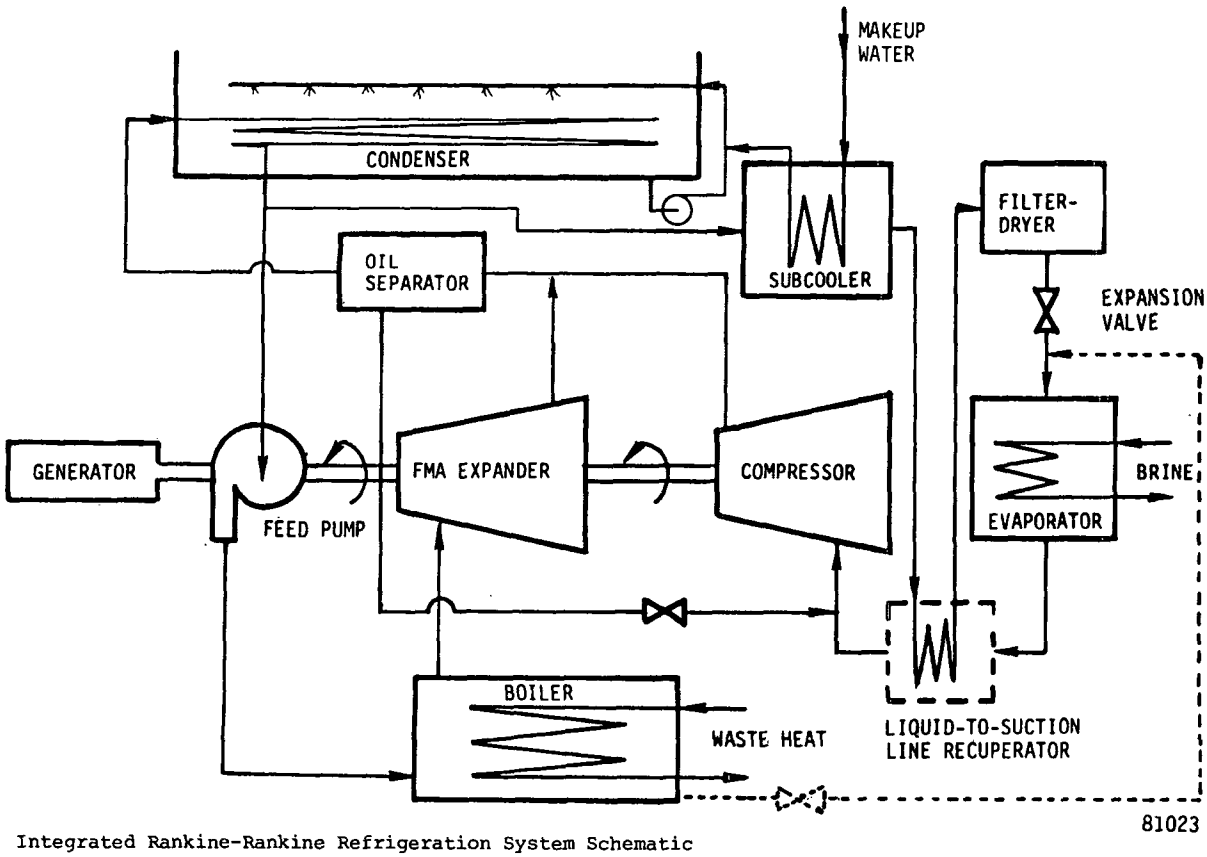


Figure 1. Integrated Rankine-Rankine Refrigeration System Schematic

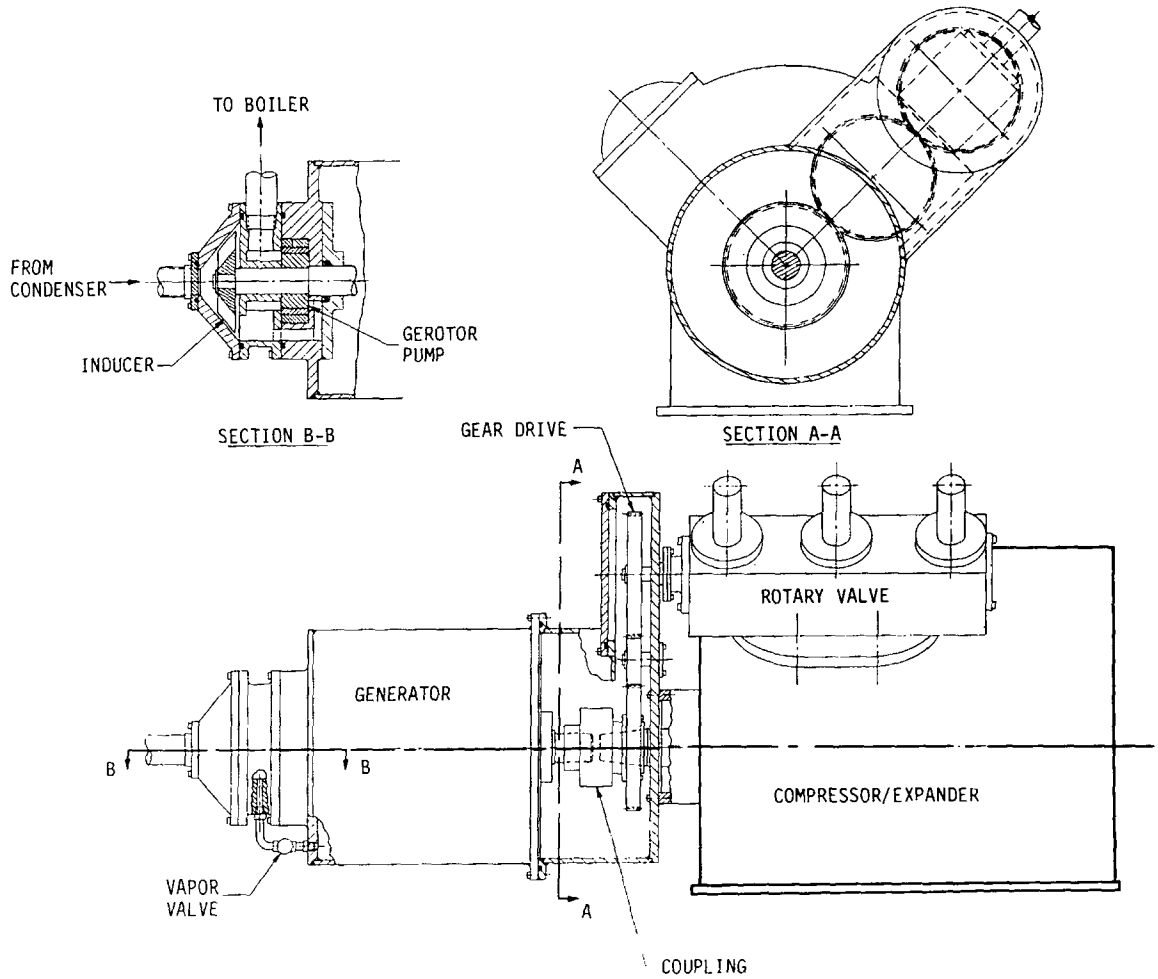


Figure 2. Mechanical Integration of Moving Parts

The boiler feed pump has been tested, and expander testing is underway. The design and construction of a complete demonstration system has begun.

TECHNICAL ACCOMPLISHMENTS

- Established Design Criterion. The sizes and configurations of the five heat exchangers are selected so as to maximize the overall system's ratio of capacity to initial cost. This capacity/cost ratio and system reliability are the major relevant criteria for judging the system.

If the waste heat energy input to the system were assigned significant value, then the overall system efficiency (ratio of evaporator and boiler heat duties) would be significant; however, waste heat is usually considered valueless and is copiously available, so system efficiency matters only insofar as it contributes to the capacity/cost ratio. Operating costs, such as pump and blower power, are to be minimized where possible; however, the typical method of rejecting this waste heat (cooling tower) has comparable operating costs, and it simplifies the optimization process to ignore any differences in these operating costs.

- Selected Condenser Configuration. Both cycles are sensitive to the condensing temperature. Table 1 shows that as the condensing temperature rises, the expander power output declines and the compressor power requirement increases. Thus, minimizing the condensing temperature is more important to this system than it would be to an electrically driven refrigeration system. This explains the large size of the condenser, and it explains why an evaporative condenser was chosen. An air cooled condenser rejects heat to the ambient dry bulb temperature, whereas an evaporative condenser rejects to the ambient wet bulb temperature; the difference between dry and wet bulb temperatures is generally significant when compared to the temperature difference that drives this Rankine Cycle. A water-cooled condenser with a cooling tower ultimately rejects heat to the ambient wet bulb, but the circulating water loop adds a temperature difference that significantly raises the condensing temperature.

- Minimized Size and Cost of Components. One approach to the design of this system would be to size the expander to provide sufficient power to drive the compressor when the ambient wet bulb is 80°F, which is exceeded only one percent of the time at the gaseous diffusion plants. Ninety-nine percent of the time the expander would be oversized, and the surplus shaft power would be utilized by the motor/generator, which would act as a motor only when starting plus one percent of the time. This approach would require six expander cylinders to drive two compressor cylinders. Another approach is to design for an ambient wet bulb temperature that is approximately 20°F lower. This allows two expander cylinders to drive two compressor cylinders. Not only does this cut the total number of cylinders in half, but it allows the boiler to be

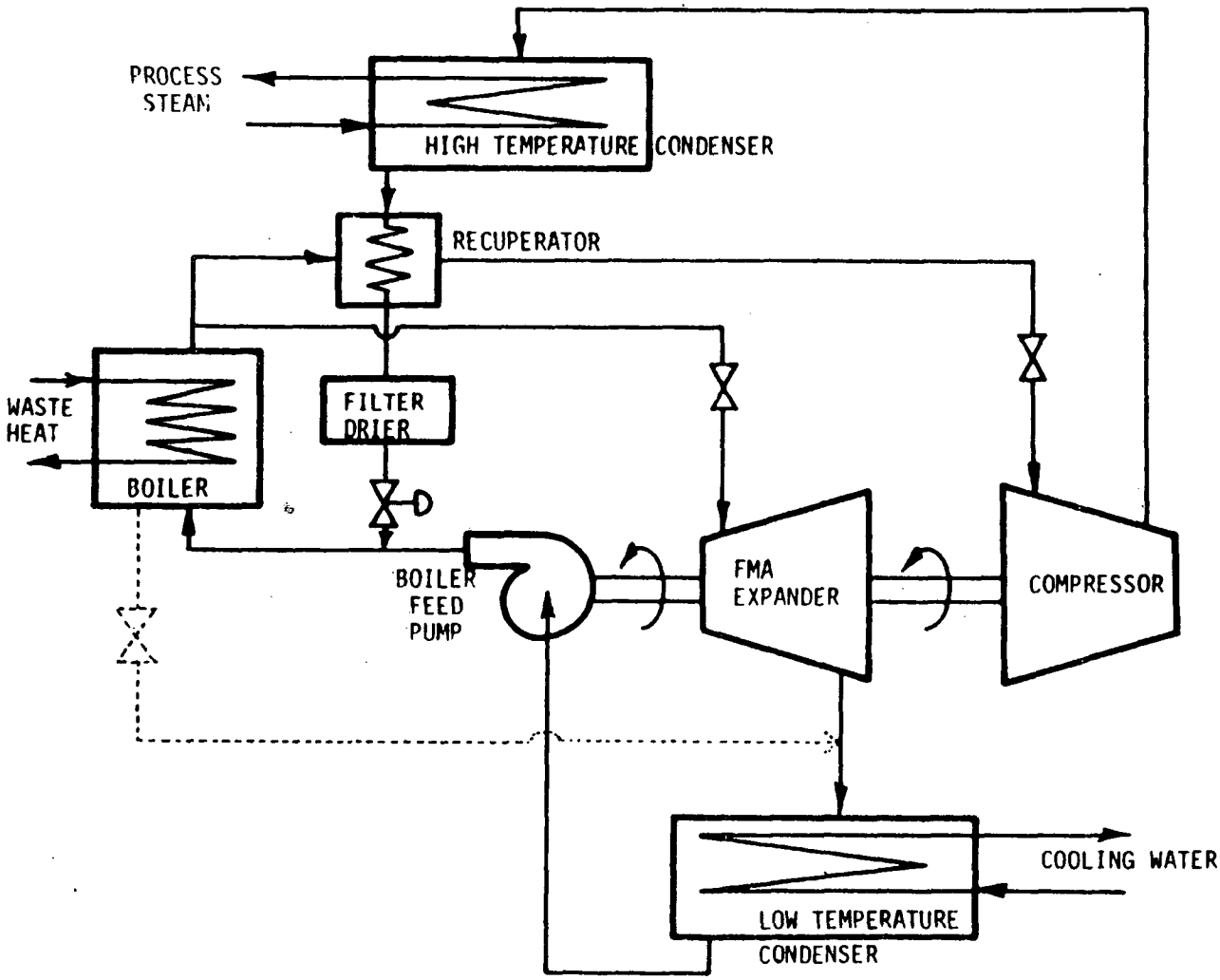


Figure 3. Flow Schematic Diagram of the Heat Pump Configuration.

| Condensing Temperature | Expander Output | Compressor Input | Feed Pump Input | Net Power To Generator (From Motor) | Balance After Pump And Fan |
|---|---|---|---|---|---|
| °C (°F) | kW | kW | kW | kW | kW |
| 35 (95) | 13.0 | 18.5 | 1.7 | (7.2) | (16.9) |
| 29 (85) | 17.7 | 17.2 | 2.1 | (1.6) | (11.3) |
| 24 (75) | 22.2 | 15.7 | 2.4 | 4.1 | (5.6) |
| 18 (65) | 25.9 | 13.8 | 2.7 | 9.4 | (0.3) |
| 13 (55) | 29.3 | 11.6 | 3.0 | 14.7 | 5.0 |

Table 1. Sensitivity of Power Balance to Condensing Temperature

one-third as large and the condenser to be one-half as large. The motor/generator will need to act as a motor more of the time this way, but it will still function as a generator most of the time.

- Selected the York RS44A Compressor. It has four cylinders, each with 3.75 in. bore and 4.5 in. stroke, and operates at 1200 rpm. One feature that makes it attractive to this system is that its discharge is manifolded externally; expander cylinders can be separated from compressor cylinders simply by removing the manifold. York makes a similar compressor with two-thirds the stroke (3 in.) and three-halves the speed (1800 rpm). Because the piston speed is the same, breathing losses would be the same. The m.e.p. is the same, so friction losses would also remain the same. The slower compressor is more attractive as an expander because its longer stroke permits a larger clearance volume, which is necessary to accommodate the expander's valves. The differential pressure across the piston is limited to 275 lb/in$^2$. Above this pressure, the connecting rod big end crankpin bearings become overloaded. The maximum pressure difference will occur across the expander piston, with the boiler pressure above and the evaporator pressure below. Under design conditions (127°F boiler, 15°F evaporator), this pressure difference is 245 lb/in$^2$. This maximum pressure difference limits the saturated boiler temperature that can be used with R-22 to 136°F (with a 15°F evaporator); boiler temperatures as high as 168°F can be accommodated by using R-12 instead of R-22.

• Designed Rotary Expander Valves. The intake and exhaust valving for both expander cylinders is accomplished by a pair of rotary valves. Both valves are on a single shaft, which is geared to the crankshaft. Figure 4 shows how the valve connects the port in the cylinder head to the intake line, or to the exhaust line, or to neither one, as required. The free breathing of the rotary valve arrangement is apparent from Figure 5. Because both intake and exhaust functions use the same port to the cylinder, it can be relatively large; the valve open area is 34 percent of the piston area. The flow paths are reasonably straightforward; a flow coefficient of 0.6 was used for the analysis of breathing losses. Like any rotary valve, this valve requires ample lubrication. This presents no particular problem to this expander. Temperatures are low enough to permit oil lubrication (which is not the case with a steam engine), and the oil is quite compabible with the working fluid. Note in Figure 1 the oil separator that connects to the exhaust of both the compressor and expander cylinders; the expander cylinders contribute more oil to this common exhaust, due primarily to rotary valve lubricating oil.

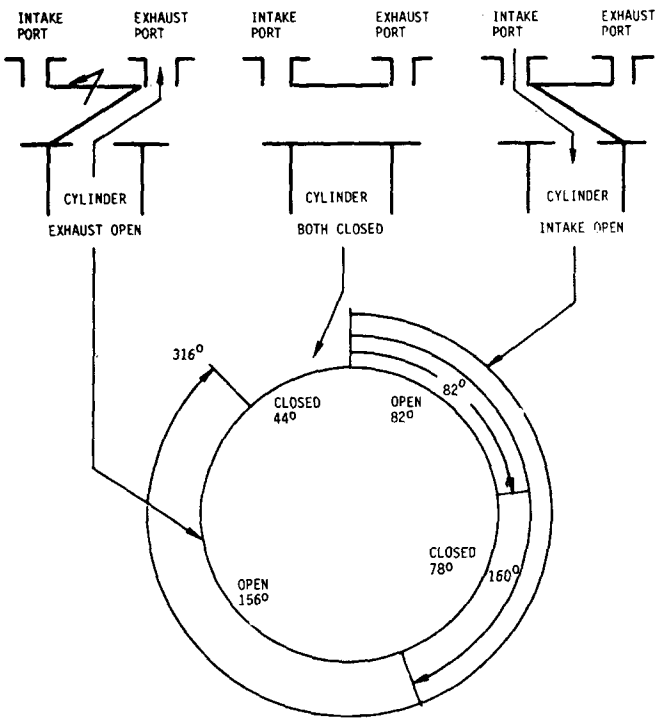


Figure 4. Operation and Timing of Rotary Valve

• Modified Lubrication System. Lubricating oil is supplied by the compressor's oil pump. The rotary valve requires oil at 300 psig, for the intake fluid is at boiler pressure (285 psig). The compressor's lubrication system is designed to supply oil to the crankshaft at crankcase pressure plus 60 psi, which is less than the rotary valve requires. The lubrication system must be modified (see Figure 6) by blocking one end of the crankshaft and adding a flow restrictor.

• Optimized Valve Timing. The efficiency and capacity of a reciprocating expander are quite sensitive to valve timing. Optimum valve timing is a function of inlet and exhaust pressures and of working fluid properties. Ideally, the intake valve should close at a point during the expansion stroke so that at bottom dead center the cylinder pressure equals the exhaust pressure. Similarly, the exhaust valve should close at a point during the compression stroke such that at top dead center the cylinder pressure equals the inlet pressure. These points are easily calculated for the ideal case, which is the isentropic expansion or compression of perfect gas, for cylinder pressure is directly calculable from volume by means of the expression

$$p\,v^{\gamma} = \text{constant} \qquad (1)$$

The *ideal* valve opening points are functions only of intake and exhaust pressures. However, the *real* case differs from this ideal case because of four factors:

• Inability to open or close valves instantaneously

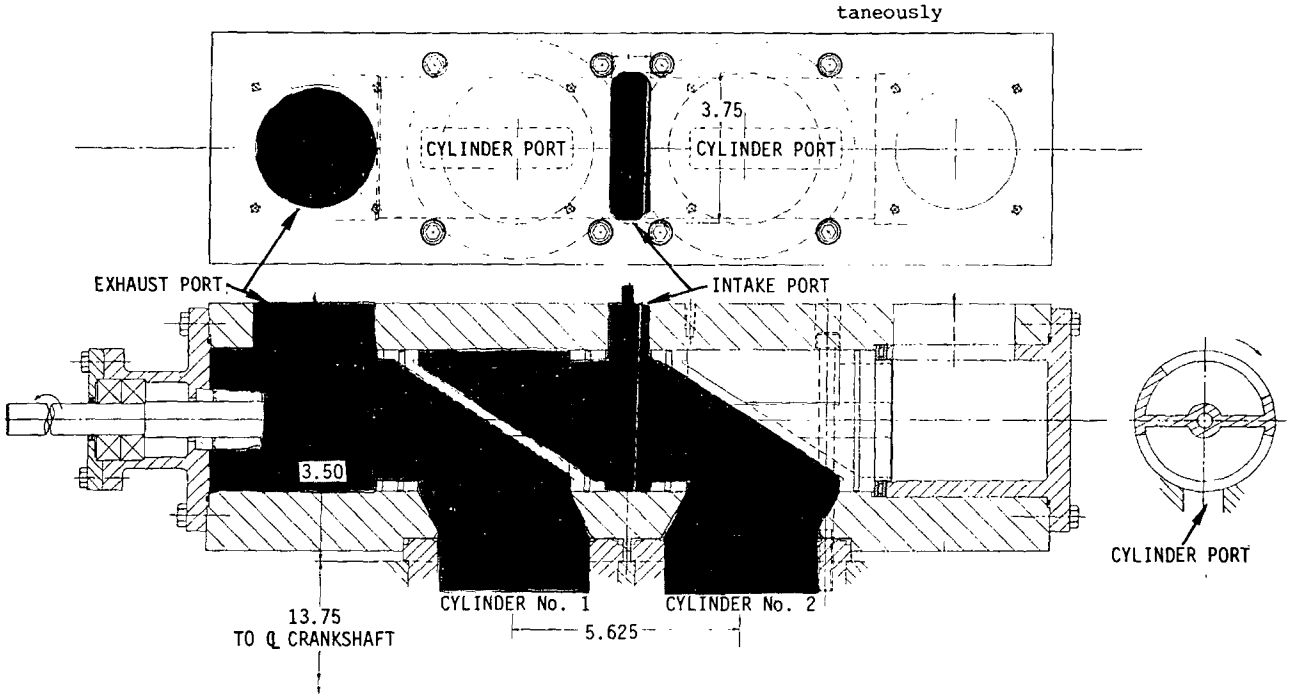


Figure 5. Layout of Rotary Valves, Showing Straightforward Flow Paths

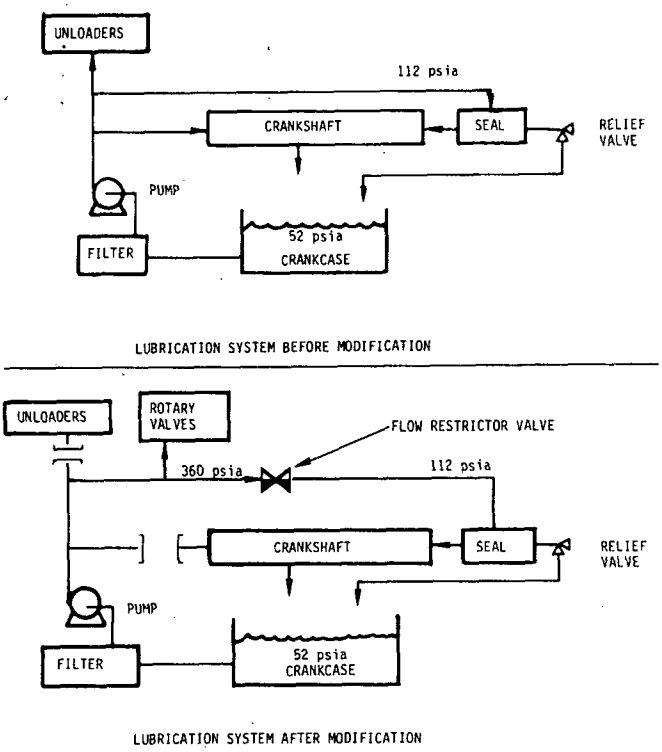


Figure 6. Modifications to Lubrication System

- Resistance to flow presented by finite flow area
- Heat transfer between working fluid and cylinder walls
- Deviation of working fluid from perfect gas.

All of these factors can be taken into account, but the analysis becomes sufficiently involved to require a finite difference computer program, which keeps track of the mass and pressure inside the expander cylinder. Flow through the valves is proportional to valve open area and to the square root of the pressure difference between the cylinder and the intake or exhaust, as appropriate. Figure 7 presents the expander P-V diagram resulting from this computer analysis, together with a tabular comparison of ideal and predicted parameters. Valve breathing losses are calculated to be 13 percent of the theoretical work; with valve area limited to that of poppet valves, breathing losses would amount to approximately 50 percent.

- Tested the Boiler Feed Pump. The pump is a gerotor pump, commonly used for hydraulic systems. Because R-22 is comparatively inviscid, it was necessary to tighten the axial clearances to limit backleakage and maximize volumetric efficiency. However, R-22 is also a poor lubricant, so they couldn't be too small. Axial clearances of 0.0015" were experimentally determined to be a good compromise. It was determined that this particular gerotor requires about 20 feet of NPSH at 120 rpm, so a centrifugal inducer was added upstream of the gerotor to provide more NPSH than could be provided by static head and subcooling alone.

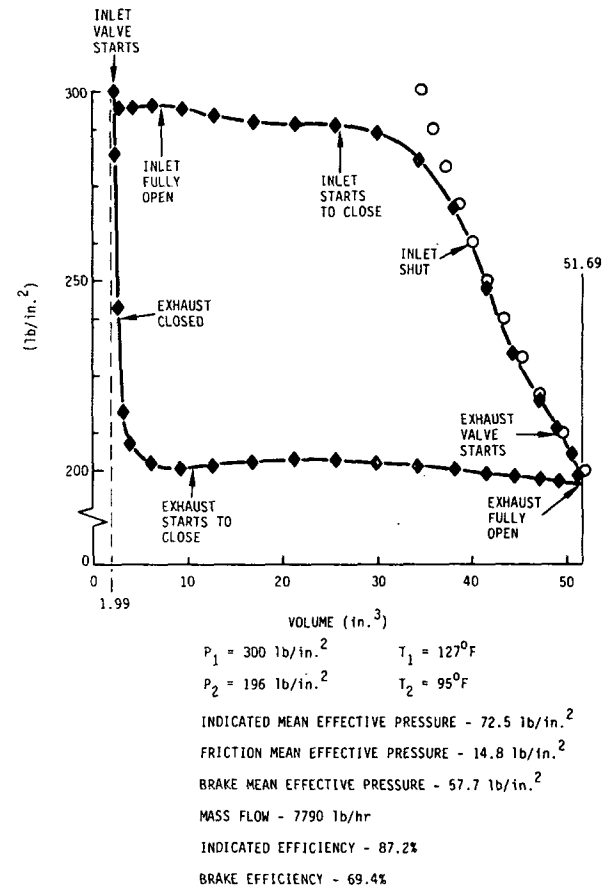


Figure 7. P-V Diagram From Computer Results

## FUTURE ACTIVITIES

Contract Activities. Expander testing will be completed, and the system design will be finalized. The demonstration system will be constructed and tested.

Post-contract activities. Further applications for this system will be sought. A teaming arrangement with a manufacturer will be sought.

## PUBLICATIONS

1. S. J. Hynek, "Industrial Refrigeration from 140°F Waste Heat," to be presented at the Waste Heat Management and Utilization Conference, Miami Beach, Florida, May 11-13, 1981.

2. S. J. Hynek, A. C. Harvey, R. L. Demler, D. H. Walker, and H. H. Fuller, "Design and Development of a Reciprocating Low Temperature Freon Expander," to be presented at the Inter-Society Energy Conversion Engineering Converence, Atlanta, Georgia, August 9-14, 1981.

# RD&D OPPORTUNITIES FOR LARGE HEAT PUMP SYSTEMS

TRW

DAVID GOLDENBERG

Begin SEPTEMBER, 1980 End JULY, 1981

## OBJECTIVE

The principal objective of this project was to evaluate the marketplace factors which constrain a more rapid implementation of energy conserving heating, ventilating, and air conditioning (HVAC) systems and system operation in commercial buildings. Recommendations for technical and non-technical research, design, and development (RD&D) opportunities which will assist and/or motivate this implementation were developed. The focus was on large air conditioning and heat pump (A/C) equipment. The building and system populations data presented are incomplete, and the conclusions and recommendations are preliminary. This work will be completed for the final report.

Air conditioning accounted for approximately two Quads ($10^{15}$ Btu) of energy resource consumption in 1979 [1], which was 2.5 percent of all energy resource consumption [2]. Heat pumps offer the potential to reduce energy resource consumption for commercial buildings heated with electricity. Electrically heated commercial buildings accounted for approximately 0.4 Quads of resource energy consumption [1].

## BACKGROUND

In recent years, a relatively successful heat pump development effort has occurred in government and industry. However, that effort is aimed at smaller (residential) equipment. In the commercial sector, A/C equipment and systems are more complicated. A/C equipment is but a single element in the successful design of building HVAC systems, and this equipment provides both heating and cooling energy in certain buildings. Therefore, this report presents the building and equipment populations data, system descriptions, energy and system cost comparisons, and discussions of non-technical constraints for all A/C equipment which is representative of the commercial sector. An understanding of this milieu is necessary to assess the RD&D opportunities that could lead to conservation of energy.

This investigation concerns electric-drive, vapor compression cycle A/C equipment and HVAC distribution and control systems used for the environmental conditioning of buildings. Heat pumps and compound uses of A/C equipment are of special concern because of the dual heating/cooling capability. Recommendations were intended to cover only larger equipment, 7 1/2 tons and larger for heat pumps and unitary equipment and 20 tons and larger for liquid chillers. These size restrictions eliminate residential-size unitary equipment and small liquid chillers (mostly process).

The commercial sector population of buildings used for this report includes eleven building types, which are based on the "Nonresidential Buildings Energy Consumption Survey" of the Energy Information Administration (EIA) of the Department of Energy (DOE)[3]. These types are:

- Assembly
- Automotive sales and service
- Education
- Food sales
- Health care
- Lodging
- Office
- Residential
- Retail/Services
- Warehouse and storage
- Other

The final recommendations to be made for this investigation cover both technical and non-technical opportunity areas for increasing energy conservation in commercial buildings which use A/C equipment for environmental conditioning. The recommendations include the development of educational, financial, psychological, regulatory, or other mechanisms to increase the use of energy conserving equipment and systems.

In accomplishing this work, TRW plans to:

- Describe existing data bases
- Estimate commercial building and A/C equipment populations
- Describe state-of-the-art HVAC equipment used in commercial buildings
- Assess the effects of important non-technical factors on the development and increased use of energy conserving A/C equipment and associated HVAC systems
- Describe A/C equipment market sizes
- Perform a basic comparison of the relative energy use and installation cost of some common HVAC systems
- Synthesize an overall assessment of the interaction of the many factors affecting energy programs for reducing energy use of A/C equipment and associated HVAC systems in the commerical milieu
- Formulate recommendations for RD&D activities aimed at reducing commercial sector energy use

The bulk of the work for all these items has been performed. The work on commercial building and A/C equipment populations is continuing, based on information released in March, 1981 [3]. Refinement and some expansion of the recommendations for RD&D activities is also continuing.

## SYSTEM/EQUIPMENT DESCRIPTIONS

Environmental control systems can be divided into the following broad categories: cooling equipment; distribution equipment and systems; and control systems. The cooling equipment can be subdivided into two categories: 1) air cooling equipment; and 2) water chilling equipment.

### Cooling (Refrigeration)/Heat Pump Equipment.

Cooling, refrigeration, and heat pump equipment is based on the refrigeration vapor compression cycle. The heat pump is a special application of this cycle because it is reversible.

Refrigeration equipment is used in unitary equipment and applied or built-up systems. The majority of unitary units and the major portion of the installed capacity of unitary units is in the range of 15 tons or less. The majority of the applied systems (i.e., built-up systems) used in the United States employ a water chiller to generate cold water.

Distribution Systems.

A single zone system is the simplest of the air distribution systems and consists of a mixing, conditioning, and fan section. Multi-zone systems condition (heat and cool) all the air at the central system (air handler). The heated and cooled air streams are mixed at the unit to satisfy the various zone loads. Dual duct systems are similar to multi-zone systems. However, with a dual duct system, the air streams are mixed at terminal mixing boxes instead of at the central unit. In a terminal reheat system, the central conditioning system supplies a fixed temperature (cold) air to the zones which can satisfy the maximum cooling load. For zone heating/cooling loads requiring less than maximum cooling, the difference, in the form of heating energy, is supplied by the terminal heating coils called reheat coils. In a variable air volume (VAV) system the amount of fixed temperature air supplied by the central conditioning section is varied at each of the zones to satisfy zone load requirements. The volume of air delivered to each zone is determined by terminal flow control units.

The VAV application is presently considered the most energy efficient distribution system for multiple zone applications. Due to the recent economic and code pressures for reduction in energy use, a rapid increase in usage of this application is occurring.

Fan coil units and unit ventilators consist of a cabinet with a heating and/or cooling coil, fan and filter. The units can be either floor or ceiling mounted and used with a two-, three-, or four-pipe water distribution system. The fan coil units use 100 percent return air, while unit ventilators use return and outside air.

The induction air unit is designed for use in multi-room buildings such as office buildings, hospitals, hotels, and apartments. The unit is particularly useful for buildings that require cooling in one room and heating in another. The use of water to provide the room cooling requirements reduces the air quantity distributed to each space when compared to the air quantity distributed in all-air systems. Fans are located in remote areas. The induction unit is supplied with the high pressure primary air stream which is discharged through nozzles in the unit to induce room air flow across the coil.

HVAC Control Systems.

The design and operation of the HVAC control system plays a major role in the success of achieving an energy efficient system. Even when properly sized to meet the design conditions, the equipment will operate at part-load a large percentage of the time. Under these part-load conditions, the control system is most important for adequate performance of the HVAC equipment.

Control systems are implemented pneumatically, electronically, or electrically, and by hybrid pneumatic/electric combinations. In some quarters, electric control systems are considered more reliable than pneumatic because of the lack of leaks and fouling by dirt or oil contamination. Others feel that pneumatic control is more reliable, if properly maintained. Pneumatic control systems are usually preferred in larger installations, while electric systems are often used in smaller installations.

Recognition of energy conservation, as a primary design parameter, requires the minimal use of heating or cooling energy in any given installation. The HVAC control system assumes a vital role in accomplishing this requirement.

STATE-OF-THE-ART ENERGY CONSERVATION CONCEPTS

Notable progress has been made in the last decade in improving the efficiency of air conditioning systems and equipment. Some manufacturers have been offering more energy efficient machinery for years at a premium first cost; however, low cost energy has minimized the available market and severely limited production and R&D investment. Recent increasing energy costs and new building energy codes have reduced the risk of investment in efficiency.

Cooling/Heat Pump Equipment.

In the last few years, advances in more efficient cooling and heat pump equipment have been achieved by almost every manufacturer. The majority of manufacturer research and development efforts are aimed at energy conservation. One notable equipment improvement which may impact all equipment is the use of higher efficiency electric motors. These motors have definite applications in pumps, fans, chillers, etc.

Manufacturers are producing new types of unitary equipment. Versions are being offered which incorporate variable speed or flow controls along with sophisticated analog computer circuitry to read sensor signals. These signals vary hot/cold deck temperatures, readjust motor speeds, and position hot/cold deck dampers. The computer will select the best method for satsifying zone demands to minimize the aggregate energy expenditure of the unit. Two-speed compressors and motors are offered as a compromise to infinite speed control for improvement in part-load efficiency. The partial load performance of the two-speed motor/compressor combination has been questioned by some.

Applied or built-up systems are also experiencing energy conserving improvements. Inlet guide vanes are an efficient method of chiller capacity modulation. One manufacturer is offering equipment which will control the speed of the compressor by means of an inverter, and simultaneously control the operation of the inlet guide vanes. The inverter controls both frequency and voltage by changing the incoming alternating current (AC) to direct current (DC) and then synthesizing the DC into the desired AC frequency and voltage. The inverter allows for variable speed control when applied to standard AC motors.

Thermal energy storage (TES) has important applications with respect to utility load leveling and consumers' cost. From the utilities viewpoint, energy storage could alleviate peak demand on the power system; thus, the utilities could generate most of the electric power by constantly using large, baseloaded nuclear and fossil units which operate at a higher efficiency and consume the least expensive fuels. TES abilities allow the building energy system operator to run chillers during the nighttime hours. The advantages of nighttime operation are: 1) possible lower electrical rates; and 2) operation of the chiller at lower condenser temperatures, due to the lower nighttime ambient air temperatures, which improves the efficiency of the chiller.

Chillers can be modified for special applications. Two of the most noted modifications are double bundle and free cooling. In double bundle chillers, the condenser is separated (hydraulically) into two sections.

One section is used with a cooling tower or other heat sink such as normally would be employed. The other set of tubes is used to capture the energy which is usually rejected through the heat sink and put it to a useful purpose, such as producing hot water or heating perimeter areas. Free cooling is another modification which adapts the normal chiller to function as a simple heat exchanger, when the condenser coolant is available at a temperature lower than the chilled water supply temperature.

Alternate arrangements of the chiller may be utilized. Cascade systems encompass a variety of possible arrangements, each with its own advantages and disadvantages. One possible arrangement of cascade chillers is for the condenser of the low stage to be used as the heat source of the higher stage. Thus, the condenser water of the high stage can be used for heating water or for space heating. The low stage is sized and operated for cooling loads only. Heat absorbed by the low stage machine can be rejected by the high stage machine to heating coils or for process use.

Although heat pumps have been studied and used for a number of years, they are just now making an impact upon the energy market and showing up in applications in increasing numbers. Most units are in the lower tonnage range (< 5 tons) and are normally used in residential or small commercial applications.

HVAC Distribution Systems.

Along with new developments in equipment, advances are being made in energy efficient system designs. One of the most prominent examples of increased system efficiency is the variable air volume system design which offers upwards of 60 percent savings. It is estimated that HVAC system fan motors consume approximately 1.0 Quad/year of resource energy in the United States, in addition to the 2.0 Quads for cooling energy. Other improvements in systems include: more efficient air foil shapes for fan blades which have potential savings of 30-40 percent over forward curved fan blades; variable inlet guide vanes on fans; improved duct construction methods to minimize duct leakage and reduce pressure drop; improved controls; and variable speed motors. Insulated return air ducts, instead of above ceiling return air plenums located underneath the roof or uninsulated return ducts, also offer energy saving potential.

Controls.

Numerous controls and control schemes exist to improve the overall efficiency of commercial HVAC systems. Some of these schemes include basic practices used in residential applications, such as thermostat setback/setup and increasing the heating to cooling thermostat deadband. Usually, other schemes are associated exclusively with commercial applications. These include:

- Scheduling and duty cycling
- Reduced air circulation
- Outdoor air reduction
- Low leakage dampers
- Hot/cold deck temperature reset
- Zone optimizing analyzers
- Economizer cycles, cold air temperature reset, and enthalpy control
- Conversion of constant volume systems to variable air volume
- Automated energy management systems

Automatic scheduling of HVAC equipment can save energy by not operating equipment unnecessarily. Four areas where scheduling can prove beneficial are:

- Fan and pump energy
- Cooling energy for outdoor air
- Heating energy for outdoor air
- Lower/higher unoccupied period space temperatures during the heating/cooling season

Energy savings are obtained by scheduling equipment to turn on and off to conform to the hours the building is occupied. Additional savings are obtainable by varying equipment starting time, based on outdoor air temperature, to minimize warm-up or cool-down after an unoccupied period. Duty cycling conserves energy by shutting down the HVAC system for a portion of its normal operational period.

Reducing the total amount of circulated air is one of the most effective means of conserving energy. The fan or "pumping" power can be a very significant percentage of total heating/cooling costs. Theoretically, the reduction in energy consumption by a fan decreases as the third power of the decrease in circulated air. Thus, a 10 percent reduction in circulated air could save $[1-(1-0.10)^3] = 27$ percent of the fan energy consumption. In practice, this calculation is usually a good estimate of the savings.

Reducing the amount of outdoor air for ventilation can reduce the cooling or heating load on HVAC equipment. Three ways to reduce the outdoor air are:

- Reduce fixed exhaust requirements
- Shut off the outdoor air damper during unoccupied periods
- Reduce the minimum outdoor air required for ventilation during occupied periods

The installation of low leakage dampers prevents unwanted outdoor air from entering a building when outdoor dampers are closed. Hot/cold deck temperature reset controls compare zone loads and adjust the hot/cold deck temperatures to match the load for the zone(s) with the largest heating/cooling load by means of a load optimizer control.

As mentioned previously, Terminal Reheat Systems can be improved by using zone optimizing load analyzers (ZOLA). The ZOLA measures the demand of each zone, and set the cold supply air temperature to minimize the amount of reheat and cooling required.

The term "economizer" can mean many different things for HVAC controls. Originally, an economizer meant HVAC central system controls that introduced outdoor air to offset as much mechanical refrigeration as possible, when the outdoor temperature was in the right range to provide cooling. While this system was intended to save energy, the opposite was often the case. Adding cold air temperature reset can significantly reduce the energy costs for the economizer system when compared to the economizer system alone or an HVAC system without cold air reset. Enthalpy controls are a more sophisticated approach. Enthalpy controls prevent the use of slightly cooler (62-70°F) air when the humidity is high, and the enthalpy of the outdoor air is higher than the enthalpy of the return air.

Conversion of constant volume air distribution systems to variable air volume (VAV) can save energy. In a constant volume reheat system, converting to a VAV system will eliminate reheat in most interior zones and reduce it in exterior zones. In a dual duct system, converting to a single duct VAV system eliminates the wasteful mixing of the heated and cooled air streams. Sometimes, other controls must be added to prevent central fans from producing excessive negative static pressures in the building.

An Automated Energy Managment System (AEMS) has the potential to save 10 to 20 percent or more of the energy consumption in a typical commercial building. Additional consumer and utility cost savings and utility energy savings can be achieved by the demand limiting function of the AEMS. By continuously and automatically monitoring all HVAC system parameters, the AEMS aids in optimizing all equipment usage and system coordination, thus saving energy cost.

By changing the required chilled water temperature as a function of chiller load, considerable energy may be saved. This saving is typically a 1-1.5 percent increase in efficiency for each degree in chilled water supply temperature increase.

## ENERGY USE AND SYSTEM COST COMPARISON

The effects of system type and control on energy consumption and system cost for a specific building type and size in Kansas City are shown in Figure 1 and Table 1. The building is a 40,500 square foot office building. The energy consumption was calculated by using the NASA Energy-Cost Analysis Program (NECAP)[4].

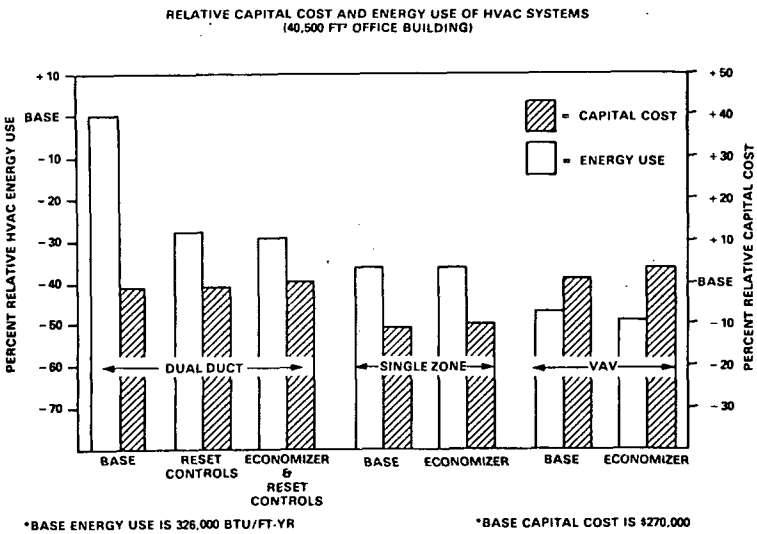


Figure 1. Percent Relative Cost and Energy Use of Dual Duct, Single Zone, and VAV Systems

These results are shown numerically for the three base systems and with the economizers and deck temperature reset controls (controls changes) in Table 1. The percentage reductions in energy use are calculated by using the unmodified dual duct system as the reference (base) system. Since the individual values are percentages, they do not sum to the totals. Also shown are the relative performances of individual systems compared to the dual duct base system with and without the controls changes.

In Table 1, under the column heading "Individual System", each system is treated as a base system. The effects of the controls changes are shown relative to each individual system base. Under the column heading "Dual Duct Base", the performance of each individual system base and of each controls change is shown relative to the dual duct reference base.

For this discussion, economizer means a fresh air, return air, exhaust air damper system with controls that modulate the dampers to introduce outdoor air for cooling, as demanded by zone thermostats. No outside air is introduced above the reasonable switchover

Table 1. Effects of System Type and Controls on Energy Use (40,500 ft² Office Building, Kansas City)

PERCENT REDUCTION IN ENERGY USE DUE TO SYSTEM CHANGES (40,500 FT² OFFICE BUILDING)

| SYSTEM | HEATING | | COOLING | | FAN | | TOTAL | |
|---|---|---|---|---|---|---|---|---|
| | INDIVIDUAL SYSTEM | DUAL DUCT BASE | INDIVIDUAL SYSTEM | DUAL DUCT BASE | INDIVIDUAL SYSTEM | DUAL DUCT BASE | INDIVIDUAL SYSTEM | DUAL DUCT BASE |
| Dual Duct | Base | | Base | | Base | | Base | |
| Deck Reset Controls | 21 | | 42 | | 0 | | | |
| Economizer & Deck Reset Controls | 21 | | 45 | | 0 | | 24 | |
| VAV | Base | 51 | Base | 31 | Base | 60 | Base | 47 |
| Economizer | 0 | 51 | 17 | 42 | 0 | 60 | 4 | 49 |
| Single Zone | Base | 37 | Base | 43 | Base | 0 | Base | 36 |
| Economizer | 0 | 37 | 1 | 44 | 0 | 0 | 1 | 36 |

temperature, which is 70-72°F for Kansas City, for typical room thermostat setpoints and return air temperatures.

For this particular building, building occupancy, and location, a small amount of additional capital results in a 50 percent reduction in energy use for the VAV system relative to the dual duct reference base system. However, by adding deck reset controls to the dual duct system, half of the savings shown for the VAV system can be achieved. Readers of this section are cautioned to remember the complexity of HVAC system/building/climate interactions. Some important variables are local energy costs and availability, building size, type and occupancy, HVAC system type and control strategy, and climatic and microclimatic variables such as temperature, humidity, solar flux, wind, shading, and wind effects.

## BUILDINGS/EQUIPMENT POPULATIONS AND USAGE PATTERNS

The total number of commercial buildings in the 48 contiguous states and the District of Columbia was estimated at 4 million[3]. The total floor space was estimated at 58 million square feet, with 78 percent of that space being cooled. The percentage of buildings with airconditioning is 64 percent[3]. For comparison, the number of industrial buildings is estimated at 240,000 [3]. Reference [3] can provide additional valuable information.

Four percent of commercial buildings (>50,000 square feet) account for 53 percent of all commercial floor space. Seventy-seven percent of these buildings (<10,000 square feet) account for 19 percent of the floor space. A breakdown of air conditioning equipment for all nonresidential buildings, including industrial is given below [3].

| A/C Equipment Type | Number of Buildings (thousands) |
|---|---|
| Window units | 856 |
| Package units | 800 |
| Central systems | 749 |
| Combination/other | 302 |
| No air conditioning | 1,530 |
| Total | 4,237 |

Equipment shipment data for the period 1965-1976 [5] show that centrifugal chillers accounted for 60 percent of total capacity (tonnage), while unitary equipment accounted for 60 percent of the total number of units shipped. Heat pumps accounted for less than 5 percent of the total units and less than 2 percent of the total capacity. All figures are for unitary and heat pump equipment 7.5 tons and up, reciprocating chillers 20 tons and up, and centrifugal chillers 60 tons and up.

## NON-TECHNICAL CONSTRAINTS ON THE DEVELOPMENT AND USE OF ENERGY CONSERVING SYSTEMS

Through contacts with many individuals across the country, an encompassing view was developed of the non-technical problems that are impeding the use of energy efficient HVAC systems. The primary impediment to the development of energy conserving systems is the low use of energy efficient HVAC systems caused by non-technical constraints, but this impediment (to development) is not significant.

For this report, non-technical constraints were divided into the following groups:

- o Institutional constraints
- o Economic constraints
- o Educational/informational/psychological constraints

Because of their overriding importance, only economic constraints will be discussed here.

### Economic Constraints

Economic constraints on the use of energy efficient HVAC systems are the most limiting. Economic constraints involve: manufacturers of HVAC equipment; the designers of HVAC systems (A&E firms); and the owners and/or users of the HVAC systems. Economic constraints are most detrimental to energy conservation in general.

HVAC system owners are the bottleneck group inhibiting the use of energy conserving systems. An HVAC system owner who pays taxes can deduct energy costs as operating expenses from gross income when calculating tax payments. Any savings in operating expenses increases revenue, which is taxable. Generally, building owners who do not pay taxes are more concerned about energy costs since operating expenses do not enter the tax picture, and the buildings usually are used or occupied completely by the owner. First costs are still critical, as will be discussed below. The Federal government, which represents this type of owner, is a major competitor for captial.

For retrofit construction, most owners are interested in those items which will save energy without costing much money. When a sizeable investment is required, two big hurdles must be overcome:

- o Capital availability
- o Credibility of the savings

Capital availability is an important part of the first cost problem. Credibility of the savings is crucial in deciding to proceed with a retrofit. A building owner must rely on a person proposing changes for saving energy costs. If the savings do not materialize, who is liable?

The limited ability of the United States' economy to generate capital affects businesses, in that all businesses compete for this capital. The Federal government also competes, and the inclusion of the Federal government in this competition limits the capital available for businesses.

To gain a better perspective of what energy costs really mean to a business, the following examples are presented, based on work presented in [5]. The first example concerns a business which:

1) has a 50 percent tax rate
2) has a ratio of energy costs to after tax profit of 20 percent

These values are representative of a typical large business. If this business reduces energy use by 30 percent in one year (assuming all forms of energy used cost an equal amount per Btu), and other factors remain constant, their after tax profit will increase 3 percent. On the other hand, if this business does not reduce energy use, and energy costs increase by 40 percent in one year, their after tax profit will decrease 4 percent. The company that reduced energy use by 30 percent would see a 0.2 percent increase in after tax profit if their price of energy increased 40 percent. Percentagewise, that is not much incentive.

The Metropolitan Medical Center (MMC) in Minneapolis, Minnesota provides a good example of economic constraints on energy conservation caused by government regulations. Medicaid and Medicare are set up to reimburse hospitals for expenses - which include energy costs - but not for capital items. Therefore, hospitals are reimbursed for consuming energy. One retrofit project involved converting the HVAC systems from constant volume to VAV systems. This project has a questionable future because of the reimbursement practices of the Federal programs. According to the director of plant operators at MMC, one-half of the expected first year's annual saving must be refunded to the government, and the operating budget used to figure government payments to MMC is reduced by the expected first year savings from the second year on. The net effect is to change a three year payback on capital improvement programs to about six years.

### Solutions to Non-Technical Constraints.

Two observations are apparent when the non-technical constraints on the development and use of energy conserving HVAC systems in buildings are put in perspective. They are:

1) Constraints involving the use of energy conserving HVAC systems completely overshadow constraints involving development

2) Building owners and their economic situations are the key to increased use of energy conserving HVAC systems and energy conservation in general

With these two observations established, approaches to overcome certain non-technical constraints will be described.

The majority of existing energy conservation programs affect the manner in which the engineer (A&E) operates, and thus, direct the building owner's decisions. However, some programs are focused on the building owners, in an attempt to affect the economic factors surrounding an energy conservation investment. Manufacturers of control systems for commercial buildings are offering creative financing for energy conservation (energy management) investments. These financing arrangements may include lower interest rates and choice of installment purchase or operating lease. The choice of owning or retaining an operating lease depends on the building owner's financial situation.

A further attempt to improve the building owner's position is made by the use of a limited warranty on energy savings, which are offered in some cases. This offer addresses the problems of "credibility of the savings."

For the future, is there a place for energy system specialists to arrange with building owners to handle the operation, maintenance, capital improvements, and billing associated with building energy systems?...to reduce the energy use and costs of building energy systems?...to assume responsibility for energy cost reductions?

Today, over 140 companies are selling space heating and hot water in France. These companies are referred to as "chauffagé" or "comfort" companies, and they control roughly 70 percent of the heating market for commerical buildings in France. The oldest companies are 11 years old.

The idea behind chauffagé companies has been extended to a concept called "energy services"[6]. Energy services represent a new business opportunity in the United States. The objective of energy services is to operate energy systems at specific output levels and at the lowest cost to the customer. Energy services represent a marketing approach to energy using systems and energy systems operation.

Implications of the building energy services company are:

1) The owner is relieved of the responsibility of maintaining the energy systems in the building

2) The energy services company should be free to invest in and make necessary modifications to reduce energy costs

3) Tenants pay for energy based on use instead of area occupied

4) The comfort, lighting, and other energy system levels must be specified prior to the service arrangement

5) Energy uses would be metered better

6) An acceptable method of accounting for the variable effects of weather on energy costs must be developed and accepted

7) Based on varying occupancy requirements, an acceptable method of charging tenants or the owner must be developed and accepted

8) Acceptable contractural obligations would have to be researched and defined

9) The owner must arrange for special services involving the building energy systems, which relieve him of the task of dealing with energy management and complicated energy systems

10) THE OWNER COMMITS NO CAPITAL AND EXPECTS TO REDUCE ENERGY COSTS

The energy services concept does not solve all the problems previously discussed. The competitive position of most businesses is not determined by whether energy is conserved.

Unless the economic situation of building owners is affected, changes in speculative, new building/HVAC design and retrofitting of existing buildings and HVAC systems are unlikely. In this project, TRW has tried to demonstrate that increasing energy costs by deregulation or other means does not sufficiently affect the competitive positions of businesses to cause serious changes in energy attitudes - justifiably so from an economic standpoint.

Energy services companies would concentrate their efforts on large buildings with high energy bills where the potential for profit is greater. Also, unless the energy services concept can be extended to cover new construction, new buildings would continue to be built with unnecessarily high energy consumptions.

An effective way for the government to change the energy use of this country would be to establish a free market for energy, so that energy decisions are not influenced by taxes or incentives. However, given the all pervasive nature of taxes and their influence on energy supplies (conservation increases available energy supplies), an effective way for government to change the energy use of this country is to change the tax structures so that saving energy is profitable, and energy use, beyond certain levels, becomes economically difficult. Gradually implementing such a program is desirable. Some ideas that could be considered are:

- Introducing energy taxes based on energy use which would be comparable to income taxes (income tax credits could be given, but using more energy should decrease after tax profit considerably)

- Introducing income tax credits based on low levels of or reductions in <u>measured</u> energy use

- Continuing to consider income tax credits for capital investment in energy saving projects

- Eliminating deductions of energy costs as operating expenses - perhaps only for energy use above certain levels

Some important considerations to remember are:

- This study concerns the commerical sector, and implementing tax structure changes for the industrial sector would take longer and be more complicated

- Businesses which use energy to produce a service ("commerical processes"), such as laundries and restaurants, will probably require special treatment

- Extensive installation of energy metering equipment would result

- Phased tax adjustments will allow capital markets, building/HVAC system designers, and manufacturers time to adjust, while gradually showing business decision makers what to expect

- Special consideration would have to be given to businesses which are already using less energy

- Standards for new construction, such as BEPS, would not require mandatory compliance. Those buildings which use no more than the required amount of energy the first year (based on actual utility or supplier billing or other data) would have no tax penalty. Using less than the required amount would lead to a tax credit, and using more would lead to a tax penalty. Businesses could choose their course of action.

- Legal considerations, such as requiring energy taxes to be paid regardless of profit and loss figures (eliminating the use of "dummy" businesses as "energy tax shelters"), would have to be researched carefully

Consideration must also be given to non-profit organizations, because taxes will not affect them. The existing schools, hospitals, and government buildings grants program, Title III of the National Energy Conservation and Policy Act (NECPA) of 1978, is an example of what can be done for this sector. One alternative

would be to offer grants to energy services companies to operate the energy systems in these buildings at specified energy consumption levels. Consideration would have to be given to existing energy system operating personnel (especially additional training).

RD&D RECOMMENDATIONS

The preliminary RD&D recommendations of this report provide an indication of future activities for DOE. Estimates of potential energy savings are still being developed. The recommendations are divided into two categories: I) high priority; and II) second priority. The high priority recommendations deal with development of the market for energy efficient HVAC systems, which is a reversal of present approaches that mandate the performance of buildings based on predicted energy consumption performance or prescribed specifications. Many of the factors constraining the existing market are nontechnical in nature and cannot be solved by straightforward, technical approaches. The high priority recommendations concern the development of a marketing approach to energy supply through conservation. This approach allows building owners to make choices about energy conservation based solely on economics. The second priority recommendations deal with technical research and education programs. The technical research work involves system design concepts and the dynamic interaction of people, buildings, and systems. The technical education programs are aimed at increasing the number of technical people capable of providing competent energy system efficiency improvement services to building owners, when owners begin to demand these services in response to market pressures derived from the high priority programs.

The recommended potential RD&D work in the energy services portion of category I, in order of priority, includes:

1) Researching and publicizing the energy services concept and the role of energy services companies as professional energy systems operators

2) Developing acceptable methods (guidelines or models) for estimating the savings resulting from the operation of buildings by energy services companies (which will be used to determine fees)

3) Developing criteria for energy services contracts

4) Educating building owners about the recommended criteria for energy services contracts

5) Developing an energy services grants program for non-taxable entities (schools, hospitals and public buildings)

6) Examining utility regulations which might hamper the energy services concept

This marketing approach to increasing the use of energy conserving systems requires business people, lawyers, and engineers to help in the development of programs. While engineering input may not be as pronounced as in energy programs of the past, more work for engineers will potentially result, and building owners will retain freedom of choice.

Changing the economics of energy use, based on metered energy use, is critical to affecting the energy use decisions of today, which will affect the economic and energy condition of this country for many years. This report has described why increasing energy prices is not as effective as such an action might seem. The most effective way to increase available energy supplies through use of energy conserving systems is to change existing tax structures. To argue that tax structure changes (tax incentives) are an artificial contrivance in a free market is one-sided. Existing tax structures for commercial building owners are often a disincentive for reducing energy use.

The recommended potential RD&D work in the energy economics portion of category I, in order of priority, includes:

1) Developing a phased-in energy tax plan for commercial buildings' energy use. Commercial process energy use could be exempted if the owner installed separate metering. At master-metered sites where offices or other commercial buildings predominate, the site owner would be responsible for collecting and paying the taxes, as a pass-through item, based on the whole site. Again, exemptions could be obtained for process use if separate metering was installed. Income tax credits would be given, but using more premium energy per square foot than a prescribed standard should affect after tax profit considerably, in relation to the deviation from the standard. Energy tax deductions should not be allowed.

2) Developing phased-in tax credits, in parallel with energy taxes, for using less premium energy per square foot than a prescribed standard, in relation to the reduction below the standard. After tax profit should benefit from good performance. For both items one and two, extensive business and legal research is required.

3) Developing the standard language and procedures by which Congress would prescribe in pertinent energy legislation that DOE determine the energy-related criteria to be used by the IRS when setting up tax regulations for the subject legislation

4) Investigating the removal of energy costs as a deductible operating expense

The energy services work and the energy economics work are equally urgent. Items with equal numbers for both programs share equivalent priority. Both programs can proceed in parallel.

The second priority recommendations are of two types: those contingent on the high priority programs; and those of an independent, technical nature. The recommendations related to the high priority programs involve educational programs and research into potential bottlenecks for the high priority programs. The independent, technical work involves research into HVAC systems, buildings, human factors, and the HVAC/building design process.

For the contingent recommendations, the high priority program that these recommendations are contingent on will be enclosed in parentheses after each. The recommended potential RD&D work for the contingent recommendations (category II), in order of priority, includes:

1) Developing seminars and educational programs for A&E firms (consultants) which offer instruction or sharing of the knowledge concerning energy efficient building/energy-system design (energy services and energy economics)

2) Developing seminars and educational programs for building owners concerning the potential energy savings and technical services they should expect (energy services and energy economics)

3) Developing and publicizing potential methods of offering energy services for buildings and marketing approaches for those methods (energy services)

4) Developing seminars and educational programs for A&E firms (consultants) which offer instruction or sharing of the knowledge concerning methods of offering energy services for buildings and successful marketing approaches for those methods (energy services)

Some of this work may be accomplished within a reasonable time by the private sector with no assistance from DOE, but a lack of qualified technical people could be a serious bottleneck for the high priority programs.

While TRW does not view the technical RD&D work recommended in this section as independently capable of increasing the use of energy efficient HVAC systems, the work is intrinsically valuable for the long-term benefits resulting from the knowledge gained concerning HVAC systems, buildings, and the design process.

The cooling equipment of most interest for RD&D work on energy conserving systems is the large chiller. The merits of large screw and centrifugal liquid chillers appear to depend on application. Also, some controversy exists concerning actual part load efficienices for all large chillers. The screw machine appears well suited for ice generation as a thermal energy storage medium and for producing chilled (42°F) and hot (120-130°F) water simultaneously (heat pump application). Heat pumps in cascade with chillers serving the building core may offer the most opportunity for heat pump and chiller uses in an energy conserving system. Further study of the best applications for screw and centrifugal liquid chillers is needed.

The overall HVAC system is crucial in any consideration of reducing energy consumption. A preferred system for heating and cooling large buildings is one which uses chiller heat reclaim. A study of building size, configuration, and type of occupancy, together with climatic effects to determine the parameters and possible change points affecting the most energy efficient and cost effective designs is needed to establish benchmarks for all designers to examine.

The ASHRAE Task Group on Dynamic Response has developed a research program aimed at defining recommended and/or standard operating procedures for buildings and associated energy systems which will maintain comfort while allowing better management of energy. The program will examine:

- Human comfort and health
- HVAC system controls
- Building and system characteristics
- Testing and building/system classification

This program could represent a major improvement in knowledge concerning the dynamic behavior of buildings. TRW recommends pursuing this work for the long-term benefits in improved buildings operation and reduced energy consumption.

An important consideration which follows from the ASHRAE work is combining the design of the HVAC system and building into a total system. This combination implies that the HVAC system and building are designed together ("total design") to take advantage of load profiles and the environment as much as possible. A further extension of this concept involves continual optimization of building performance to maximize the possible energy savings. A continual monitoring system would be necessary.

The HVAC system design process should be examined by technical and behavioral investigators to determine the effects of existing, common design process flows on present HVAC system design and on the "total design" concept.

The order of priority for these technical recommendations of category II, highest to lowest, is:

1) ASHRAE dynamic response program

2) HVAC system benchmark equipment configurations

3) Further implications of HVAC systems "total design" beyond the ASHRAE work and continual optimization systems design

4) Technical and behavioral study of the HVAC systems design process

REFERENCES/EXPECTED PUBLICATIONS*


1) Jackson, Jerry R. and William S. Johnson, "Commercial Energy Use: A Disaggregation by Fuel, Building Type, and End Use," Oak Ridge National Laboratory Report ORNL/CON-14, February 1978.

2) Department of Energy, "Annual Energy Balance," Energy Data Report, DOE/EIA 0181(79), June 1980.

3) Department of Energy, "Nonresidential Buildings Energy Consumption Survey," Report No. DOE/EIA-0246, March, 1981.

4) Robert E. Lyman, Jr., "Energy and Cost Analysis of Commercial HVAC Equipment: Offices and Hospitals," M.S. Thesis, University of Tennessee, March 1979.

5) Theodore C. Gilles, P.E. "An Approach to Innovative Tax Programs to Stimulate Energy Conservation in Existing Commercial Buildings," proceedings of the Conference on Improving Efficiency and Performance of HVAC Equipment and Systems for Commercial and Industrial Buildings, Vol. 1. West Lafayette, Indiana: Purdue Research Foundation, 1976.

6) Roger W. Sant, The Least-Cost Energy Strategy Minimizing Consumer Cost Through Competition, Arlington, Virginia: The Energy Productivity Center, 1979.

7) Oak Ridge National Laboratory, "Commercial Energy Use Model."


*Final Report, to be published by Oak Ridge National Laboratory.

MULTIPLE SLIDE SCREW COMPRESSOR

DUNHAM-BUSH, INC.

DE-AC03-79CS30035

DAVID N. SHAW, P.E.

BEGIN OCTOBER 1979; END MARCH 1982

OBJECTIVE

The objective of this project is to develop the best possible compressor for use in Series Solar Heat Pump Systems. The compressor must be able to operate at extremely high efficiency levels from a compression ratio as low as 1.5 up to a compression ratio as high as 6.5. The compressor must have the capability to efficiently adjust to the widely varying inlet flow requirements as dictated by the operating conditions of the Series Solar Heat Pump System.

DESCRIPTION OF WORK

Figs. 1 and 2 should be studied when considering the compression concepts developed herein. Following is a listing of the basic concepts involved:

Positive displacement compression, only two rotating parts.

No intake or discharge valves.

Flow and head independently variable.

Allows near-ideal liquid expansion process.

Allows bi-level intake.

Highest efficiency and system flexibility.

Simultaneous earth and solar coupling allowable.

This program is structured to develop helical screw compressors that will encompass all of the above.

In order to commercialize the solar assisted heat pump concept, it is necessary to develop high compression efficiencies over a wide range of operating conditions. The program is intended to demonstrate the potential of advanced concept helical screw compressors for that purpose up to a maximum of 25-ton heating capacity.

Presently, there are no compressors on the market that can obtain the performance level estimated herein for a two-speed, vapor-injected 25-ton helical screw compressor.

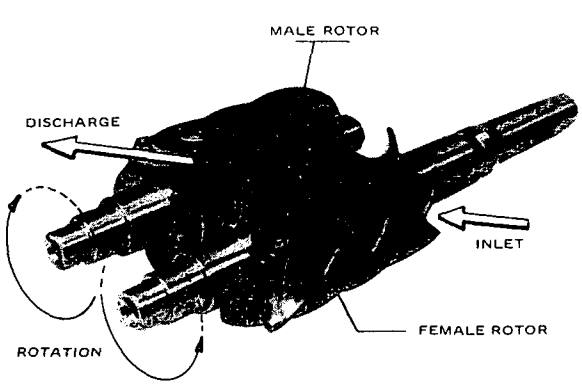


Figure 1. Helical Screw Rotor Set

Explanation of Fig. 1. In Fig. 1, compression of the vapor is started at the top rear of the diagram and high pressure discharge vapor exists as shown in the top front. The right hand rotor rotates in a counterclockwise fashion, whereas the left hand rotor rotates in a clockwise fashion as shown on the diagram.

Explanation of Fig. 2. To show in detail the discharge port and the suction bypass ports, Fig. 2 is shown in a different orientation than Fig. 1. In this figure, the inlet is at the top right while discharge is at the bottom left. The rotors have been left out for the sake of clarity. The central valve is the compression ratio slide valve and can be moved axially along its axis. If the compression ratio slide is moved to the right, the discharge port formed by the valve will open to discharge sooner and less compression of the pumped gas will take place. If the compression valve is moved to the left, the discharge port will open to discharge later and more compression of the pumped gas will take place.

Also shown in Fig. 2 is a capacity slide. The method demonstrated is called suction bypass. That is, during part load operation a portion of the gas taken in the inlet port is allowed to return back to the inlet before any appreciable work is expended on it. If the capacity slide is moved axially to the right, the bypass ports will be sealed and full capacity operation will result. As the slide moves to the left, the bypass ports progressively increase in size which progressively reduces the compressor capacity.

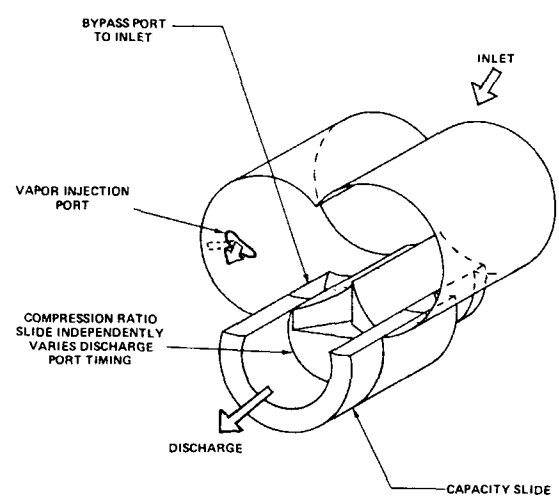


Figure 2. Slide Valve Concept (Potential)

Also shown in Fig. 2 is a vapor injection port which will allow the near-ideal liquid expansion process as shown in Fig. 5. This port will also allow the compressor to accept vapor from an evaporator but is operating at a higher pressure level than that evaporator that is feeding the compressor suction. This means that the compressor has the ability to ac-

cept energy simultaneously from two different sources. An example of this would be a lower level ground source and a higher level solar source.

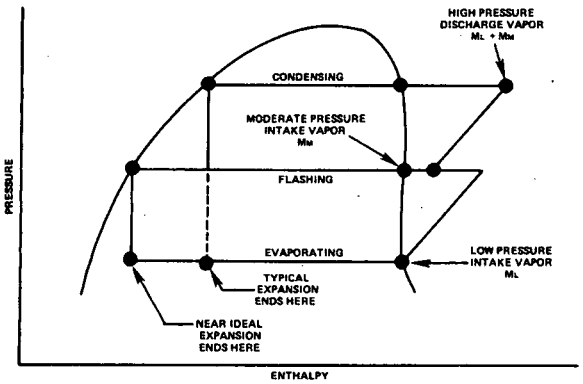


Figure 3. Pressure-Enthalpy Diagram

Explanation of Fig. 3. Fig. 3 demonstrates one of the advantages of the injection port. In a typical pressure enthalpy diagram, condensing takes place as shown. In a conventional system, the condensed liquid then passes through some type of expansion device before entering the evaporator. The conventional process is constant enthalpy and therefore appears as first a solid and then a dashed line on the diagram. Obviously, a considerable amount of flash gas is generated during this expansion process and this flash gas must enter the evaporator and accomplishes no useful refrigeration effect. Yet, in the conventional process, this flash gas must be compressed all the way from the lower evaporating level up to the condensing level thus consuming compression energy with no useful benefit at all. With the injection port, we can allow a process as shown. The condensed liquid from the condenser, instead of being flashed all the way down to the evaporating level, is now flashed approximately half way down (considering temperature differential between condensing and evaporating). The flash vapor thus generated is separated and injected into the compressor by the injection slide at the appropriate point along the compression process. Obviously, the temperature of the remaining liquid has now been reduced to the pressure corresponding to the flashing pressure. There are two ways to look at the thermodynamic cycle advantages of doing this. As can be seen from the diagram, a considerable increase in refrigeration effect is seen with the near-ideal expansion as compared to the typical expansion. Obviously, this increase in refrigeration effect has been gained at no additional mass flow required by the compressor at its low pressure intake point. What has occurred is that a moderate amount of flash vapor has been generated at a moderate pressure as shown in the diagram. This flash vapor now enters the compressor after compression has already progressed significantly and therefore only has to be compressed part way up to the condensing level. This cycle as demonstrated is similar to the flash economizer cycle used in most two-stage centrifugal air conditioning systems. Another way of looking at the cycle from the vantage point of heating is as follows. A typical cycle would have compressed an amount $M_L$ as shown all the way to condensing level. The heating effect would then be $M_L$ times the enthalpy change occurring in the heating condenser. The coefficient of performance would be the heating effect divided by the compression energy requirement. With the near-ideal liquid expansion process, an increased mass flow now passes into the condenser. The new heating effect is now $M_L$ plus $M_M$ as shown. Obviously, the energy compression requirements for the additional mass $M_M$ is less than the compression energy requirement per pound for $M_L$. Therefore, it can be seen that the heating energy delivered to the condenser compared to the compression energy required to deliver it is less if we consider the near-ideal liquid expansion process. This near-ideal liquid expansion process is one of the thermodynamic cycle advantages made possible by the utilization of the multislide screw compressor.

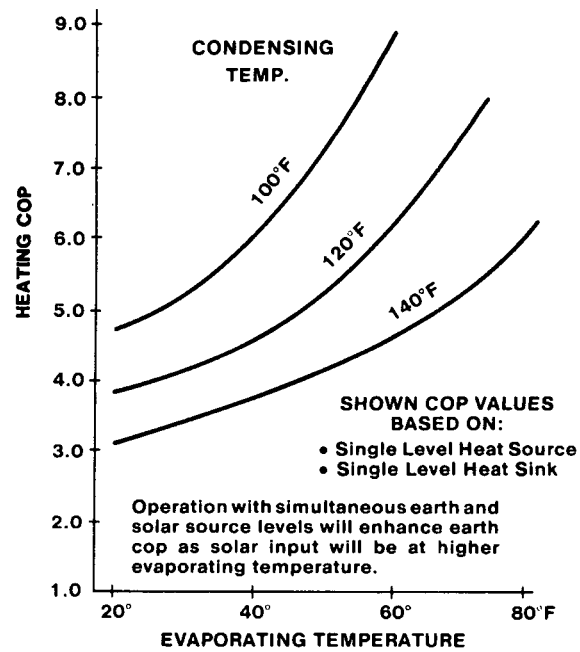


Figure 4. Estimated Heating COP

Explanation of Fig. 4. Fig. 4 is included to show the estimated heating coefficient of performance for evaporating levels from 20°F. through 80°F. and condensing levels from 100°F. through 140°F.

The curve as drawn considers the suction bypass slide to be in the fully loaded position. However, the flexibility allowed by the compression ratio slide and the suction bypass slide will allow the part load COP to be quite close to the full load COP at the same evaporating and condensing levels.

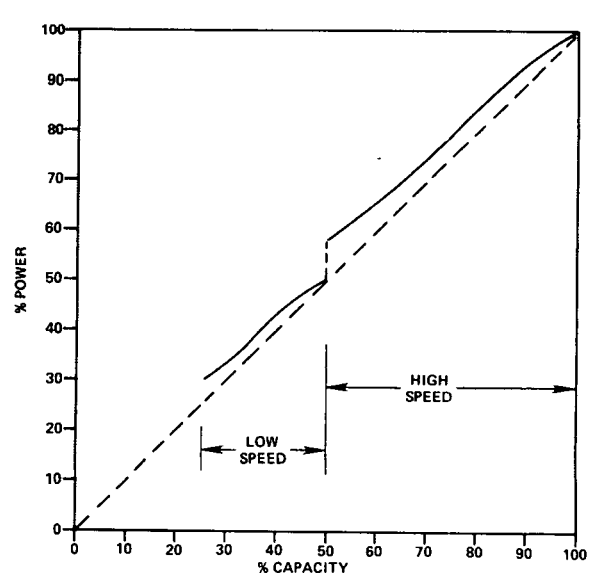


Figure 5. Part Load Performance

Explanation of Fig. 5. Fig. 5 is included to show the part load 70 power. This is the power at part load shown as a percent of the power at full load. A two-speed motor is utilized as well as the suction bypass slide valve. There is no rebalance of system operating conditions considered in these curves. The evaporating and condensing temperatures at part load are considered to be identical with those at the full load condition.

The multi-slide screw compressor is a device capable of operating at various inlet flows and any desired compression ratio. The machine is inherently reliable; only two rotors are employed, both supported by anti-friction bearings. Oil injection is employed and near-isentropic compression levels are always enjoyed. In addition to this, a secondary or moderate pressure intake level is allowed thus allowing for the near-ideal liquid expansion process as explained, as well as allowing for an additional evaporating source level higher than the primary source level. This allows such items to be considered as simultaneous coupling with both earth and solar sources. The earth source obviously would be developing vapor at the lower evaporating level, whereas the solar source would be developing vapor at the higher evaporating level. The actual mass flow involved in both streams could be approximately equal or could be widely varying as the demands and the sources allow. This simultaneous dual source capability would certainly allow the low side heat exchangers to be utilized most effectively and will avoid overpulling them. The potential capacity of such a compressor is significantly enhanced through utilization of the basic compressor displacement twice.

TECHNICAL ACCOMPLISHMENTS

- Development of computer models to simplify compressor performance analysis.
- Proving by actual test that the originally developed concept for compressor capacity control was not completely satisfactory from an energy reduction standpoint.
- Development of alternate unloading concepts that still allow necessary independent variation of internal compression ratio. The new concept is shown in this paper.
- Overall simplification of original compressor and system concepts, i.e., current approach favors a two-speed two-slide machine.
- Generation of predicted performance maps for various configurations of compressors.
- Prototype design has been completed yet some simplification redesign is now taking place.

FUTURE ACTIVITIES

Contract activities. The major tasks to be completed on this contract are as follows:

Performance estimating of the second generation prototype -- this is the 25-ton compressor and as the estimated performance is developed based on input from the first generation prototype, the delivered data base will be enhanced.

Procurement/building of second generation prototype -- this task will result in a second generation 25-ton prototype ready for test.

Testing of the second generation prototype -- it is expected that this task will bear out the estimated performance predictions pertaining to this 25-ton compressor.

Post-contract activities. It is expected that various performance estimates and experimental results can be used in order to provide a basis for synthesis of advanced solar systems. A fundamental key to Series Solar Heat Pump Systems of the future are the best possible compression concepts. It certainly is expected that this project will result in the proper development of these concepts. As such, commercialization of the various ideas should present no problem as the cost levels of the developed machines are not expected to be significantly above those of less capable production type compressors. It is expected that it will not be difficult to therefore identify the overall cost effectiveness of the enhanced performance of the developed machines as they relate to Series Solar and other potential Heat Pump Systems.

DEVELOPMENT OF A HIGH TEMPERATURE SOLAR POWERED WATER CHILLER
ENERGY SYSTEMS DIVISION, CARRIER CORPORATION
DE-AC-3-77CS31590
RICHARD A. ENGLISH
PERIOD OF CONTRACT: Begin OCTOBER 1977 End SEPTEMBER 1981

## OBJECTIVE

The objectives of this program are: to develop a high temperature solar powered air cooled 25 ton chiller utilizing 250 to 300°F solar hot water suitable for commercial and multi-family applications; to study, design, and build a prototype Rankine powered vapor compression cycle; and to demonstrate and evaluate performance through steady state and dynamic laboratory testing.

## DESCRIPTION OF WORK

The design concept chosen as a result of Phase I work is a dual loop Rankine power cycle driving a Rankine vapor compression cycle. A single working fluid, R-113, is used and both loops are connected to a common condenser (Figure 1). The prime mover is a high speed turbo compressor being developed under subcontract by Mechanical Technology, Inc. This hermetic machine consists of a single stage centrifugal compressor driven by a radial inflow turbine. An auxiliary motor is mounted on the shaft between the turbine and compressor and will be powered by a variable frequency inverter. The shaft assembly is supported on gas film bearings to eliminate oil from the system (Figure 2).

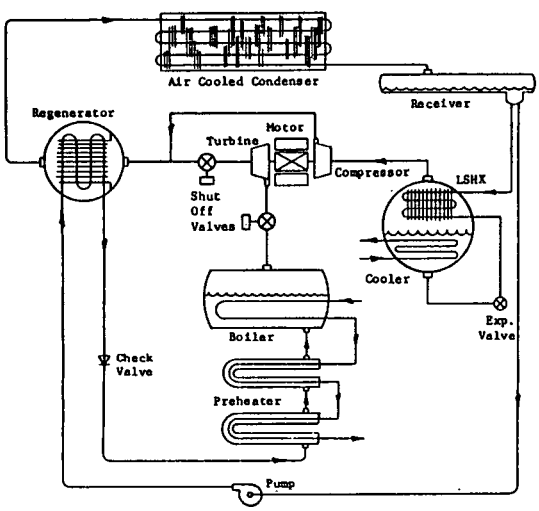


Figure 1. Air Cooled Dual Loop Rankine Driven Vapor Compression Cycle.

The chiller will be an air cooled outdoor packaged unit similar to conventional rooftop air conditioning equipment. Under design conditions of 45°F leaving chilled water, 95°F entering condenser water and 290°F entering solar water, the chiller will produce 25 tons of cooling at a COP predicted to be above 0.70 with low parasitic power.

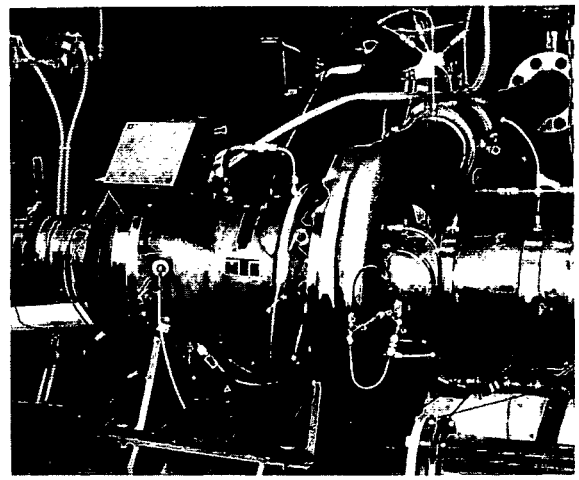

Figure 2. Carrier/MTI Solar Powered Water Chiller Turbo Machine Instrumented for Test.

Under Phase II, the detailed design of all of the components and the final assembly was completed; preliminary thermal stability tests on R-113 were carried out successfully. The project is now in Phase III. The chiller has been built in a water cooled configuration for initial turbo machine testing, and has been installed and instrumented in a special test facility. The turbo machine was built and successfully tested on air at MTI. The air tests verified the dynamic stability of the rotor running in the gas bearings at all speeds up to the design speed of 24,000 rpm. Initial spin testing now underway with R-113 in the solar chiller is directed at reproducing the air tests and confirming the rotor dynamics predicted for R-113 (Figure 3).

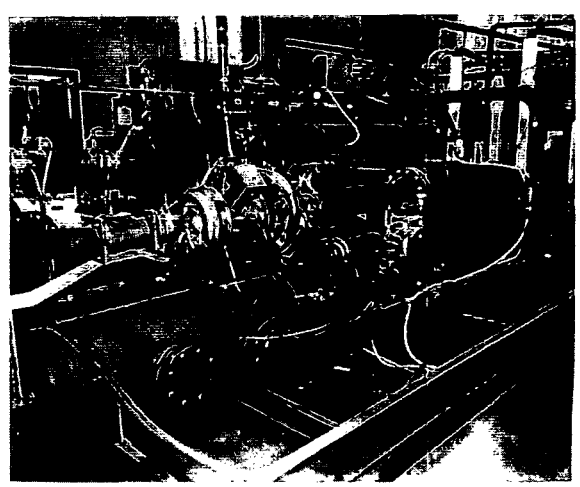

Figure 3. Carrier Solar Chiller Water Cooled Test Bed.

Another aspect of this testing is to verify motor/bearing cooling gas flow and the resulting cooling effectiveness. At the same time, experience with the boiler feed pump and its piping design is being gained. (Figure 4).

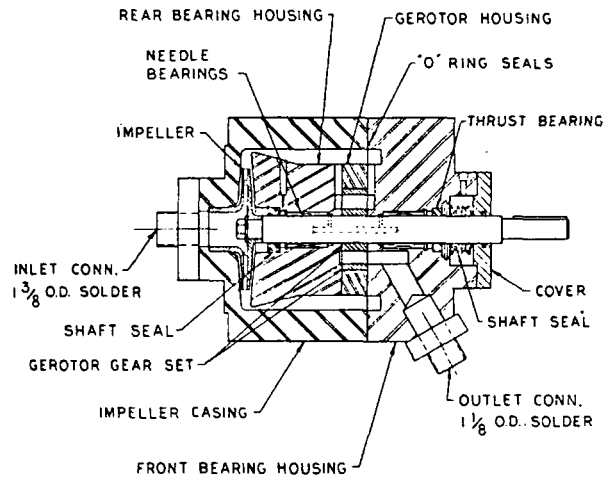


Figure 4. Cross Section of Prototype Boiler Feed Pump.

When these initial qualification tests are completed satisfactorily, performance mapping of the turbine and compressor will be undertaken. At the same time, a performance evaluation of the major heat transfer components and the overall cycle can be accomplished. Control strategies will also be explored and evaluated.

The air cooled configuration will then be built, controls will be finalized, and tests will be run to verify performance of the final chiller concept.

ACCOMPLISHMENTS

Cycle studies and preliminary turbo machine studies were completed under Phase I establishing the final conceptual approach and anticipated cost/performance.

The evaluation of the working fluid thermal stability has satisfactorily shown that R-113 has excellent life potential in an oil-free steel boiler at the maximum expected temperature, 320°F, for this application.

The detailed design of the turbo machine and the chiller has been completed.

The turbomachine has been completed and has successfully passed its qualification tests on air.

The chiller has been built in the water cooled configuration, has been installed in a test facility, instrumented and charged.

A two stage boiler feed pump has been developed and successfully tested on R-113 in a separate loop.

FUTURE ACTIVITIES

Initial qualification tests of the turbo machine are underway in the chiller test loop to evaluate the dynamic behavior of the turbo machine.

After dynamic testing, the turbo machine performance will be mapped over the expected operating range and evaluated.

The chiller performance will be mapped over the expected operating range and evaluated.

The chiller will be converted to the final air cooled configuration and performance testing will be undertaken to complete the scope of work of the original contract.

PUBLICATIONS/REPORTS

1. Development of a High Temperature Solar Powered Water Chiller, Phase I Technical Progress Report for Period September 1977 to June 1978. Volumes I, II, III, and IV. R. A. English, SAN-1590-1/314 (1978).

2. Development of a High Temperature Solar Powered Water Chiller, Phase II Technical Progress Report for Period June 1978 to March 1979. Volumes I and II. R. A. English.

3. Development of a High Temperature Solar Powered Water Chiller, Phase III, Technical Progress Report for Period April 1979 to September 1979, Volume I, R. A. English.

4. Development of a High Temperature Solar Powered Water Chiller, Proceedings of DOE Active Solar Heating and Cooling Contractors Meeting, March 26-28, 1980, R. A. English.

5. Development of a High Temperature Solar Powered Water Chiller, Proceedings of the Annual DOE Active Solar Heating and Cooling Contractors Review Meeting, March 26-28, 1980, CONF 800340

# LABORATORY DETERMINATION OF FROSTING AND DEFROSTING LOSSES FOR A HIGH EFFICIENCY AIR-SOURCE HEAT PUMP*

OAK RIDGE NATIONAL LABORATORIES
CONTRACT NUMBER W-7405-eng-26
W. A. MILLER AND R. D. ELLISON

## OBJECTIVE

The objectives of the air-source heat pump experiments at ORNL are to provide detailed system and component performance data, to quantify the dynamic losses, and to seek and evaluate methods for reducing these losses.

## BACKGROUND

Performance data for air-source heat pumps has customarily been presented as steady-state capacity and COP tabulated for a range of outdoor dry-bulb temperatures. Estimates of performance degradation in the "frosting range" of outdoor temperatures, [i.e., between 7 and -8°C (45 and 17°F)] are sometimes included, but losses due to on/off cycling of the heat pump are seldom noted. More recently, there has been increasing interest in the cycling and *de*frosting losses, but little attention has been paid to the losses incurred while frost is accumulating on the outdoor coil, the *frosting* losses. Detailed information on the cycling and frosting losses is sparse.

The results described in this paper are from the third of a series of laboratory tests (1, 2, 3, 4) performed at ORNL aimed at providing detailed characterization of the frosting losses as well as steady-state performance data. Cycling losses will be studied in our next series of tests with the goal of adding more detailed information to that provided in the pioneering experiments at the National Bureau of Standards(5).

## SUMMARY

A high efficiency split-system heat pump was installed in two separate air loops, with one loop housing the indoor unit and the other housing the outdoor unit. Sharp-edged orifice plates were installed in both test loops for the measurement of airflow rate across each unit. Thermocouple grid networks were used for the measurement of average air temperature entering and exiting each individual unit. Moisture level content was automatically controlled in the air loop containing the outdoor unit. A centrifugal fan, powered by a variable drive motor, was placed in series with the fan of the outdoor unit in order to maintain a zero static pressure drop, and thus to simulate free flow conditions across the outdoor unit. Maintenance of air temperature within the loop was accomplished through the use of auxiliary heaters controlled with a variable voltage transformer.

Calculations of the heat pump's performance based on air-side measurements were within 3% of that based on refrigerant side measurements. Refrigerant flow rate was measured using a turbine flow meter. Refrigerant temperatures and pressures were measured with thermocouples and pressure transducers connected at various strategic locations in the refrigeration circuit. Electric power consumption for all three motors was measured with Thermal-watt converters.

Performance of the heat pump was measured under steady-state, dehumidification, and frosting-defrosting conditions with major emphasis placed on the dynamic frosting operation of the system. The study encompassed an evaluation of the system and component performance for ambient temperature levels of 8.3, 4.4, 1.7, -1.1 and -8.3°C and for discrete humidity levels ranging from 50 to 90%.

For non-frosting dehumidification conditions, the performance of the heat pump improved somewhat with increasing relative humidities greater than 60%. At 4.4°C the performance of the system initially improved for relative humidities greater than 70% as a result of higher mass transfer rates of moisture to the outdoor heat exchanger. At ambients less than 4.4°C the rate of performance degradation due to frost formation increased with increasing relative humidity due to the increased rate of frost accumulation on the outdoor heat exchanger. Cumulative frosting loss coefficients were determined as a function of time, temperature and relative humidity. Losses due to defrosting were measured and found to be minimal for typical winter ambient conditions. However, for relative humidities greater than 80%, defrosting losses produced a 28% reduction in COP. Even under these severe frosting conditions, defrosting required only 3.17 minutes.

A time and temperature defrost sensor based on a 90 minute cycle would not adequately control frost build-up under severe frosting conditions. The outdoor coil was observed to be 90% blocked with frost within 50 minutes for ambient temperatures of 1.6°C and -1.1°C with high relative humidities. However under mild frosting conditions, unnecessary defrosting operations would occur. Measurements of air and coil surface temperatures indicate that a defrost scheme based on the temperature difference between ambient air and coil surface would avoid these problems and provide better protection for the compressor under severe frosting conditions.

## TECHNICAL ACCOMPLISHMENTS

- Demonstrated and characterized the improvement in system performance under non-frosting dehumidifying conditions. System performance improved slightly under conditions of high humidity in the non-frosting range. With 8.3°C ambient air temperature and relative humidity of 60% or greater, the outdoor coil surface temperature is below the dew point of the air, but above the freezing point of water. Thus, the coil surface is wet, but not frosted. The time averaged COP at this temperature was observed to increase from 2.9 at 60% relative humidity to 2.95 at 70% relative humidity and 3.05 at 80% relative humidity. The heating capacity responded similarly to COP with corresponding capacities of 9.1 kW, 9.6 kW and 10.0 kW. The improvements in COP and heating capacity are the result of increased latent heat contribution.

- Observed an atypical improvement in performance at 4.4°C for high relative humidities. Intuitively, it would seem that the increase in ambient relative humidity under frosting conditions should increase the rate of frosting. However, test results at 4.4°C for relative humidities greater than 70% reveal initial improvements in system performance as typified in Fig. 1. The improvement in system performance is

*This research was sponsored by the Office of Buildings and Community Systems, U.S. Department of Energy, under contract W-7405-eng-26 with the Union Carbide Corporation.

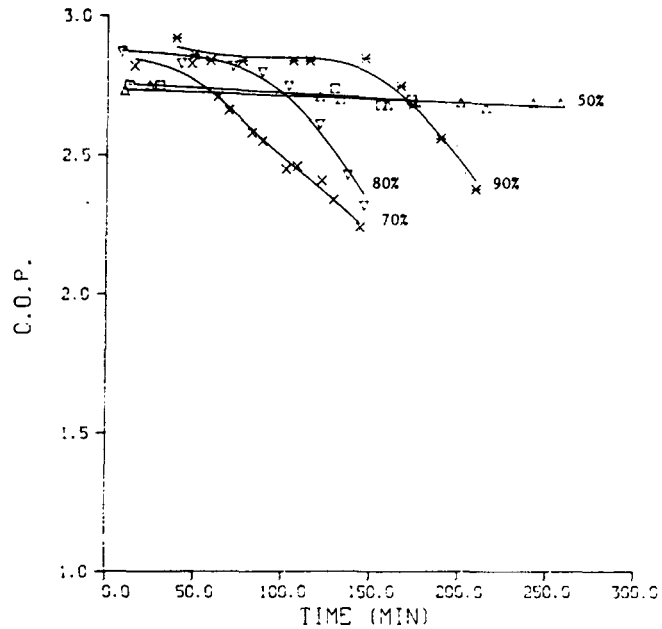


Fig. 1 COP observed at 4.4°C (40°F).

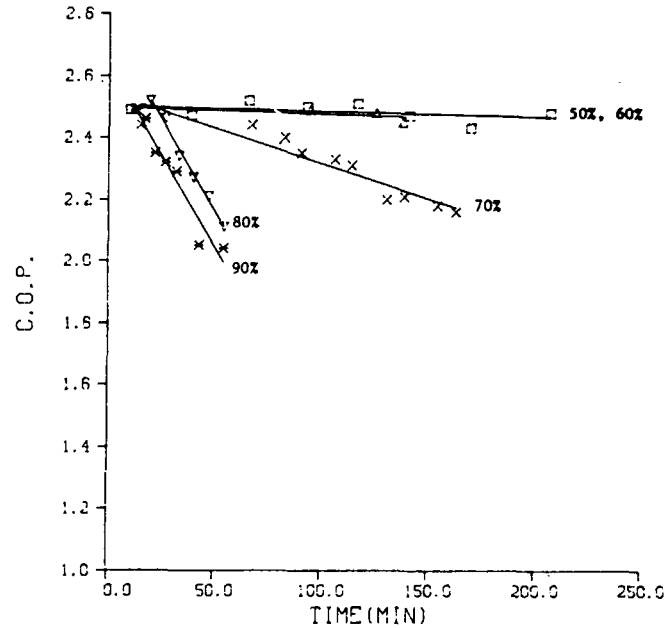


Fig. 2 COP observed at -1.1°C (30°F).

due to the increased mass transfer rate as the moisture content of the air is enhanced above moisture levels present at 60% humidity. The augmented mass transfer increased both evaporator tube wall temperature and compressor inlet pressure. This increase in tube wall temperature caused a delay in the onset of frosting as the relative humidity was increased. Frosting commenced 35 minutes into the 70% test run, while at 80% and 90% relative humidities frosting did not begin until 85 minutes and 160 minutes, respectively, of run time had elasped.

- Characterized heat pump performance degradation due to frost accumulation. For tests conducted at ambient temperatures of 1.6°C, -1.1°C, -6.7°C, and -8.3°C, increasing the relative humidity increased the rate of frost accumulation on the outdoor heat exchanger. The threshold of frosting was observed at 60% relative humidity. The temperatures at which frosting was observed ranged from 4°C to -7°C (the frosting zone). The trends of performance degradation are similar within the frosting zone, however, the *rate of performance degradation decreases as ambient* temperature decreases for a given relative humidity. Lowering the ambient temperature for a given relative humidity lowers the moisture content of the air and necessitates higher relative humidities for the accumulation of frost on the outdoor heat exchanger.

For relative humidities less than 70%, frosting was observed to be light during the typical tests at -1.1°C ambient air temperature. Incrementing the relative humidity to 70% produced a noticeable frost accumulation on the outdoor heat exchanger; however, the coil was only moderately frosted at termination of the experiment. The increase of relative humidity above 70% augmented the rate of frost accumlation on the outdoor heat exchanger, and 90% of the coil was blocked with frost within 50 minutes of operation. The COP dropped from approximately 2.5 to values of 2.1 and 2.0, respectively, for the 80% and 90% humidity tests, as seen in Fig. 2. Frost accumulation on the evaporator coil increased the thermal resistance from air to refrigerant within the evaporator. The drop in heat flow then lowered the evaporator tube wall temperature and refrigerant suction pressure at inlet to the compressor. The reduction of suction pressure yielded similar reductions in compressor inlet refrigerant density. This caused a decline in refrigerant flow rate and compressor power consumption with flow rate lowering proportionally more than power consumption. As a result, the COP and heating capacity both declined below observed steady-state values.

- Calculated frosting loss coefficients for use in a seasonal performance model. Cycling and defrosting have, in the past, been considered the major losses to efficient heat pump operation. However, very little attention has been given to the losses incurred due to frosting of the outdoor heat exchanger. As discussed previously, the accumulation of frost resulted in noticeable reductions of COP and heating capacity. The deviation in performance from steady-state operation results in losses, termed here as frosting losses. Using observed performance data, a frosting loss coefficient was calculated, and is usable in calculating the cumulative loss in capacity output as follows:

$$Q_{FROST\ LOSS} = (\dot{Q}_{HSS})(\tau)(1 - \eta) \ ,$$

where

$\dot{Q}_{HSS}$ = steady-state capacity

$\tau$ = time increment

$\eta$ = frosting loss coefficient

Similar frosting loss coefficients were calculated for the degradation of COP. Both COP and capacity degradation coefficients can be used in the formulation of a seasonal performance model which includes losses due to frosting.

- *Defrost losses were observed to be minimal* except under severe frosting conditions. Compressor power consumption and chilling of the indoor air stream during defrost operation lowers the integrated COP and heating capacity of a heat pump. These

losses in COP and capacity were found to be minimal for relative humidity conditions of 70% and less within the frosting zone. However, under high humidity conditions, the defrost losses result in degradations of COP greater than 15% and losses in capacity greater than 8% for ambients of 1.6 and -1.1°C. The degradations of COP and capacity, due to frosting and defrosting for an ambient of -1.1°C, are listed in Table 1, and as seen, the losses increase with increasing relative humidity. At the higher humidities of 80% and 90%, the defrost losses become significant. However, only those regions that experience long periods of relative humidities greater than 80% would incur defrosting losses comparable to cycling and frosting losses.

Table 1. Degradation of performance due to frosting/defrosting at an ambient temperature of -1.1°C

| Relative humidity | Steady-state COP | COP with Frosting | COP with Frosting/ Defrosting | Total % COP Degradation | Total % Capacity Degradation |
|---|---|---|---|---|---|
| 70% | 2.54 | 2.36 | 2.25 | 11% | 8% |
| 80% | 2.75 | 2.39 | 1.99 | 27.6% | 24.7% |
| 90% | 2.63 | 2.31 | 1.87 | 28.8% | 25.8% |

NOTE: COP with frosting is the average COP calculated over the respective period of the frosting test.

- Demand defrost sensors will reduce losses in heat pump efficiency. The time and temperature defrost control is considered a reliable defrost sensor for periodic removal of frost accumulation on the outdoor heat exchanger. However, if the sensor is set for a 90 minute cycle, it will not defrost soon enough to prevent heavy frost accumulations under conditions of high relative humidity within the frosting zone. The sensor also allows the occurrence of false defrost operations when little, if any, frost has accumulated on the outdoor coil.

A demand defrost sensor, based on a temperature difference between ambient air and evaporator tube wall, would probably eliminate severe frost build-up and false defrosting of the outdoor coil. This style sensor, using temperature difference as its logic, would better sense the appropriate time to initiate a defrost cycle since it directly senses the effect of frost build-up. Experimental results for ambients, ranging from 4.4°C to -6.7°C, showed a 8.3C° (15F°) temperature difference to correspond well to the time at which defrosting should commence. The plots of temperature difference in Fig. 3, for an ambient temperature of -1.1°C, reveals the above observation. The more timely sensing of defrost initiation would reduce the frosting/defrosting losses, and would also better protect the compressor during the severe frosting operations.

## FUTURE ACTIVITIES

Presently an experimental test stand is being assembled for the upcoming cycling loss study of a conventional air-to-air split system heat pump. A continuous modulation heat pump was selected for the study because of the experimental versatility of the unit. The primary objective of the study will be the determination and understanding of cycling losses. The continuous modulation heat pump will be modified to operate as a conventional single-speed unit for the determination of these losses. Upon completion of the cycling study, other tentative studies could commence as follows:

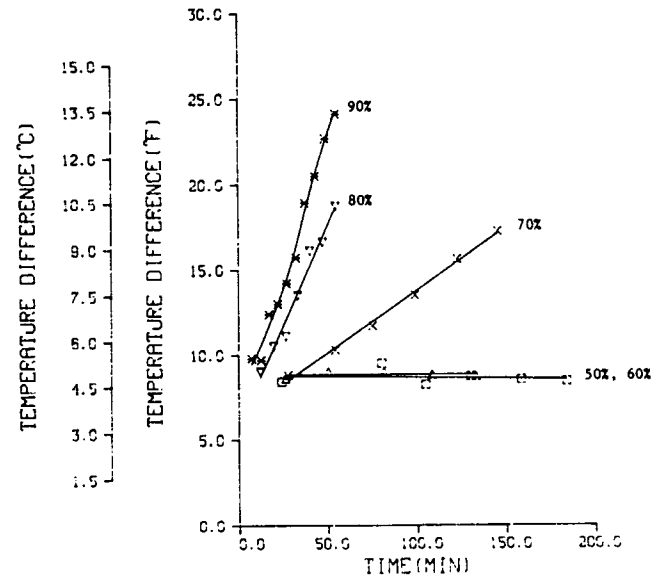


Fig. 3 Temperature difference between ambient air and evaporator tube wall as observed at -1.1°C (30°F).

- Effect of variable capacity on cycling losses
- Efficiency of variable capacity systems at different speeds and ambients
- Typify frosting losses for continuous modulation systems
- Typify performance for a two-speed heat pump.

## REFERENCES


1. Domingorena, A. A. and Ball, S. J., Performance Evaluation of a Low-First-Cost Three-Ton, Air-to-Air Heat Pump in the Heating Mode, Oak Ridge National Laboratory, ORNL/CON-18, October 1978.

2. Domingorena, A. A. and Ball, S. J., Performance Evaluation of a Selected Three-Ton Air-to-Air Hea Pump in the Heating Mode, Oak Ridge National Laboratory, ORNL/CON-34, January 1980.

3. Miller, William A., The Influence of Ambient Temperature and Relative Humidity on the Heating Mode Performance of a High-Efficiency Air-to-Air Heat Pump, M.S. Thesis, Dept. of Mechanical Engineering, University of Tennessee, Knoxville, Tennessee, 1980.

4. Miller, W. A. and Ball, S. J., Performance Evaluation at Various Ambient Relative Humidities of a High-Efficiency Air-to-Air Heat Pump in the Heating Mode (working title), ORNL/CON-69 (in preparation).

5. Parken, W. H., Beusoliel, R. W., and Kelly, G. E., Factors Affecting the Performance of a Residential Air-to-Air Heat Pump, National Bureau of Standards, Washington, D.C.

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