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ENERGY

CONSERVATION

MASTER

RESEARCH AND DEVELOPMENT OF AN AIR-CYCLE HEAT-PUMP WATER HEATER

FINAL REPORT

October 1, 1979

Prepared by:

John T. Dieckmann
Alve J. Erickson
Andrew C. Harvey
William M. Toscano

FOSTER-MILLER ASSOCIATES, INC.
350 SECOND AVENUE
WALTHAM, MASSACHUSETTS 02154

work performed for
OAK RIDGE NATIONAL LABORATORY

Operated by
UNION CARBIDE CORPORATION

for the

U. S. DEPARTMENT OF ENERGY

Division of Buildings and Community Systems

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Foster-Miller Associates, Inc.
350 Second Avenue
Waltham, Massachusetts 02154

Prepared under Subcontract 7226 for the

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830

Operated by:

UNION CARBIDE CORPORATION
for the
U.S. Department of Energy
Contract No. W-7405-ENG-26

ABSTRACT

A prototype reverse Brayton air cycle heat pump water heater has been designed and built for residential applications. The system consists of a compressor/expander, an air-water heat exchanger, an electric motor, a water circulation pump, a thermostat, and fluid management controls.

The prototype development program consists of a market analysis, design study, and development testing. The results of a market study of the reverse Brayton cycle water heater revealed that a potential residential market for the new high efficiency water heater is approximately 480,000 units per year. The retail and installation cost of the new high efficiency water heater is estimated to be between $500 and $600 which is approximately $300 more than a conventional electric water heater. The average payback per unit is less than 3-1/2 years and the average recurring energy cost savings after the payback period is approximately $105 per year at an average seasonal coefficient of performance (COP) of 1.7.

As part of the design effort, a thermodynamic parametric analysis was performed on the water heater system. It was determined that to obtain a coefficient of performance of 1.7, the isentropic efficiency of both the compressor and the expander must be at least 85 percent. The selected mechanical configuration is a reciprocating compressor/expander system with reed valves for the compressor, and poppet valves for the expander. The crosshead drive mechanism consists of a simple crank driving a double-acting piston through a crosshead. The heat exchanger configuration is a helical coil of fin tube wrapped around a hollow, blocked center core. The overall dimensions of the water heater are a diameter of 25 in. and a height of 73 in.

The prototype water heater system was fabricated and tested for the thermodynamic performance. The results of the development testing are summarized as follows:

- The electrical motor maximum efficiency is 78 percent
- The compressor isentropic efficiency is approximately 95 to 119 percent and the volumetric efficiency is approximately 85 percent
- The expander isentropic efficiency is approximately 58 percent and the volumetric efficiency is 92 percent

- A significant heat transfer loss occurred for the expander; approximately 16 percent of the refrigeration developed by the expander (extracted) was lost to the ambient

- For the expander, the heat transfer losses are of the same order of magnitude as the flow leakage losses

- The prototype heat pump system COP is 1.26 which is less than the design goal of at least 1.7.

From the results of the development testing, a number of recommendations were made for future development work:

- A small scale valve development program dedicated to reduce significantly the valve losses in the expander

- An investigation of the transient heat transfer phenomena

- A second generation design of the heat pump system.

FORWARD

Foster-Miller Associates has contracted with Union Carbide Corporation, Nuclear Division to perform the following work in the development of a Brayton air cycle heat pump for use in residential electric water heaters:

- Determine the potential market
- Design a prototype Brayton cycle water heater including a compressor/expander, an air-water heat exchanger, and electric motor, a water circulation pump, a thermostat, and fluid management controls
- Fabricate and perform development testing of the prototype system
- Perform endurance testing of the prototype system
- Prepare a Phase II program plan for the production, fabrication, and field demonstration of prototype systems.

As each of the above tasks was completed, a task summary report was prepared. Because the results of the development testing revealed a low system COP of 1.26 as compared to a design COP of 1.7, it was decided not to perform the endurance testing and the Phase II program plan preparation. Instead, it was believed to be more appropriate to disseminate the Phase I development information to the industrial and engineering community.

The final report consists of an executive summary and the three task summary reports:

- Market Survey
- Design Study
- Development Testing.

ACKNOWLEDGEMENTS

W.M. Toscano was the Program Manager and John Dieckmann, Alve Erickson, Andy Harvey, Hans Hug, and Hal Fuller were co-researchers. Messers Robert P. Jackson and P.G. Para of W.L. Jackson Manufacturing Company provided invaluable marketing information and projections for the marketing study.

Considerable technical and management support was provided by Messers Virgil Haynes of Oak Ridge National Laboratory (ORNL) and George Courville of DOE. Finally, Dr. Donald Walukas, the ORNL technical monitor for the program contributed significantly to the technical direction of the total program.

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SCOPE OF WORK

Task 1.1

Submit a detailed project plan for review and approval by the ORNL TM. This plan shall indicate, in more detail than the proposal program plan, final allocation of financial and personnel resources, timing of principal events that are to occur during execution of the project, decision points and progress milestones, technical approach, and other items of direct relevance to timely and successful accomplishment of the project objectives.

Submit a final project plan reflecting resolution of comments from the ORNL TM review of Deliverable 1D.

The Contractor shall not proceed with Task 1.2 or beyond until this plan is approved by the ORNL TM. No changes shall be made to the approved plan without approval of the ORNL TM.

Task 1.2

Using the approach presented in Section 3 of the seller's proposal, perform the studies necessary to determine the potential market for the improved water heater. These studies should address both the residential and commercial sectors in a disagrregated (by type of building and region) manner, as appropriate. Size of units, purchasers, and end user should be identified. All problems which impede commercialization of the improved water heater are to be identified, along with the solutions planned to overcome the problems. These problems should include, but not be limited to the anticipated selling price or price differential and to institutional and other factors that have a strong effect on buyer acceptance, manufacturer capital requirements, applicability to conventional manufacturing and installation practice, maintenance, safety, and regional factors. Develop and apply a rating method to indicate the best target market(s) for demonstrating the improved water heater. Rating criteria should include the potential for National energy savings, the time schedule on which such energy savings might realistically be achieved, and the difficulty of solving the problems impeding commercialization.

Submit, for review by the ORNL TM, a report documenting the basic information, methodology, criteria, other pertinent information, and results of the Task 1.2 study.

Submit a final report for Task 1.2, reflecting resolution of comments from the ORNL TM review of Deliverable 2D.

Task 1.3

Perform the R&D necessary to develop, fabricate, and test a unit(s) which is optimized for the target market identified in Task 1.2. Evaluations should be made of the tradeoff between size of equipment and energy storage capability, the reliability and cost-effectiveness of components, and the modifications required to adapt the equipment to other portions of the potential market. Testing should be performed under conditions which are realistic to the target market and which are compatible with NBS-FEA-FTC* testing and labeling requirements for water heaters to the greatest extent possible.

Submit for review by the ORNL TM, a preliminary design report including the test procedure and evaluations supporting the design of the improved water heater and the components selected.

---

* National Bureau of Standards-Federal Energy Administration-Federal Trade Commission

Submit a design, test, and evaluation report of the compressor/expander and the heat exchanger.

Task 1.4

Prepare a final summary report covering all aspects of the Phase I work, reflecting resolution of comments from the ORNL TM based on review of draft copy. Include Task Report 1.2 and ORNL TM reviewed and approved versions of report from Task 1.3.

Submit to ORNL TM a photo-ready copy and 200 copies.

RESEARCH AND DEVELOPMENT OF AN AIR-CYCLE HEAT-PUMP WATER HEATER

Executive Summary

October 1, 1979

Prepared Under Subcontract No. 7226 by

Foster-Miller Associates, Inc.
350 Second Avenue
Waltham, Massachusetts 02154

for

Oak Ridge National Laboratory
Oak Ridge, Tennessee 37830

Operated by:

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Contract No. W-7405-ENG-26

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1. INTRODUCTION

Energy consumption in the United States is increasing at a rate of over 4 percent a year [1]. In the search for new fuel alternatives and energy sources, there is a strong national concern to conserve the existing available energy sources. At present, the water heater accounts for approximately 15 percent of the energy consumption in a typical household. On the national level this represents 4 percent of the total energy consumption in the United States [2]. The purpose of this project is to develop and test electric water heaters whose efficiency is substantially greater than those units anticipated for the near term residential market.

A reverse Brayton air cycle heat pump water heater has been designed for residential applications. The system consists of a compressor/expander, an air-water heat exchanger, an electric motor, a water circulation pump, a thermostat, and fluid management controls.

There are several advantages to a reverse Brayton air cycle heat pump for water heating.

- The air cycle system is independent of an outdoor heat exchanger installation with fan, refrigeration lines, and defrosting mechanism

- The electric motor and compressor/expander need not be hermetically sealed or thermally coupled. Consequently, each component may be optimally designed and maintained independently

- The working fluid is air which has no adverse effects on the ozone layer surrounding the earth

- The thermal load which is the sensible heat of the water is easily matched with the sensible heat of the air in the air-water counterflow heat exchanger

- The coefficient of performance (COP) and heat pump capacity do not vary significantly with the ambient temperature. Therefore, if the system proves to be effective at standard ambient conditions, it can be considered for operation in any climate.

The program consists of a market analysis, design study and development testing. Each of these efforts is summarized in the following sections.

2. CONCLUSIONS AND RECOMMENDATIONS

The first phase of the research, development and demonstration of a high efficiency electric water heater has been completed. The tasks in Phase I included:

- Market survey
- Design study
- Prototype fabrication
- Prototype development testing.

The conclusions of the first phase are the following:

- The development program resulted in a prototype heat pump system with a COP of 1.26 which is less than the design goal of at least 1.7.

- The differences between experimental and design thermodynamic performance can be attributed to mass transfer losses and heat transfer losses.

- A heat pump system with a seasonal coefficient of performance (SCOP) of 1.7 and an initial capital cost of $300 more than a conventional electric water heater will exhibit an average payback of 3 years.

The results of a market study of the reverse Brayton cycle water heater revealed that a potential residential market for the reverse Brayton air cycle heat pump exists in all major regions in the United States. Based on initial projections of market penetration, it is expected that the high efficiency water heater will capture 25 percent of the electric water heater market in newly constructed homes and 10 percent of the electric water heater replacement market within the first

3 to 4 years of the market penetration effort. Consequently, it is estimated that the potential residential market for the new high efficiency water heater is approximately 480,000 units per year.

The retail and installation cost of the new high efficiency water heater is estimated to be between $500 and $600 which is approximately $300 more than a conventional electric water heater. The average payback per unit is less than 3-1/2 years and the average recurring energy cost savings after the payback period is approximately $105 per year at an average seasonal coefficient of performance of 1.7.

Because of the small unit volume and large variation in heater size and requirements, the commercial market is not as attractive as the residential market; however, the commercial market does represent approximately 13.8 percent of the total water heater market.

Test market cities were selected for the commercial demonstration and testing of the new high efficiency water heater. The country was divided into four major regions and a city was selected from each region. The cities selected are the following:

Region	Test Market City
Northeast	Philadelphia
South	Jacksonville
North Central	Chicago
West	Los Angeles

These cities offer the greatest potential for energy cost savings, new housing starts, replacement market, temperature range variation and water quality characteristics. It is recommended that new high efficiency electric water heaters

The valve losses are comprised of throttling losses and lift response time losses. Depending on the mechanical design configuration, the compressor/expander isentropic efficiency estimates vary between 78 and 92 percent. The mechanical design configuration was optimized in order to minimize the thermodynamic losses occurring in the compressor/expander system. The piston diameter and stroke are 6 and 2 in., respectively, and the valve diameter and lift are 2 and 0.3 in., respectively. The engine speed is 1150 rpm.

A preliminary mechanical design study was performed in order to reduce the losses in the compressor/expander. Various valve types, valve drive mechanisms and crosshead mechanisms were investigated. The selected mechanical configuration is a reciprocating compressor/expander system with reed valves for the compressor, and poppet valves for the expander. The reed valves are pressure actuated and the poppet valves are operated by levers which are directly coupled to cam lobes located on the crankshaft. The crosshead drive mechanism consists of a simple crank driving a double-acting piston through a crosshead.

The air-water heat exchanger has been designed for a counterflow heat exchanger effectiveness of at least 85 percent. The heat exchanger configuration is a helical coil of fin tube wrapped around a hollow, blocked center core.

A system package has been designed. The heat pump fits beneath the water storage tank; the overall dimensions of the water heater are a diameter of 25 in. and a height of 73 in.

A prototype water heater system was fabricated and tested for the thermodynamic performance. The results of the development testing are summarized as follows:

be placed in Philadelphia, Jacksonville, Chicago and Los Angeles, and that one new high efficiency electric water heater remain at FMA for long-term testing and performance evaluation. The total number of the new high efficiency electric water heaters to be tested will be determined during the preparation of the program plan for Phase II of the research, development, and demonstration of a high efficiency electric water heater.

It is recommended that at the conclusion of Phase II, the market study report be updated and expanded. The revised market study report would include the thermodynamic and reliability performance results of the testing of the sample units as well as marketing data, such as manufacturing cost, installation cost, operating cost, consumer acceptance attitude, city and state government attitudes, etc. The report would include the development of a model code for the installation and operation of a reverse Brayton air cycle heat pump water heater. Finally, the market study report would include a more detailed analysis of the commercial market and a re-evaluation of the feasibility of employing a high efficiency water heater in the commercial market place.

As part of the design effort, a thermodynamic parametric analysis was performed on the water heater system. The effects of ambient temperature, air-water heat exchanger effectiveness, pressure ratio and compressor/expander isentropic efficiency on the coefficient of performance of the complete system were investigated. It was determined that to obtain a coefficient of performance of 1.7 the isentropic efficiency of both the compressor and the expander must be at least 85 percent.

An analysis was performed on the compressor/expander in order to quantify the valve losses, heat transfer losses and mechanical losses. It was determined that the major sources of irreversibility are the valves and valve drive mechanism.

- The electrical motor maximum efficiency is 78 percent at a motor speed of 1140 rpm and a motor capacity of approximately 1.3 hp

- The prototype heat pump system COP is 1.26 which is less than the design goal of at least 1.7

- The results of the valve tests indicate that the experimental compressor and expander valve pressure drops are less than the predicted valve pressure drops

- The compressor isentropic efficiency is approximately 95 to 119 percent and the volumetric efficiency is 85 percent

- The expander isentropic efficiency is approximately 58 percent and the volumetric efficiency is 92 percent

- The flow leakage past the poppet valves during operation is approximately 3.5 to 4.3 scfm which represents 15 to 17 percent of the total expander capacity

- A significant heat transfer loss occurred for the expander; approximately 16 percent of the refrigeration developed by the expander (extracted work) was lost to the ambient

- For the expander, the heat transfer losses are of the same order of magnitude as the flow leakage losses

- From the results of the development testing, it is projected that with new valve designs, a system COP of approximately 1.57 may be achieved.

From the results of the development testing, a number of recommendations were made for future development work:

- A small scale valve development program should be initiated to develop valves such as rotary valves which should significantly reduce the valve losses in the expander and mechanical losses. In addition, the initial capital cost of the heat pump system would be reduced.

- An investigation of transient heat transfer phenomena should be undertaken. The results will provide heat pump/engine designers with better information to predict quantitatively the effects of transient heat transfer on the system's overall thermodynamics performance.

- In the event the alternate valve development proves to be successful, a second generation design of the heat pump system should be contemplated. The decision should be based on the updated projected heat pump system COP, revised market study results, and the energy environment of the United States.

3. MARKET STUDY

A market survey of a reverse Brayton air cycle heat pump water heater for residential use was performed. A significant portion of the input to the study was obtained from Jackson Manufacturing, Inc. of Chattanooga, TN. For the study, it was expected that the SCOP of the heat pump will be 1.7 and the water storage capacity will be 80 gal. (The experimental results of the project resulted in a SCOP of the prototype of 1.26.)

A market study was performed in order to:

- Determine the potential residential and commercial markets
- Identify a mechanism for commercialization
- Select suitable target market cities for the commercialization, demonstration and testing of the new high efficiency electric water heater.

A potential residential market for the reverse Brayton air cycle heat pump exists in all major regions in the United States. The residential market consists of the new housing market and the replacement market, of which 60 percent is for the replacement market. With respect to the new housing market, the South Atlantic and Western states will lead the remainder of the country in the growth of housing starts. However, for the replacement market, the Northeast and North Central will lead the remainder of the country.

Based on initial projections of market penetration, it is expected that the high efficiency water heater will capture 25 percent of the electric water heater market in newly constructed homes and 10 percent of the electric water replacement market within the first 3 to 4 years of the market penetration effort.

Consequently, it is estimated that the potential residential market for the new high efficiency water heater is approximately 480,000 units per year.

Because of the small sales volume and large variation in heater sizes and requirements, the commercial market is not as attractive as the residential market. However, the commercial market does represent approximately 13.8 percent of the total water heater market in terms of sales dollars.

The successful commercialization of the water heater will depend on factors such as initial capital investment for manufacturing facilities, appropriate government incentives or the existence of a "market pull" program, consumer acceptance, and potential energy cost savings.

An investment of approximately $7,025,000 will be required for a new facility for the manufacture of the new high efficiency water heater. However, many of the existing water heater manufacturers have the majority of the equipment, space, and inventory. Consequently, only a capital investment of the order of $2,500,000 will be required.

The government could create "market pull" by:

- Offering tax incentives to the consumers
- Inititating legislation favorable for the sales of the high efficiency water heater
- Creating favorable building codes
- Offering low interest loans.

Government incentives for the manufacturer are low interest loans for the required initial investment capital, tax credit, accelerated depreciation credits in new facilities and equipment, and legislation favorable to the successful market penetration and high volume sales of the new high efficiency water heaters.

The retail and installation cost of the new high efficiency water heater is estimated to be between $500 and $600 which is approximately $300 more than a conventional electric water heater. The average payback per unit is less than 3-1/2 years. Also, the average recurring energy cost savings after the payback period is approximately $105 per year at an average seasonal coefficient of performance (SCOP) of 1.7.

Test market cities were selected for the commercial demonstration and testing of the new high efficiency water heater. The country was divided into four major regions and a city was selected from each region. The cities selected are the following:

Region	Test Market City
Northeast	Philadelphia
South	Jacksonville
North Central	Chicago
West	Los Angeles

These cities offer the greatest potential for energy cost savings, new housing starts, replacement market, temperature range variations, and water quality characteristics. It is recommended that new high efficiency electric water heaters be placed in Philadelphia, Jacksonville, Chicago, and Los Angeles, and that one new high efficiency electric water heater remain at Foster-Miller Associates for long term testing and performance evaluation. The total number of the new high efficiency electric water heaters to be tested will be determined during the preparation of the program plan for Phase II of the Research, Development and Demonstration of a High Efficiency Electric Water Heater.

It is recommended that at the conclusion of Phase II, the market study report be updated and expanded. The revised market study report would include the thermodynamic and reliability performance results of the testing of the sample units as well as marketing data, such as manufacturing cost, installation cost, operating cost, consumer acceptance attitude, city and state government attitudes, etc. The report would also include a development of a model code for the installation and operation of a reverse Brayton air cycle heat pump water heater. Finally, the market study report would include a more detailed analysis of the commercial market and a reevaluation of the feasiblity of employing a high efficiency water heater in the commercial market place.

4. DESIGN STUDIES

A design study was performed to determine the mechanical/ economic viability of a reverse Brayton air cycle water heater. A summary of the prototype design specifications selected for the successful introduction of the new water heater product are presented in Table ES-1. Also described is a conventional electric water heater. The significance of the thermal efficiency of the conventional electric water heater and the coefficient of performance of the high efficiency water heater is that for a given water mass and temperature rise, approximately twice as much input electrical energy is required for the conventional water heater.

Prior to undertaking the detailed design and specification of the system components, an overall system design study was performed, quantifying the effect of the performance characteristics of the individual components on overall system performance.

The system configuration that was studied, an open polytropic, regenerative, reverse Brayton cycle, is shown in Figure ES-1. Ambient air is drawn into the system and sprayed with water. The water spray reduces the air temperature but increases the air moisture content, leaving the total enthalpy of the air unchanged. This reduction in temperature increases the amount of heat that can be recovered in the recuperator. The air then passes through a recuperator where it is preheated by the compressed outlet air from the air-water heat exchanger. The preheated air passes through a second water spray, where the resulting reduced air temperature and specific volume results in less compression work per unit mass flow of air. The air is then compressed, thus increasing its temperature. The hot compressed air then passes through the air-water heat exchanger, counterflow to the water, cooling the air and heating the water to the desired use temperature. The latent heat of the moisture from both

Table ES-1. Design Specifications of the High Efficiency Water Heater System and a Conventional System

Parameters	High Efficiency Systems	Conventional Systems
Water usage rate	80 gal per day	80 gal per day
Hot water temperature	140°F	150°F
Recovery time	8 hr	
Maximum temperature rise	100°F	100°F
Water storage tank capacity	80 gal	52 gal
Compressor/Expander mass flow capacity	33 scfm	--
Heat pump heat rate	3.0 kW	--
Auxiliary electric heating element heat rate	2.5 kW	--
Conventional electric heating element heat rate	--	4.5 kW
Maximum water heater system dimensions	76 in. long by 26 in. diam	50 in. long by 22 in. diam
Thermodynamic coefficient of performance	1.7 or greater	--
Thermal efficiency	--	83 percent
Reliability	Mean time between failure (MBTF) of 25,000 hr or more	
Retail price	$600 or less	$300 or less

water sprays is then recovered via condensation in the air-water heat exchanger. The warm compressed air is passed through the recuperator where it preheats the incoming ambient air. The compressed air is then expanded, with the recovered work helping to drive the compressor. Air is exhausted from the expander at temperatures between -70°F and 20°F. It can be exhausted to ambient during the winter and can be used to augment air conditioning during the summer. (Air conditioning credit was not used in any of the analyses in this portion of the project).

System performance was evaluated over ambient conditions ranging from 0°F to 80°F and 0 to 100 percent relative humidity (RH). A wide range of component performance characteristics (including the elimination of the water sprays and recuperator) was considered.

The conclusions of the design study can be summarized as follows:

- Water sprays produce an insignificant increase in COP for RH values between 0 and 100 percent

- A recuperator will produce a small (less than 4 percent) but not economical increase in COP under low ambient temperature conditions

- Compressor/expander isentropic efficiency strongly affects the heat pumping COP, so that the compressor/expander was designed to achieve highest efficiency. The effect of the compressor/expander isentropic efficiency, cycle pressure ratio, and ambient temperature on COP is illustrated in Figure ES-2. An average annual COP of 1.7 can be attained with an 85 percent efficiency compressor/expander; an annual COP of 2.0 can be attained with a 90 percent efficient compressor/expander.

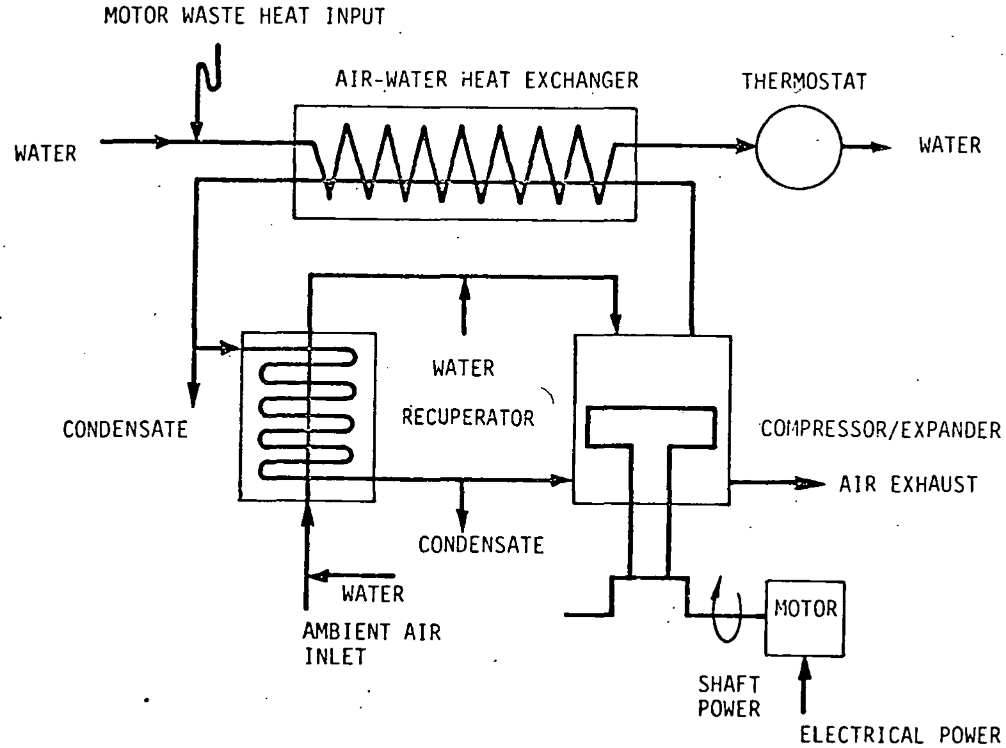


Figure ES-1. Flow Diagram for the Regenerative Brayton Cycle Considered for the High Efficiency Water Heater

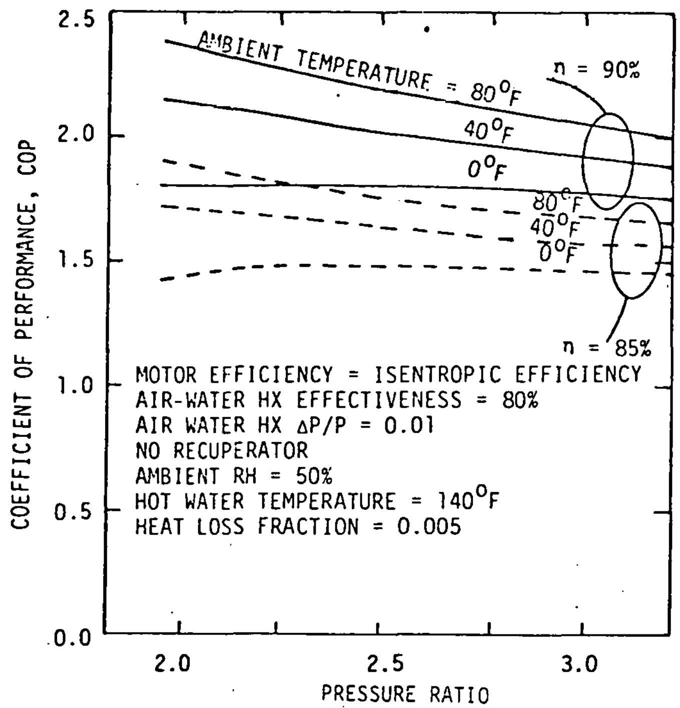


Figure ES-2. The Effect of Pressure Ratio and Ambient Temperature on the Coefficient of Performance

- The optimum pressure ratio of 3.0 represents a compromise among the value of higher COPs attained at lower pressure ratios, the cost of the higher compressor/expander displacement required at lower pressure ratios, and the required hot water temperature

- It is cost effective to design the air-water heat exchanger for high heat transfer effectiveness in excess of 85 percent.

The marketing study revealed that to make the high efficiency water heater viable in the market place, it is necessary that the system COP be 1.7 or greater. In order to achieve this performance goal it is necessary that the individual system components be designed to minimum design specifications. The ultimate system design goal specifications are as follows:

Sytem Component	Performance Goals
Compressor/Expander Isentropic Efficiency	$\geq$ 85 percent
Air-Water Heat Exchanger Effectiveness	$\geq$ 85 percent
Pressure Ratio	= 3.0
Electrical Motor Efficiency	$\geq$ 85 percent
Coefficient of Performance	$\geq$ 1.7

The reciprocating type of machine was selected for the first demonstration of the air-cycle water-heating application for two important reasons; the time and risk associated with the development of this type machine is small, and the possibility of achieving very high efficiency is highly likely.

A high coeffcienct of performance for the reverse Brayton cycle water heater requires high isentropic efficiencies for the compression and expansion processes. The major categories of losses which lower the overall efficiencies of a compressor or expander may be grouped as mechnical losses due to the bearings, seals, ring-friction, etc. and aerodynamic/thermodynamic losses due to pressure losses in the flow path, heat transfer, valve losses, etc. In order to optimize the aerodynamic performance of the air compression and expansion processes subject to the constraints of finite valve opening time and limited flow areas for inlet and exhaust valves, the governing equations were developed and programmed for solution on a digital computer.

A summary of the range of values for the thermodynamic losses occurring in the compressor/expander system for various mechanical design configurations is shown in Table ES-2. It is evident that the major contributors to total losses are the valves and the valve drive mechanism. Depending on the design, the isentropic efficiency of the compressor/expander system may be expected to fall between 78 and 92 percent.

The mechanical design configuration was optimized in order to minimize the thermodynamic losses occurring in the compressor/expander system. The piston diameter and stroke are 6 inches and 2 inches, respectively, and the valve diameter and lift are 2 inches and 0.3 inches, respectively. The engine speed is 1150 rpm. The selected compressor/expander configuration for the system is summarized in Table ES-3.

The functional parts of the heat pump water-heater are the water tank, the air-water heat exchanger, the reverse Brayton cycle compressor/expander, a water circulation pump, a water thermostat flow control, and tank thermostat switch controls. All of these items are typical commercial hardware except the compressor/expander and the air/water heat exchanger. The heat

Table ES-2. Summary of Thermodynamic and Mechanical Losses Occurring in the Compressor/Expander

Type	Percent Loss
Valve losses	
• Valve area	1.6
• Time of opening	2.2
• Valve leakage	1.0 to 5.0
Internal heat transfer loss	1.0
Mechanical losses	
• Piston ring	0.4 to 1.6
• Crosshead drive mechanism	0.5 to 1.0
• Rod bearings	0.1 to 1.0
• Main bearings	0.1 to 1.0
• Valve drive mechanism	0.75 to 7.5
Total Losses	7.65 to 21.9

Table ES-3. Summary of the Selected Characteristics for the Compressor/Expander

Variable	Value
Suction pressure	15 psia
Discharge pressure	45 psia
Engine speed	1150 rpm
Piston diameter	6 in.
Piston stroke	2 in.
Valve diameter	2 in.
Valve lift	0.3 in.
Clearance volume	5 percent

exchanger can be designed and fabricated similar to existing hardware, but the compressor/expander has no known predecessor, although there is a vast background of reciprocating gas-dynamic machinery. A flow diagram of the prototype Brayton cycle water heater is shown in Figure ES-3.

A preliminary mechanical design study was performed in order to reduce the losses in the compressor/expander. Various valve types, valve drive mechanisms and crosshead mechanisms were investigated. The selected mechanical configuration is a reciprocating compressor/expander system with reed valves for the compressor and poppet valves for the expander. The reed valves are pressure actuated and the poppet valves are operated by levers which are directly coupled to cam lobes located on the crankshaft. The crosshead drive mechanism consists of a simple crank driving a double-acting piston through a crosshead. An external view of the compressor/expander is shown in Figure ES-4.

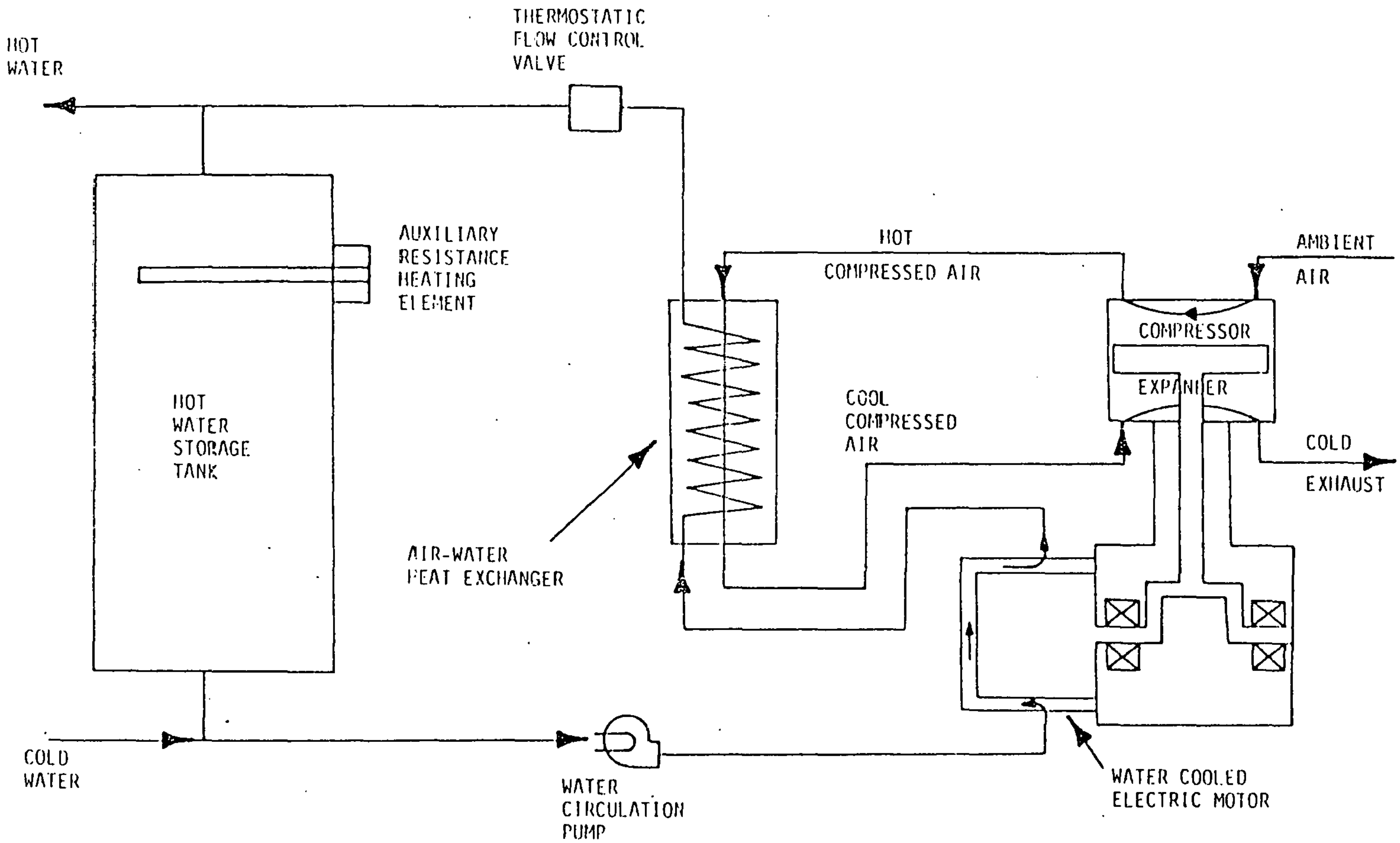


Figure ES-3. Flow Diagram of Prototype Brayton Cycle Water Heater

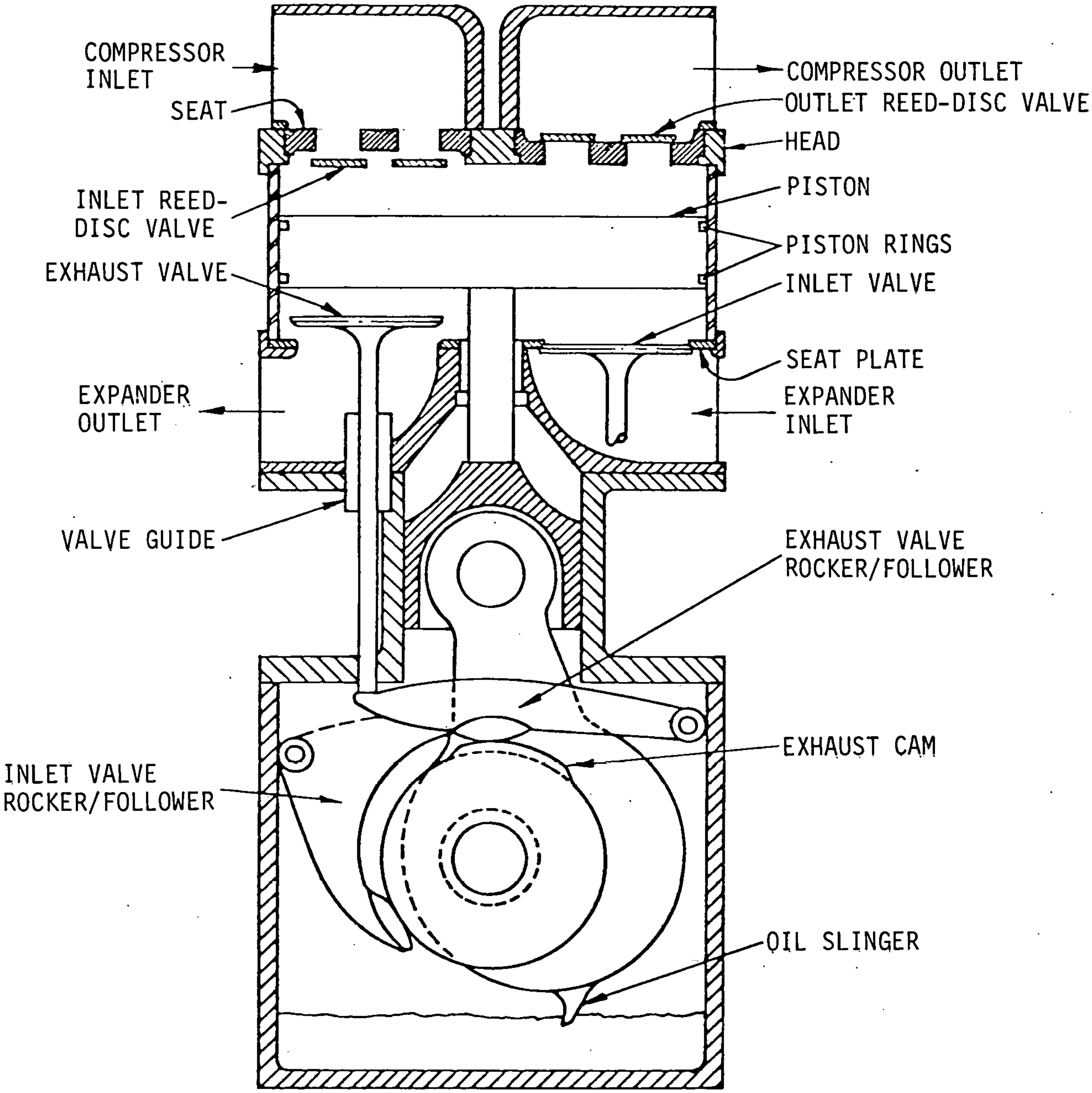


Figure ES-4. Schematic of the Compressor/Expander System

The water circulation loop consists of a water circulation pump, the air-water heat exchanger, and a thermostat to regulate the water through flow in response to changing operating conditions. The air-water heat exchanger has been designed for a counterflow heat exchanger effectiveness of at least 85 percent. The heat exchanger configuration is a helical coil of fin tube wrapped around a hollow, blocked center core.

The system will fit into a squat cylindrical package under a large water tank of usual proportions. The three major components, motor, compressor/expander, and heat exchanger lie in the horizontal plane mounted upon a shallow drum that serves both as structural support and also as the volume required for intake and exhaust silencing. Figure ES-5 shows the basic dimensions and proportions, to scale. The overall dimensions are a diameter of 25 inches and a height of 73 inches.

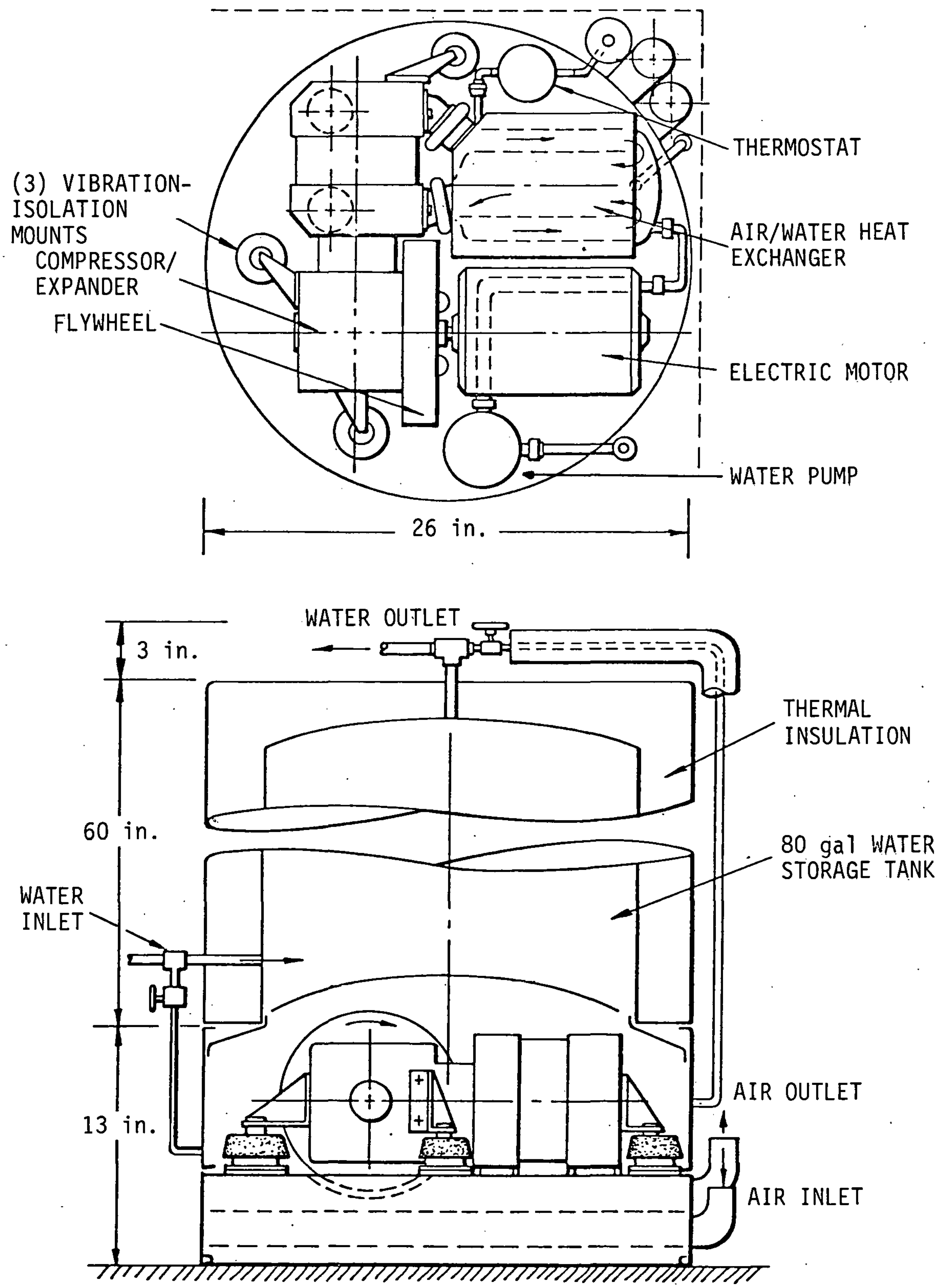


Figure ES-5. Schematic of the Brayton Air Cycle Heat Pump Water Heater System

5. DEVELOPMENT TESTING

A development testing program was conducted for the Brayton cycle water heater system. The test program consisted of the following:

- Performance measurements of the electrical motor
- Heat pump system preliminary performance tests
- Reed and poppet valve tests
- Expander test
- Heat pump system final performance tests.

A photograph of the experimental prototype is shown in Figure ES-6.

A 1.3 hp electrical motor was purchased from Gould, Inc. The motor was modified to include proprietary high efficiency features. A performance test was performed by Gould engineers and a summary of the test results is shown in Figure ES-7. The maximum motor efficiency is approximately 78 percent at a motor speed of 1140 rpm and a motor capacity of approximately 1.3 hp.

A series of preliminary tests were performed on the Brayton cycle water heater system. A summary of the preliminary test results is shown in Figure ES-8. The system COP is 1.26 compared to the design COP of 1.62. The experimental PV and Pθ diagrams for both the compressor and expander are presented in Figure ES-9. The experimental PV diagram very nearly correlates to the computer predictions.

From the preliminary experimental results, it was revealed that either a heat transfer or flow leakage loss mechanism occurs in the compressor and expander. The total heat transfer includes a steady-state heat transfer contribution and a transient heat transfer contribution. The steady-state heat transfer may occur along the cylinder walls or across the piston. The transient heat transfer is due to the fact that the piston

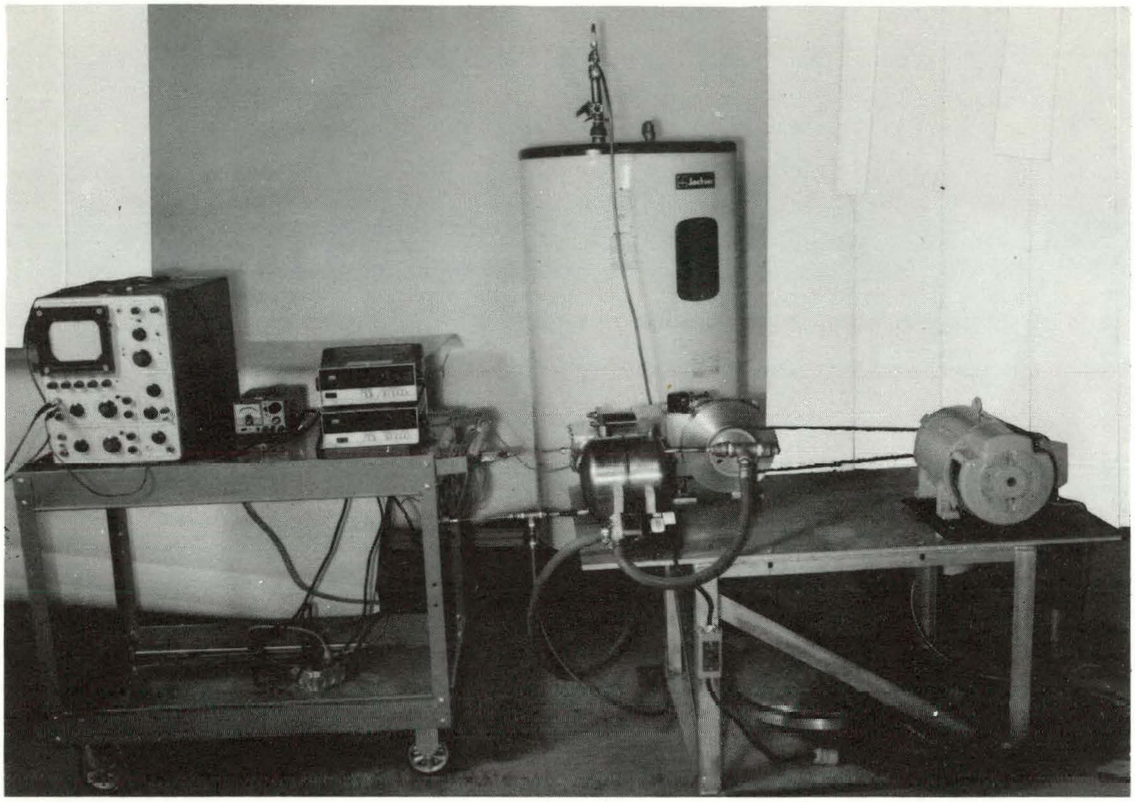


Figure ES-6. Photograph of the Prototype Brayton Air Cycle Heat Pump Water Heater System

and cylinder walls are not able to respond rapidly to the temperature fluctuations of the gas in both the expander and compressor. Superimposed on the steady-state heat transfer flux is the unsteady-state heat transfer contribution. The amplitude of the oscillating temperatures of the piston and cylinder walls adjacent to the gas is smaller than the oscillating temperature of the gas in both the expander and compressor. As a result of the large differences in the temperature amplitudes, there occurs an unsteady-state heat transfer between the gas and piston/cylinder walls. The consequence of the unsteady-state heat transfer is the presence of local transient heat transfer coefficients which may be 2 to 10 times larger than the steady-state heat transfer coefficients [3]. In addition, flow leakage may occur across the valves as well as the piston.

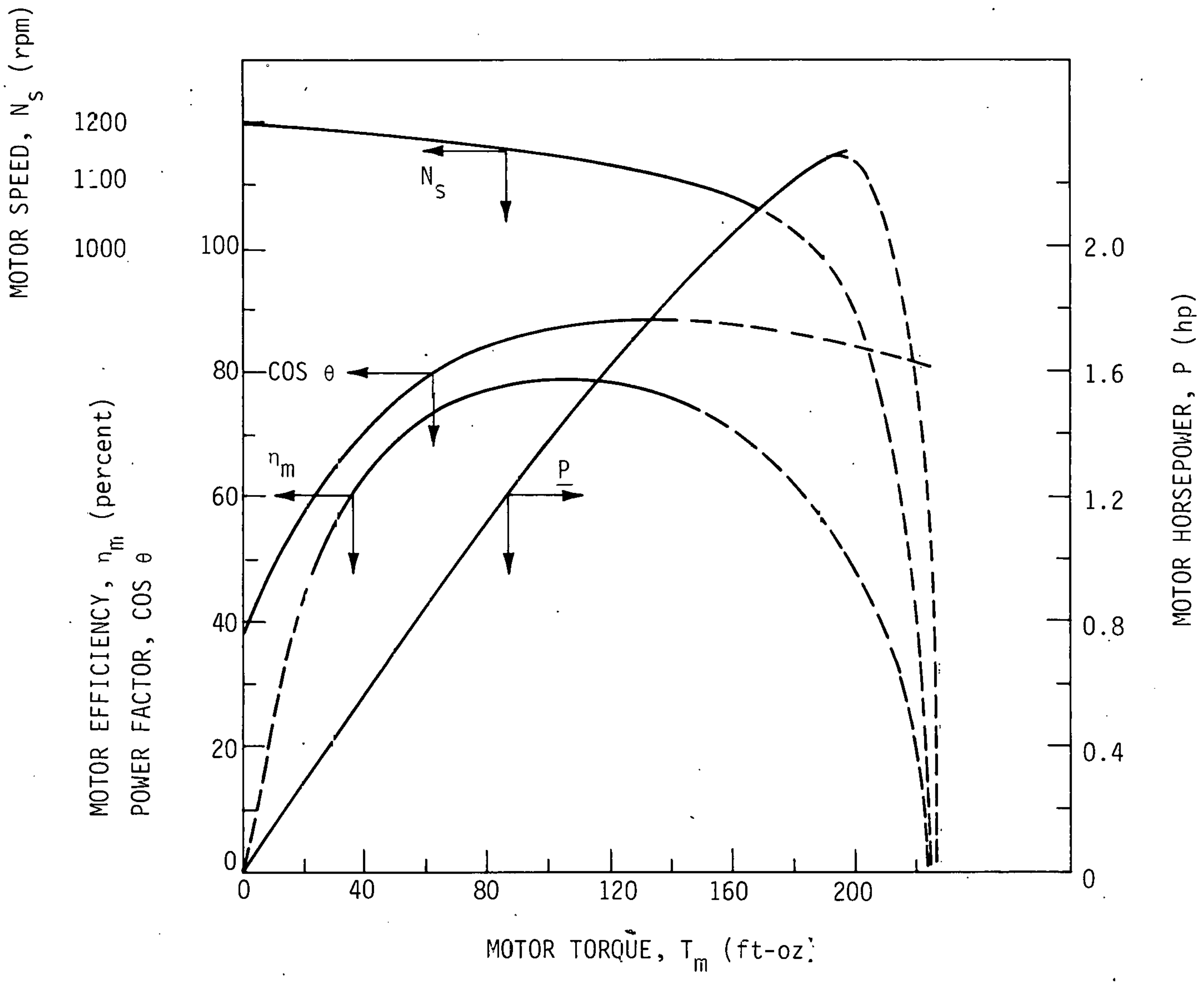
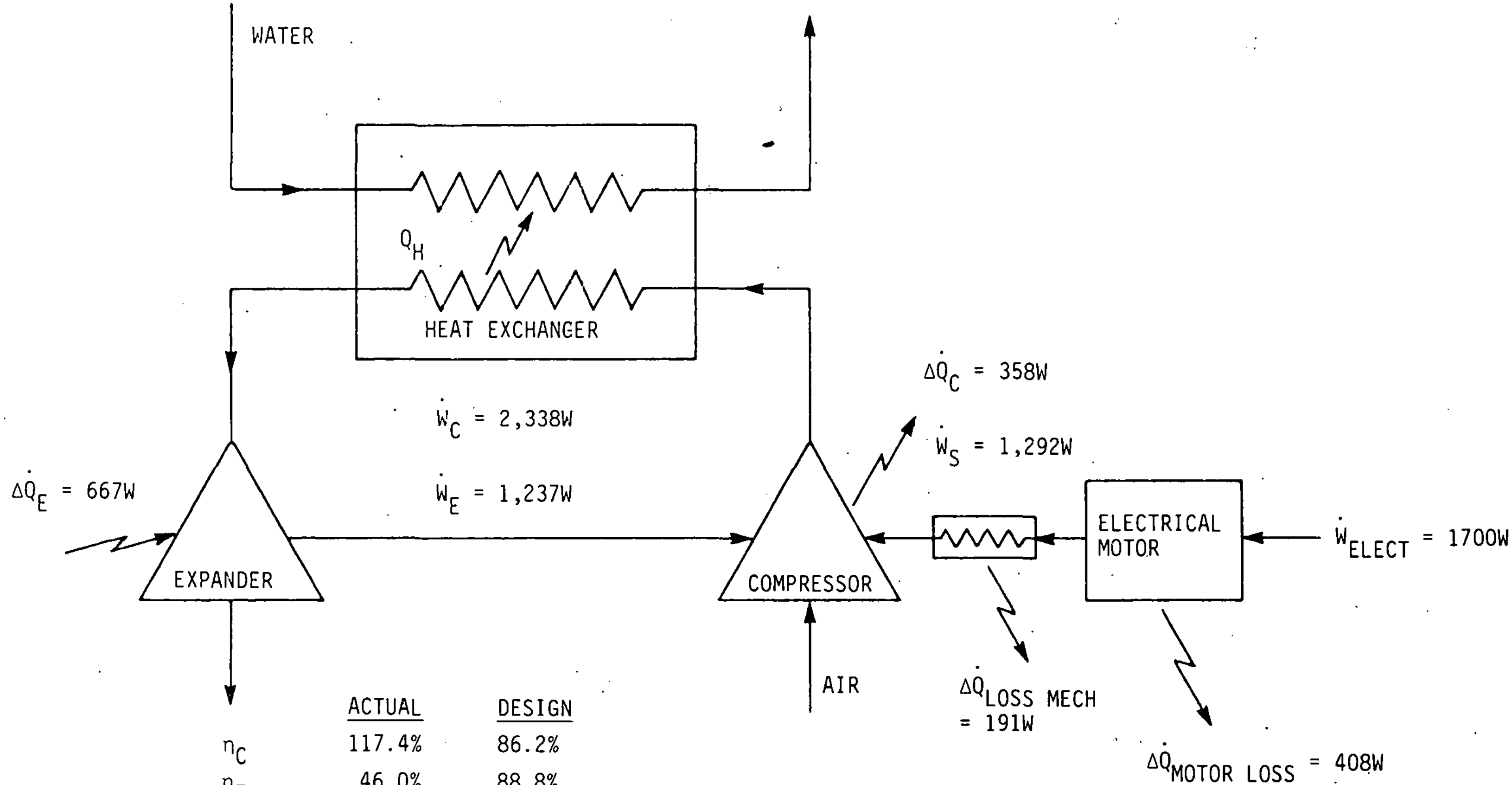
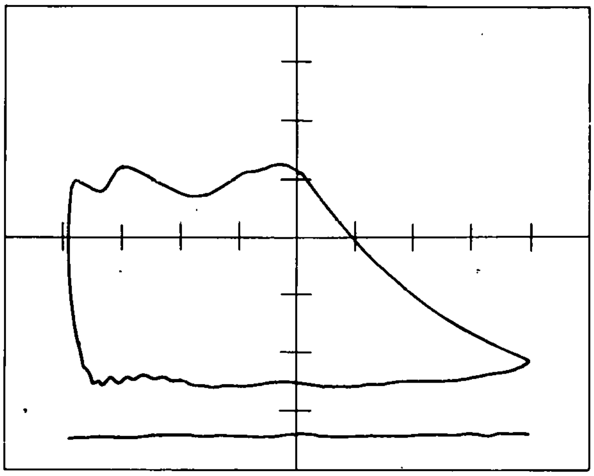


Figure ES-7. Motor Efficiency, Power Factor, Speed and Power Characteristics of the Gould Motor

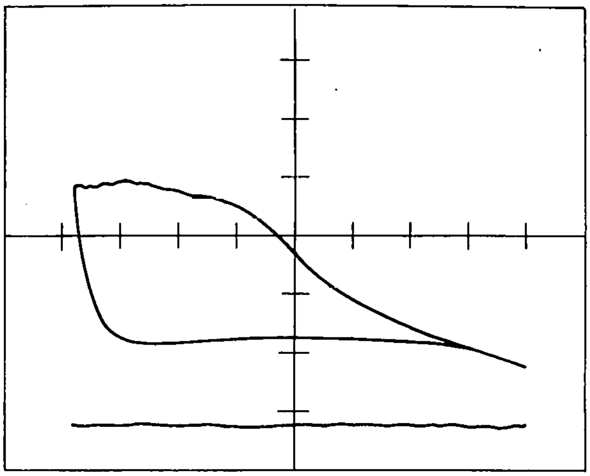


	ACTUAL	DESIGN
η_C	117.4%	86.2%
η_E	46.0%	88.8%
η_m	76.0%	80.8%
ε	90.8%	87.0%
$\dot{Q}_{H/}$	1,733W	2,302W
$\dot{Q}_H$	2,141W	2,626W
COP	1.26	1.62
COP_S	1.34	1.776

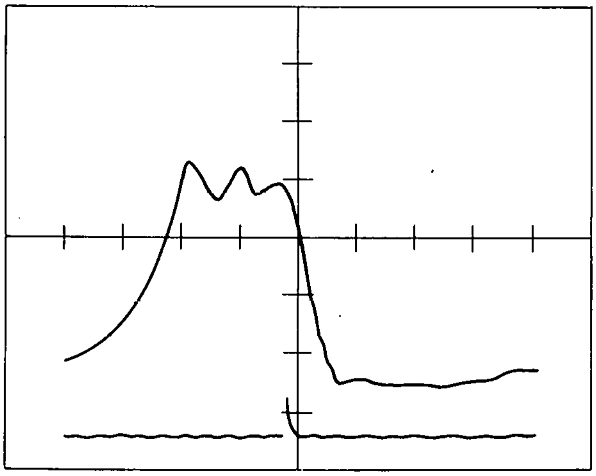
Figure ES-8. Preliminary Test Results



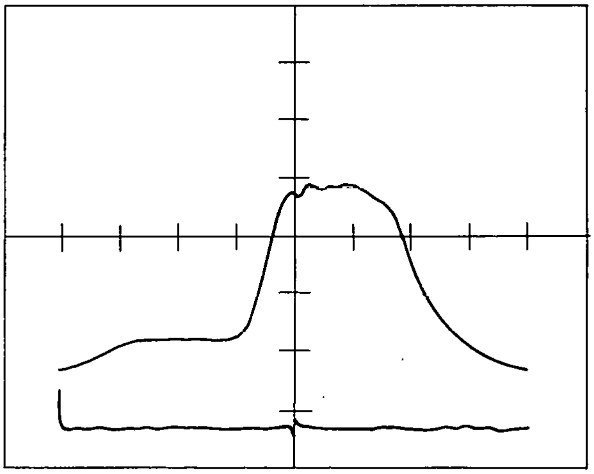
COMPRESSOR PV DIAGRAM



EXPANDER PV DIAGRAM



COMPRESSOR P-θ DIAGRAM



EXPANDER P-θ DIAGRAM

Figure ES-9. Compressor and Expander PV and Pθ Diagrams

To resolve the uncertainty of the sources and magnitude of the irreversibilities in the compressor and expander components, tests were performed. The components tested were valves, the compressor and the expander.

A series of reed and poppet valve tests were performed to determine the pressure drop losses with steady flow. The results of the valve tests revealed that the experiment flow discharge coefficients are greater than the flow discharge coefficient used for estimating the pressure drop across the valves.

The compressor was isolated from the expander and a series of tests were performed. The results of the compressor tests are:

- The flow leakage past the piston ring is approximately 1 scfm or 3 percent of the total compressor capacity
- The compressor indicated isentropic efficiency is 95 to 119 percent and the volumetric efficiency is 85 percent
- For all cases, the presence of water cooling of the compressor cylinder head, improves the isentropic efficiency because of an increase in the heat transfer
- At slower speeds the compressor isentropic and volumetric efficiencies are reduced because the ratio of the losses to compressor capacity is increased
- For most of the experimental runs, the heat transfer losses are greater than the flow leakage losses; however, both loss mechanisms contribute to the total compressor losses.

The expander was isolated from the compressor and a series of tests were performed. The results of the expander tests are as follows:

- The flow leakage past the poppet valves during operation is approximately 3.5 to 4.3 scfm which represents 15 to 17 percent of the total expander capacity.

- The expander isentropic efficiency is approximately 58 percent and the volumetric efficiency is approximately 92 percent.

- A significant heat transfer loss occurs for the expander. Approximately 16 percent of the refrigeration (extracted work) developed by the expander is lost to the ambient.

- The heat transfer losses are of the same order of magnitude as the flow leakage losses.

Upon completion of the components testing, a series of final system tests were performed. The results of the final system tests are summarized as follows:

- Compressor isentropic efficiency is 96.8 percent
- Expander isentropic efficiency is 70.5 percent
- The estimated COP is 1.2 to 1.4
- The heating capacity is approximately 2 kW.

The Brayton cycle water heater was designed to achieve an overall COP of 1.7 or greater. To satisfy this performance goal, it was necessary to design a compressor and expander with an isentropic efficiency of 85 percent or greater. Hence, the major potential sources of irreversibility were identified and the system was designed to minimize the losses. Except for the electric motor, the major losses for the heat pump system have been measured and are presented in Table ES-4. The first column represents the predicted or designed heat pump system losses. The second column represents the experimentally measured/calculated heat pump system losses. As can be observed from the total heat pump losses, the actual total losses exceed the design total losses by 500 to 700W. The reason for the large discrepancy is that both the actual compressor/expander losses and motor losses are more than double their respective estimated losses.

It is anticipated that with a new valve design, the expander isentropic efficiency may be increased from 70 to 75 percent which will result in a projected system COP of 1.57. The value is less than the design goal of 1.7 which according to the market survey represents a viable COP for a payback period of 3 years or less.

Table ES-4. Summary of Heat Pump System Losses

Component/Type	Design (W)	Actual (W)
Heat exchanger	(ε = 85 percent)	(ε = 90 percent)
Subtotal	104	25
Compressor/expander	($\eta_E = \eta_C$ = 87 percent)	(η_E = 70.5 percent, η_C = 96.8 percent)
Valve losses	178.9	312
Internal heat transfer losses	37.3	254
Mechanical losses	96.9 - 298.2	250
Subtotal	313.1 - 541.4	816
Motor	(η_M = 80 percent)	(η_M = 81 percent)
Subtotal	177.7	458*
Total heat pump losses	594.8 - 796.1	1300

*Estimated

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2. S.H. Dole, Energy Use and Conservation in the Residential Sector: A Regional Analysis; R-1641-NSF, PB 254, 468, Rand Corporation, Santa Monica, CA, June 1975.

3. K.P. Lee and J.L. Smith, Jr., "Influence of Cyclic Wall-to-Gas Heat Transfer in the Cylinder of the Valved Hot-Gas Engine," Proceedings of the 13th Intersociety Energy Conversion Engineering Conference, Society of Automotive Engineers, Inc., Warrendale, PA, 1978, pp. 1798-1804.

RESEARCH AND DEVELOPMENT OF AN
AIR-CYCLE HEAT-PUMP
WATER HEATER

Market Survey

October 1, 1979

Prepared Under Subcontract No. 7226 by

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1. INTRODUCTION

The purpose of the present market study is to determine the potential residential and commercial markets for the reverse Brayton air cycle heat pump water heater, to identify a mechanism for commercialization of the new high efficiency water heater, and to select suitable target markets for the demonstration and testing of the new high efficiency water heater.

The present study discusses the design requirements of the new high efficiency water heater such as the water storage tank capacity, practical water temperature, and the thermal insulation. The commercialization requirements for the successful market penetration such as projected unit cost, initial capital investment required for manufacturing facilities, and manufacturing, selling, and installation considerations are presented.

The residential market which consists of the new housing market and the water heater replacement market is examined in detail. The potential market as well as the potential payback period for the new high efficiency water heater are estimated.

Finally, a semequantitative cumulative qualitative evaluation procedure used to select test market cities for the testing and commercial demonstration of the high efficiency water heater is presented.

2. DESIGN REQUIREMENTS

The design constraints which must be considered for the successful introduction of a new water heat product are water heater size, water storage tank capcacity, practical water temperature, and thermal insulation.

2.1 Water Heater Size

The study for residential application was restricted to five standard water heater sizes between 50 and 120 gallons as shown in Table 2-1. A water heater with a storage tank less than 50 gallons would not adhere to the minimum storage capacity requirements specified by the Housing and Urban Development (HUD) property standards [2]. Conversely, a water heater with a storage tank larger than 120 gallons would have to be redesigned in order to meet the American Society of Mechanical Engineers (ASME) Pressure Vessel Codes [3]; resulting in an unacceptable increase in manufacturing cost.

Table 2-1. Summary of Existing Residential Water Heater Sizes

Water Heater Size (gal)	Water Tank Diameter (in.)	Jacket Diameter (in.)	Length (in.)
50	18	22	50-1/4
66	20	24	52-3/8
80	20	24	61-3/8
100	24	28	59-5/8
120	24	28	69-5/8

The maximum possible diameter of the water heater is dictated by the dimensions of standard size doors in houses. The nominal door size is 32 inches. A maximum width of 28 inches is recommended for the new high efficiency water heater. In order to maintain a high level of thermodynamic performance, it is necessary to design the water heater for low standby losses. A minimum of two inches of insulation between the water tank and jacket surrounding the water tank is required. This limits the maximum diameter of the water tank to 24 inches. The heat pump is approximately 10 inches in diameter and 20 inches long. Consequently, if the heat pump and the water storage tank are combined into one package for a 28 inch jacket diameter, the theoretical height for a 50 and 100 gallon water heaters are approximately 50 and 80 inches, respectively. A system larger than 80 inches would cause installation problems in some homes.

2.2 Water Storage Tank Capacity

The HUD minimum property standards 4900.1 and 4910.1 (Table 6015.2) [4] present storage capacity, input, and recovery rates for typical direct fired water heaters and the minimum one hour draw required for different size homes. The one hour draw is defined as the sum of the water storage in gallons and the recovery in gallons per hour provided by the power input of the water heater. The recovery rate is based on a 100°F water temperature rise. To meet the minimum HUD requirements, it is necessary that the water heater provide the minimum one hour draw. These standards are reproduced in Table 2-2. For example, a home with one bedroom and one to one and a half baths must have a water heater which provides a minimum of 30 gallons per hour for a one hour draw. With the same number of bathrooms, a two-bedroom house must have a water heater which provides a minimum of 44 gallons per hour for a one hour draw. The minimum one hour draw for various house configurations are also summarized in Table 2-2.

Table 2-2. Summary of HUD Requirements for Direct Fired Water Heater Capacities [4]

	Fuel	Gas	Electric	Oil	Gas	Electric	Oil	Gas	Electric	Oil	Gas	Electric	Oil
	Number of Bedrooms		1			2			4			–	
1 to 1½ baths	Storage (gal)	20	20	30	30	30	30	30	40	40	–	–	–
	Input (Btu/hr or kw)	27k	2.5	70	36k	3.5	70k	36k	4.5	70	–	–	–
	Draw (gal/hr)	43	30	89	60	44	89	60	58	89	–	–	–
	Recovery (gal/hr)	23	10	59	30	14	59	30	18	59	–	–	–
	Minimum one hour draw (gal/hr)	30	30	30	44	44	44	58	58	58		–	
	Number of Bedrooms		2			3			4			5	
2 to 2½ baths	Storage (gal)	30	40	30	40	50	30	40	50	30	50	66	33
	Input (Btu/hr or kw)	36k	4.5	70k	36k	5.5	70k	38k	5.5	70k	47k	5.5	70k
	Draw (gal/hr)	60	58	89	70	72	89	72	72	89	90	88	89
	Recovery (gal/hr)	30	18	59	30	22	59	32	22	59	40	22	59
	Minimum one hour draw (gal/hr)	58	58	58	70	70	70	72	72	72	88	88	88
	Number of Bedrooms		3			4			5			6	
3 to 3½ baths	Storage (gal)	40	50	30	50	66	30	50	66	30	50	80	40
	Input (Btu/hr or kw)	38	5.5	70k	38k	5.5	70k	47k	5.5	70k	50k	5.5	70k
	Draw (gal/hr)	72	72	89	82	38	89	90	88	89	92	102	99
	Recovery (gal/hr)	32	22	59	32	22	59	40	22	59	42	22	59
	Minimum one hour draw (gal/hr)	72	72	72	82	82	82	88	88	88	92	92	92

Since the high efficiency water heater is designed to provide a maximum recovery rate of 18 gallons per hour for a 100°F temperature rise, a 50 gallon water storage tank operating in conjunction with the reverse Brayton air cycle heat pump will provide 68 gallons per hour for a one hour draw. Similarily, for the 66, 80, 100 and 120 gallon water storage tanks, operating in conjunction with the heat pump, the maximum one hour draw values are 84, 98, 118 and 138 gallons per hour, respectively.

2.3 Practical Water Temperature

Even though the thermostats on residential electric water heaters are capable of being set at higher temperatures, most manufacturers and utilities recommend that the homeowner keep his water heater thermostat setting between 140°F and 150°F. Hot water is used for cleaning dishes (130°F to 150°F), washing clothes (100°F to 140°F), and personal use (80°F to 110°F). Consequently, because of the dishwater and clothes washer requirements, the minimum thermostat setting on the high efficiency water heater must be 140°F.

From the results of the "Mississippi District" water heater tests [5], it was found that with standard 50 gallon electric water heaters with no additional insulation, reducing the thermostat setting from 150°F to 140°F resulted in a monthly savings of 15 kilowatt hours in energy necessary to make up standby losses representing a five percent savings in energy consumption. This result is in rough agreement with other studies [6-9]. The savings are significant enough to justify having lower maximum thermostat settings in a water heater that is to be specifically sold as a high efficiency water heater.

2.4 Thermal Insulation

For electric water heaters, heat loss through the insulation and heat leaks through the fittings amount to 15 to 17 percent of the total daily electrical input [6], most of which, between 12 and 15 percent, is due to thermal losses through the thermal insulation. In general, for a 50 gallon water heater, the total heat loss amounts to 17 percent of the total daily electrical input [6]. A summary of standby losses (heat flux per unit surface area) in water heaters is shown in Table 2-3 [10]. The present industry standard (ASHRAE Standard 90-75) for the maximum allowable standby loss is four watts per square foot of the water storage tank surface area which represents a significant reduction from the previous standard of six watts per square foot of the jacket surface area [11].

Adding thermal insulation reduces thermal losses. According to Mutch [7], an increase in insulation from two to four inches results in an energy savings of eight percent for electric water heaters. Similarily, the American Gas Association [12] reported a six percent savings, Arthur D. Little [6] an eight percent savings, and ORNL [13] a seven percent savings with an additional two inches of thermal insulation.

In the past, fiberglass has been used as the thermal insulation material; however, for the same thickness, polyurethane is approximately twice as effective as fiberglass. Polyurethane will be used in the new high efficiency water heater.

Table 2-3. Summary of Standby Losses in Water Heaters [10]

Nominal Storage Capacity U.S. Gallons		Total Unit Shipment 1972 in Survey	Typical Tank Surface Area Square Foot	Standby Loss Total Watts Loss	Watts Per Square Foot 1972
20		108,888	11.5	96.85	8.42
30	Standard Tall	450,422	15.5	128.95	8.54*
	Stubby		13.5		
40	Standard Tall	567,266	20.0	153.77*	7.87*
	Stubby		17.5		
52	Standard Tall	400,018	22.8	167.79*	7.44*
	Stubby		19.9		
66		43,725	26.2	188.75	7.20

Watts allowed by GAS Specification = 4 watts per square foot

* These have been calculated using the 1972 shipment statistics to weigh the tank surface area by the percentage of standard tall models versus stubby models.

3. COMMERCIALIZATION REQUIREMENTS

3.1 Projected Unit Cost

A summary of the projected manufacturing costs of the high efficiency water heater is presented in Table 3-1. The most expensive item is the compressor/expander. The cost for this item will not vary significantly with size since the majority of the cost is attributed to labor rather than materials. The heat exchanger cost will also remain constant with size. The jacket and component costs include the materials and labor cost for the manufacture of the jacket, thermal insulation, anode, thermostat, dip tube, drain valve, and shipping carton.

Depending on the size, the total manufacturing cost of the high efficiency water heater varies between $256 and $356. Table 5 presents a summary of the manufacturing costs of a conventional water heater. Comparing the total manufacturing costs for the high efficiency water heater and the conventional water heater, one will observe that differential in the costs is approximately $195. The differential retail and installation cost is projected to be approximately $300.

3.2 Capital Investment Required

Table 3-3 presents an estimate of the capital equipment, manufacturing space, and inventory which must be purchased in order to commence the volume manufacture of the high efficiency water heater. To initiate a new manufacturing venture, an investment of approximately $7,025,000 will be required. However, for a typical water heater manufacturer having the majority of the equipment, space, and inventory required, an additional capital investment of the order of $2,500,000 will be required for the high efficiency water heater.

Table 3-1. Summary of the High Efficiency Water Heater Estimated Manufacturing Cost (1975 Dollars)

Water Heater Size (gal)	Water Storage Tank Cost ($)	Air/Water Heat Exchanger Cost ($)	Compressor/ Expanded Cost ($)	Piping & Fittings Cost ($)	Jacket & Components Cost ($)	Total Cost ($)
50	28.45	35.00	150.00	10.00	32.48	255.93
66	35.86	35.00	150.00	10.00	34.27	265.13
80	38.70	35.00	150.00	10.00	40.89	274.59
100	54.00	35.00	150.00	10.00	89.90	338.90
120	63.29	35.00	150.00	10.00	97.67	355.96

Table 3-2. Summary of the Conventional Electric Water Heater Manufacturing Costs

Water Heater Size (gal)	Water Storage Tank Cost ($)	Jacket and Components Cost ($)	Total Cost ($)
50	28.45	32.48	60.93
66	35.86	34.27	70.13
80	38.70	40.89	79.59
100	54.00	89.90	143.90
120	63.29	97.67	160.96

Table 3-3. Summary of Costs Required for Capital Equipment, Manufacturing Space, and Inventory Required for the Initial Manufacturing Phase of the High Efficiency Water Heater

Items		Costs ($)
Capital Equipment		
1. Press Equipment		
600 ton press for tank heads	$150,000	
150 ton press for light gauge parts	75,000	
300 ton press for jacket top, bottom and light gauge tank parts	125,000	
SUBTOTAL		$ 350,000
2. Tank shell roll (non automated)		10,000
3. Tank side seam welder		15,000
4. Glasslining dryer (installed)		150,000
5. Glasslining furnace (installed)		400,000
6. Ball mill, spray machines, conveyor lines, etc.		600,000
7. Tank bottom press		10,000
8. Tank head press		10,000
9. Tank head welder		15,000
10. Tank bottom welder		15,000
11. Assembly department		150,000
12. Jacket shell assembly capital equipment		275,000
13. Foam insulation capital equipment		175,000
14. Paint lines (booth, cleaning, drying, baking oven, conveyor)		300,000
15. Heat exchanger capital equipment		50,000
16. Compressor/expander manufacturing capital equipment		1,000,000
SUBTOTAL		$3,525,000
Manufacturing Space ($13 per square foot)		3,000,000
Inventory		500,000
TOTAL INITIAL COST		$7,025,000

3.3 Manufacturing, Selling, and Installation Considerations

The single largest constraint for the manufacture of the reverse Brayton air cycle heat pump water heater will be the successful manufacturing of the compressor/expander. There are two possible routes which a water heater manufacturer may pursue. The first route is to license the manufacture of the compressor/expander system to an existing compressor manufacturer. The second route is for the water heater manufacturer to invest capital for necessary compressor/expander tooling required for the manufacture of the compressor/expander. The technology required of a compressor manufacturer is much more sophisticated than that of the water heater manufacturer. As a result, if the water heater manufacturer decides to pursue the latter route, he will not only need to acquire the necessary tooling for the manufacture of the compressor/expander, but he will also need to acquire skilled laborers, manufacturing and engineering staff, sales and marketing staff, and field service staff who are familiar with compressor manufacturing technology.

In order to make the new water heater available to the maximum number of consumers, it would seem reasonable to attempt to have the product available through the maximum number of qualified distributors and installers. It is suggested that soon, after the initial marketing stage of the program, other water heater manufacturers and distributors are offered the opportunity to participate in the marketing of this new product. The business practice of one water heater manufacturer buying a specialized product from another manufacturer and marketing it under the buying manufacturer's name is not unknown to the water heater industry. For example, large commercial electric water heaters in the size range of 150 to 10,000 gallons and 15 to 10,000 kilowatts input are manufactured by National Steel Construction Company and are sold to almost all the major water heater manufacturers for resale under their own brand names. This has the effect of increasing competition by providing the same product to all of the major water heater manufacturers and reducing prices by increasing the manufacturing output of these specialized water heaters. The more manufacturers that offer this product, the more distributors throughout the country will have it available. This distribution system would increase the competition among the wholesalers and the installers and would increase the manufacturing volume produced by the sole manufacturer. As a result, the price of the new high efficiency water heaters would be kept low.

The eventual design of the new water heater will determine who will be capable of installing it. The installer will certainly be either a plumber or plumbing contractor, or an air conditioning/ refrigeration specialist. In addition, if the proposed water heater requires special skills for installation, it will be necessary to provide some sort of an educational program for the installers. These programs will be developed and initiated during the prototype and test market stages of the program.

3.4 Maintenance, Safety, and Regional Factors

The maintenance requirements of a conventional water heater include the periodic draining of the water storage tank. However, for the new high efficiency water heater, not only will the consumer be required to drain the water storage tank, but he will also be required to periodically change the oil in the crank case, check the oil level, and replace the intake air filter to the compressor. The compressor/expander system may require an overhaul after 25,000 hours of operation which corresponds to 8.5 years of operation.

In the event the heat pump fails, the auxiliary heater will turn on to maintain continuity of hot water supply. A red light located on the water heater will turn on to indicate heat pump failure. The heat pump will be as safe as a typical household refrigerator, air conditioner, or space heating heat pump.

3.5 Consumer Acceptance

Consumer acceptance of a new product will certainly depend partially upon existing familiarity with the product and confidence of the successful operation of the product. Over the past few years, the public's confidence in heat pumps has risen dramatically. The unfavorable publicity and product rejection which the manufacturers of heat pumps earned during the 1950's have been tempered by the effects of time, the development of new reliable products and the economic necessity of reducing electric power usage [14]. Heat pumps for space heating are now being used throughout the United States and in southern Canada [15, 16]. The marketing program to promote the new high efficiency electric water should capitalize on the public's confidence in heat pumps and should not emphasize the fact that the reverse Brayton air cycle heat pump is a new heat pump concept for water heating applications.

It is unlikely that there may be some resistance from the utilities in approving the use of an electric heat pump for water heating. The potential advantages of better electric power load management, reduced energy consumption, and the general acceptance by the utilities of heat pumps used for space heating, are factors likely to favor acceptance by the utilities of the reverse Brayton air cycle heat pump water heater.

3.6 Codes

The success of both the replacement market as well as the new installation market is dependent on meeting the requirements of building codes. At present, the three major bodies are the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) Building Codes [17], the Building Officials and Code Administrators (BOCA) Basic Plumbing Code [18], and the Southern States Building Code [19]. There does not appear to be any conflict between the new high efficiency water heater and the existing building and electrical codes.

However, at present, codes regulating the installation and operation of heat pump water heaters do not exist. On completion of Phase II of the program, after many high efficiency water heaters have been installed and operated over a period of time at various geographical locations in the United States, it may be useful to develop a model code during Phase II which the various code agencies could incorporate into their existing building codes. At this time, many of the potential installation and operation difficulties such as piping for the cold exhaust air and ambient air, compressor/expander noise, thermal insulation, safety devices, air purity level etc., will have been successfully dealt with.

4. RESIDENTIAL MARKET

4.1 Future Housing Growth

For the years 1980 through 1985, a gradual increase in home building throughout the United States is expected [20]. Total housing starts are projected to exceed 2.5 million per year during the first half of the 1980's. A summary of the United States regional housing starts is shown in Table 4-1. The total housing starts will consist of 1.3 million single-family home starts per year, 0.7 million multi-family home starts per year and 0.5 million mobile home starts. The South Atlantic and Western states should lead the remainder of the country in the growth of housing starts.

4.2 History of the Water Heater Market

A summary of the number of residential gas-fired, electric, and oil-fired water heaters sold in the United States from 1967 through 1976 is shown in Table 4-2. The figures shown are derived from the regular statistical highlights printed by the Gas Appliance Manufacturers Association (GAMA) [21] which obtains its date from voluntary reporting by manufacturers of gas-fired, electric, and oil-fired water heaters. It is apparent from the figures in Table 4-2 that sales of electric water heaters are rapidly approaching the sales of gas-fired water heaters. Over the ten year period, shipments of gas-fired water heaters increased by 21.8 percent and shipments of electric water heaters increased by 107.7 percent. Also for the same ten year period, the shipments of oil-fired water heaters increased by 137.0 percent. It must be noted that the significant increases in sales of water heaters occurred between two intervals, 1967 to 1969 and 1970 to 1972. It should be noted that oil-fired water heaters are insignificant energy consumers, nationally, and that the majority of oil-fired water heaters manufactured are sold in the Northeastern states.

Table 4-1. Summary of the Housing Starts Per Year in the United States

Region	Housing Starts Type	Time Period					
		1960-1964	1965-1969	1970-1974	1975-1979	1980-1984	1984-1989
Northeast	Conventional	262,000	234,000	258,000	250,000	286,000	310,000
	Mobile	16,000	34,000	46,000	36,000	49,000	56,000
	Sub Total	278,000	268,000	304,000	286,000	335,000	366,000
South	Conventional	532,000	568,000	810,000	731,000	892,000	923,000
	Mobile	49,000	128,000	260,000	193,000	268,000	281,000
	Sub Total	581,000	696,000	1,070,000	924,000	1,160,000	1,204,000
North/Central	Conventional	314,000	349,000	390,000	398,000	371,000	366,000
	Mobile	34,000	74,000	109,000	81,000	96,000	98,000
	Sub Total	348,000	423,000	499,000	479,000	467,000	464,000
West	Conventional	363,000	264,000	411,000	442,000	498,000	482,000
	Mobile	30,000	55,000	102,000	85,000	106,000	113,000
	Sub Total	393,000	319,000	513,000	527,000	604,000	595,000
Total	Conventional	1,471,000	1,414,000	1,869,000	1,821,000	2,047,000	2,081,000
	Mobile	129,000	291,000	517,000	395,000	519,000	548,000

4.3 Potential Market

Based on initial projections of market penetration, it is expected that the high efficiency water heater will capture 25 percent of the electric water heater market in newly constructed houses and 10 percent of the electric water heater replacement market within the first 3 to 4 years of the market penetration effort.

The potential market penetration into the sales of replacement residential gas-fired water heaters does not appear to be feasible. Even though the high efficiency electric water heater will be more efficient (based on end use energy) than the conventional gas-fired water heater, the high efficiency electric water heater will not save enough money for the consumer to justify the replacement of a gas-fired water heater with an electric water heater. However, if the gas utilities begin to limit new gas connections for residences or the United States Congress deregulates the price of natural gas, it is possible that the new high efficiency electric water heater will be able to replace the conventional gas-fired water heaters.

It is projected that the potential market for the high efficiency electric water heater is approximately 480,000 units per year. This estimate was obtained by assuming the following statistical factors:

- For the next 2 years it is expected that approximately 6,000,000 gas-fired and electric water heaters will be shipped in the United States.

- Within the next 2 years, electric water heaters should account for approximately 50 percent of all water heaters shipped in the United States.

- According to the Gas Appliance Manufacturers Association, approximately 60 percent of the water heaters shipped each year are for the replacement market.

Of course the final number of units shipped per year will depend on factors such as selling price, government incentives for the consumer, electrical energy costs and payback period.

There are many avenues which both government and private industry can pursue in order to expedite the successful introduction and acceptance of a high efficiency water heater into the market place. The federal and state governments could promote "market pull" by:

- Offering tax incentives to the consumer for purchasing a high efficiency water heater,

- Providing government subsidies, similar to that already initiated by HUD for solar water heaters,

- Initiating legislation promoting the sales of high efficiency water heaters by requiring that all water heaters meet a certain minimum coefficient of performance that would exclude conventional water heaters,

- Creating a strict building code which would dissuade the consumer purchase of a conventional water heater, and

- Offering low interest loans to the purchasers of the high efficiency water heaters.

Possible government incentives for the manufacturer are low interest loans for the required initial investment capital, tax credit, accelerated depreciation credit on new facilities and equipment, and legislation favorable for the successful market penetration and continued high volume sales of the new high efficiency water heaters.

Since it is expected that for an average hot water consumption rate of 80 gallons per day and an average water temperature rise of 80^{O}F, each high efficiency water heater can save 2,336 kilowatt hours of electricity per year, the overall potential energy savings for 480,000 units is 1,121,280,000 kilowatt hours per year. Assuming an average electrical energy cost rate of $0.045 per kilowatt hour, the potential cost savings could be as high as $50,458,000 per year. Table 4-3 presents a projected number of the new high efficiency electric water heaters to be sold in the United States between 1980 and 2000. Without better marketing forecasting information, it is assumed that the potential market for the high efficiency electric water heater will remain at a constant value of 480,000 units per year. The cumulative energy savings and cumulative energy cost savings by 2000 would be 40,336 million kilowatt hours and 1,817 million dollars, respectively.

Water heaters consume approximately 4 percent of the U.S. total energy budget and represent a large market. From Table 4-2, the potential market for electrical water heaters may be as high as six million water heaters a year. If wholesale capture of the residential market were successful, more than 630 million dollars per year, representing 1.4×10^{10} kilowatt hours per year in energy savings, would occur. To capitalize on this large potential energy savings with the installation of high efficiency water heaters nationally, an agressive, widespread marketing approach to capture the largest possible share of the market in the shortest period of time must be instituted.

4.4 Potential Payback

Based on cost estimates for the current design, the new high efficiency water heater can be sold to the consumer and installed

Table 4-2. Number of Residential Water Heaters Shipped Between 1967 and 1976

Year	GAS-FIRED		ELECTRIC		OIL-FIRED		TOTAL	
	Number	Percent of Market	Number	Percent of Market	Number	Percent of Market	Number	Percent Growth
1967	2,554,510	66.7	1,260,500	32.9	13,500	0.4	3,828,510	-
1968	2,756,300	64.9	1,479,600	34.8	10,400	0.3	4,26,300	10.9
1969	2,742,430	63.7	1,549,000	36.0	13,800	0.3	4,305,230	13.9
1970	2,785,210	62.1	1,683,600	37.6	13,600	0.3	4,482,410	4.1
1971	3,088,480	61.5	1,922,000	38.2	14,700	0.3	5,025,180	12.1
1972	3,162,820	58.1	2,264,800	41.6	15,600	0.3	5,443,220	8.3
1973	3,080,440	54.0	2,591,100	45.4	30,400	0.6	5,701,940	4.8
1974	2,568,650	50.6	2,486,600	48.9	26,700	0.5	5,081,950	-10.9
1975	2,645,110	54.4	2,183,400	44.9	32,000	0.7	4,860,510	-4.6
1976	3,112,130		2,618,200		N/A			

Table 4-3. Projected Number of New High Efficiency Water Heaters to be Sold in the United States Between 1980 and 2000

Year	Cumulative Number of New High Efficiency Water Heaters to be Sold in U.S. (thousands)			Cumulative Energy Savings ($x10^5$ Kw-hr)	Cumulative Energy Cost Savings ($x10^6$ dollars)
	New Housing Market	Replacement Market	Total		
1980	0	0	0	0	0
1985	750	450	1,200	5,046	227
1990	2,250	1,350	3,600	16,319	757
1995	3,750	2,250	6,000	28,592	1,287
2000	5,250	3,150	8,400	40,366	1,817

in his home for a price between $500 and $600. Currently, the purchase and installation cost of a conventional 80 gallon electric water heater is approximately $300. Consequently, the high efficiency water heater would cost approximately $300 more than a conventional water heater. With regards to the replacement market, the $300 difference would not be true. Instead, an added installed cost between $50 and $100 should be added to the difference. As a result, the difference would be between $350 and $400. In either case, the expected savings in electricty should save money for the user when compared with the conventional electric water heater.

The potential payback period is defined as the installed cost differential of a high efficiency water heater versus a conventional water heater divided by the operating and maintenance cost differential.

$$\text{PAYBACK PERIOD} = \frac{\text{INSTALLED COST DIFFERENTIAL}}{\text{OPERATING AND MAINTENANCE COST DIFFERENTIAL}} \quad (1)$$

The operating cost differential is a function of the product of the energy consumption and the electrical energy cost rate. The energy consumption is a function of the hot water usage rate, the average temperature rise of the water, and the efficiency of the water heater. Figures 4-1 through 4-4 represent the payback period for the high efficiency water heater relative to a conventional electric water heater as a function of the hot water usage rate, average temperature rise of the water, and electrical energy costs. It is assumed that the average seasonal coefficient of performance (SCOP) is 1.7, the installed cost differential is $300 and the maintenance cost differential is negligible (approximately less than one percent of the installed cost differential). The reason for such a small maintenance cost is that it is projected

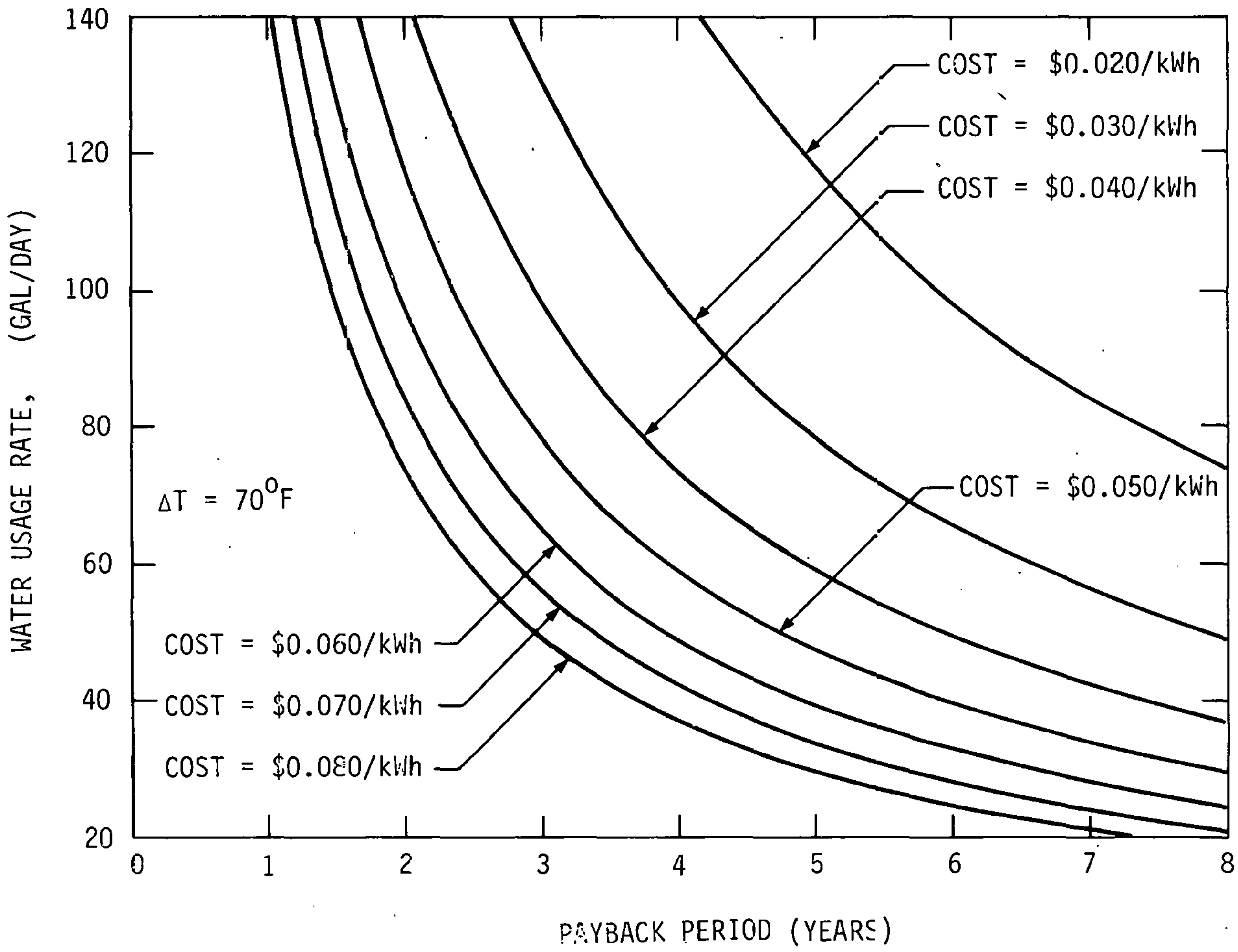

Figure 4-1. The Payback Characteristics of the High Efficiency Water Heater Operating at a Water Temperature Rise of 70°F

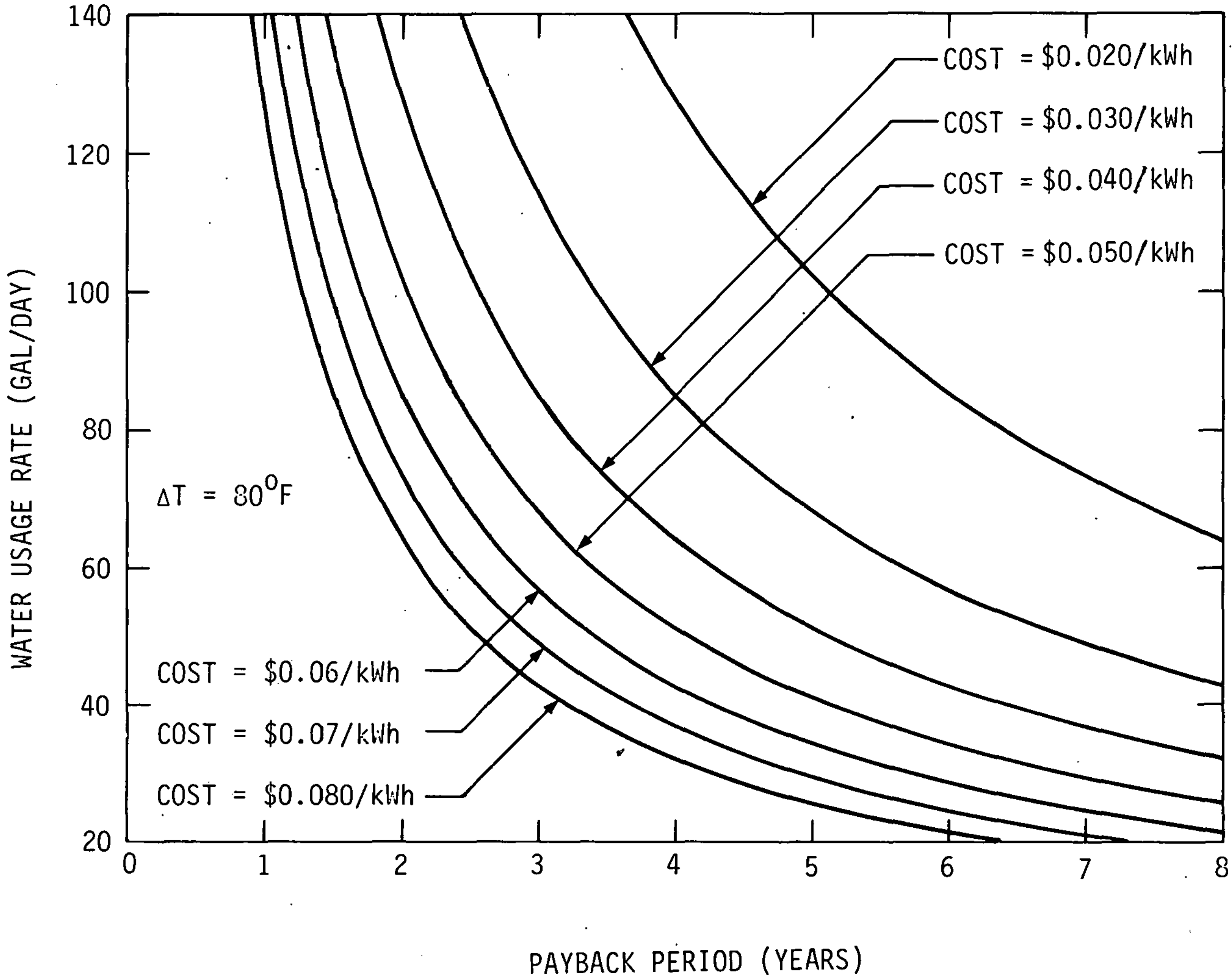

Figure 4-2. The Payback Characteristics of the High Efficiency Water Heater Operating at a Water Temperature Rise of 80°F

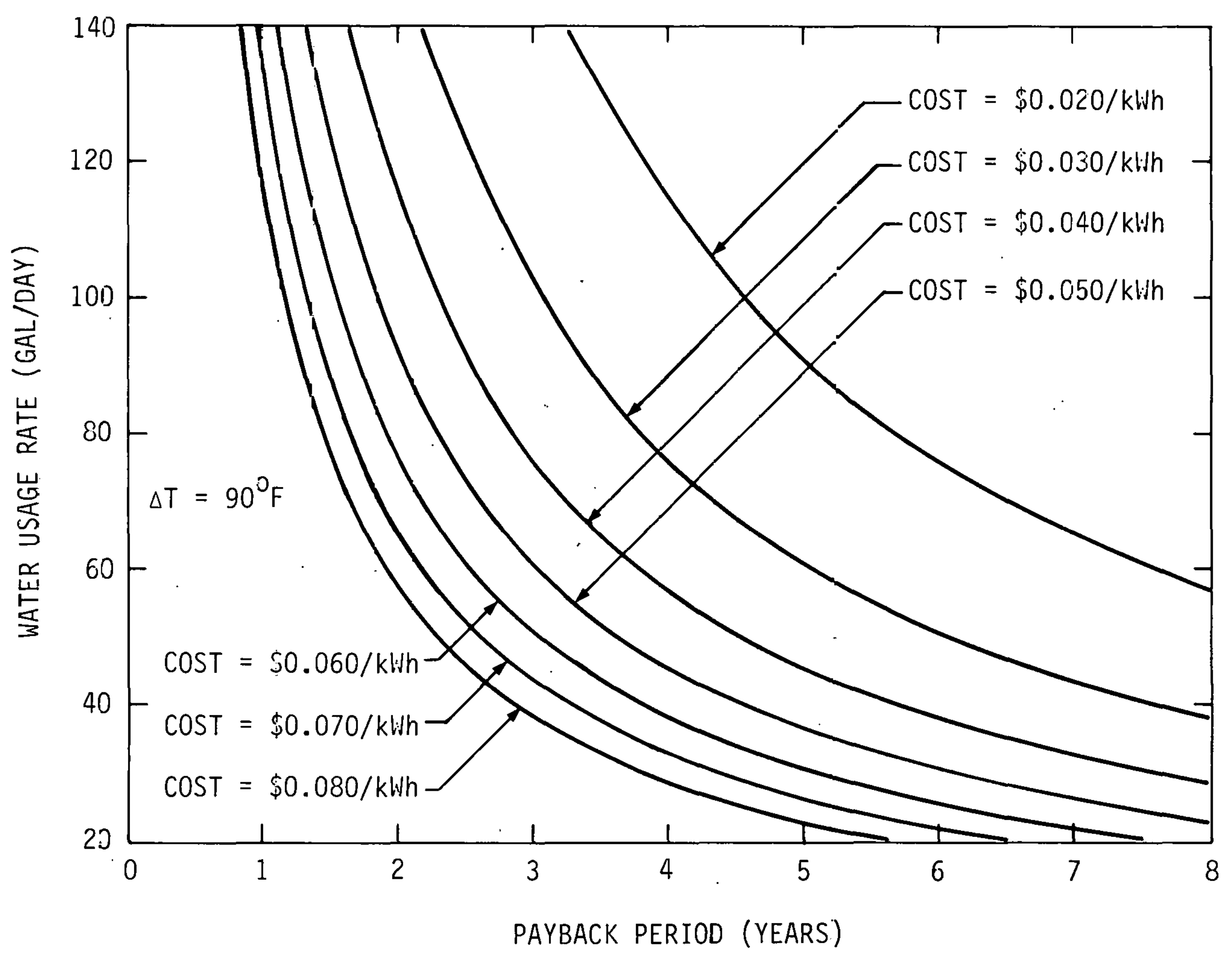

Figure 4-3. The Payback Characteristics of the High Efficiency Water Heater Operating at a Water Temperature Rise of 90°F

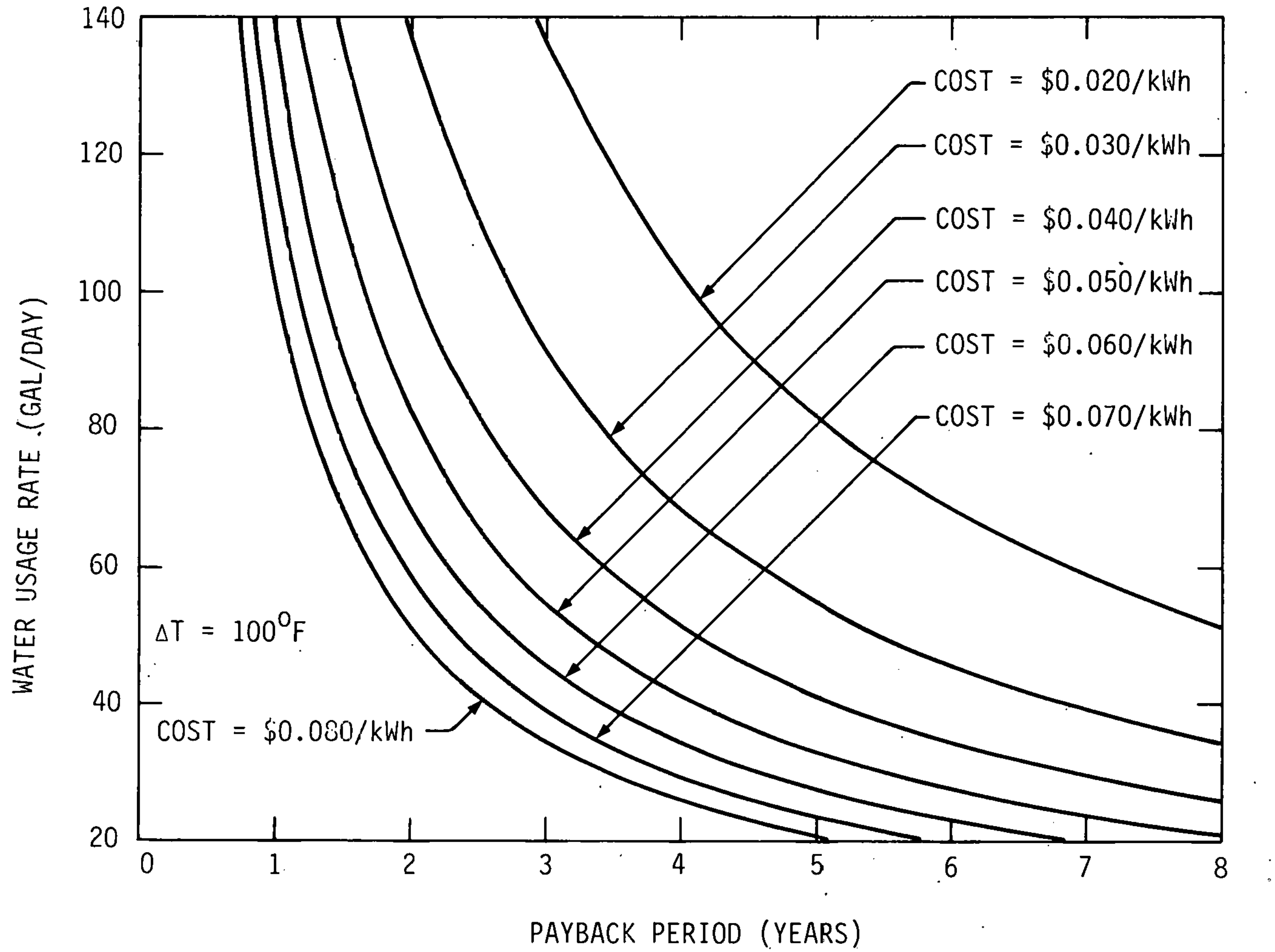

Figure 4-4. The Payback Characteristics of the High Efficiency Water Heater Operating at a Water Temperature Rise of 100°F

that the compressor/expander will be as reliable as a household refrigerator or air conditioner. Both of these appliances require minimal maintenance and exhibit excellent reliability. A better estimate of the installed cost differential and the maintenance cost differential will be obtained during the field testing of the prototype high efficiency water heaters during Phase II.

The average hot water usage rate for a single family household has been reported at values between 60 and 80 gallons per day. A few values are the following:

Average Hot Water Usage Rate (gal/day)	Source
64.5	NBS [22]
71.4	ADL [6]
75.0	ORNL [13]
81.0	Rand Corp. [7]

For an installation having an average hot water usage rate of 80 gallons per day, an average water temperature rise of 80°F, and an electrical energy cost rate of $0.03 per kilowatt hour and greater, the payback period is expected to be 4 years or less. A summary of the payback characteristics of the high efficiency water heater for a water usage rate of 80 gallons per day is presented in Table 4-4.

Because of the recurring energy cost savings that occur through use of a high efficiency water heater, the consumer can recoup his initial added investment for a high efficiency water heater versus a conventional water heater during the payback period. After the payback period, the consumer will obtain the recurring energy cost savings for the life of the high efficiency water heater.

Table 4-4. Summary of the Payback Characteristics of the High Efficiency Water Heater for a Water Usage Rate of 80 Gallons per Day

Average Water Temperature Rise (°F)	Electrical Energy Cost Rate ($kW-hr)	Payback Period (Year)
70	0.020	7.34
	0.030	4.89
	0.040	3.67
	0.050	2.94
80	0.020	6.42
	0.030	4.28
	0.040	3.21
	0.050	2.57
90	0.020	5.71
	0.030	3.81
	0.040	2.85
	0.050	2.28
100	0.020	5.14
	0.030	3.43
	0.040	2.57
	0.050	2.05

It is important that a payback period by short in order to promote the selling of the high efficiency water heater. At present, the water heater manufacturers sell water heaters with five year and ten year warranties. Because of the difference in cost and the average time a user remains in one location, the majority of water heaters sold are the ones with a 5 year warranty [23]. For the present case of the marketing of the high efficiency water heater, it is felt that long life expentancy as well as high performance should be promoted. Consequently, the consumer must be convinced of the advantages of purchasing a ten year warranty water heater rather than a five year warranty water heater. The obvious method for accomplishing this task is to offer a water heater with a short payback period.

5. TEST MARKET FOR RESIDENTIAL WATER HEATER

To encourage the widest possible acceptance for the new high efficiency water heater, test models should be distributed throughout all regions of the United States. However, this strategy may require more water heaters than are available for the Phase II of the development program. Consequently, a technique for determining possible test market locations for the initial residential testing of the reverse Brayton air cycle heat pump water heaters is developed.

5.1 Test Market Rating Procedures

The United States is divided into four major regions; Northeast, South, North Central and West, and nine subregions; New England, Mid Atlantic, South Atlantic, West South Central, East North Central, West North Central, Mountain, and Pacific, as defined by the U.S. Census Bureau. These regions with their respective states are defined in Table 5-1 and are illustrated in Figure 5-1.

Twenty-eight cities were initially selected to be representative cities for the various major regions and subregions. These cities and their respective locations are shown in Table 5-2. The initial selection method was based on the following factors:

- population
- geographical location
- utility rates
- average seasonal ambient temperatures
- average seasonal water temperatures.

Table 5-1. Major Subregions of the United States

Regions	States
Northeast	
New England	Maine, New Hampshire, Vermont Massachusetts, Connecticut, Rhode Island
Mid Atlantic	New Jersey, New York, Pennsylvania
South	
South Atlantic	Delaware, Maryland, Washington, D.C., Virginia, West Virginia, North Carolina, South Carolina, Georgia
East South Central	Kentucky, Tennessee, Alabama, Mississippi, Florida
West South Central	Arkansas, Louisiana, Oklahoma, Texas
North Central	
East North Central	Ohio, Indiana, Michigan, Illinois, Wisconsin
West North Central	Minnesota, North Dakota, South Dakota, Iowa, Kansas, Nebraska, Missouri
West	
Mountain	Montana, Wyoming, Idaho, Colorado, Utah, Nevada, Arizona, New Mexico
Pacific	California, Washington, Oregon, Alaska, Hawaii

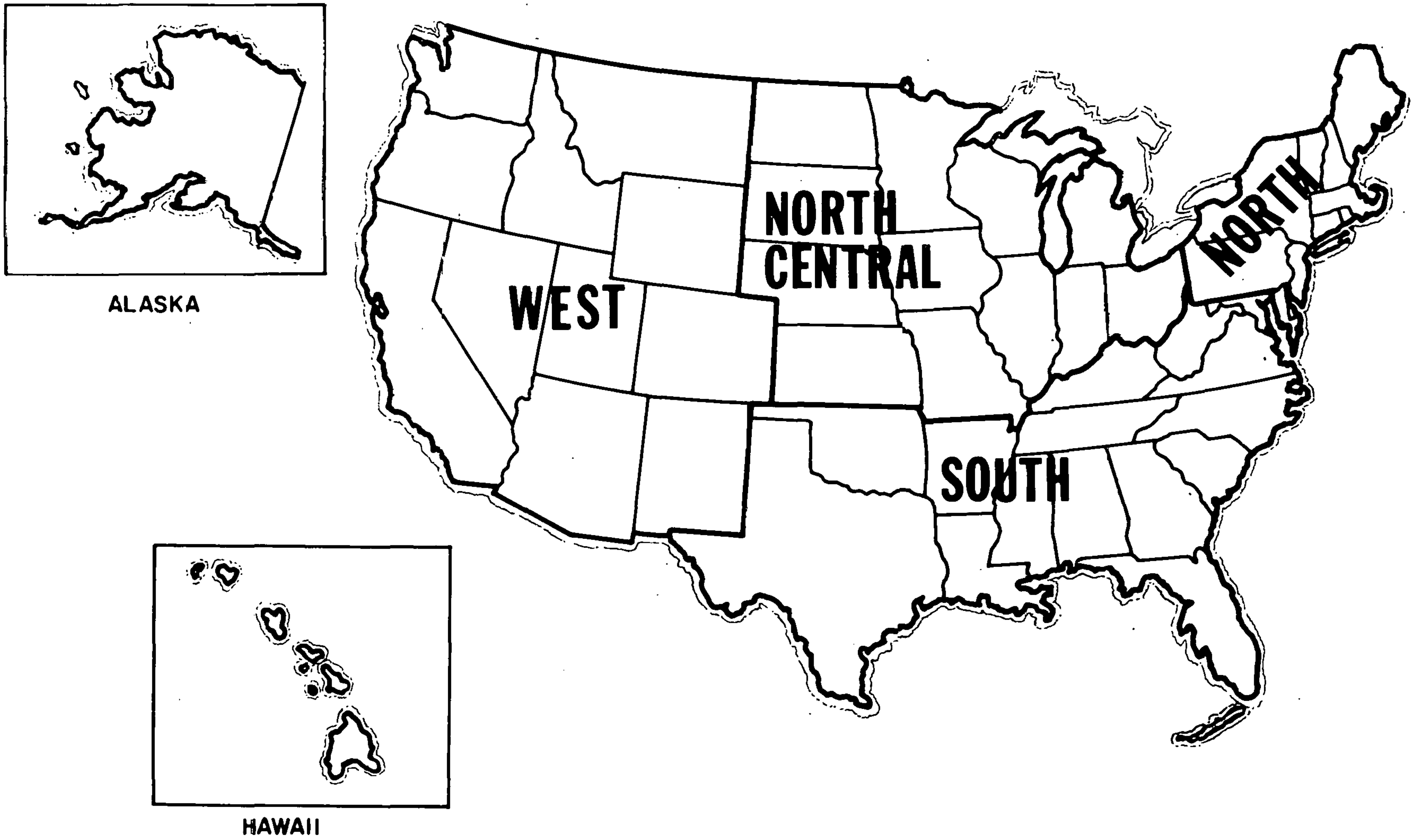

Figure 5-1. Major Regions of the United States

Table 5-2. Major Cities Considered for the Test Market

Region	Cities Selected
Northeast	
New England	Portland, ME, Boston, MA, Hartford, CT
Mid Atlantic	Buffalo, NY, New York City, NY, Philadelphia, PA, Pittsburgh, PA
South	
South Atlantic	Washington, D.C., Norfolk, VA, Charlston, SC, Atlanta, GA
East South Central	Miama, FL, Tampa, FL, Jacksonville, FL, Birmingham, AL, Nashville, TN
West South Central	Little Rock, AR, New Orleans, LA, Dallas, TX, Dallas, TX, Houston, TX, San Antonio, TX
North Central	
East North Central	Chicago, IL, Cincinnati, OH, Cleveland, OH
West North Central	St. Louis, MO, Minneapolis, MN, Kansas City, MO
West	
Mountain	Cheyenne, WY, Alburquerque, NM, Denver, CO, Phoenix, AZ, Salt Lake City, UT, Las Vegas, NV
Pacific	San Francisco, CA, Los Angeles, CA, Seattle, WA, Anchorage, AK, Honolulu, HI

A semi-quantitative cum qualitative evaluation procedure was developed to evaluate these 28 cities and select four cities for test marketing. The quantative data used in the evaluation process includes the fuel cost, the average seasonal ambient temperature, the average seasonal water temperature, and the rate of water usage. The average seasonal coefficient of performance (SCOP) is predicted from the average seasonal ambient temperature [24]. Table 5-3 summarizes this data for the 28 cities selected and includes a quantative rating of the cities presented as a function of the potential energy costs savings occurring over a period of 10 years. The energy cost savings consists of the differential in operating costs for the high efficiency water heater and a conventional electric water heater.

Energy Cost Savings (ECS) =

$$mc_p \Delta T_{avg} \left[1.0 - \left(1.0/SCOP \right) \right] \times \text{ELECTRICAL ENERGY COST RATE} \times \text{TIME} \tag{2}$$

where

(1) The term $mc_p \Delta T_{avg}$ is the average heat input required for heating 80 gallons of water per day from the average seasonal ambient water temperature to $140^{o}F$. It is assumed that the difference in standby losses for the high efficiency water heater and the conventional electric water heater are negligible.

(2) The electrical energy cost rate is assumed to be constant over the 10 year period because of the difficulty in predicting the inflation rates for each utility company.

(3) The time (TIME) is taken to be equal to 10 years.

Table 5-3. Summary of Quantitative Data Used for Test Market City Evaluation

Region	Cities	Average Seasonal Ambient Temperature (°F) [25]	Temperature Extremes Winter (°F)	Temperature Extremes Summer (°F) [26]	Average Seasonal Water Temperature (°F) [27,28]	Average Seasonal SCOP [24]	Electrical Energy Cost ($/kW-hr) [23,29,30]	10 Year Energy Cost Savings ($)	Pay Back Period (Years)
NORTHEAST									
• New England	Portland	52.9	-5	88	45	1.69	0.03627	997.70	3.01
	Boston	51.4	6	91	45 (Est)	1.69	0.04520	1243.34	2.41
	Hartford	49.8	1	90	45	1.69	0.03788	1041.99	2.65
• Mid Atlantic	Buffalo	46.7	3	88	45	1.69	0.02390	657.43	4.55
	New York City	54.5	12	93	48	1.69	0.07741	2062.12	1.45
	Philadelphia	53.5	11	93	50	1.69	0.5514	1436.94	2.09
	Pittsburgh	50.3	7	90	50	1.69	0.03200	833.91	3.61
SOUTH									
• South Atlantic	Washington D.C.	57.0	16	94	60	1.70	0.02526	590.12	5.08
	Norfolk	59.7	14	96	50	1.70	0.03156	829.46	3.62
	Charleston	59.0	9	92	65	1.70	0.02562	561.12	5.35
	Atlantic	61.4	18	95	60	1.70	0.03300	770.94	3.89
• East South Central	Miami	75.1	44	92	70	1.71	0.03485	718.34	4.18
	Tampa		36	92	70	1.70	0.04154	849.14	3.53
	Jacksonville	69.5	29	96	60 (Est)	1.70	0.04338	1013.43	2.95
	Birmingham	61.0	19	97	60 (Est)	1.70	0.04120	962.51	3.12
	Nashville	60.0	12	97	62	1.70	0.02460	560.33	5.35
• West South Central	Little Rock	61.7	19	99	60	1.70	0.03450	805.98	3.72
	New Orleans	68.6	32	93	70	1.71	0.04500	927.56	3.23
	Dallas	65.8	19	101	67	1.70	0.03965	845.24	3.55
	Houston	69.2	29	96	65	1.71	0.02999	662.32	4.53
	San Antonio		25	99	65	1.71	0.04321	954.28	3.14
NORTH CENTRAL									
• East North Central	Chicago	50.8	-3	94	48	1.69	0.04257	1134.02	2.65
	Cincinnati	55.2	8	94	55	1.69	0.02540 (1972)	625.15	4.80
	Cleveland	49.7	2	91	45	1.69	0.04676	1286.26	2.33
• West North Central	St. Louis	55.3	7	96	55	1.69	0.03701	910.89	3.39
	Minneapolis	43.7	-14	92	45	1.68	0.03530	962.64	3.12
	Kansas City	56.8	4	100	55	1.70	0.30200	749.62	4.00
WEST									
• Mountain	Cheyenne	45.9	-6	89	45 (Est)	1.68	0.02020	550.86	5.45
	Albuquerque	56.6	14	96	72	1.69	0.02000 (1975)	393.79	7.62
	Denver	49.5	-2	92	51	1.69	0.03117	803.26	3.73
	Phoenix	69.0	31	108	59.4	1.71	0.03545 (Avg)	841.36	3.57
	Salt Lake City	50.9	5	97	43	1.69	0.03180	893.16	3.36
	Las Vegas		23	108	73	1.71	0.01835	362.03	8.29
• Pacific	San Francisco	56.8	42	80	60	1.69	0.03165	733.15	4.09
	Los Angeles	64.4	42	94	64	1.71	0.04065	909.71	3.30
	Seattle	51.1	28	81	51	1.69	0.01280	329.86	9.09
	Anchorage	37.0	-25	73	37 (Est)	1.68	0.02150	635.69	4.72
	Honolulu	75.9	60	87	75	1.71	0.04220	807.71	3.71

The differential in the initial capital and installation costs for the high efficiency water heater and a conventional electric water heater has been considered a constant value of $300 for all 28 cities. Also shown in Table 5-3 is the payback time period in years. The recurring energy cost savings per year is simply the energy cost savings divided by 10 years.

Qualitative factors considered in the test market rating evaluation include in order of importance:

- potential housing market
- population
- fossil fuel costs comparisons
- seasonal temperature range
- water quality
- distribution and installation availability.

Population is an important criteria to consider as it influences the potential housing market, the number of new sites for the use of the high efficiency water heater, and the replacement water heater market which is expected to be as high as 60 percent of the total residential water heater market.

Comparison of various energy fuel type costs are important since a gas-fueled consumer would be unlikely to purchase an electric high efficiency water heater if the price of natural gas is significantly less than the price of electricity.

The new high efficiency water heater should be tested over a broad range of temperatures so that its performance can be predicted for operation at different geographical locations in the United States. For example, it would be undesirable to test the water heater in Honolulu where the maximum temperature range is only 27^{o}F, whereas in Minneapolis the maximum temperature range is 106^{o}F.

Water quality is an important factor since in certain regions of the country large deposits of lime can build up in the water storage tank which can affect the recovery rate, storage capacity, system life, and thermodynamic performance of the high efficiency water heater.

It is important that qualified water heater distributors and installers be available for the successful marketing and selling of this new high efficiency water heater.

A rating method for the qualitative criteria is based on a scale of one to five where five represents excellent and one represents poor. Also, each criterion is given a weighting function so that the various criteria are not judged equally. The total weighting factor is defined as follows:

$$W_f = \sum_{i=1}^{6} a_i x_i \qquad (3)$$

where x_i is the individual qualitative criteria and a_i is the respective weighting factor.

Table 5-4 summarizes the qualitative criteria and their respective basis for rating. The basis for the fuel cost comparison is determined by assuming that the differential of costs for electricity, natural gas and oil is reflected in the price of electricity. One reason for this assumption is that throughout the United States the price of natural gas is cheaper than the price of electricity. The temperature range for the cities is determined by subtracting the minimum winter temperature from the maximum summer temperature. The water quality is based upon the untreated public water inpurity level in parts per million (ppm). The reason for only three rating levels for the water quality is that it was consistently found that the water quality was either excellent, good or poor.

Table 5-4. Summary of Rating Basis for the Qualitative Criteria

Annual Housing Starts in the early 1980's	x_1
More than 300,000	5
230,000 - 300,000	4
180,000 - 230,000	3
110,000 - 180,000	2
Less than 100,000	1
Population	x_2
More than 1,000,000	5
750,000 - 1,000,000	4
500,000 - 750,000	3
250,000 - 500,000	2
Less than 250,000	1
Electrical Energy Cost (\$/kW-hr)	x_3
Less than 0.020	5
0.020 - 0.030	4
0.030 - 0.040	3
0.040 - 0.050	2
More than 0.050	1
Ambient Temperature Range (°F)	x_4
More than 90	5
80 - 90	4
70 - 80	3
60 - 70	2
Less than 60	1
Water Hardness Impurity Level (ppm)	x_5
0 - 60	5
60 - 120	4
More than 120	1
Distributor Availability	x_6
Assumed Constant	5

Table 5-5 presents the values of x_i for the 28 cities. Note that x_6 was given a value of 5 for all cities; it is assumed that the distributor and installed availability will exist in all of the 28 cities during the early 1980's.

It is decided that the overall weighting factor, W_f, should only have a maximum influence of 20 percent on the energy cost savings. Consequently, values are attached to weighting factors so that for the case of all x_i equal to five, the energy cost savings is 1.2 times its original quantative value and for the case of all x_i equal to one, the energy cost savings is 1.04 times its original quantative value. The weighted energy cost savings is defined as follows:

$$\text{WECS} = \text{ECS}\ (1 + W_f) \tag{4}$$

where W_f varies between 0.04 and 0.2 The values chosen for the weighting factors are the following:

Criteria	Weighting Factor	Relative Importance Percent	Weighting Factor Value
Annual Housing Starts	a_1	30	0.012
Population	a_2	25	0.01
Electrical Energy Costs	a_3	20	0.008
Ambient Temperature Range	a_4	20	0.008
Water Hardness Impurity Level	a_5	3	0.0012
Distributor Availability	a_6	2	0.0008

The values of W_f and WECS for the 28 cities are shown in Table 15. Also shown in Table 15 are values of WECS normalized with respect to the maximum value of WECS which occurs for New York City. Because the final value of WECS is subjective, a 10 percent variation of WECS is not significant for the present analysis.

Table 5-5. Summary of Qualitative Data Used for Test Market City Evaluation

Regions	Cities	X_1 Potential Housing Market [20]	X_2 Population [31]	X_3 Fuel Type Cost Comparison	X_4 Ambient Temperature Range	X_5 Water Quality [32]	X_6 Distributor & Installer Availability	W_f	WECS ($)	$WECS/WECS_{max}$
NORTHEAST										
• New England	Portland	1	1	3	5	5	5	0.096	1093.48	0.462
	Boston	1	3	2	4	5	5	0.100	1367.67	0.578
	Hartford	1	1	3	4	5	5	0.088	1133.69	0.479
• Mid Atlantic	Buffalo	4	2	4	4	5	5	0.142	750.79	0.317
	New York City	4	5	1	4	5	5	0.148	2367.31	1.000
	Philadelphia	4	5	1	4	4	5	0.1458	1647.88	0.696
	Pittsburgh	4	3	3	4	4	5	0.1428	952.99	0.403
SOUTH										
• South Atlantic	Washington D.C.	5	4	4	3	5	5	0.166	688.08	0.291
	Norfolk	5	2	3	4	5	5	0.146	950.56	0.402
	Charleston	5	1	4	4	5	5	0.144	641.92	0.271
	Atlantic	5	3	3		5				
• East South Central	Miami	4	2	3	1	1	5	0.1052	793.91	0.335
	Tampa	4	2	2	1	1	5	0.0972	931.68	0.394
	Jacksonville	4	3	2	2	1	5	0.1152	1130.18	0.477
	Birmingham	3	2	2	3	5	5	0.106	1064.54	0.450
	Nashville	3	2	4	4	4	5	0.132	534.29	0.268
• West South Central	Little Rock	2	1	3	4	5	5	0.108	893.03	0.377
	New Orleans	2	3	2	2	4	5	0.0948	1015.49	0.429
	Dallas	4	4	3	4	1	5	0.1492	971.35	0.410
	Houston	4	5	3	2	1	5	0.1432	757.16	0.320
	San Antonio	4	3	2	3	1	5	0.1232	1071.85	0.453
NORTH CENTRAL										
• East North Central	Chicago	5	5	2	5	1	5	0.1712	1328.16	0.561
	Cincinnati	5	2	4	4	1	5	0.1492	718.42	0.303
	Cleveland	5	4	2	4	1	5	0.1532	1463.32	0.618
• West North Central	St. Louis	2	3	3	4	1	5	0.1152	1015.82	0.429
	Minneapolis	2	2	2	5	1	5	0.1052	1063.91	0.449
	Kansas City	2	3	3	5	1	5	0.1232	841.97	0.356
WEST										
• Mountain	Cheyenne	3	1	4	5	1	5	0.1232	618.73	0.261
	Albequerque	3	2	4	4	1	5	0.1252	443.09	0.187
	Denver	3	3	3	5	4	5	0.1388	914.75	0.386
	Phoenix	3	3	3	3	1	5	0.1192	941.65	0.398
	Salt Lake City	3	1	3	4	1	5	0.1152	996.05	0.421
	Las Vegas	3	1	5	4	1	5	0.2412	449.35	0.190
• Pacific	San Francisco	5	3	3	1	1	5	0.1272	826.41	0.349
	Los Angeles	5	5	2	1	1	5	0.1392	1036.34	0.438
	Anchorage	2	1	4	5	4	5	0.112	706.89	0.299
	Honolulu	2	2	2	1	5	5	0.078	870.71	0.368

5.2 Selection of Test Market Cities

It is important that the cities which are selected for the test market offer an energy cost savings which allow a payback period of 3.5 years or less. The high efficiency water heater will become a viable product only if the potential payback period and recurring energy cost savings are attractive to the consumer. Only 15 cities of the total 28 cities satisfy the target payback period and were considered as candidates. Many of these 15 cities in the four major regions can be eliminated for a number of reasons.

For the Northeast region of the country, the candidate cities are New York City, Philadelphia, Boston, Hartford and Portland. Portland and Hartford display a poor rating for both potential housing market and replacement market. Consequently, these cities are eliminated. Boston offers a large energy cost savings; however, it has a poor rating for potential new housing. Therefore, it is eliminated. New York City offers the largest energy cost savings but the majority of existing hot water heaters are gas-fired type because of the large difference in the price of natural gas and electricity and the availability of natural gas. Consequently, an electric high efficiency water heater would not be able to compete commercially with a gas-fired water heater. Also, New York City is a better market place for commercial water heaters than residential water heaters. As a result, New York City is eliminated. For the Northeast region of the country, Philadelphia is a suitable test market candidate city. In terms of energy cost savings potential, it is the second highest in the nation. Also, Philadelphia exhibits excellent population, housing growth potential, and temperature variation range characteristics.

For the South region of the country, the candidate cities are Jacksonville, San Antonio, Birmingham and New Orleans. New Orleans and Birmingham have average or poor potential housing growth, population, and temperature range variation characteristics. Consequently, these cities are rejected. Jacksonville and San Antonio are very similar for all qualative criteria; however, because of its higher value of energy cost savings potential, Jacksonville is selected as the test market candidate city for this region.

For the North Central region of the country, the candidate cities are Cleveland, Chicago and Minneapolis. Chicago is the only city in the country that exhibits excellent population, housing growth potential, and temperature variation range characteristics. Consequently, Chicago is selected as the test market city candidate of the North Central region even though Cleveland displays a slightly higher energy cost savings potential (nine percent difference).

For the West region of the country, the only candidates are Los Angeles, Salt Lake City and Phoenix. Of these three cities, Los Angeles is selected because of its higher energy cost savings potential and its excellent ratings for potential housing market and population. Unfortunately, it has a poor rating for temperature range variation characteristics.

The results of the analysis are summarized as follows:

Regions	Cities	ECS ($)	W_f	WECS ($)	$\frac{WECS}{WECS_{max}}$
Northeast	Philadelphia	1436.94	0.1468	1647.88	0.696
South	Jacksonville	1013,43	0.1152	1130.18	0.477
North Central	Chicago	1134.02	0.1712	1328.16	0.561
West	Los Angeles	909.71	0.1392	1036.34	0.438

Of the four selected cities, Philadelphia offers the greatest potential for energy cost savings, whereas Los Angeles offers the least potential.

For the test market phase of the program, for the development of a reverse Brayton air cycle heat pump water heater, it is recommended that prototype water heaters be placed in Philadelphia, Jacksonville, Chicago and Los Angeles, and one prototype water heater remain at Foster-Miller Associates for long term performance and reliability testing. The total number of the new high efficiency electric water heaters to be tested will be determined during the preparation of the program plan for Phase II of the Research, Development and Demonstration of a High-Efficiency Electric Water Heater. The reason for suggesting the distribution of a large number of prototype water heaters in a small number of cities rather than a small number of prototype water heaters in a large number of cities is that for the first phase of the residential testing of the water heaters, it will be logistically easier if the prototype water heaters are concentrated at a few focal points.

6. CONCLUSIONS AND RECOMMENDATIONS

A potential residential market for the reverse Brayton air cycle heat pump water heater will exist in various geographical locations in the United States. The residential market consists of the new housing market and the replacement market of which 40 percent is new housing market. The regions which seem to be the most attractive with respect to new housing starts are the South Atlantic and West.

The retail and installation cost of the new high efficiency water heater is estimated to be between \$500 and \$600, which is approximately \$300 more than a conventional electric water heater. The successful commercialization of the water heater will depend on factors such as initial required capital investment for manufacturing facilities, appropriate government incentives or "market pull" program, consumer acceptance, and potential energy cost savings.

It is estimated that the potential residential market for the new high efficiency water heater will be approximately 480,000 units per year and the average payback per unit will be less than 3-1/2 years, based on an average seasonal coefficient of performance of 1.7.

Because of the small sales volume and large variation in heater sizes and requirements, the commercial market is not as attractive as the residential market; however, the commercial market does represent approximately 13.8 percent of the total water heater market in terms of sales dollars.

Test market cities were selected for the commercial demonstration and testing of the new high efficiency water heater. he country was divided into four major regions and a city was selected from each region. The cities selected are the following:

Region	Test Market City
Northeast	Philadelphia
South	Jacksonville
North Central	Chicago
West	Los Angeles

These cities offer the best combination of potential for energy cost savings, new housing starts, replacement market, temperature range variation, and water quality characteristics. It is recommended that new high efficiency electric water heaters be placed in Philadelphia, Jacksonville, Chicago, and Los Angeles, and one new high efficiency electric water heater remain at FMA for long term testing and performance evaluation. The total number of the new high efficiency electric water heater to be tested will be determined during the preparation of the program plan for Phase II.

It is recommended that at the conclusion of the field demonstration of the new high efficiency electric water heaters, the market study report be updated and expanded. The revised market study report would include the thermodynamic and reliability performance results of the testing of the sample units as well as marketing data, such as manufacturing cost, installation cost, operating cost, consumer acceptance attitude, city and states government attitudes, etc. The report would include the development of a model code for the installation and operation of a reverse Brayton air cycle heat pump water heater. Finally, the market study report would include a more detailed analysis of the commercial market and a reevaluation of the feasibility of employing a high efficiency water heater in the commercial market place.

APPENDIX A. MANUFACTURING FACILITIES

Existing water heater manufacturers contain facilities for the manufacture of conventional electric water heaters; however, in order to manufacture the new high efficiency water heater, the manufacturers may have to acquire new facilities. The manufacturing constraints affecting the type of new facilities required for the successful introduction of the new high efficiency water heater into the market place are listed in order of importance:

- Complexity of the water heater design
- Material used
- Annual volume
- Degree of automation required
- Desired level of quality
- Targeted manufacturing costs of assembly.

In order to determine whether a capital investment is necessary for new manufacturing facilities, the components must be evaluated on an individual basis. The components to be considered are the water storage tank, air/water heat exchanger, compressor/expander, jacket shell assembly, auxiliary heating element, thermostat, interconnecting piping necessary for the compressor/expander and air/water heat exchanger, and the thermal insulation.

A.1 Water Storage Tank

The water storage tank is a standard unit which is glass-lined and capable of withstanding a 150 psia working pressure and a 300 psia hydrostatic pressure. These requirements are in accordance with both the Underwaters Laboratories [33] and the merican Gas Association Laboratories [34] standards for non-code pressure vessels. The purpose of the glasslining is to prevent and retard rust formation on the interior and exterior surfaces of the water storage tank. Regardless of the different environmental and water conditions present in the various geographical areas in the United States, the water storage tank must be operational for at least 10 years. The above requirements are accomplished by means of a two-coat, two-fire glass coating on the surfaces of the water storage tank.

Specialized equipment for spraying or tank fabrication is not required if the manufacturer is presently producing a glass-lined vessel.

A.2 Air/Water Heat Exchanger

The air/water heat exchanger is designed to meet several criteria.

- Counter flow with heat transfer effectiveness between 80 and 85 percent for an air flow of 33 scfm
- Air pressure loss through the exchanger less than 0.1 psi
- 150 psia air side working pressure
- Compact
- Compatability with arrangement of other system components
- Compatible with low cost automated mass producing techniques
- Maximum operating temperature of 200^{O}F.

The heat exchanger is an externally finned tube and shell arrangement. The diameter of the heat exchanger package is six inches and the length is 12 inches. The heat exchanger is designed so that it will fit the arrangement of the other system components without requiring special means of mechanical support or awkward piping configurations.

The method of manufacture will be to produce a "core" subassembly of externally finned tubes or continuously folded strips either in a circular or straight configuration. This "core" would thus be located inside a small containment vessel manufactured in a manner very similar to that used in producing a water heater vessel. Since the working fluid in this case is compressed air, there would be no significant problems of potable water contamination; therefore, there is no need for a double wall heat exchanger. The prevention of rust is paramount in this application. Dissimilar metals may be used as long as galvanic action is prevented. Becasue of the pressure and temperature requirements, a welded unit is necessary.

The capital costs for the manufacturing equipment would depend upon the design but the containment vessel would use the same fabrication techniques as a water heater and, thus, basically the same equipment. It would be necessary to alter the welding process (weld speed, weld filler wire, etc.) if stainless steel were used, but this is not a significant manufacturing problem.

The core would necessitate a header-tube configuration which could be resistance welded or silver soldered to the containment shell. The header would be die punched using a standard OBI punch press. The tubes would be purchased item ordered "to length" so cutting would be unnecessary. The tubes could be silver soldered or roll formed around a header plate with flanged openings. A core design utilizing a folded strip would be more difficult to manufacture because of the greater skill required to produce uniformity. This could be accomplished with a small capital expenditure for equipment. An approximation for equipment costs plus tooling and fixtures would be $50,000.

A.3 Compressor/Expander

The small, high efficiency, reciprocating air compressor/ expander is the most important component of the high efficiency water heater system. The air compressor/expander is designed to meet several criteria.

- Isentropic efficiency of approximately 90 percent for both the compressor and expander
- Pressure ratio of 3:1
- Volumetric flow rate of 33 scfm
- Compact
- Compatibility with arrangement of the other system components
- Compatible with low cost automated mass production techniques
- Maximum operating temperature of 200°F.

Unfortunately, to meet some of these criteria, the compressor/ expander must be manufactured to very close tolerances. The body of the compressor/expander will probably be a single piece die cast part with the internal valve seats, inlet and outlet ports, cylinder bore, and cylinder head mating surfaces machined to the desired tolerances. Also there will be a single piston, seals, connection rod, bearings, crosshead drive mechanism, and electrical motor. The compressor/expander unit must be thermally insulated and provided with a housing.

The machining of the various compressor/expander items would be the most expensive and the most time-consuming operation as compared to the other items in the high efficiency water heater system. Compressor/expander components such as the crank, balance linkage, connecting rod, plunger rod, and cross head linkages would be forged, whereas the cam lobes, piston springs, seals, bearings, piston, and electrical motor would be purchased from original equipment manufacturers.

If the water heater manufacturer does not have the facilities and equipment necessary for the manufacture of the compressor/expander, then there are two viable options available to the manufacturer; contract the manufacture of the compressor/expander to an existing compressor manufacturer, or make a substantial capital investment ($1,000,000 or more) to "tool up" for the manufacture of the compressor/expander.

A.4 Jacket Shell Assembly

The jacket shell assembly consists of the jacket, jacket top, jacket bottom, and required closure doors. The jacket is made from light gauge cold rolled steel, cut to length, rolled, and welded or lock seemed. The jacket top and jacket bottom are formed by a press.

The appearance of the jacket parts is solely dependent upon the painting operation. To the customer, a poor paint job reflects poor craftsmanship; as a result, sales will decline. To obtain the best quality, the paint must be baked dry. The resulting finish will be a uniform hard surface that can withstand the most extreme weather conditions as was the case for the water storage tank.

All the water heater manufacturers presently have the equipment to produce the jacket shell components. As the demand for the new high efficiency water heater approaches 200,000 units per year, more complex equipment such as electrostatic spray, degreasing lines, automatic unloading, etc., should be considered for reduced labor costs and increased production capacity.

However, as a minimum for an automated operation, an investment of $275,000 would be necessary for a press, feeder, forming dies, jacket roll, lock seam, and painting equipment.

A.5 Other Auxiliary Equipment

An auxiliary electric water heater element will be added to the water heater system. In the event that the reverse Brayton air cycle heat pump fails or the instantaneous hot water demand is greater than the heat pump hot water recovery rate, the auxiliary electric water heating elements will be employed. In this manner, more hot water will be available to the water heater owner. A control system for the conversion from the heat pump to the auxiliary electric water heating element will also be provided.

Other components that are to be installed in the high efficiency water heater system are the thermostat, interconnecting piping and thermal insulation. All of the aforementioned items will be purchased.

At the present time, blanket type fiberglass insulation of thickness between one and two inches and densities between 0.5 and 2.5 pounds per cubic foot is being used on water heaters. In the past, fiberglass insulation has been adequate; however, as the need for energy conservation increases, the requirement to provide better thermal insulation on water heaters becomes much more important. So a better substitute for the blanket type fiberglass is a polyurethane foam which has a density of 2.5 pounds per cubic foot and a thermal conductivity of approximately half of that of fiberglass. However, the initial capital required for the purchase of equipment and assembly line equipment required to place the foam on the exterior surface of the water storage tank is in the order of $175,000 or more.

APPENDIX B. COMMERCIAL MARKET

B.1 Description of a Typical Commercial Water Heater

The commercial water heater sizes range from 50 to 10,000 gallons of water storage capacity; however, 99 percent of commercial water heaters range from 50 to 150 gallons and the heating input to the water requirements vary between 22 to 146 kilowatts (75,000 to 500,000 Btu per hour). Because of the small water storage capacity and the large heating input value, the typical commercial water heater operates in an instantaneous mode rather than a storage mode. As a result, the recovery rates are very large. The commercial water heaters are classified into four categories:

- gas integral tank type
- gas copper tube or coil type
- electric
- oil-fired

Except for the large heating input value, the gas integral tank type and electric water heaters are very similar in design to the residential water heaters. The gas copper tube or coil type does not have a storage tank, but contains a heat exchanger which is directly heated by a gas flame. The gas copper tube or coil type is used for commercial buildings and swimming pools. There are some oil-fired commercial water heaters; however, the total number is very small.

B.2 Past Performance of the Commercial Water Heater Market

A summary of the number of commercial gas-fired and electric water heaters sold in the United States between 1975 and September, 1977 is shown in Table B-1. The figures shown are obtained from the regular statistical reports printed by GAMA which obtains

Table B-1. Number of Commercial Water Heaters Shipped Between 1975 and September 1977 [21]

Year	Gas Integral Tank Type	Percent of Market	Gas Copper Tube or Coil Type	Percent of Market	Electrical	Percent of Market	Total	Percent Growth
1975	68,146	51.1	50,817	38.1	14,431	10.8	133,394	-------
1976	81,762	51.8	61,918	39.2	14,288	9.0	157,968	18.4
Sept. 1977	82,784	-----	53,933	----	------	----	-------	-------

its data from voluntary reporting by the manufacturers of commercial water heaters [21]. The sales of commercial gas-fired water heater exceeds signficantly the sales of electric water heaters which comprises only nine to eleven percent of the total commercial water heater market. The sales of oil-fired commercial water heaters is so small that the manufacturers do not report them.

B.3 Potential Market

In 1976, the average manufacturer's selling price of residential electric and gas-fired water heaters was $77 and the average selling price of a commercial water heater was $450 [23]. Consequently, the residential water heater market was approximately $444 million and the commercial water heater market was approximately $71 million which represented 13.8 percent of the total water heater market in terms of sales dollars.

Because of the smaller market for commercial water heaters, and because of the small market share percentage which commercial electric water heaters serve, there is less potential for volume-sales of a reverse Brayton air cycle heat pump water heater, it is very unlikely that a significant number of gas-fired water heaters will be changed to electric water heaters, unless the price of gas increases or the availability of gas decreases. However, compared to the average residential consumer, businessmen are more likely to understand the economic advantages of employing a high efficiency water heater for commercial use. Thus, consumer acceptance is expected to be high.

Large heat pumps suitable for commercial use could be developed after the application of the present reverse Brayton air cycle heat pump water heater has been demonstrated satisfactorily for residential use. Size and space requirements would not be as signficant in the commercial sector as in the residential sector, but more stringent codes and regulations would have to be satisfied.

Information regarding the future commercial building prospects for various geographical locations of the country, is not readily available.

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RESEARCH AND DEVELOPMENT OF AN
AIR-CYCLE HEAT-PUMP
WATER HEATER

Design Study

October 1, 1979

Prepared Under Subcontract No. 7226 by

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1. INTRODUCTION

This report describes the design of a new efficient electric water heater which employs a reverse Brayton air cycle heat pump. A T-S diagram of the Brayton cycle is shown in Figure 1-1. The water heater consists of a reciprocating compressor and expander, a counterflow water-air heat exchanger, a water circulation pump, an electrical motor, and a water storage tank. A flow schematic of the water heater system is shown in Figure 1-2. The open reverse Brayton air cycle may be described as follows: Ambient air is compressed adiabatically from state point 1 to state point 2, a pressure corresponding to the water heating temperature. The compressed air is then passed through an air-water counterflow heat exchanger where heat, $\dot{Q}_H$, is transferred from the air to the water which is stored in the water storage tank. The cooler high pressure air is then expanded from state point 3 to state point 4 which corresponds to ambient pressure. The expander shaft work, W_E, is used to drive the compressor. In addition, shaft work from an electric motor, W_M, is used to supply the remaining work input required for the compressor.

There are several advantages for the application of a reverse Brayton air cycle heat pump for water heating.

- The air cycle system is independent of an outdoor heat exchanger installation with fan, refrigeration lines, and defrosting mechanism

- The electric motor and compressor/expander need not be hermetically sealed or thermally coupled. Consequently, each component may be optimized, lubricated, and repaired or replaced if necessary

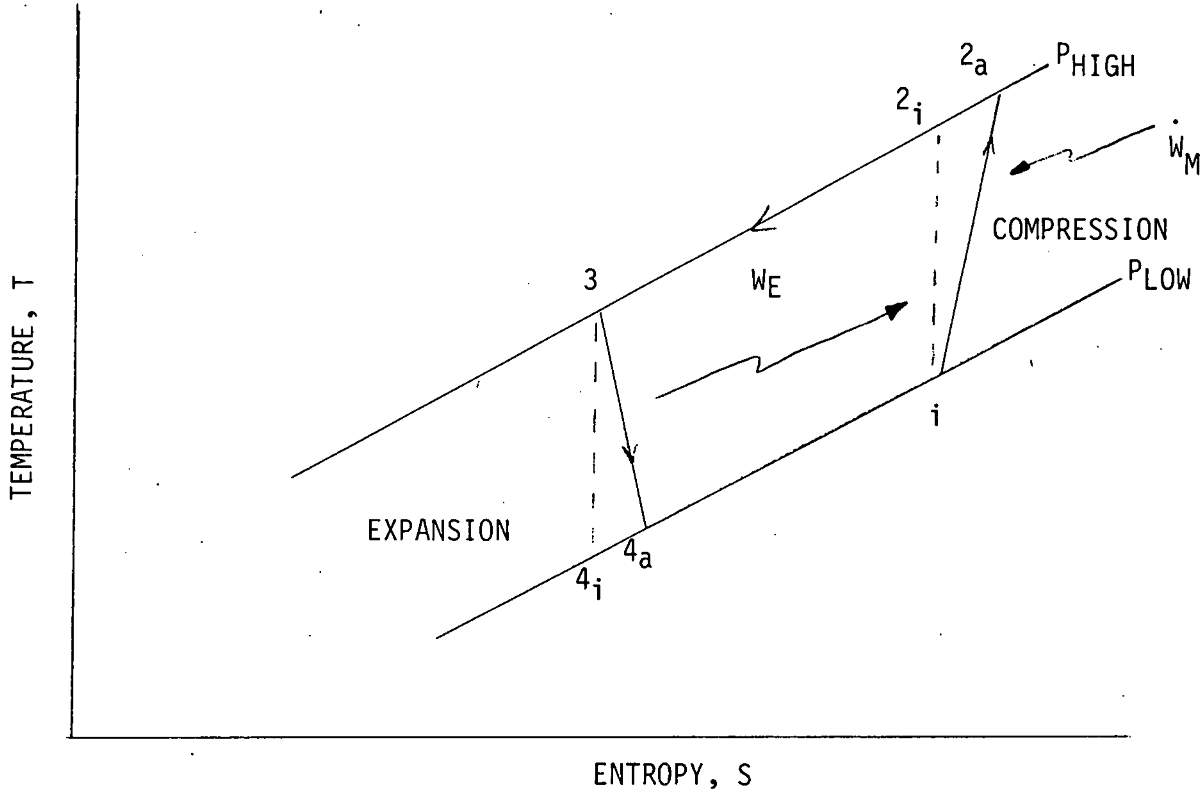


Figure 1-1. T-S Diagram Depicting the Reverse Brayton Air Cycle

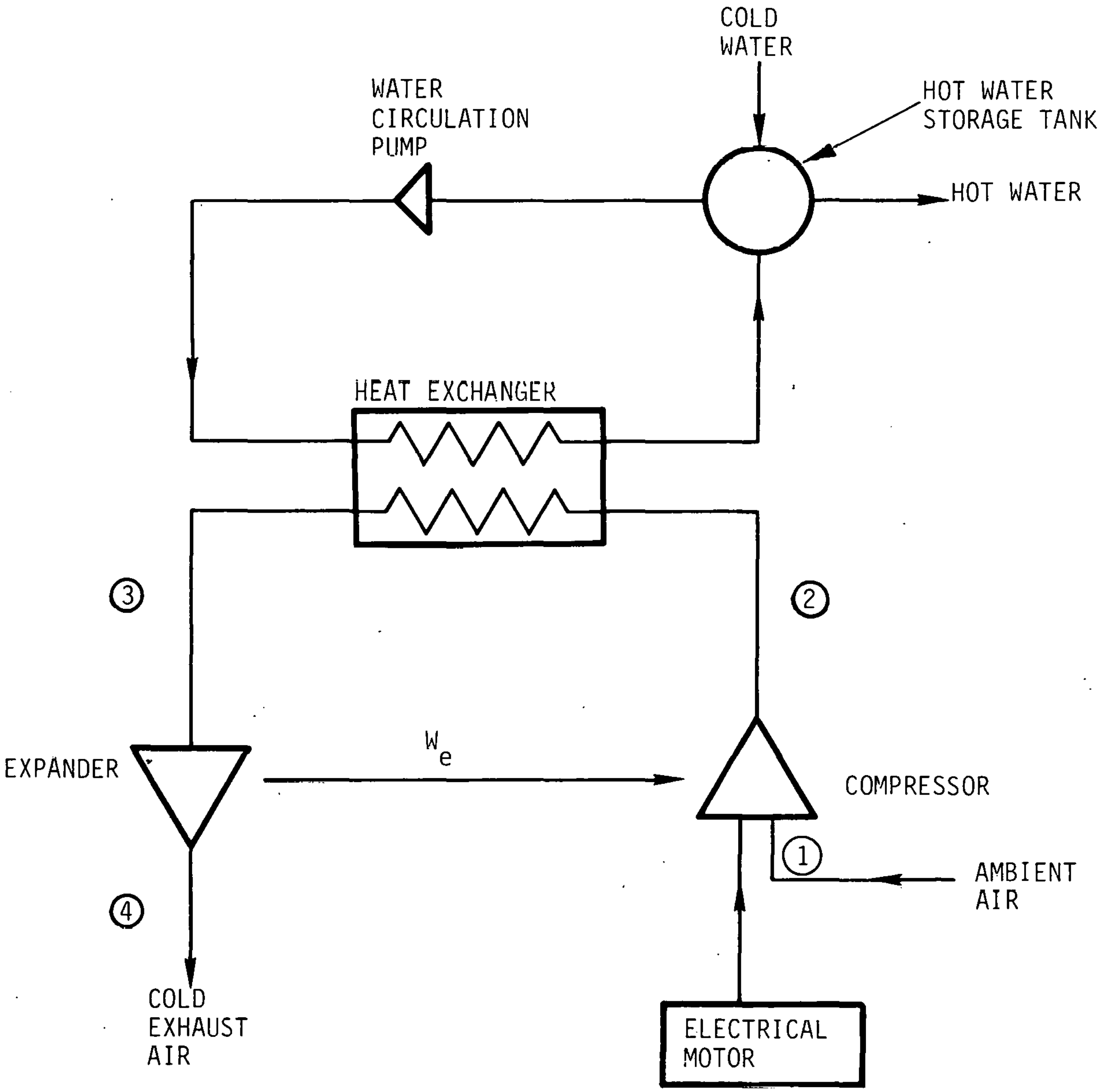


Figure 1-2. Flow Schematic of The Brayton Cycle Water Heater

- The thermal load which is the sensible heat of the water is easily matched with the sensible heat of the air in the air-water counterflow heat exchanger

- The coefficient of performance (COP) and heat pump capacity are insensitive with the ambient temperature. So if the system proves to be effective at standard ambient conditions, it can be considered for operation in any climate

- The working fluid is air.

The present design development included the following:

- The selection of water heater system design specifications

- A thermodynamic parametric analysis of the reverse Brayton cycle to determine the optimum operating thermodynamic state points of the water heater system

- A thermodynamic model to identify the losses occuring in the compressor/expander

- A preliminary and final mechanical design of the compressor/expander and remaining system, and

- The preparation of a test plan for the development and endurance testing of the prototype high efficiency water heater.

2. DESIGN SPECIFICATION

2.1 Results of the Market Study

Prior to the selection of the design specifications, a market study of the reverse Brayton cycle water heater was performed. The conclusions are:

- A potential residential market for the reverse Brayton air cycle heat pump will exist in all major regions in the United States.
- The high efficiency water heater is expected to capture 25 percent of the electric water heater market in newly constructed homes and 10 percent of the electric water heater replacement market, approximately 480,000 units per year, within the first 3 to 4 years of the market penetration effort.
- The payback of the unit is estimated to be 3-1/2 years based on an initial first cost premium of $300 and a seasonal coefficient of performance of 1.7.
- A maximum width of 28 in. and a maximum height of 80 in. are recommended for the unit.
- A water storage tank capacity between 50 and 100 gal is recommended for residential applications.

2.2 Prototype Design Specifications

To minimize the cost of the Brayton cycle water heater requires that the heat pump be as small as possible and the water tank be as large as possible. This is due to the fact that the incremental cost of increasing tank size is less than the incremental cost of providing a higher capacity heat pump. This results in a water heater with a relatively slow recovery rate with a large capacity holding tank. The available hot water, however, will be equivalent to that available from competing conventional water heaters.

The prototype design specifications are listed in Table 2-1. The heating rate of the heat pump is 2.5 kW and a reserve resistance heating element rated at 2.5 kW is to be placed in the water storage tank. As a result, the design capacity of the compressor/expander is 33 scfm. Also listed in Table 2-1 are the specifications for a conventional electric water heater. The significance of the thermal efficiency of the conventional electric water heater and the coefficient of performance of the high efficiency electric water heater is that for given water mass and temperature rise approximately twice as much input electrical power is required.

2.3 Preproduction Design Goals

The marketing study revealed that to make the high efficiency water heater viable in the marketplace, it is necessary that the system COP be 1.7 or greater. In order to achieve this performance goal it is necessary that the individual system components are designed to minimum design specifications. In Table 2-2 are the preproduction system design goal specifications.

Table 2-1. Design Specifications of the High Efficiency Water Heater System and a Conventional System

Parameter	High Efficiency Systems	Conventional Systems
Water Usage Rate	80 gallons per day	80 gallons per day
Hot Water Temperature	140°F	150°F
Recovery Time	8 hr	
Maximum Temperature Rise	100°F	100°F
Water Storage Tank Capacity	80 gallons	52 gallons
Compressor/Expander Mass Flow Capacity	33 scfm	-
Heat Pump Heat Rate	2.5 kW	-
Auxiliary Electric Heating Element Heat Rate	2.5 kW	-
Conventional Electric Heating Element Heat Rate	-	4.5 kW
Maximum Water Heater System Dimensions	76 in. long by 25 in. diameter	50 in. long by 22 in. diameter
Thermodynamic Coefficient of Performance	1.7 or greater	-
Thermal Efficiency	-	83 percent
Reliability	Mean time between failure (MBTF) of 25,000 hrs or more	5 to 10 year Guarantee
Retail Price	$600 or less	$300 or less

Table 2-2. Preproduction System Design Goal Specifications

System Component	Performance Values
Compressor/Expander Isentropic Efficiency	$\geq$ 85 Percent
Air-Water Heat Exchanger Effectiveness	$\geq$ 85 Percent
Pressure Ratio	= 3.0
Electrical Motor Efficiency	$\geq$ 85 Percent
Coefficient of Performance	$\geq$ 1.7

3. SYSTEM DESIGN STUDY

A process flow diagram of the Brayton cycle portion of the water heater appears in Figure 3-1. All of the system components under consideration are included in the flow diagram. Several of these components were subsequently eliminated from the system because they have proven to provide little or no improvement in the overall performance of the system. The system shown in Figure 3-1 is a regenerative open cycle Brayton cycle system. Ambient air is drawn into the system and sprayed with water, reducing the air temperature (and specific volume) and increasing the air humidity. The enthalpy of the air stream remains constant. The air passes through the recuperator, exchanging heat with the expander inlet air (generally increasing in temperature). The air is again sprayed with water, reducing the temperature (and specific volume), increasing the humidity, and not changing the enthalpy. The air is then compressed, substantially increasing its temperature. The hot compressed air then passes through the air-water heat exchanger, counterflow to the water, cooling the air and heating the water to the desired water temperature. If the air is cooled below its dewpoint (at the operating pressure of the air), moisture condenses, transferring additional heat to the water. The cool compressed air passes through the recuperator, being reduced further in temperature and in moisture content, preheating the inlet ambient air. Then the compressed air is expanded, with the recovered work being used to help drive the compressor. The cold discharge air from the expander is exhausted to ambient or may be used to augment air-conditioning during the cooling system.

Driving the compressor requires more work than is recovered by the expander. The extra work is provided by an electric motor. The motor is less than 100 percent efficient with the losses being dissipated as heat that can be recovered to contribute to the water heating load.

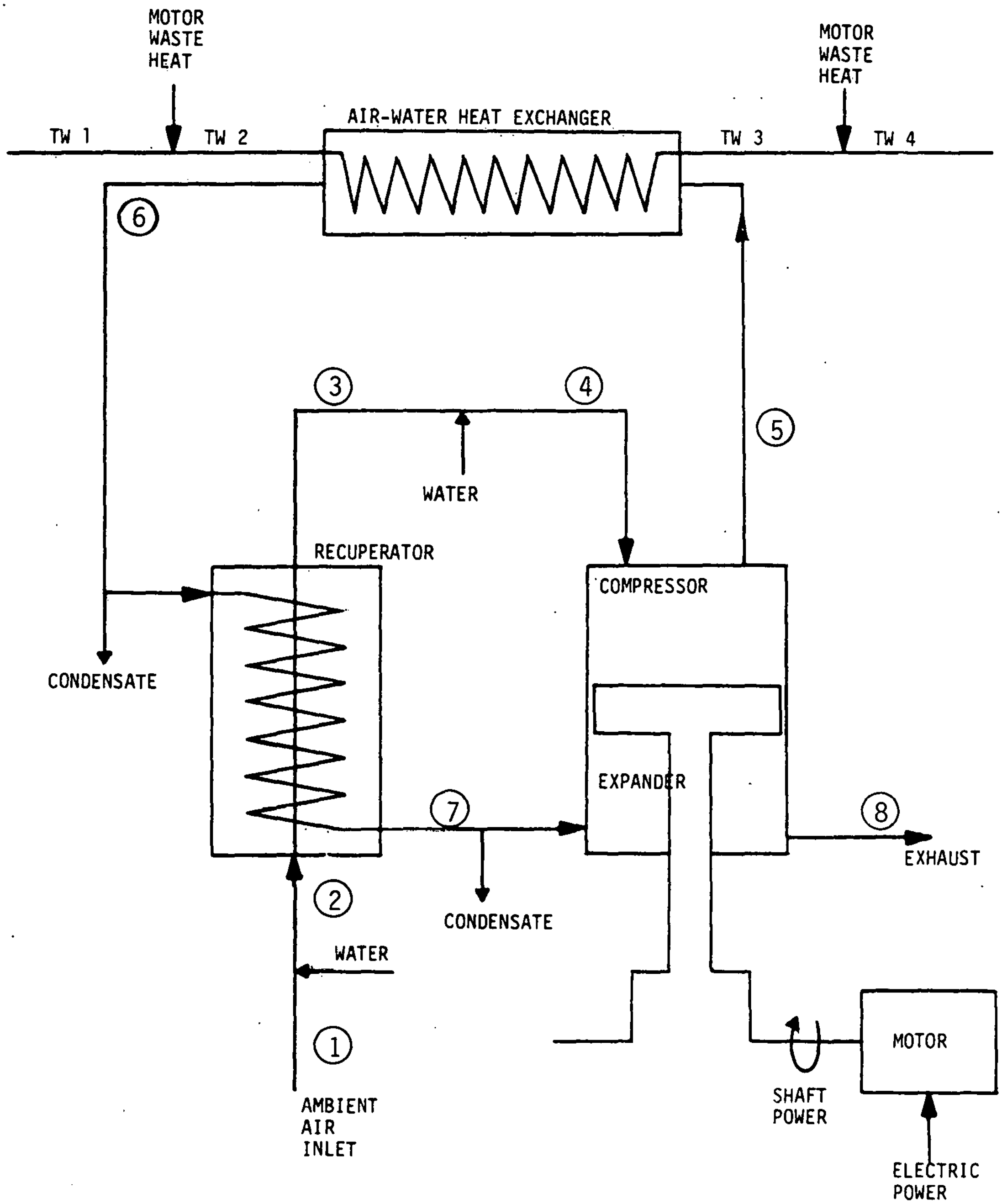


Figure 3-1. Polytropic Regenerative Brayton Cycle Considered for the High Efficiency Water Heater

The water is heated in the motor cooling jacket and the air-water heat exchanger. The water can pass through the two in either order. Passing the water through the motor first results in cooler motor operation, placing less demand on the quality of the motor insulation, bearings, and lubricants. Passing water through the air-water heat exchanger first increases the amount of heat that can be recovered from the hot compressed air, increasing the overall cycle COP.

The design of the water heater system and the design and sizing of the individual components have been directed toward two overall objectives:

- Design for maximum practical heat pumping coefficient of performance, maximizing the energy savings offered by the system

- Design an attractive, simple, reliable, marketable, cost effective system, offering a short enough investment payback period to attract a substantial share of water heater sales.

The design options for a given capacity water heater fall into three categories:

a. Selection of system components

b. The sizing and performance of the system and the components

c. Component design to meet the sizing and performance requirements.

To reach an optimum water heater design, it has been necessary to follow an orderly process of evaluating these design options. First, overall water heating capacity and performance goals presented in Section 2 were established, based largely on conclusions drawn from the marketing study and a preliminary assessment of hardware cost constraints. Next, a thorough study of the effect of design option categories (a) and (b) on the overall system performance was conducted. The results of this system design study, documented in this section of the report, have provided the basis for selection of the system configuration. The design study has determined quantitatively the effect of individual component performance on the overall system performance. This has provided the basis for establishing cost effective component performance specifications.

The effect on system performance of the following component performance measures has been addressed in the design study.

- Compressor/expander isentropic efficiency
- Compressor/expander pressure ratio
- Electric motor efficiency
- Air-water heat exchanger effectiveness
- Air-water heat exchanger pressure drop
- Water spray effectiveness
- Recuperator effectiveness
- Recuperator pressure loss.

The study has considered the effect of the variables in operating conditions that will affect the system performance. These variables are:

- Ambient temperature
- Ambient relative humidity
- Water supply temperature
- Hot water temperature set point.

The first three of these vary due to climatic changes and the hot water temperature setpoint can be changed by the owner of the water heater. A major advantage of the Brayton cycle water heater is the relative insensitivity of cycle performance to changes in these variables. The design study has served to provide quantitative demonstration of this characteristic of the Brayton cycle water heater.

In the following subsections, the system model and optimization technique is described and the results are presented. The resulting system configuration is then presented.

3.1 System Model and Optimization Technique

The effect of individual component performance on overall system performance was assessed using a computer model.

The system is defined in terms of the performance of the individual components, the state of the working fluid (ambient air), and the state of the water being heated at the connection points between components. Thus, the system state variables used are the water temperatures and flow rate through the heating loop and air temperature, humidity, pressure, and flow rate throughout the air circuit. These variables are listed in Table 3-1.

From an assumed set of component performance specifications and operating conditions, the following system performance parameters are calculated and given on a unit SCFM compressor inlet air flow.

a. Compressor work
b. Expander work
c. Compressor/expander net work
d. Motor electric power
e. Heat delivered to water
f. Quantity of water heated
g. COP, heat delivered to water per electric power

3.1.1 Individual Component Models

3.1.1.1 Compressor/Expander

The compressor and expander are modeled to take into account the respective isentropic effiencies and relative displacement of each component. The psychometric properties of air are included. The thermodynamic variables used to describe each component are presented in Table 3-1.

Table 3-1. Variables in the Brayton Cycle Water System Model

Variable	Description
TW 1	Water cold inlet temperature
TW 2	Water temperature at the motor cooling jacket outlet
TW 3	Water temperature at the air-water heat exchanger outlet
TW 4	Water temperature at the motor cooling jacket outlet
T1, 2, etc.	Air temperature at each of 8 points in the system (see Figure 3-1)
H1, 2, etc.	Air humidity ratio
P1, 2, etc.	Air absolute pressure
1	Ambient air
2	Air after first water spray
3	Air after recuperator
4	Air after second water spray
5	Air after compressor
6	Compressed air after air-water heat exchanger
7	Compressed air after recuperator
8	Air at expander exhaust

Estimates of conduction heat loss from the compressor to the expander are calculated. The calculated enthalpy increase for the compressor is reduced by the estimated heat loss fraction. This heat is then added to the inlet enthalpy of the air entering the expander prior to calculation of the expansion work.

Compression of Air/H_2O Vapor

The reversible adiabatic work required to compress a perfect gas in steady flow is given by:

$$W_{rev_c} = c_p (T_2 - T_1) = c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \quad (3.1)$$

where

c_p	=	specific heat at constant pressure
c_v	=	specific heat at constant volume
k	=	c_p/c_v
P_1	=	inlet pressure
P_2	=	outlet pressure
T_1	=	inlet temperature
T_2	=	outlet temperature

The actual compressor work is then given by:

$$W_c = \frac{W_{rev_c}}{\eta_c} = \frac{c_p T_1}{\eta_c} \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \quad (3.2)$$

where

W_c = actual compressor work per unit mass flow

η_c = compressor efficiency

The properties of the gas mixture are given by:

$$c_p = \frac{(c_p)\ \text{air} + \omega\ (c_p)\ H_2O}{1 + \omega} \tag{3.3}$$

$$c_v = \frac{(c_v)\ \text{air} + \omega\ (c_v)\ H_2O}{1 + \omega} \tag{3.4}$$

where

ω = humidity, pound of water vapor per pound of air

Expansion of Air-Water Vapor Mixture

The composition of the gas stream leaving the recuperator is that of air saturated with water vapor. When this mixture expands doing work on the piston, some of the water vapor condenses. The mass of water condensed is small (because the entering humidity or moisture content is limited by the air temperature and pressure) and therefore the change in mass of gas mixture is small. The primary effect of the condensing vapor is a heat addition to the remaining gas. The following simple analysis treats the expansion process as a standard expansion of air with a heat addition term.

The relation between air properties during the reversible expansion can be written as:

$$Tds = dh - vdP$$

where

h = specific enthalpy

P = pressure

s = specific entropy

T = air temperature

v = specific volume

If $Tds = dQ$, $dh = c_p dT$ and the perfect gas law $Pv = RT$ applies, then

$$dQ = c_p dT - \frac{RT}{P} dp \tag{3.6}$$

where R is the gas constant.

The total amount of heat, Q_o, added to the air during the expansion process is determined by the amount of water condensed as

$$Q_o = \omega_c \left(h_{w_1} - h_{w_2}\right) \tag{3.7}$$

where

ω_c = pound of water vapor condensed per pound of air

h_{w_1} = specific enthalpy of water vapor at inlet to expander

h_{w_2} = specific enthalpy of liquid water at outlet of expander

The addition of heat to the gas during the expansion process is taken to vary as:

$$Q = Q_o \frac{T_1 - T}{T_1 - T_2} \tag{3.8}$$

where

T_1 = initial temperature

T_2 = final temperature

Differentiating equation (3.8) and substituting into equation (3.6) yields

$$\frac{-Q_o}{T_1 - T_2} dT = c_p dT - RT \frac{dP}{P} \qquad (3.9)$$

or

$$\left(c_p + \frac{Q_o}{T_1 - T_2}\right) \frac{dT}{T} = R \frac{dT}{P} \qquad (3.10)$$

Integrating equation (3.10) between inlet and outlet states (1) and (2) yields

$$\frac{T_1}{T_2} = \left(\frac{P_2}{P_1}\right)^{\alpha} \qquad (3.11)$$

where

$$\alpha = \frac{R}{c_p} \left[\frac{1}{1 + \frac{Q_o}{c_p (T_1 - T_2)}}\right] = \frac{k-1}{k} \left[\frac{1}{1 + \frac{Q_o}{c_p (T_1 - T_2)}}\right] \qquad (3.12)$$

For the case of $Q_o = 0$ the exponent α in equation (3.11) reduces to the familiar value of $\alpha = k/(k-1)$.

The reversible adiabatic expansion work per unit is then calculated from

$$W_{reve} = h_1 - h_2 + Q_o = c_p(T_1 - T_2) + Q_o \qquad (3.13)$$

where W_{reve} is the reversible adiabatic expansion work per unit mass flow and the actual expansion work per unit mass flow is obtained using the expander effieiency, η_e, as

$$W_e = \eta_e W_{reve} = \eta_e (h_1 - h_2 + Q_o) \qquad (3.14)$$

where W_e is the actual expander work per unit mass flow.

The values of h_2 and T_2 in the above equations result from reversible expansion from the inlet state. If the efficiency, η_e, is not 100 percent then the actual air exit enthalpy h_2' is determined from the steady flow energy equation as

$$h_1 - h_2' = W_e - Q_o \qquad (3.15)$$

and the actual air discharge temperature T_2' is given by

$$c_p(T_1 - T_2') = h_1 - h_2' \qquad (3.16)$$

In the calculational procedure outlined above the value of m_{cwv}, the mass of condensed water vapor, is not known numerically a priori. Once the calculation has been carried out, m_{cwv} and h_{w2} can be calculated using the values of P_2 and T_2 and the inlet humidity. An iterative procedure works very well for this calculation in which m_{cwv} is first taken equal to zero and the procedure is repeated with improved values of m_{cwv}. Three or four iterations are sufficient for obtaining the outlet properties.

3.1.1.2 Air-Water Heat Exchanger

The air-water heat exchanger performance is modeled in terms of its heat transfer effectiveness and air side pressure drop. For an assumed heat transfer effectiveness, ε,

$$\varepsilon = \frac{T_5 - T_6}{T_5 - TW_2} \qquad (3.17)$$

where TW_2 is the water inlet temperature to the heat exchanger

or

$$T_6 = T_5 - \varepsilon (T_5 - TW_2) \qquad (3.18)$$

The air side pressure drop is expressed in the non-dimensional form $\Delta P/P$, where $\Delta P/P$ is the pressure drop and P is the absolute air pressure at the inlet to the air-water heat exchanger (P_5).

The steady state water side heat transfer is equal to the steady state air side heat transfer. The water side inlet and outlet temperatures are specified : TW_2 is the cold water temperature plus the temperature rise in the motor cooling jacket and TW_3 is the specified hot water temperature set point, regulated by the thermostat. Thus, for the water side the water flow rate is calculated so that the air side and water side heat transfer is in balance. Heat is recovered from the air side via the reduction in air sensible heat and recovery of latent heat if water is condensed as the air passes through the heat exchanger. Air side sensible heat transfer, per CFM of air flow, q_{sens}, is calculated as

$$q_{sens} = \rho_{air} \; c_p (T_5 - T_6) \qquad (3.19)$$

where

ρ_{air} = density of air at standard conditions, 0.075 lbm/ft^3

c_p = specific heat of air at constant pressure, 0.24 Btu/lbm - ^oF.

If the air side inlet humidity ratio, H_5, is greater than the saturation humidity ratio at T_6, P_6 at the heat exchanger outlet, the air is assumed to exit the heat exchanger saturated with water vapor, with the extra moisture and the associated latent heat of vaporization having been deposited in the air-water heat exchanger. The latent heat recovered per CFM of air flow, q_{lat}, is calculated as

$$q_{lat} = \rho_{air} \; h_{fg} \; (H_5 - H_6) \tag{3.20}$$

where

h_{fg} = specific latent heat of vaporization of water at T_6, BTU/lbm

H_6 = saturation humidity ratio at T_6, P_6 if this is less than H_5

The rate of water heating, Q_{wat}, is then calculated in lbm/hr of water heated per CFM of air flow as

$$Q_{wat} = \frac{(q_{sens} + q_{lat})}{c_{p_w} (TW_3 - TW_2)} \tag{3.21}$$

where

c_{p_w} = specific heat of water, 1 Btu/lbm - ^{o}F

3.1.1.3 Recuperator

The recuperator is modeled in terms of its heat transfer effectiveness and the pressure drop through each side. For an assumed heat transfer effectiveness, ε,

$$\varepsilon = \frac{T_3 - T_2}{T_6 - T_2} \qquad (3.22)$$

or

$$T_3 = T_2 + \varepsilon (T_6 - T_2) \qquad (3.23)$$

Thus the heat transfer effectiveness of the recuperator is defined in terms of the ambient air side. The pressurized air side heat transfer may involve condensation as well as sensible heat transfer. The pressurized air outlet temperature from the recuperator is calculated so that the decrease in total enthalpy of the pressurized air equals the increase in sensible heat of the ambient air side. Thus,

$$c_p (T_3 - T_2) = c_p (T_6 - T_7) + h_{fg} (H_6 - H_7) \qquad (3.24)$$

where

H_7 = the saturation humidity ratio at P_7, T_7 if this is less than H_6

h_{fg} = latent heat of vaporization of water at T_u, Btu/lb

The pressure drop through each side is expressed in non-dimensional form $\Delta P/P$, where ΔP is the pressure loss through a side of the heat exchanger and P is the absolute pressure of the air at the inlet. Thus,

$$\Delta P_{\text{ambient side}} = (\Delta P/P)\ P_2 \qquad (3.25)$$

and

$$\Delta P_{\text{compressed air side}} = (\Delta P/P)\ P_6 \qquad (3.26)$$

with $(\Delta P/P)$ specified for each side.

3.1.1.4 Water Sprays

The water sprays simultaneously reduce the air temperature and increase the air moisture content, leaving the air total enthalpy unchanged. The air temperature is reduced by the rise of the corresponding sensible heat reduction to supply the heat of vaporization for the increased moisture content. It is assumed that this process is complete, that is, the moist air exits the water spray at a saturation state (100 percent relative humidity).

The computer program has an air psychometric chart stored within it. The specific enthalpy, h, of the inlet air is calculated as

$$h = c_p T_1 + h_{fg}\ H_1 \qquad (3.27)$$

for the first water spray or

$$h = c_p T_3 + h_{fg} H_3 \tag{3.28}$$

for the second water spray. The psychometric chart is then consulted to find the air temperature with the same saturation enthalpy. This temperature and the corresponding saturation humidity ratio specify the outlet conditions from the water spray. It is assumed that no air pressure is lost while passing through the water spray.

3.1.1.5 Electric Motor

The electric motor is modeled as a device that converts electric power to shaft power at a specified efficiency. Thus,

$$\dot{W}_s = \eta_m \dot{E} \tag{3.29}$$

where,

$\dot{W}_s$ = shaft power
η_m = motor efficiency
$\dot{E}$ = electric power

The electric power supplied to the motor that is not converted to shaft power is dissipated as heat. This heat is recovered and used to heat water. The amount of heat recovered, $\dot{Q}$, is given by

$$\dot{Q} = (1 - \eta_m)\dot{E} \tag{3.30}$$

3.1.2 Solution Technique for System Model

The Brayton cycle water heater is inherently a non-linear system. Even with the relatively simple component models used, no convenient closed form solution for the steady-state values of the main system variables exists. Several hundred cases (individual combinations of component performance parameters and operating conditions) had to be worked out to provide sufficient data base for system optimization and evaluation. A numerical solution technique was devised, making use of digital computer computation to efficiently generate solutions to the numerous cases that were treated.

The solution technique employed uses an educated initial guess of the steady value of certain variables - then exact component by component computation provides a set of values of the system variables with which to replace the original guess. By repetition of this process, an accurate solution for the steady state values of the system variables is reached. From these values, the important system performance measures - heating capacity, electric power consumption, COP, etc. - can be calculated.

The solution is calculated by following these steps:

a. The air temperature at the outlet to the air-water heat exchanger (T6) is guessed

b. The first water spray outlet conditions are calculated

c. The heat balance for the recuperator is calculated

d. The second water spray outlet conditions are calculated

e. The ambient air side pressure loss in the recuperator is calculated

f. The cycle pressure ratio, compressor discharge temperature, work, and heat loss to the expander are calculated

g. The expander discharge temperature and work are calculated

h. The net compressor/expander shaft work is calculated

i. The electric motor electric power consumption and waste heat are calculated

j. The water temperature rise through the motor cooling jacket is calculated

k. The heat balance for the air-water heat exchanger is calculated, providing an updated outlet temperature estimate

l. Steps c. through k. are repeated several times, with the values of the variables converging to several decimal place accuracy.

3.1.3 Optimization Technique

The optimum Brayton Cycle Water Heater would provide the purchaser with the lowest life cycle water heating costs. The water heater design would represent a compromise between the maximum physically attainable coefficient of performance (and electric energy saving) and the lowest possible manufacturing cost. In theory, the individual component sizing and performance would result in a system design wherein the difference between the customer's savings in electric costs and the customer's increased cost of amortizing the purchase of the water heater (over a conventional resistance electric water heater) would be a maximum.

The purpose of the design study was to determine an engineering optimum design. The model has been used to establish reasonable component sizes and quantify the consequences on overall system performance of the individual component performance and variable operating conditions. From the standpoint of cost optimization, the optimization process has necessarily proceeded with estimated available cost data, and has relied on the experience and judgement of the members of the project team.

The prototype system design specifications represent a reasonable compromise between energy savings and expected manufacturing cost. A data base was developed by running computer calculations for water heating systems built up from components having a wide range of performance specifications. This provided a quantitative basis to evaluate the effect the performance of an individual component has on the overall COP and capacity. This data base was used in trial and error fashion to specify a high performance system with individual components that will be manufacturable at a low enough cost to be saleable.

Based on the prototype testing experience, the individual component design specifications will be reassessed in light of accurate component and system performance measurements and more reliable cost data.

3.2 Results of Design Study

The effect the design variables have on system performance was examined on a variable by variable basis. This facilitated establishing optimum component sizes and performance specifications and allowed a thorough examination of the effect of variable operating conditions on the performance of the system. The results of the design study are presented in two groupings, the effect of individual component performance on the system performance and the effect of variable operating conditions on the system performance.

Several measures of system performance are of primary interest:

- COP
- Net heat transferred to the service water
- Electric power consumption.

The COP is the ratio of the net heat transferred to the service water to the electric power consumed by the heat pump, in the same units, so that the COP is dimensionless. The heat transfer capacity and electric power consumption are presented throughout this section on a unit compressor displacement basis. The cost of the compressor/expander is largely determined by the air throughput capacity of the compressor. Therefore, it is desirable to maximize the heating capacity for a given compressor capacity. The variation of electric power consumption over the expected range of operating conditions is of particular interest because electric motor efficiency is a maximum value at the design point.

3.2.1 Effect of Component Performance on System Performance

3.2.1.1 Compressor/Expander Efficiency

The overall performance of the water heater is significantly affected by the efficiency of the compressor and expander. The compressor/expander efficiency is defined in terms of isentropic efficiency and the compressor-to-expander conduction heat loss (a discussion of this is presented in 3.1.1.1).

The system COP is plotted versus compressor/expander isentropic efficiency in Figure 3-2 for electric motor efficiencies of 80 percent, 85 percent and 90 percent. The performance of the other components and the operating conditions are held at the constant values tabulated in Figure 3-2. The effect of compressor/expander isentropic efficiency on the COP is substantial so the design goal for the compressor/expander isentropic efficiency is 90 percent to attain a system COP of 1.7.

The effect of compressor to expander conduction heat loss on the COP is shown in Figure 3-3 for compressor/expander isentropic efficiencies and motor efficiencies of 85 and 90 percent. Conduction heat losses cause corresponding reductions of the system COP. A major design specification for the compressor/expander is to limit the conduction heat loss to less than one percent of the compression work.

The heating capacity per compressor displacement is plotted as a function of compressor/expander isentropic efficiency in Figure 3-4 for motor efficiencies of 80, 85 and 90 percent. The effect of compressor/expander efficiency on the heating capacity is not substantial, so this is not an important design consideration.

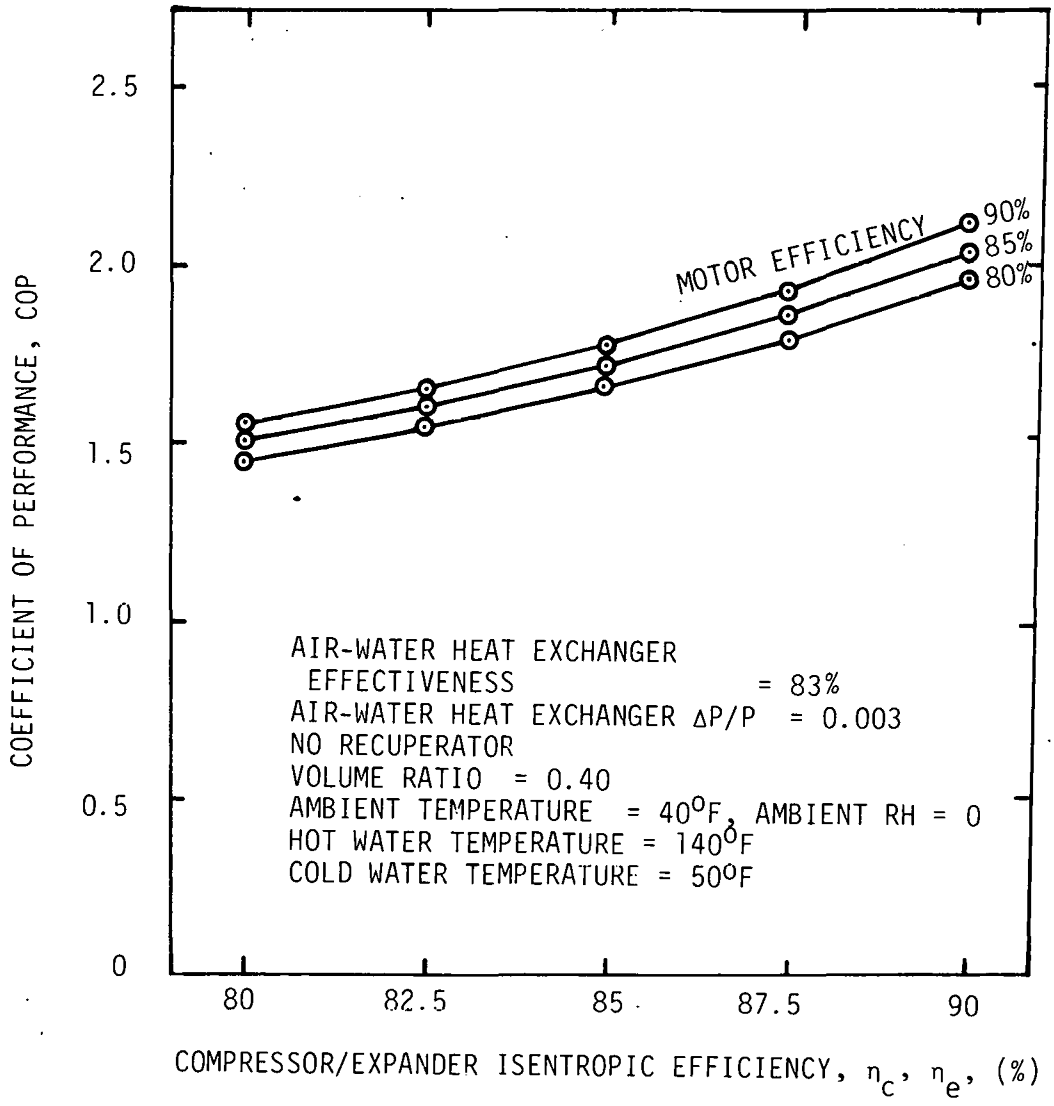


Figure 3-2. Effect of Compressor/Expander Isentropic Efficiency on the Heat Pumping Coefficient of Performance

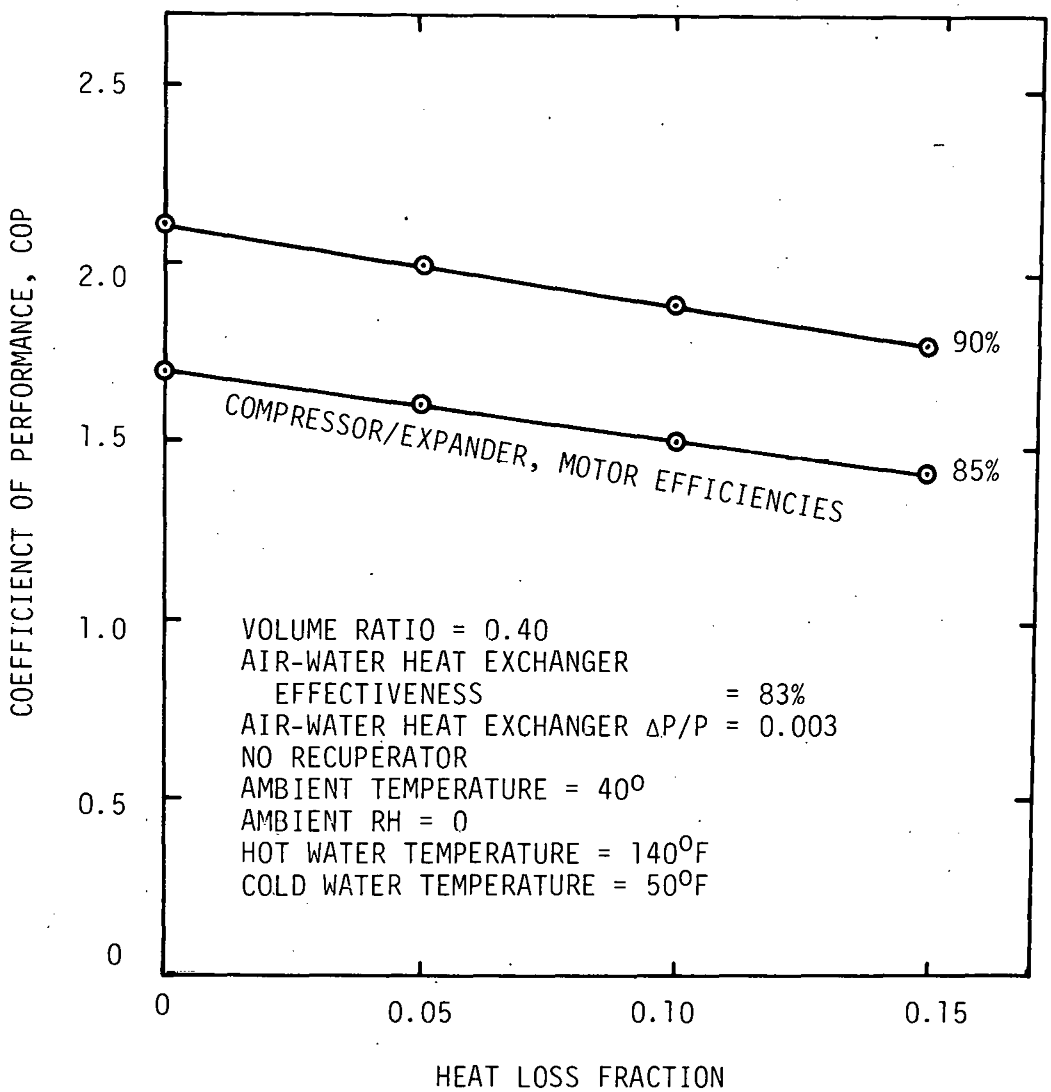


Figure 3-3. Effect of Compressor to Expander Conduction Heat Loss on Heat Pumping Coefficient of Performance

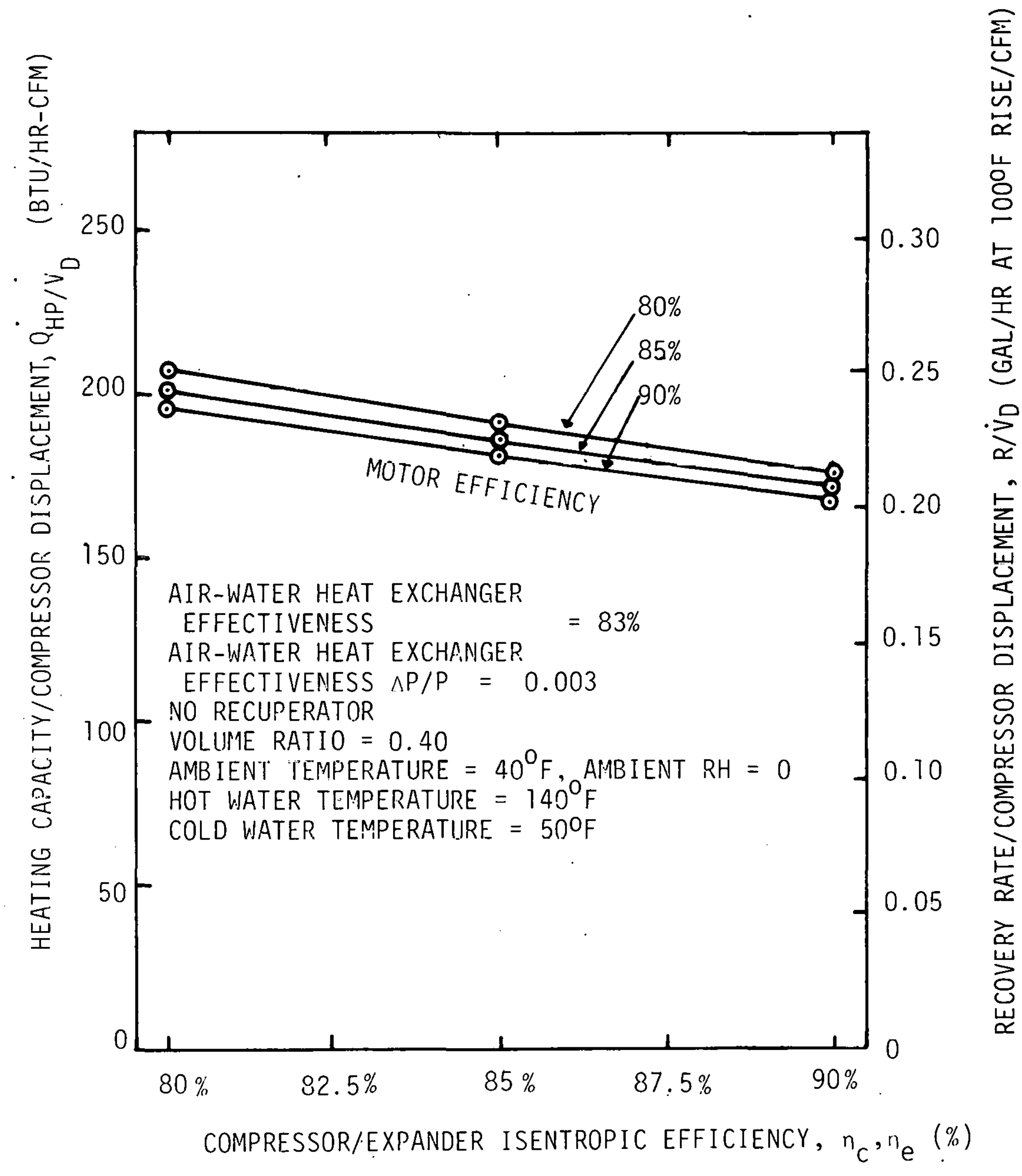


Figure 3-4. Effect of Compressor/Expander Isentropic Effieincy on Heating Capacity Per Compressor Displacement

The heating capacity per compressor displacement is plotted as a function of compressor to expander heat loss fraction in Figure 3-5. Substantial conduction heat losses cause heating capacity losses that are approximately proportional to the reductions in COP illustrated in Figure 3-3. Thus, minimizing the heat loss fraction is consistent with maximizing the heating capacity of the system as well as maximizing the COP.

3.2.1.2 Pressure Ratio

The pressure ratio is determined primarily by the ratio of the compressor displacement to the expander inlet cut-off volume. Temperature variations in the cycle have a secondary effect on the pressure ratio. The pressure ratio has opposite effects on the COP and heating capacity, with increasing pressure ratio increasing the heating capacity, but decreasing the COP. The heating COP is plotted versus pressure ratio in Figure 3-6. The heating capacity is plotted versus pressure ratio in Figure 3-7. The COP shows a mild decline with increasing pressure ratio. The heat pumping capacity shows substantial increases with increasing pressure. Higher design pressure ratios, in the 3.0 - 3.5 range are favored, minimizing the size and cost of the compressor/expander for a given heat pumping capacity, accepting a mild penalty in the COP.

3.2.1.3 Electric Motor Efficiency

The effect of electric motor efficiency on overall system performance is illustrated in Figures 3-8 and 3-9.

In Figure 3-8 is a plot of the overall COP versus motor efficiency for several compressor/expander efficiencies. The motor losses result in heat generation in the motor. This waste heat i-

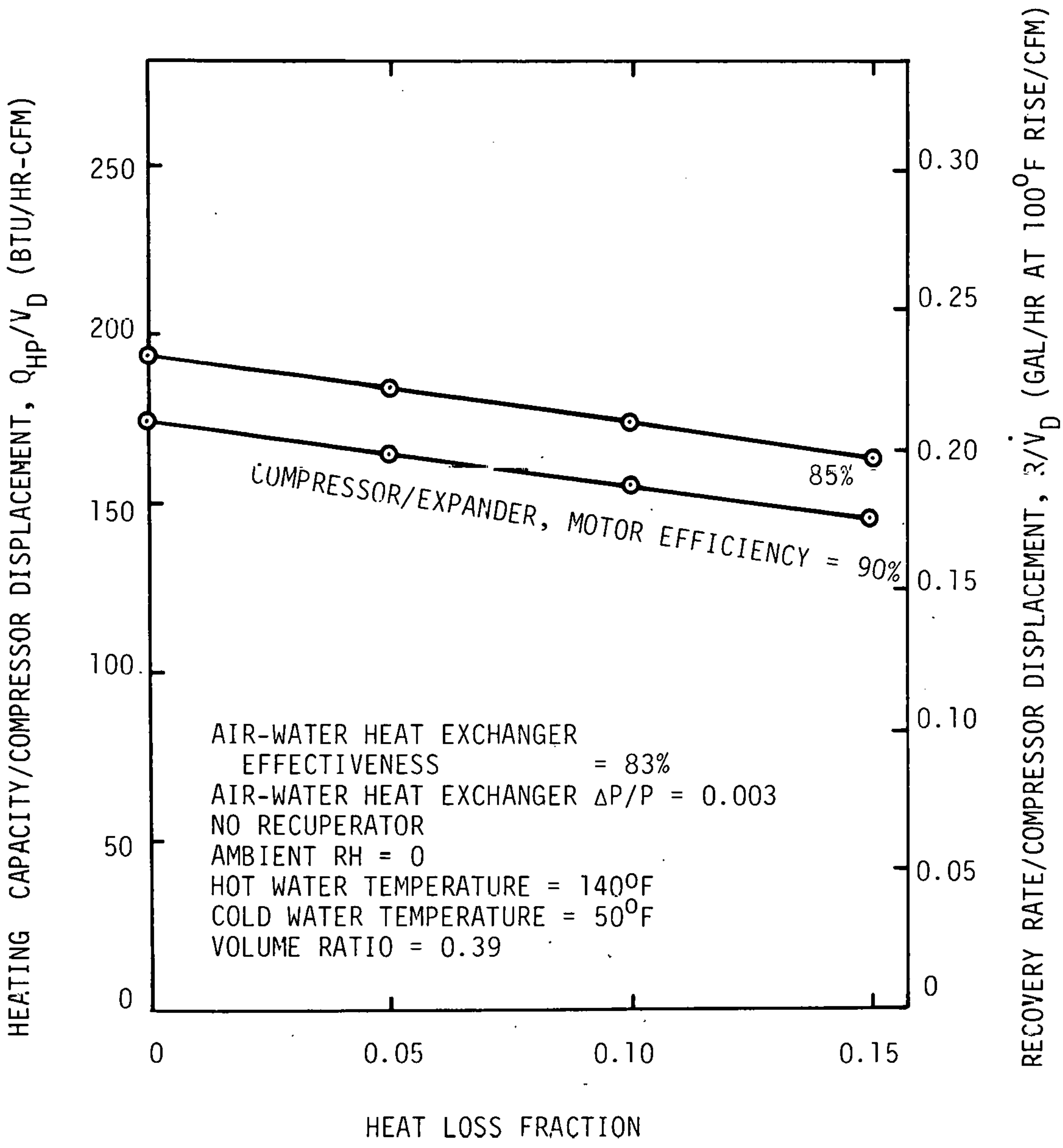


Figure 3-5. Effect of Compressor to Expander Conduction Heat Loss on Heating Capacity per Compressor Displacement

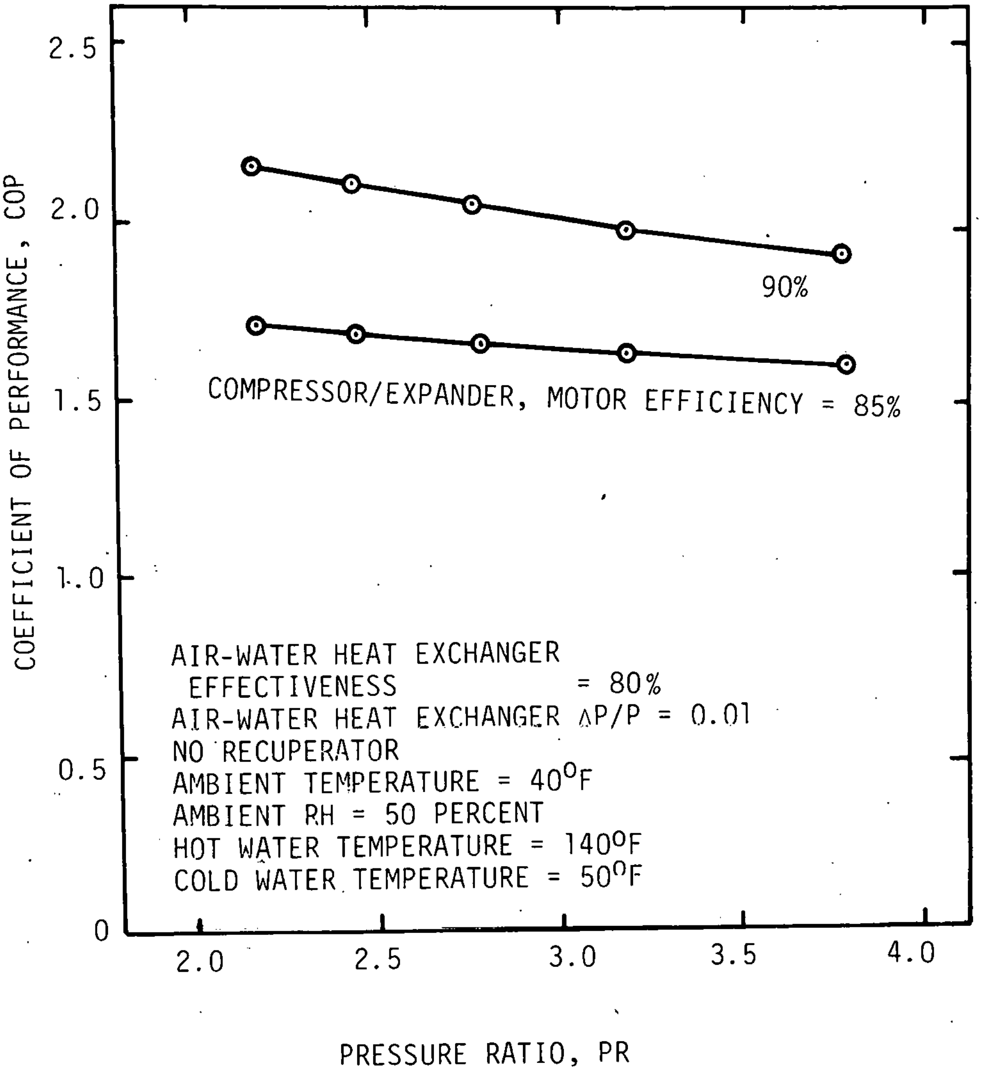


Figure 3-6. Effect of Pressure Ratio on the Coefficient of Performance

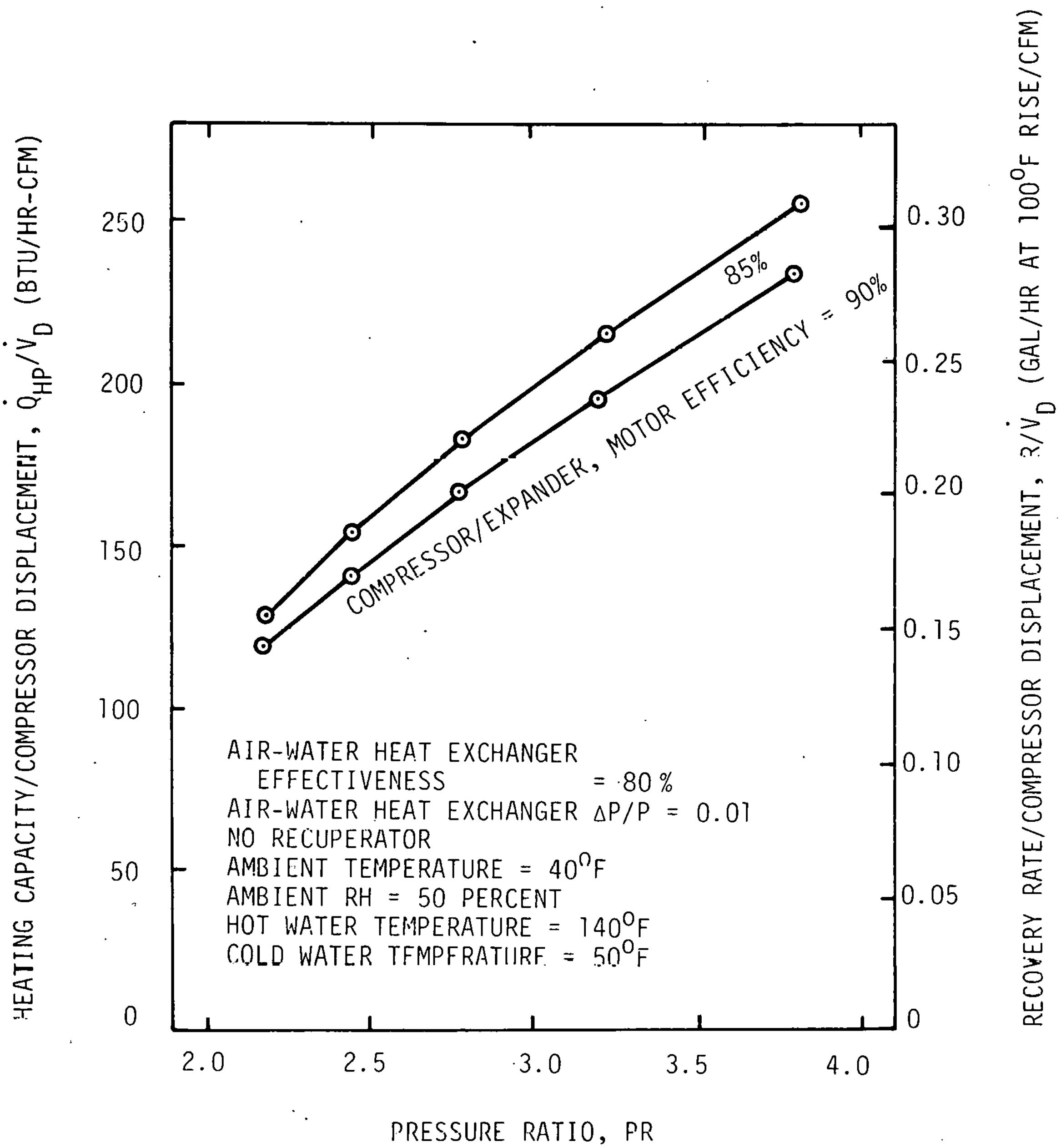


Figure 3-7. Effect of Pressure Ratio on Heating Capacity

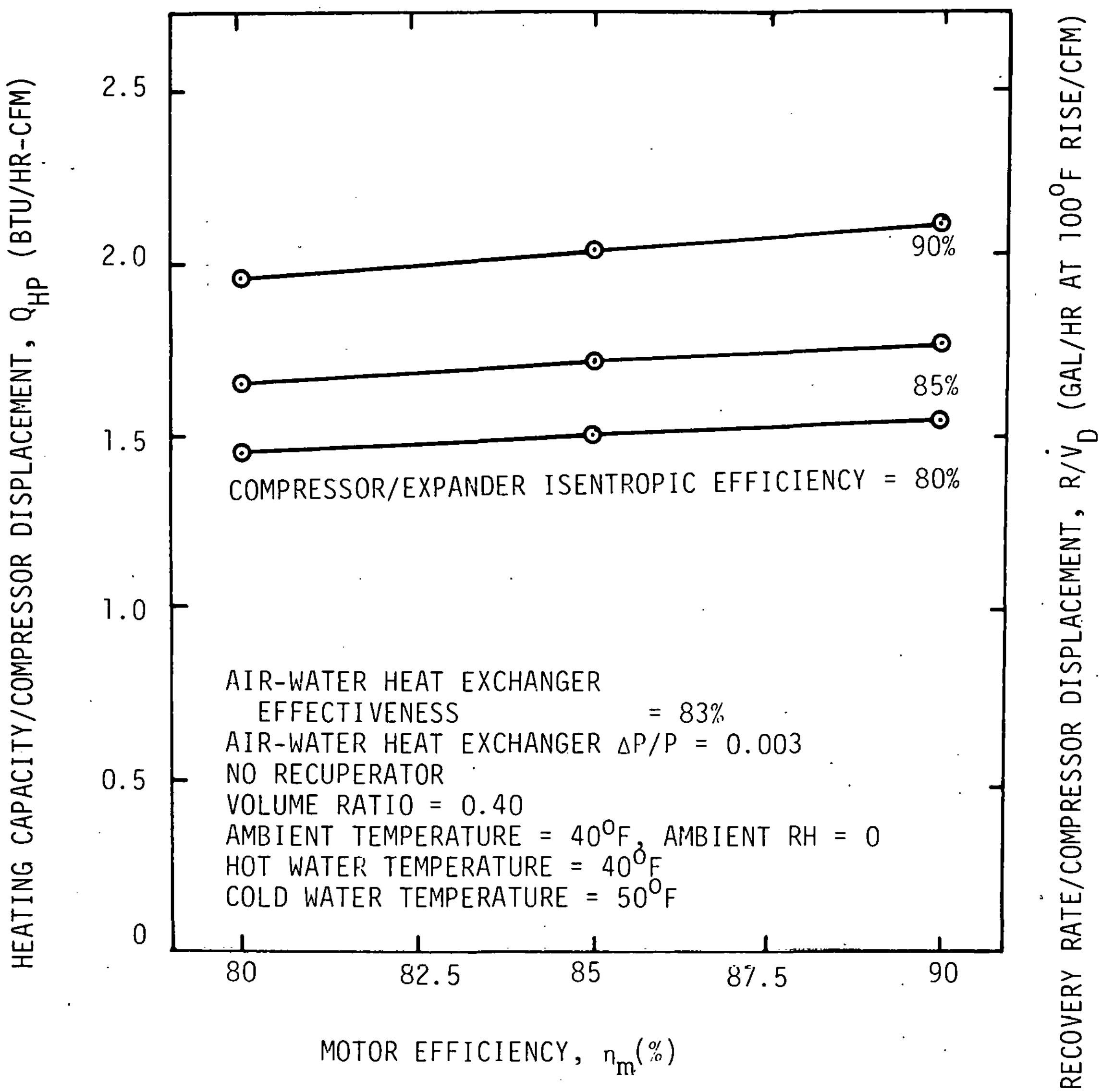


Figure 3-8. Effect of Motor Efficiency on Heating Coefficient of Performance

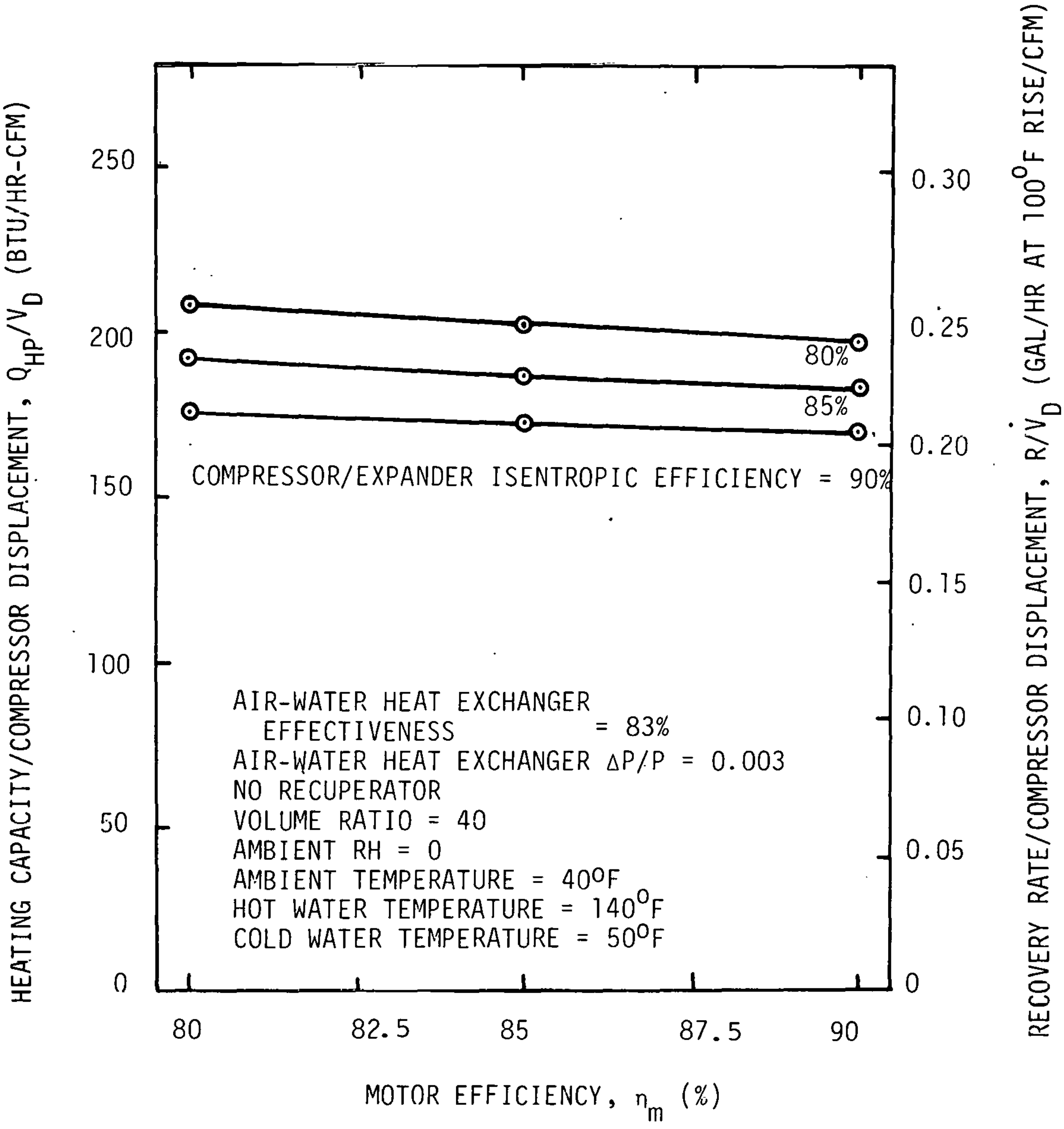


Figure 3-9. Effect of Motor Efficiency on Heating Capacity per Compressor Displacement

used for water heating, so a given motor inefficiency level causes a reduction in overall COP approximately proportional to one half of the motor inefficiency.

In Figure 3-9 a plot of the heating capacity per compressor displacement versus motor efficiency is presented. The heating capacity increases as the motor efficiency decreases. The increased heating capacity relfects the increased amount of motor waste heat recovered for water heating at decreased motor efficiency. Because this additional heat input is effectively supplied by dissipation of input electric power, there is no advantage to a capacity increase provided in this manner.

3.2.1.4 Air-Water Heat Exchanger Performance

The heat transfer effectiveness of the air-water heat exchanger and the air side pressure loss through the air-water heat exchanger each affect the overall system performance.

Increased air-water heat exchanger effectiveness increases the COP by increasing the heat transferred to the water relative to the losses of the compressor/expander. Representative data illustrating this effect is plotted in Figure 3-10.

The heat pumping capacity for a fixed compressor displacement is also increased by the increased heat transfer per air mass flow that results from increased heat exchanger effectiveness. Representative data illustrating this effect is plotted in Figure 3-11.

Increased pressure loss through the air side of the air-water heat exchanger reduces the COP by reducing the amount of work that is available for the expander to recover. Representative data illustrating this effect is plotted in Figure 3-12.

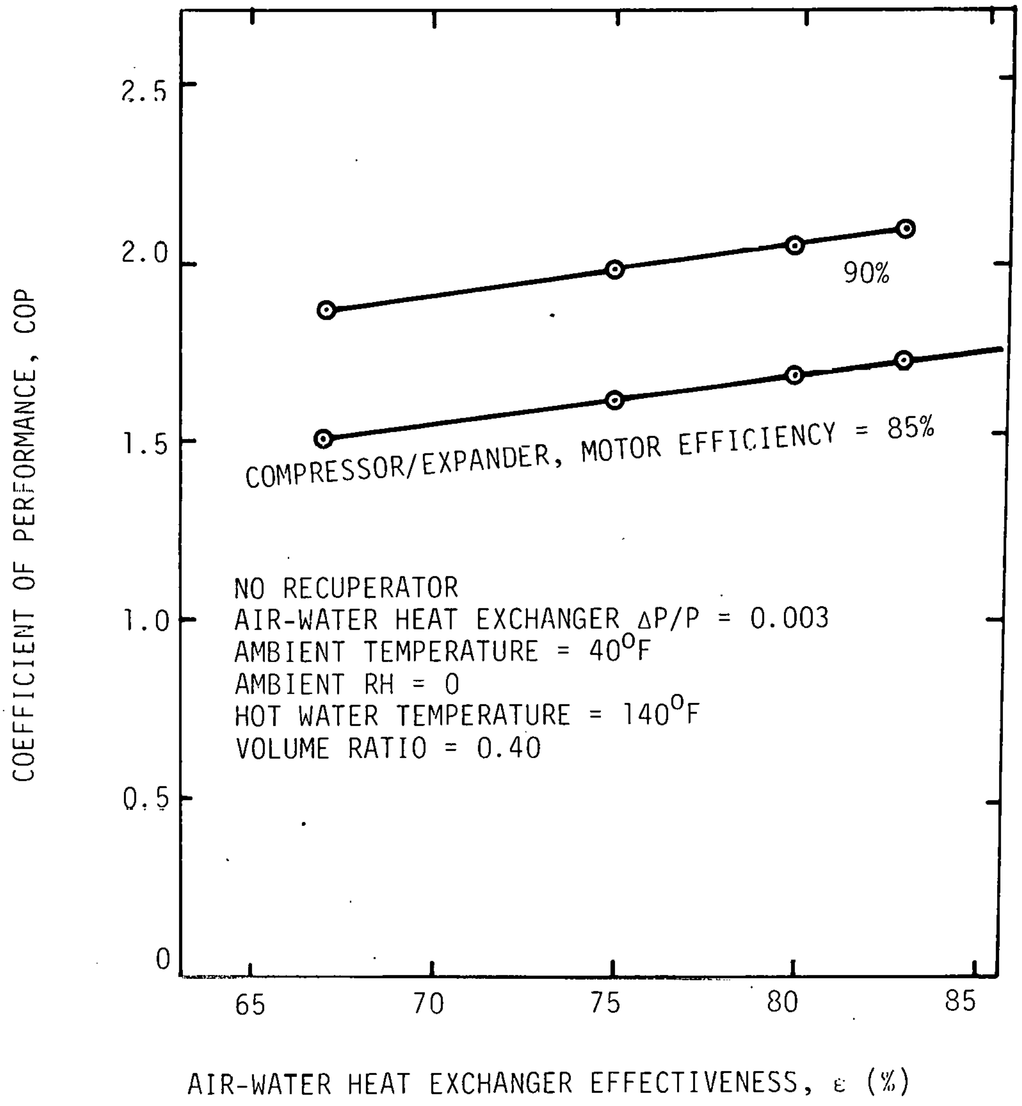


Figure 3-10. Effect of Air-Water Heat Exchanger Effectiveness on Heating COP

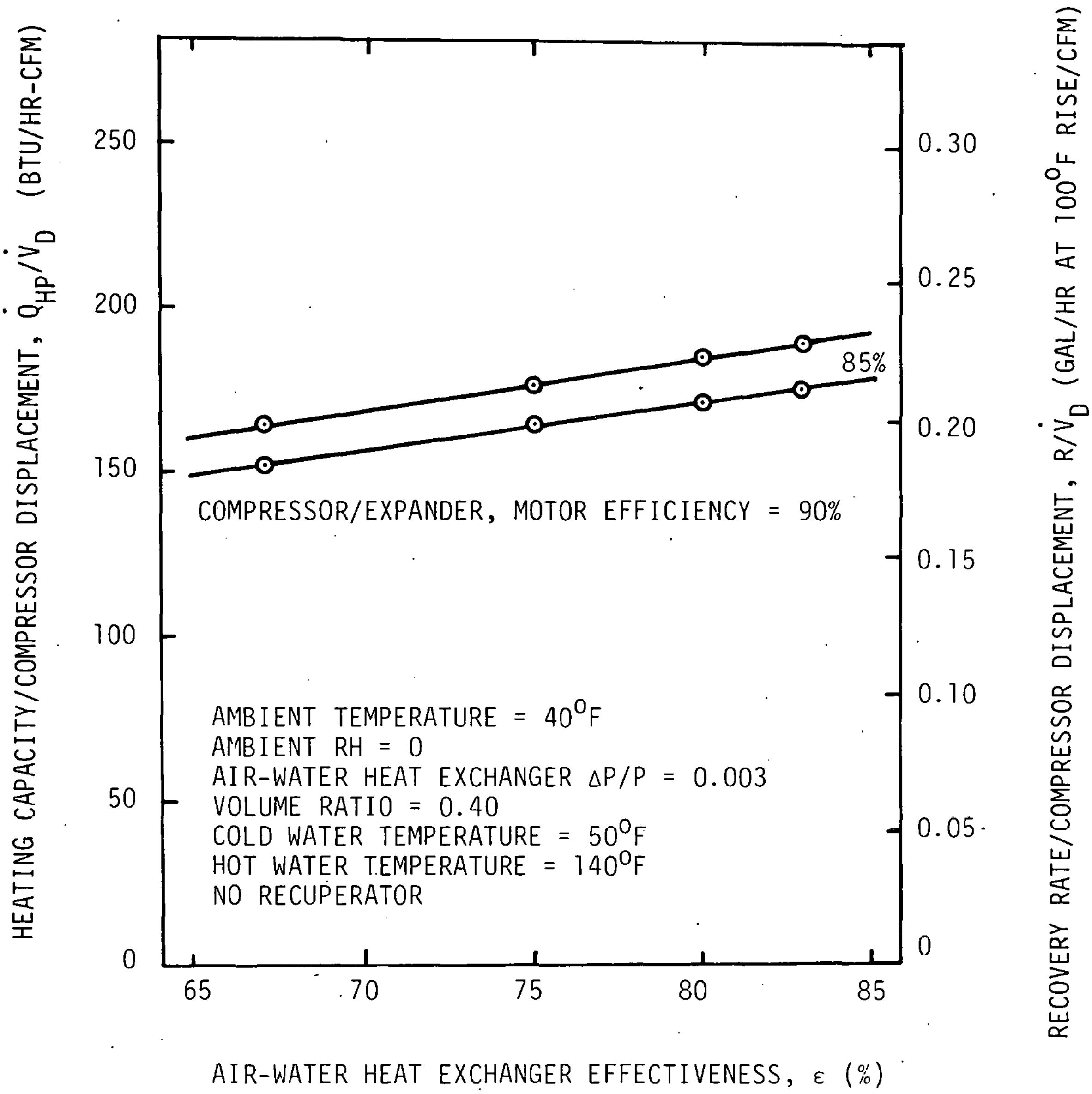


Figure 3-11. Effect of Air-Water Heat Exchanger Effectiveness on Heating Capacity per Compressor Displacement

The air-water heat exchanger air side pressure loss has a negligible effect on the heating capacity at fixed compressor displacement.

Preliminary design studies for the air-water heat exchanger (described in Section 5.1.1) indicated that a compact, inexpensive air-water heat exchanger could be designed with both a high heat transfer effectiveness and a low pressure loss. The heat exchanger design specification was set accordingly for an 83-85 percent heat transfer effectiveness with a pressure loss $\Delta P/P$ less than 0.005 (6 inches of water at the design pressure).

3.2.1.5 Recuperator and Water Sprays

The recuperator serves to improve the overall COP when the ambient temperature is low. By preheating the incoming air prior to compression and precooling the compressed air prior to expansion, a relatively large increase in the heat available for water heating is accomplished. The required compression work undergoes a small increase and the available expansion work undergoes a small decrease, slightly increasing the cycle net work. The net result is an increase in COP. The first water spray increases the amount of heat that can be recovered by the recuperator. The second water spray reduces the required compression work. To obtain an improvement in the COP from the water sprays requires that the water vapor latent heat content of the air be recovered via condensation in the air-water heat exchanger. The COPs that result from all combinations of using or not using a 67 percent effective recuperator and the two water sprays are tabulated in Table 3-2. The system for this set of results has a 90 percent efficient compressor/expander and motor and an 83 percent effective air-water heat exchanger, the design goals for these components. The water sprays cause small decreases in the COP in most cases. The recuperator provides a 7 percent increase in COP

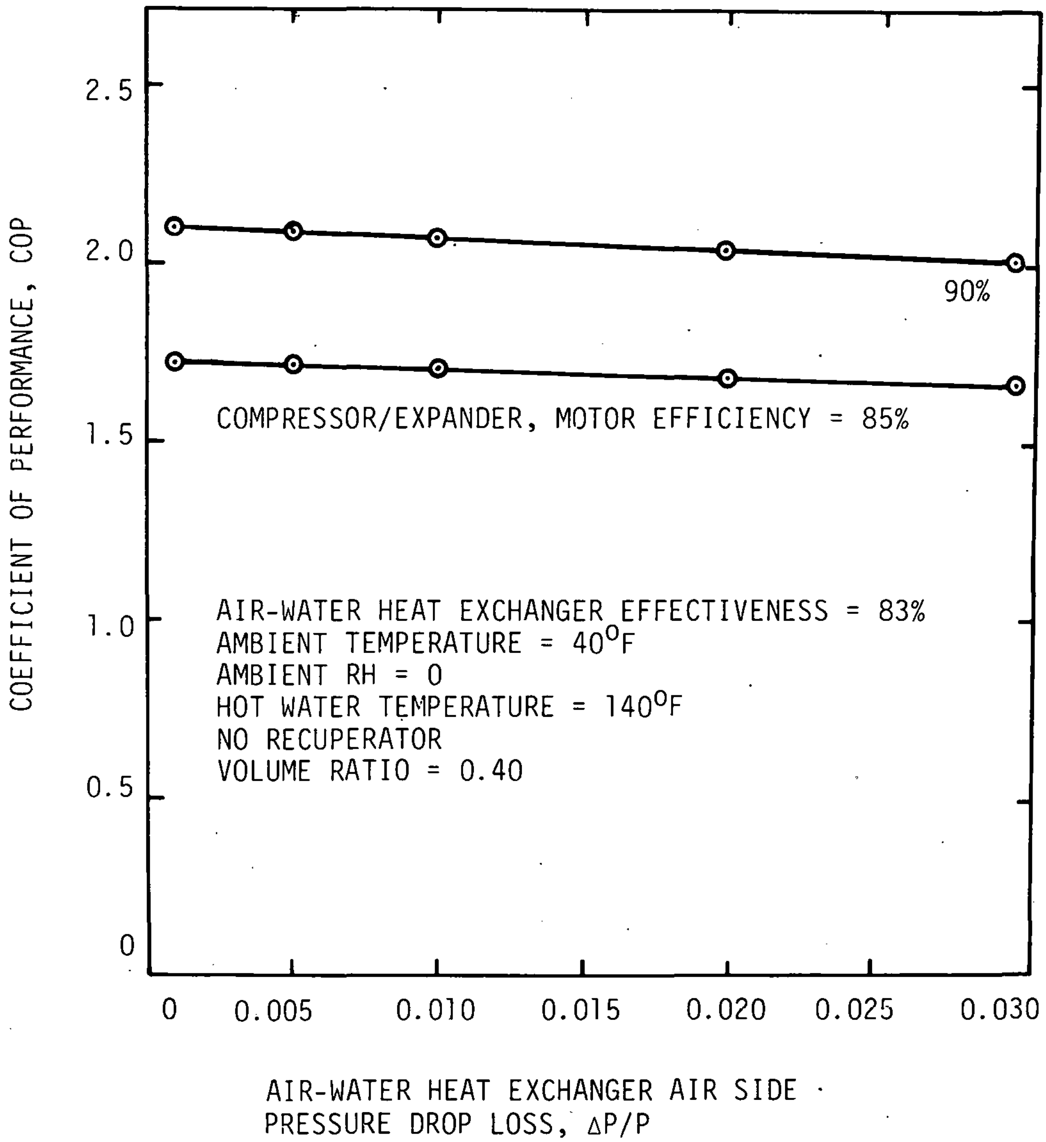


Figure 3-12. Effect of Air-Water Heat Exchanger Air Side Pressure Loss on Heating COP

Table 3-2. Effect of Water Sprays and 67% Effective Recuperator on COP with Other System Components At Design Goal Performance

Air-Water HX $\Delta P/P$	0.003
Air-Water HX Effectiveness	83%
Compressor/Expander, Motor Efficiency	90%
Compressor/Expander Volume Ratio	0.39
Recuperator 67% Effective	
Recuperator $\Delta P/P$	0.002
1 only - Water spray at inlet of recuperator	
2 only - Water spray at exit of recuperator	

		Coefficient of Performance					
		No Recuperator		67 Percent Effective Recuperator			
Amb Temp	Amb R.H.	H_2O No Inj.	H_2O Inj.	No H_2O Inj.	H_2O 1 only	H_2O 2 only	H_2O 1 & 2
0	0	1.807	1.785	1.931	1.914	1.807	1.812
	50%	1.807	1.796	1.930	1.921	1.815	1.818
	100%	1.807	1.807	1.929	1.929	1.825	1.825
40	0	2.104	2.000	2.156	2.073	1.960	1.973
	50%	2.095	2.043	2.144	2.109	2.005	2.010
	100%	2.077	2.077	2.141	2.141	2.043	1.043
80	0	2.369	2.109	2.369	2.179	2.079	2.104
	50%	2.320	2.323	2.259	2.320	2.347	2.417
	100%	2.813	2.813	2.696	2.696	2.749	2.749

Table 3-3. Effect of Water Sprays and 100% Effective Recuperator on COP for System with 100% Effective Air-Water Heat Exchanger

Air-Water HX	100% Eff
Compressor/Expander, Motor Efficieny	90% Eff
Compressor/Expander Volume Ratio	0.39
Recuperator 100% Eff	
Recuperator 100% Eff, No Pressure Loss	
1 only - Water spray at inlet of recuperator	
2 only - Water spray at exit of recuperator	

		Coefficient of Performance					
		No Recuperator		100 Percent Effective Recuperator			
Amb Temp	Amb R.H.	No H_2O Inj.	H_2O Inj.	No H_2O Inj.	H_2O 1 Only	H_2O 2 Only	H_2O 1 & 2
0	0	1.992	1.967	2.176	2.171		2.105
	50%	1.992	1.979	2.176	2.176	2.102	2.113
	100%	1.991	1.991	2.182	2.812	2.121	2.121
40	0	2.337	2.213	2.362	2.314	2.257	2.336
	50%	2.325	2.307	2.357	2.370	2.370	2.409
	100%	2.418	2.418	2.454	2.454	2.484	2.484
80	0	2.659	2.590	2.579	2.593	2.488	2.604
	50%	3.000	3.060	2.951	3.025	2.930	2.979
	100%	3.547	3.547	3.500	3.500	3.388	3.388

at $0^{O}F$ ambient, a 2 percent increase in COP at $40^{O}F$ ambient, and a 3 percent *decrease* in COP at $80^{O}F$ ambient.

In Table 3-3, the system is changed to include a 100 percent effective air-water heat exchanger. The recuperator, when included in the system, is also 100 percent effective. The objective of these runs was to assess the upper limit of performance improvements achievable with the use of water sprays. Again, the water sprays result in small decreases in the COP in most cases and minimal increases (less than two percent) in other cases.

In Table 3-4, results are tabulated comparing a system with an 83 percent effective air-water heat exchanger, no water sprays, and a 100 percent effective recuperator with an otherwise identical system having no recuperator. This set of results served to define the upper limit of performance improvement possible with the recuperator, given a realistically sized air-water heat exchanger. In this case, approximately a 15 percent increase in COP is attained at $0^{O}F$ ambient temperature, a 6 percent increase at $40^{O}F$ ambient temperature and a 2 - 6 percent decrease at $80^{O}F$ ambient temperature, depending on the humidity level. Even at this upper limit of recuperator performance, the overall system performance improvements are not substantial.

It is concluded, based on the data presented, that water sprays result in no net improvement to system performance. A realistically sized recuperator (Table 3-2 results) provides a modest increase in the COP at low ambient temperature, however the average annual increase in COP attained is minimal, not adequate to justify the cost of including a recuperator in the system. The final system configuration, therefore, does not include a recuperator or water sprays.

Table 3-4. Effect of 100% Effective Recuperator on COP for System with 83% Effective Air-Water Heat Exchanger

Amb Temp	Amb RH	Coefficient of Performance: 100% Effective Recuperator	Coefficient of Performance: No Recuperator
0%	0%	2.112	1.807
	50%	2.111	1.807
	100%	2.116	1.807
40%	0%	2.228	2.104
	50%	2.222	2.099
	100%	2.242	2.092
80%	0%	2.350	2.369
	50%	2.315	2.321
	100%	2.536	2.813

Hot Water Temperature = 140°F

Air-Water Heat Exchanger Effectiveness = 83%

Air-Water Heat Exchanger $\Delta P/P = 0.003$

Volume Ratio = 0.039

Recuperator $\Delta P/P = 0$

3.2.2 Effect of Variable Operating Conditions on System Performance

3.2.2.1 Ambient Temperature

Changes in ambient temperature have two effects on the performance of the heat pump.

- Reduced ambient temperature increases the heat pumping temperature differential, decreasing the Carnot cycle COP
- Reduced ambient temperature increases the air density, increasing the air mass flow through the system when a positive displacement compressor is used.

The net effect of these factors is that the COP and heating capacity undergo mild declines with decreasing ambient temperature. For a constant speed, positive displacement compressor/expander with fixed expander valve timing, the electric power consumption and pressure ratio remain almost constant. Thus, the compressor/expander motor package runs close to the design point over a wide range of ambient temperatures.

These results are illustrated quantitatively in Figures 3-13 through 3-16. In these Figures, COP, heating capacity, electric power consumption and pressure ratio are plotted against ambient temperature for various volume ratios for the system defined by Table 3-5.

The volume ratio is defined as the ratio of the expander cutoff volume to the compressor displacement. The pressure ratio of the compressor is inversely related to the volume ratio through the thermodynamic relationships for the compression process. For small volume ratios (large pressure ratios) the size of the heat pump system will increase.

Table 3-5. System Definition For Figures 3-13 Through 3-16

Motor Efficiency, η_m	90 percent
Compressor Efficiency, η_c	90 percent
Expander Efficiency, η_e	90 percent
Volume Ratio*, VR	Variable
Heat Loss Fraction	0.005
Air-Water HX Effectiveness, ε	80 percent
Air-Water HX $\Delta P/P$	0.01
Recuperator Effectiveness, ε_R	0
Recuperator, $\Delta P/P$	0
Water Injection	None
Ambient Temperature, T_o	Variable
Ambient RH	50 percent
Cold Water Temperature, TW_1	40°F at 0° Ambient, 60°F at 80° Ambient
Hot Water Temperature, TW_2	140°F

* Ratio of expander inlet cut-off volume to compressor displacement

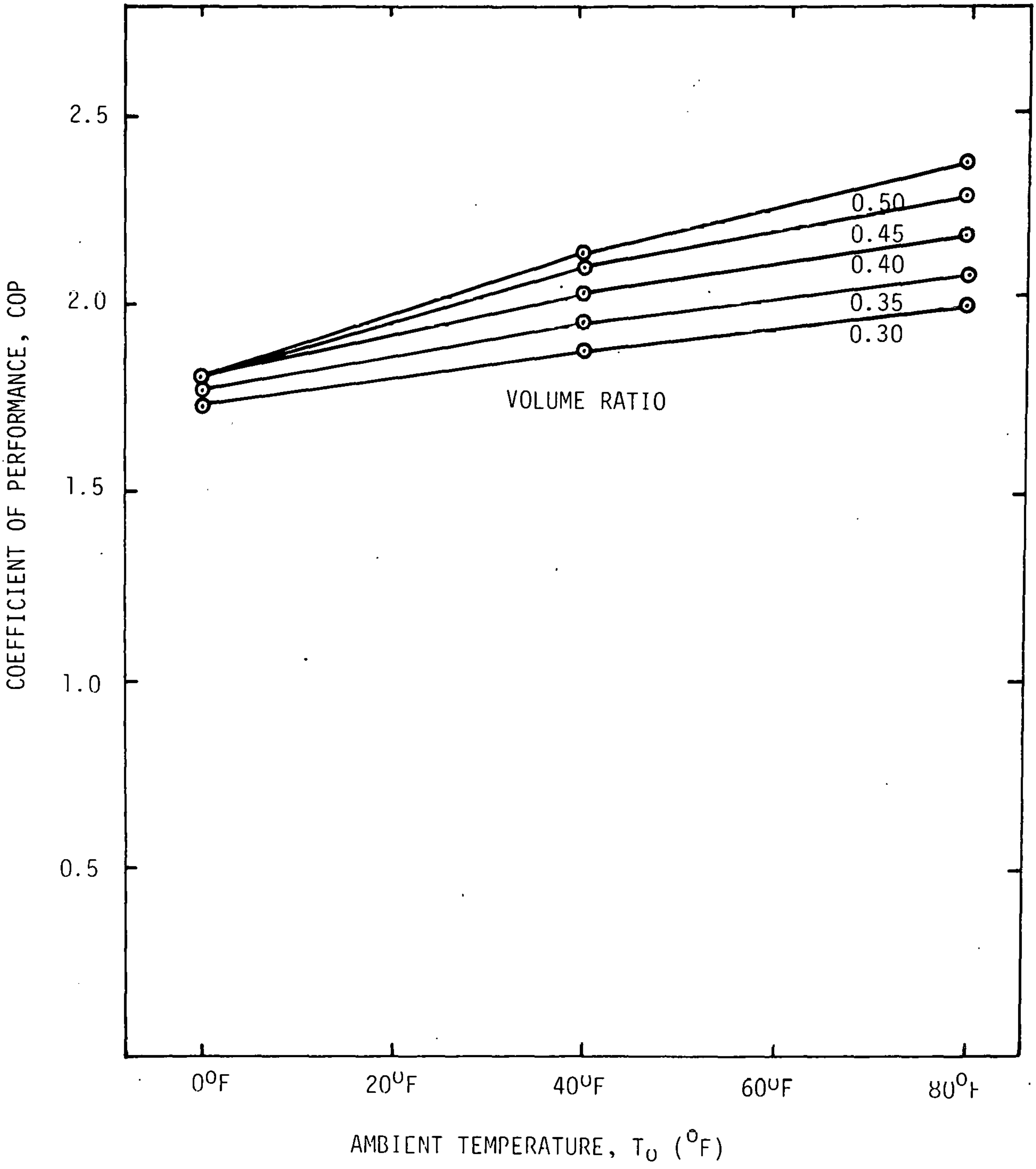


Figure 3-13. Effect of Ambient Temperature on Heating COP

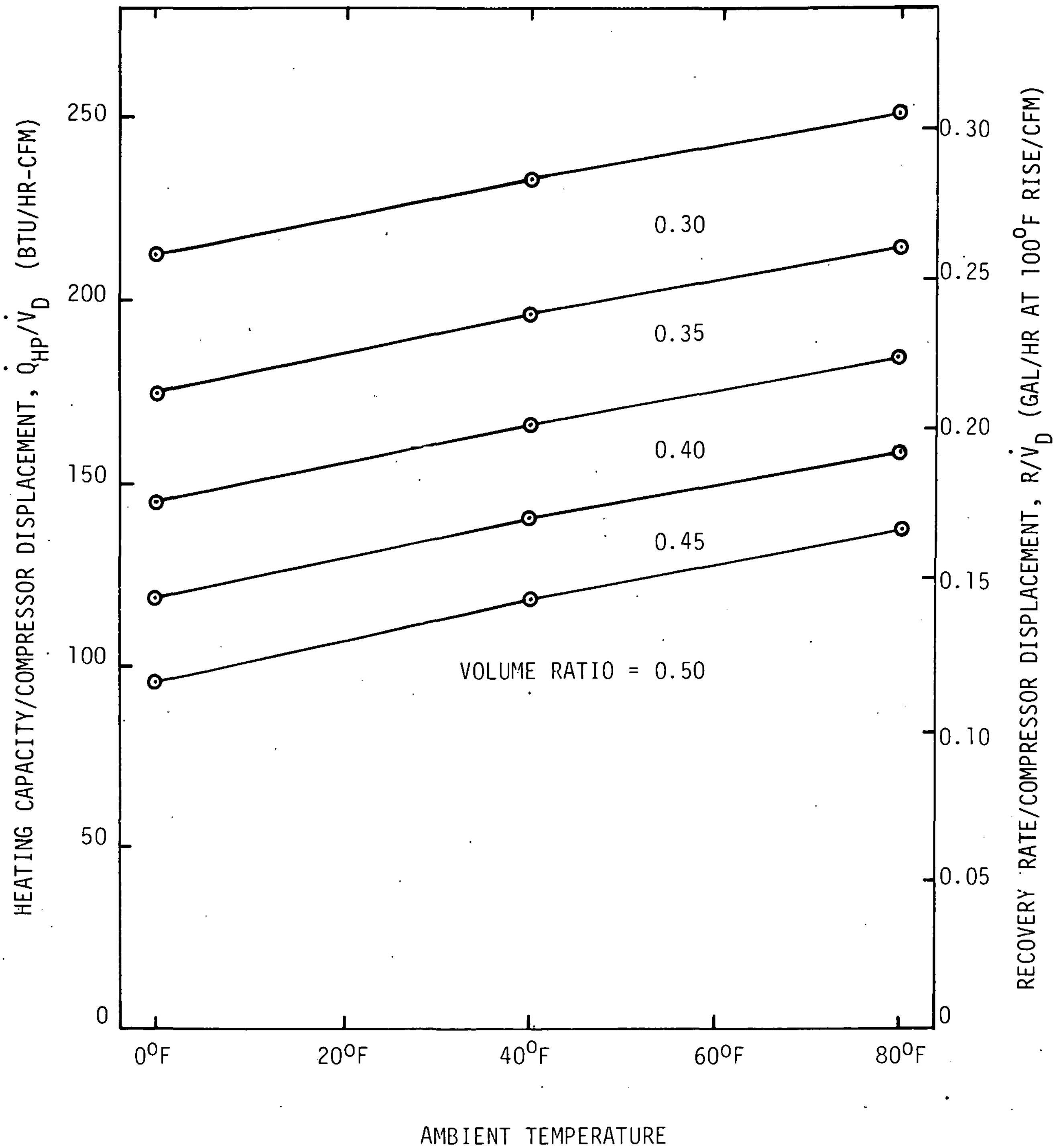


Figure 3-14. Effect of Ambient Temperature on Heating Capacity per Compressor Displacement

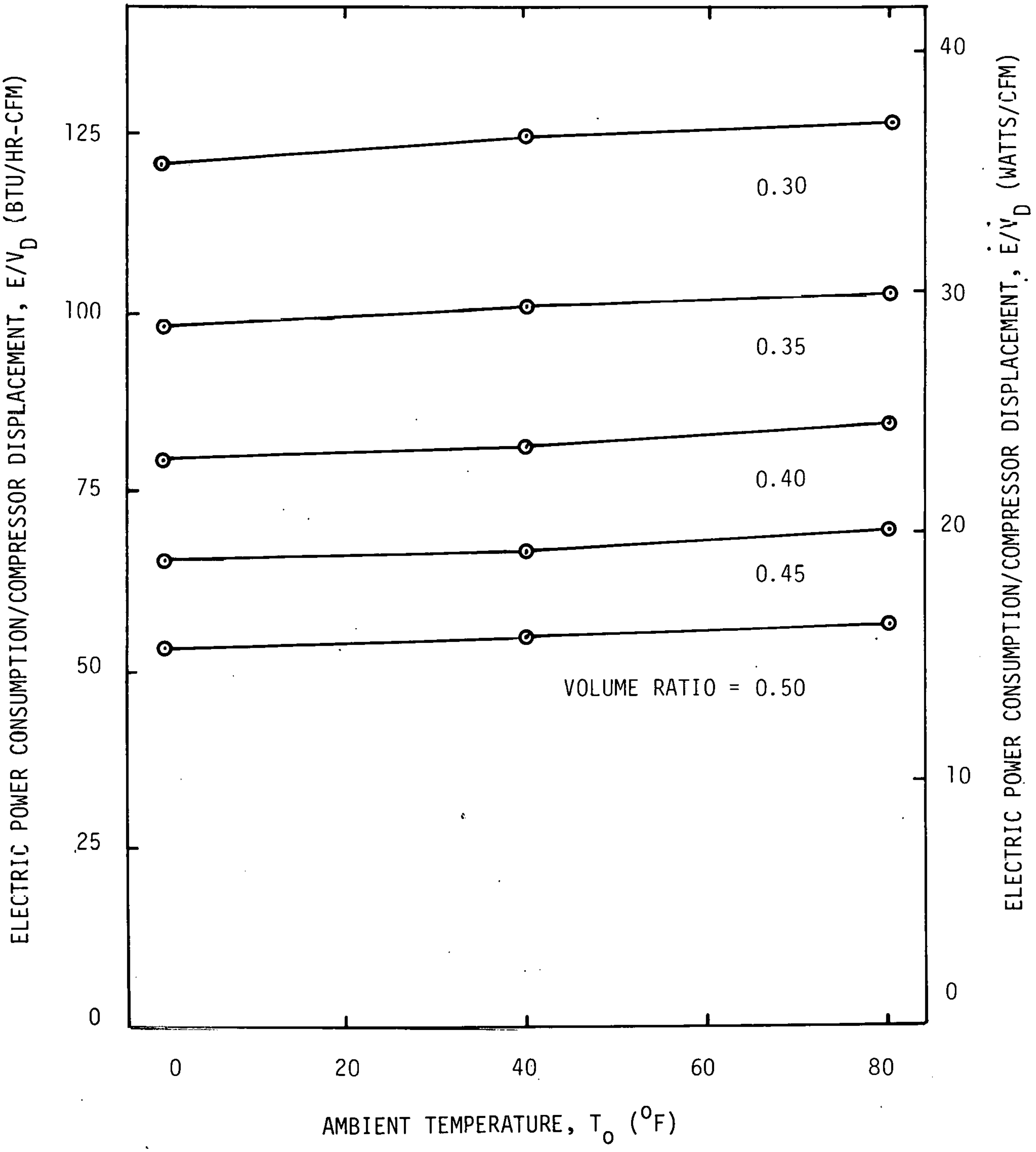


Figure 3-15. Effect of Ambient Temperature on Electric Power Consumption

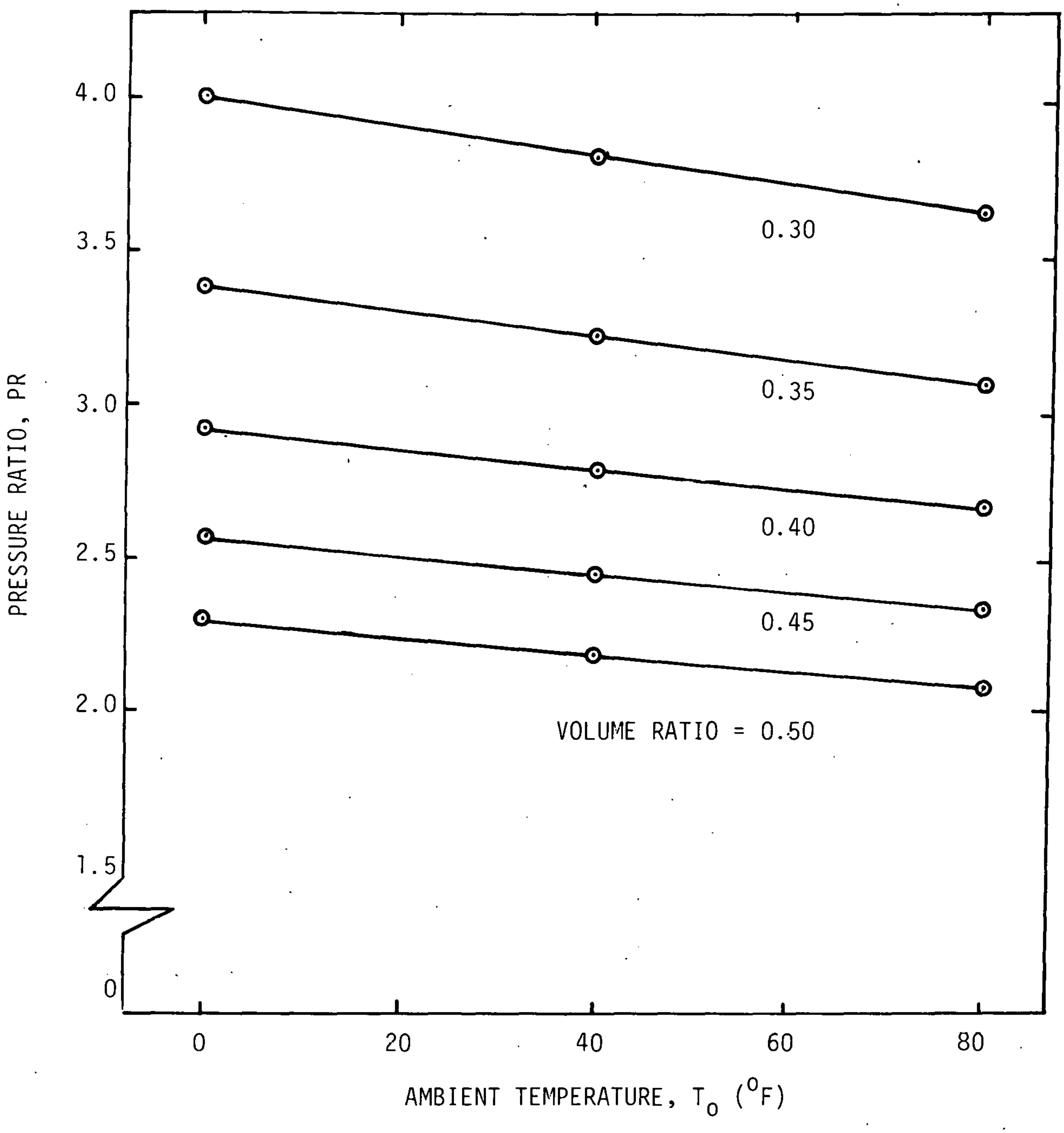


Figure 3-16. Variation of Pressure Ratio with Ambient Temperature With Fixed Compressor/Expander Displacements

3.2.2.2 Ambient Relative Humidity

Changes in ambient relative humidity have only a small effect on the average annual performance of the water heater. The COP and heat pumping capacity are increased substantially by very high relative humidities with ambient temperatures above 75°F. Under these conditions, enough moisture is present in the air for substantial condensation heat transfer to occur in the air-water heat exchanger. Data showing the effect of variations in relative humidity on heating COP is included in Tables 3-2, 3-3 and 3-4 under the "no water injection, no recuperator" column.

3.2.2.3 Cold Water Supply Temperature

As the cold water supply temperature increases, the inlet temperature to the air-water heat exchanger is also increased. The amount of heat that is transferred from the compressed air to the water is consequently reduced. This reduces both the COP and the heating capacity of a fixed volumetric machine. Typical data illustrating the effect of variation of the water supply temperature on the COP and the heating capacity is plotted in Figures 3-17 and 3-18, respectively. The volume flow rate of water heated to a fixed temperature of 140°F (recovery rate) increases as the water supply temperature increases. This is illustrated in Figure 3-19.

3.2.2.4 Making Up Standby Losses

When the water heater is operating to make up standby losses, the COP and heat pumping capacity are reduced. These reductions occur because the high water inlet temperature (120°F versus 40°F to 60°F) to the air-water heat exchanger reduces the amount of heat that can be recovered from the hot compressed air. Figure 3-20 present plots of COP versus compressor/expander

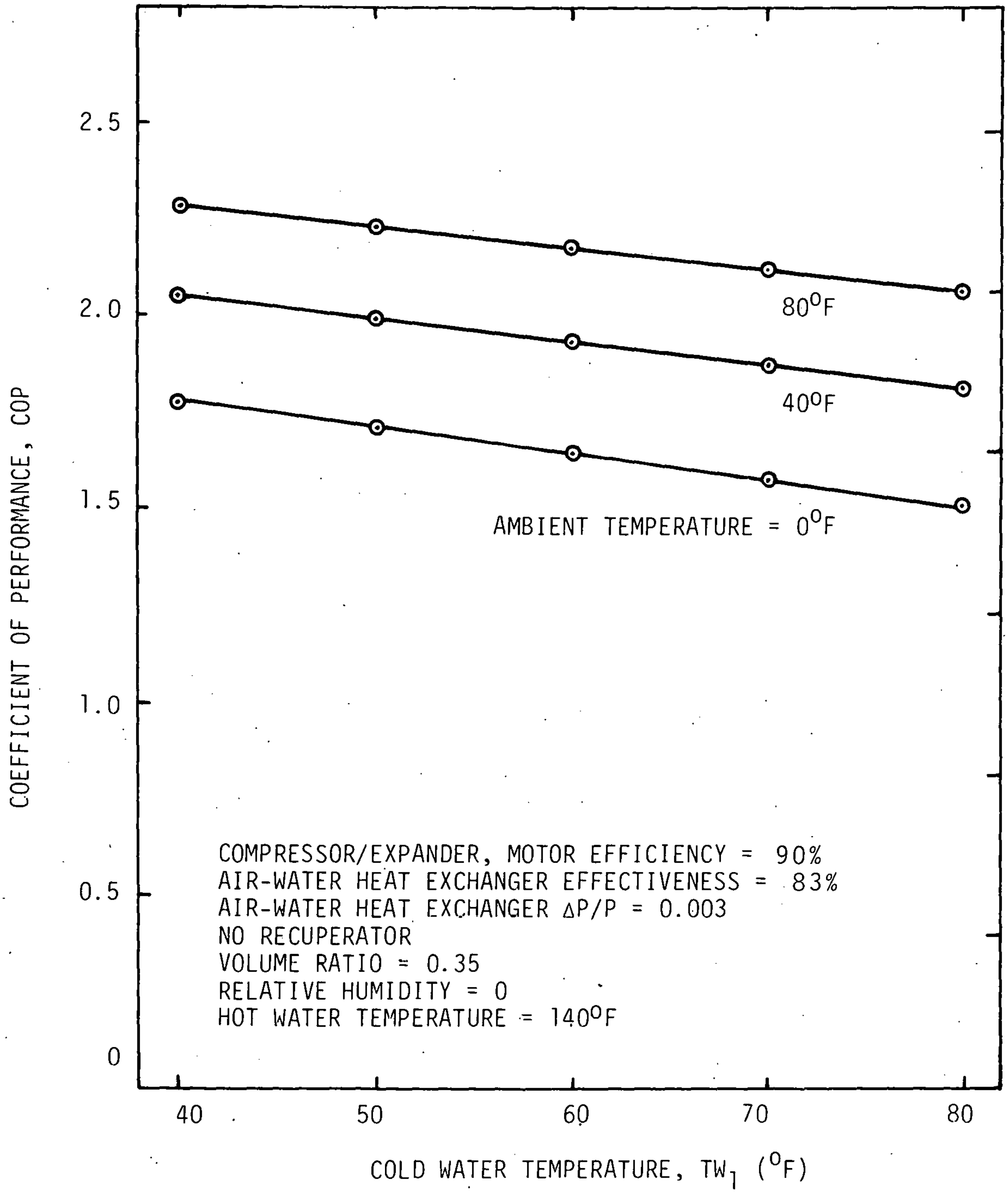


Figure 3-17. Effect of Cold Water Supply Temperature on Heating Coefficient of Performance

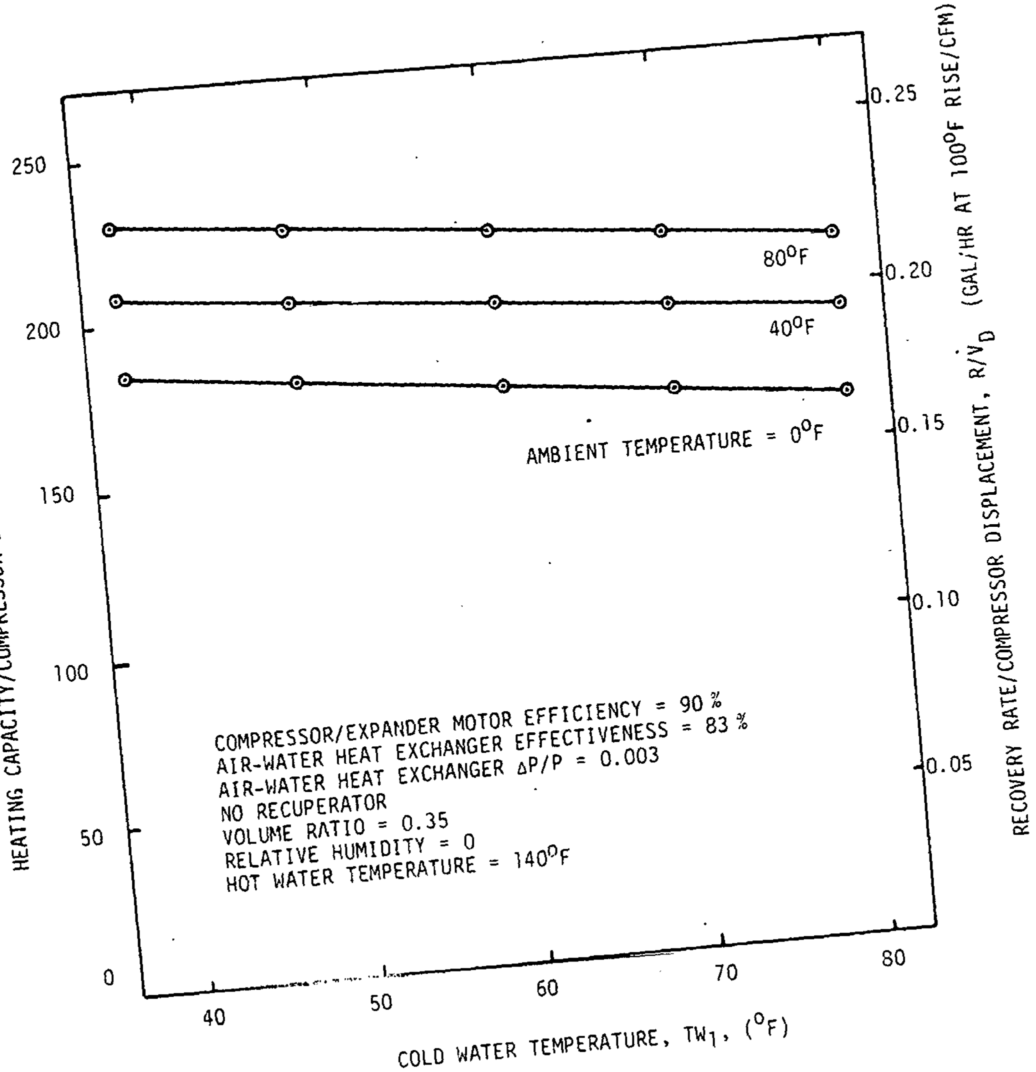


Figure 3-18. Effect of Cold Water Supply Temperature on Heating Capacity

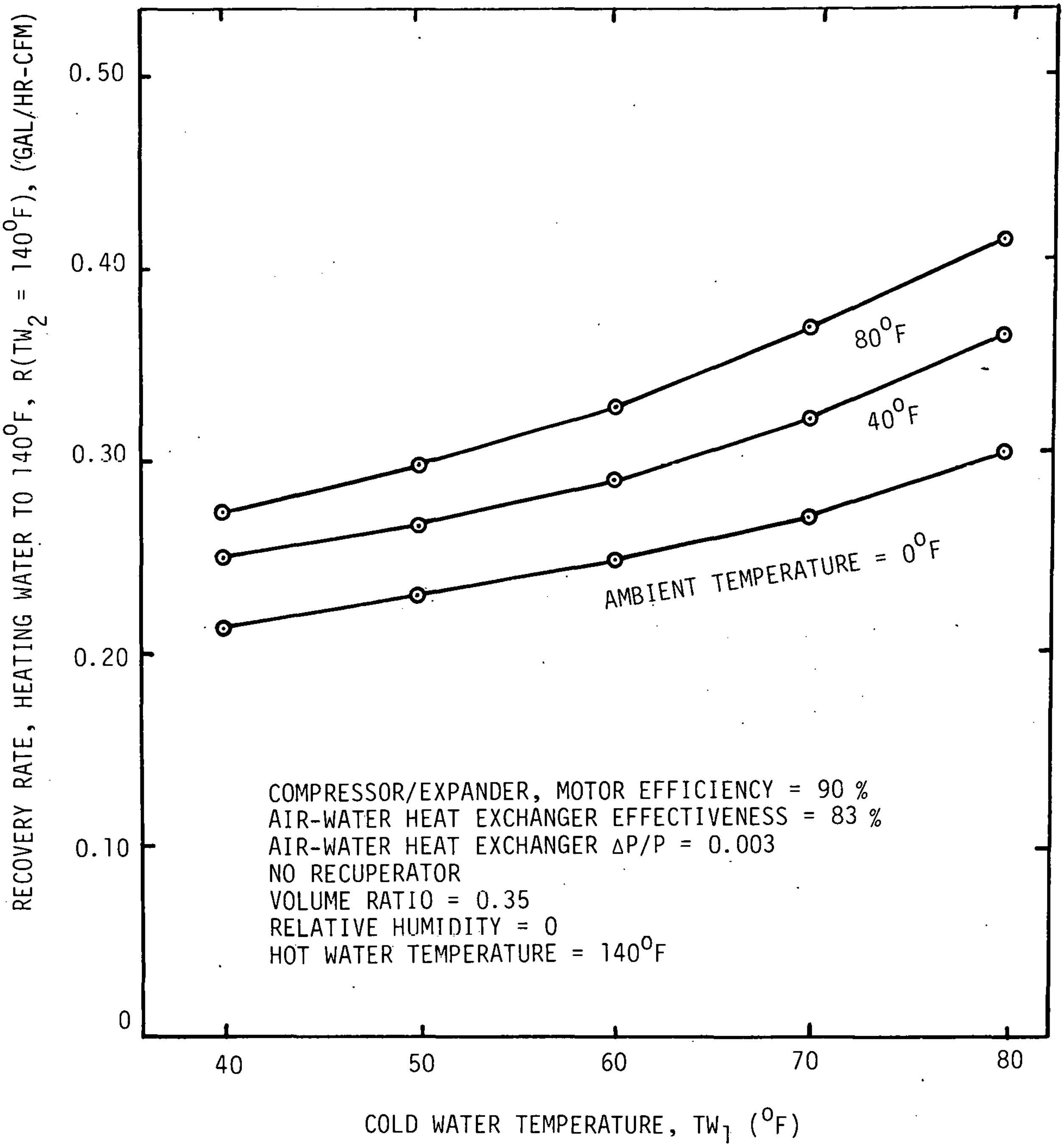


Figure 3-19. Effect of Cold Water Supply Temperature on Recovery Rate When Heating Water to 140°F

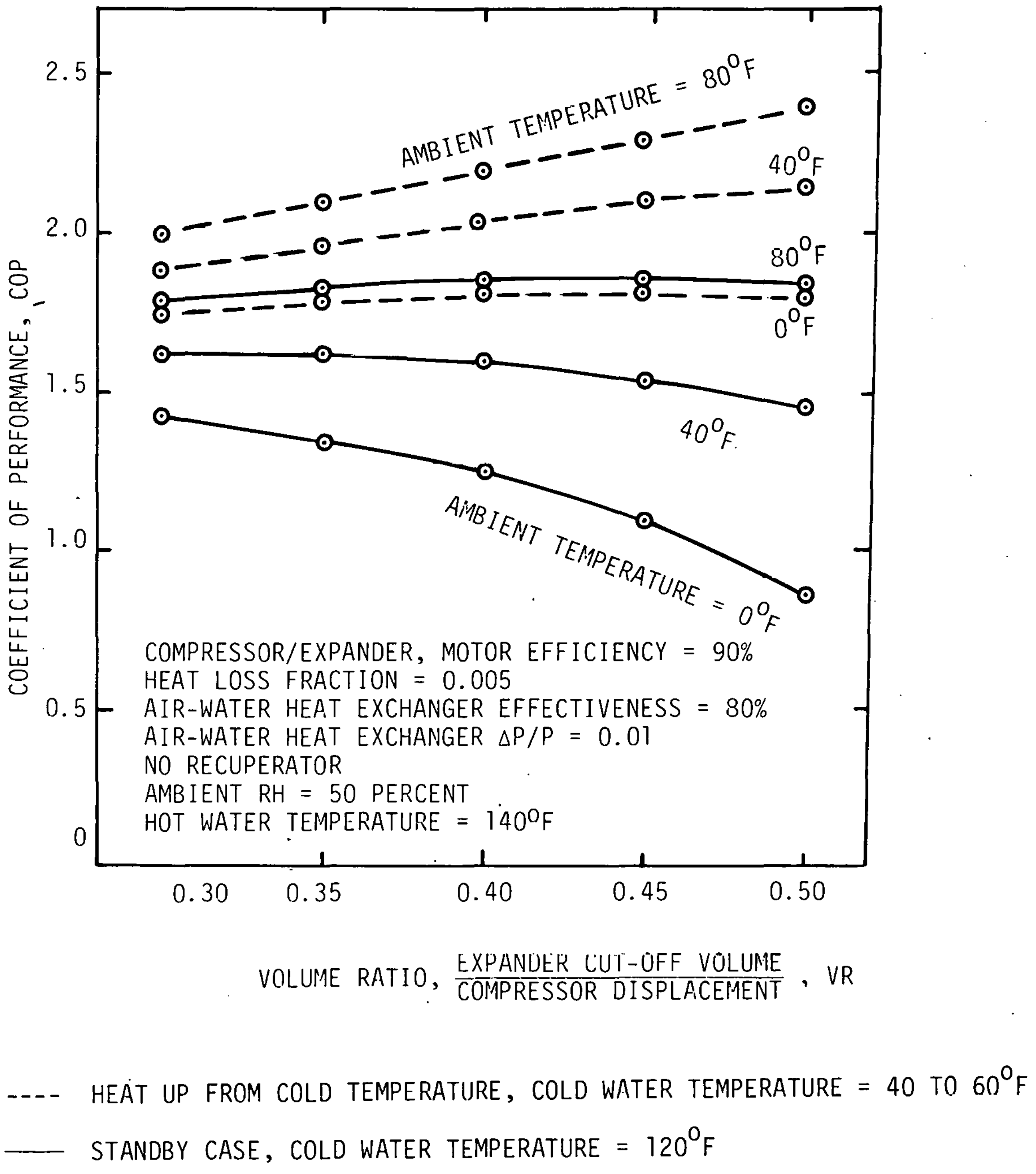


Figure 3-20. Comparison of the Heating COP During Heat Up From the Cold Temperature with the COP During Heating to Make Up Standby Losses

volume ratio for various ambient temperatures comparing heating cold water with heating to make up standby losses. Substantial losses of COP result, particularly at high volume ratio (low nominal pressure ratio). Although make up of standby losses typically account for less than 10 percent of the total heat input, this consideration favors designing the system to operate at a higher pressure ratio.

3.2.2.5 Hot Water Temperature Set Point

Changes in the hot water temperature set point result in changes in the heat transfer effectiveness of the air-water heat exchanger. An increase in the hot water temperature increases the average water temperature in the air-water heat exchanger, decreasing the effectiveness of the air-water heat exchanger. This effect is plotted for a typical case in Figure 3-21. The resulting effect of changing hot water temperature set point on the COP and heating capacity is also shown for a design effectiveness of 83 percent with the hot water temperature set point at 140°F. As shown by the plots, the effect on COP and heating capacity over the normal range of desired hot water temperature is not particularly significant.

3.2.3 Conclusions

Based on the above discussion of the results, the following overall conclusions can be drawn.

- Water sprays do not offer any improvement in performance and should not be included in the system

- A recuperator provides a marginal increase in COP, but not enough to justify the cost of including it in the system

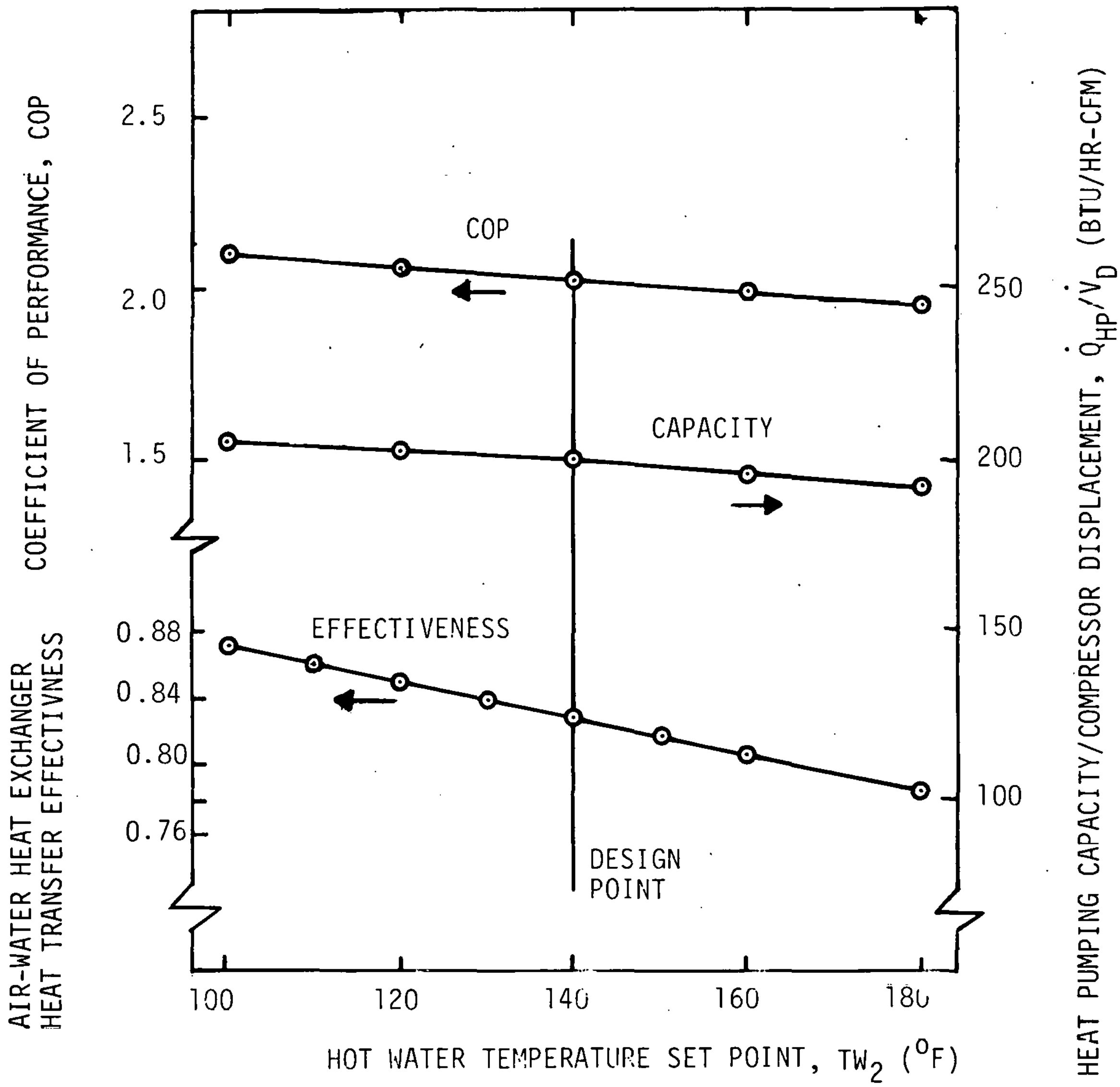


Figure 3-21. Effect of Hot Water Temperature Set Point on Air-Water Heat Exchanger Heat Transfer Effectiveness and Heating COP and Capacity

- The heat transfer effectiveness of the air-water heat exchanger has a substantial effect on the COP and heating capacity. It appears cost-effective to design for an effectiveness in excess of 83 percent

- The air-side pressure loss through the air-water heat exchanger does not substantially affect the COP or heating capacity. Modest pressure losses (up to 1/2 psi) can be tolerated to minimize the size of the heat exchanger

- The isentropic efficiency of the compressor/expander has the largest effect on the COP. The prototype compressor/expander is, therefore, being designed for an isentropic efficiency in excess of 90 percent in order to assure a system COP of at least 1.7

- The pressure ratio of the cycle has strong and opposite effects on the COP and heating capacity. Increased pressure ratio mildly decreases the COP and strongly increases the heating capacity per compressor air throughflow

- The Brayton cycle shows a high degree of insensitivity to variations in ambient temperature and humidity and to changes in cold water temperature and hot temperature set point. COP and capacity vary within 15 percent over a 0° to $100^{\circ}F$ ambient temperature range. Electric power consumption varies by only plus or minus 3 percent over this range, so the motor will always operate near its design point and thus its maximum efficiency.

The specifications for the prototype system, presented in 3.3, are based on these conclusions.

3.3 Configuration and Performance Characteristics of the Final System

A flow diagram of the prototype Brayton cycle water heater is presented in Figure 3-22. The system is identical to that of Figure 3-1, except for the removal of the recuperator and water sprays. The components have been designed to meet the performance specifications or goals listed in Table 3-6.

The expected prototype system performance at the design conditions (ambient temperature of $47^{\circ}F$, cold water temperature of $50^{\circ}F$, hot water temperature set point of $140^{\circ}F$) is summarized in Table 3-7. Off design performance is plotted in Figures 3-23 through 3-27. In Figure 3-23 the plot of the COP and heat pumping capacity versus ambient temperature for heating water from the cold water temperature is presented. In Figure 3-24 a plot of the COP and heat pumping capacity versus ambient temperature for heating to make up standby losses is presented. In Figure 3-25 a plot of the variation of actual pressure ratio with ambient temperature is presented. In Figure 3-26 a plot of the variation of electric power consumption with ambient temperature is presented. In Figure 3-27 a plot of the COP and heat pumping capacity versus hot water temperature set point, is presented.

Table 3-6. Individual Component Performance Specifications/ Goals for the Prototype Brayton Cycle Water Heater

Component	Performance Measure	Specification/Goal
Compressor	Inlet Air Flow	37 CFM
	Isentropic Efficiency	90 Percent
Expander	Isentropic Efficiency	90 Percent
Compressor/Expander	Expander Inlet Cut-Off to Compressor Inlet Volume Ratio	0.33
	Pressure Ratio at Design Conditions	3.0
	Compressor to Expander Heat Loss	$\leq$ 1 Percent of Compression Work
Electric Motor	Efficiency	80 Percent
Air-Water Heat Exchanger	Heat Tranfer Effectiveness	83 - 85 Percent
	Air Side Pressure Loss	$<$4 inch W.C.

Table 3-7. Performance Specification of Prototype Brayton Cycle Water Heater at Design Conditions

Design Conditions	Specificaion
Ambient Temperature	$47^{\circ}F$
Cold Water Temperature	$50^{\circ}F$
Hot Water Temperature Set Point	$140^{\circ}F$
Performance Specifications	**Specification**
COP	1.7
Heating Capacity	2.5 kW
Recovery Time	8 hr
Pressure Ratio	3.0
Electric Power Input	1.37 kW

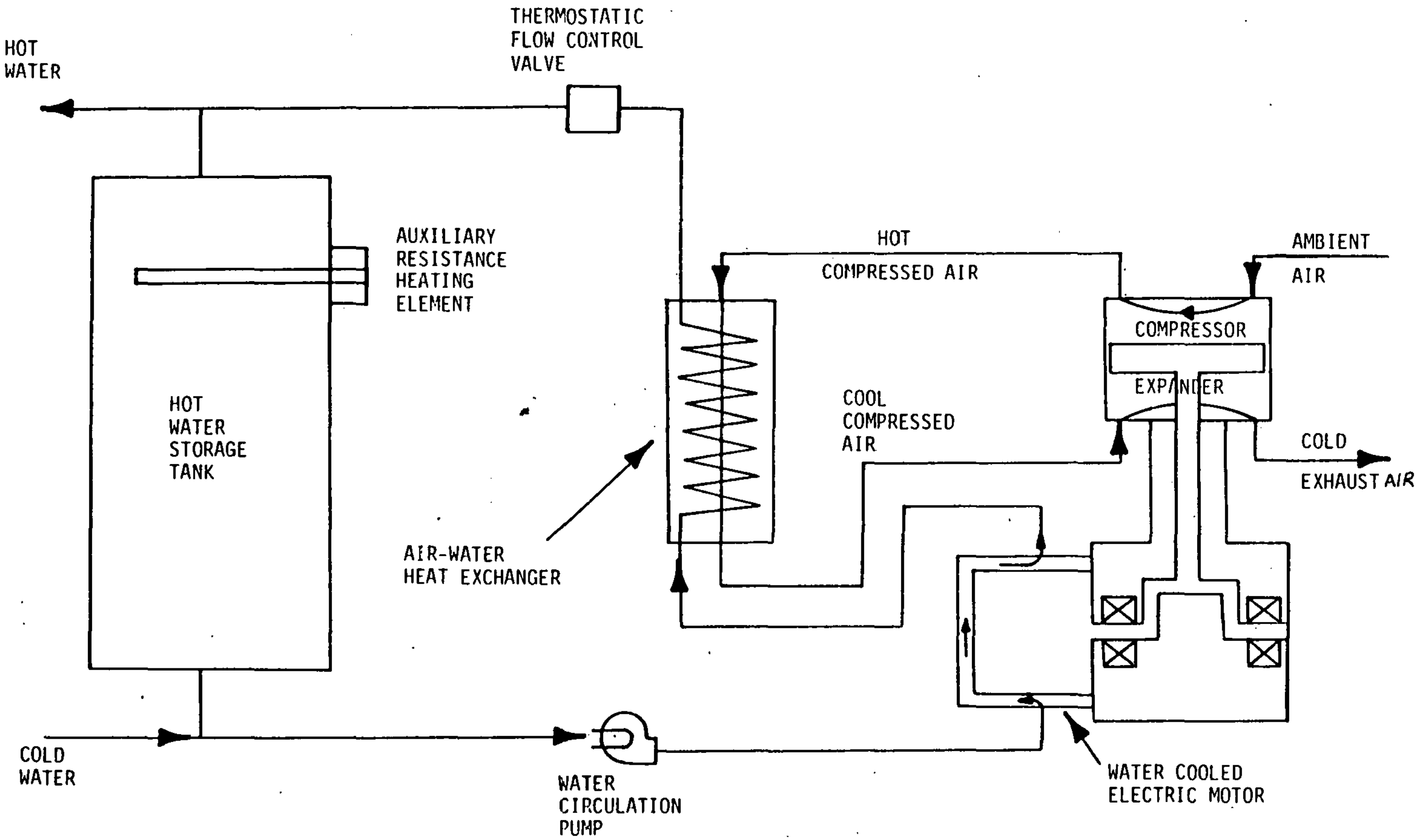


Figure 3-22. Flow Diagram of Prototype Brayton Cycle Water Heater

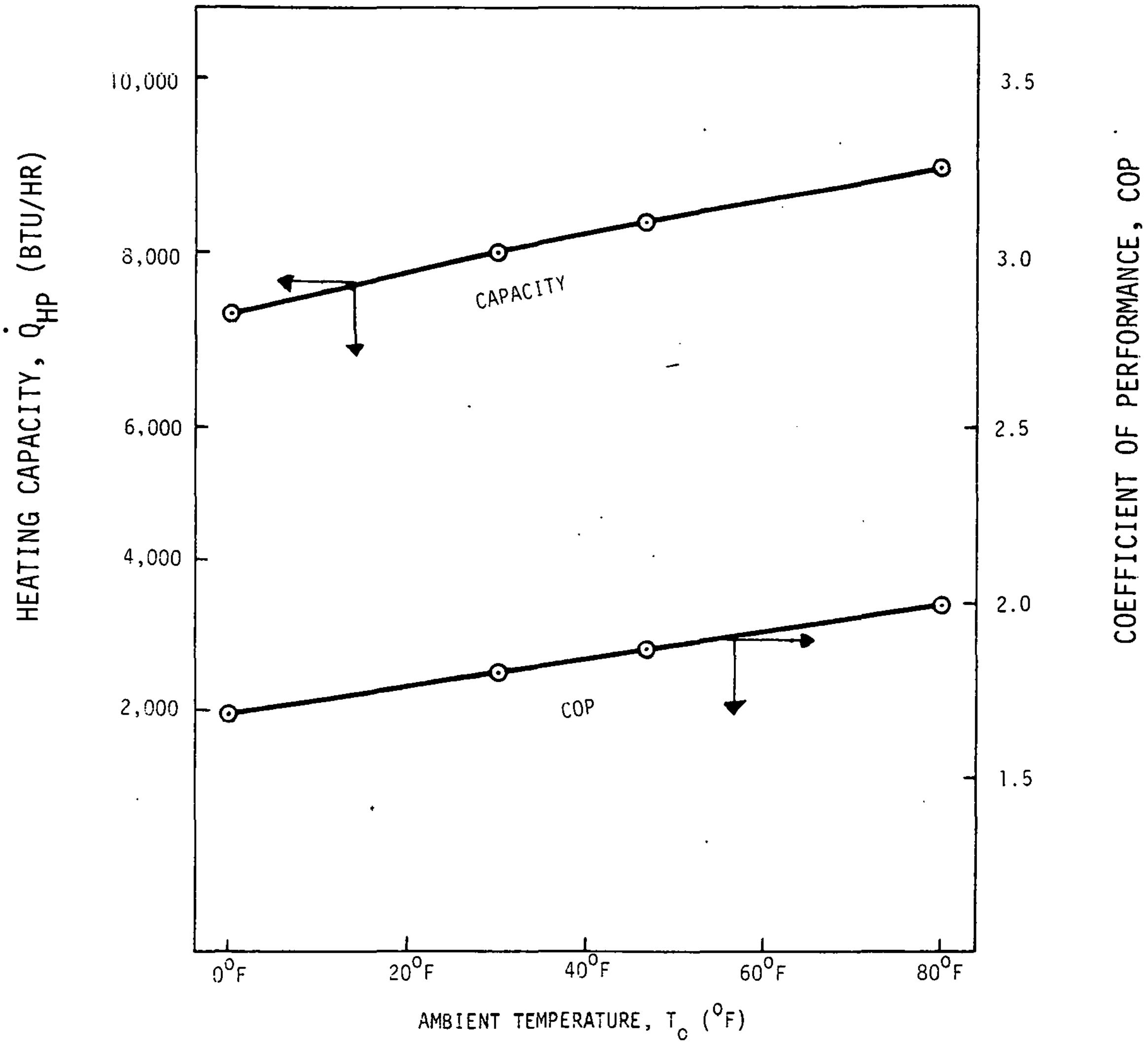


Figure 3-23. Variation of COP and Heating Capacity of Prototype Water Heater with Changes in Ambient Temperature When Making Up Standby Losses

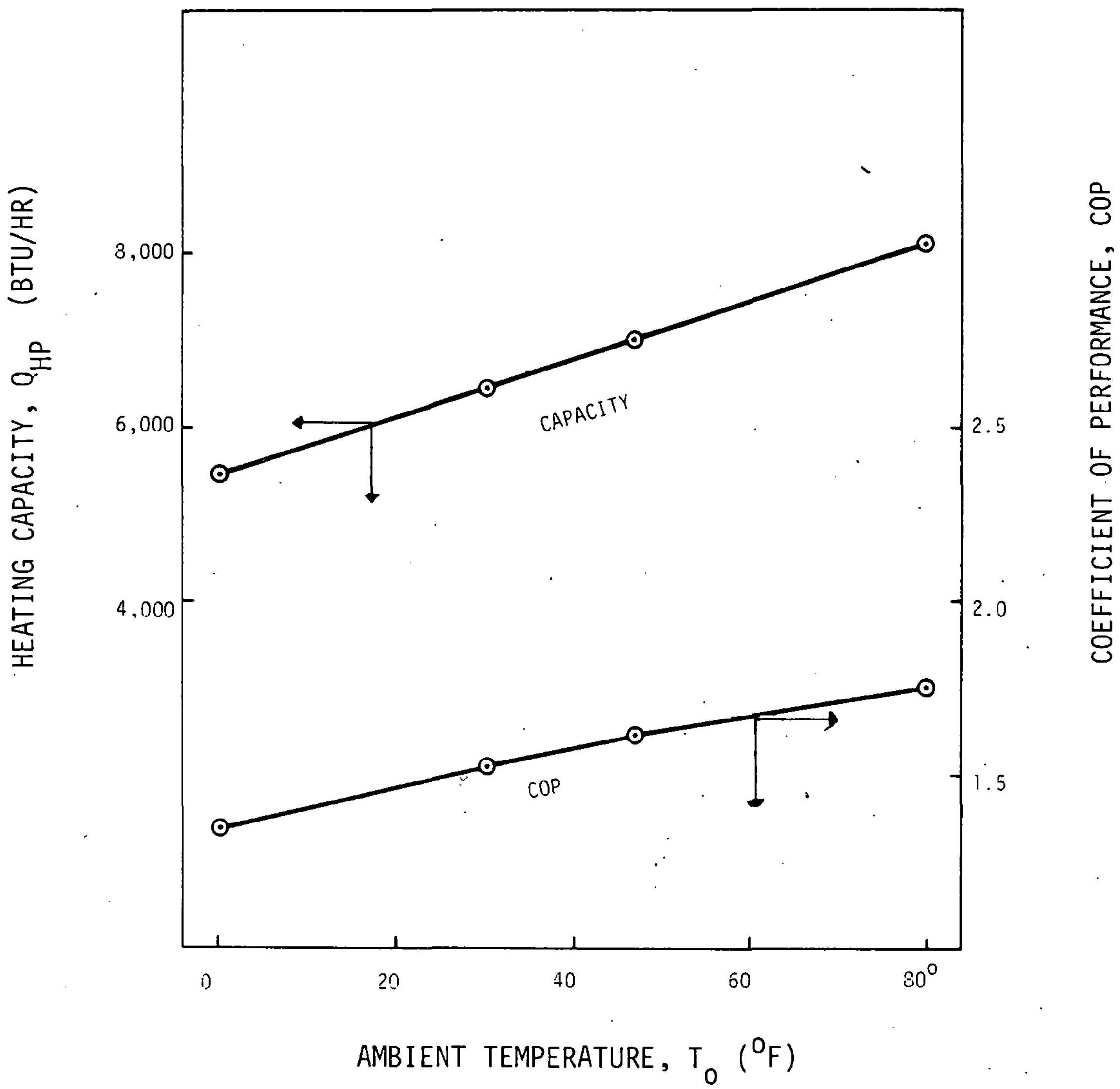


Figure 3-24. Variation of COP and Heating Capacity of Prototyp Water Heater with Changes in Ambient Temperature When Making up Standby Losses

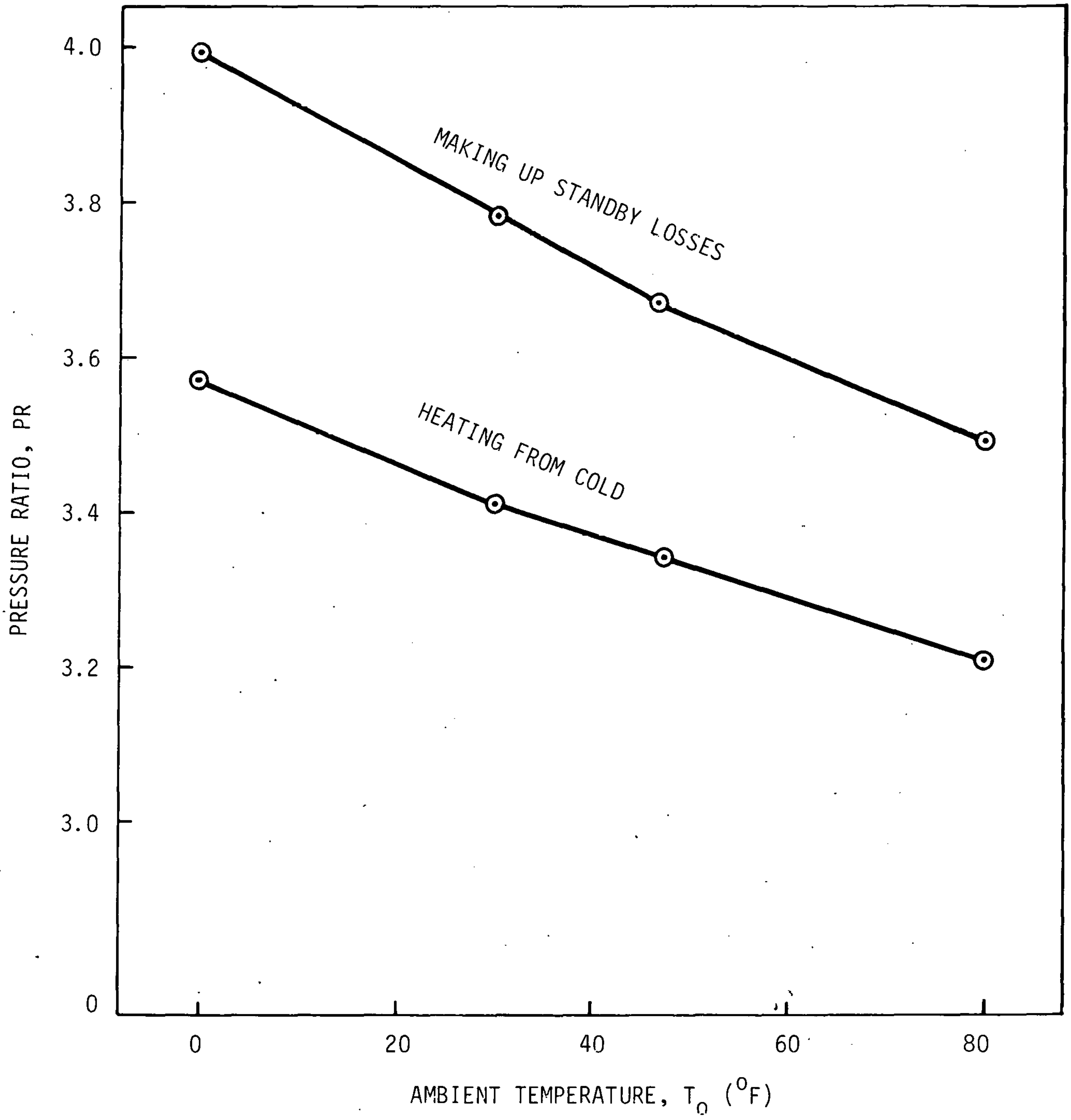


Figure 3-25. Variation of Pressure Ratio in Prototype Water Heater with Changes in Ambient Temperature

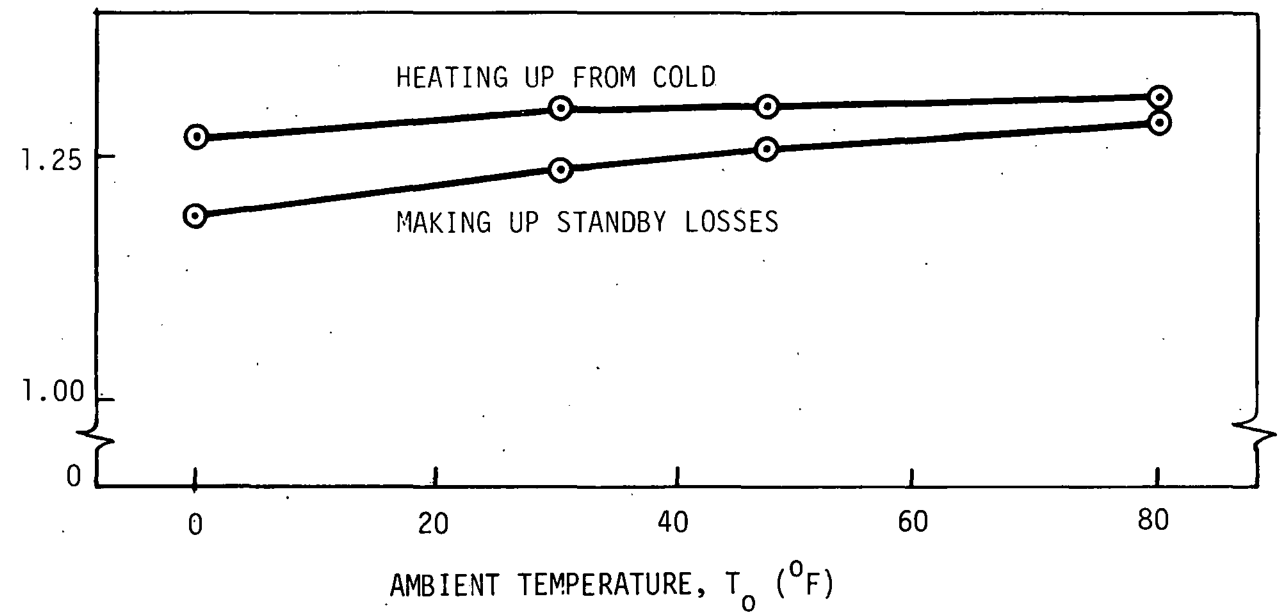


Figure 3-26. Variation of Electric Power Consumption in Prototype Water Heater with Changes in Ambient Temperature

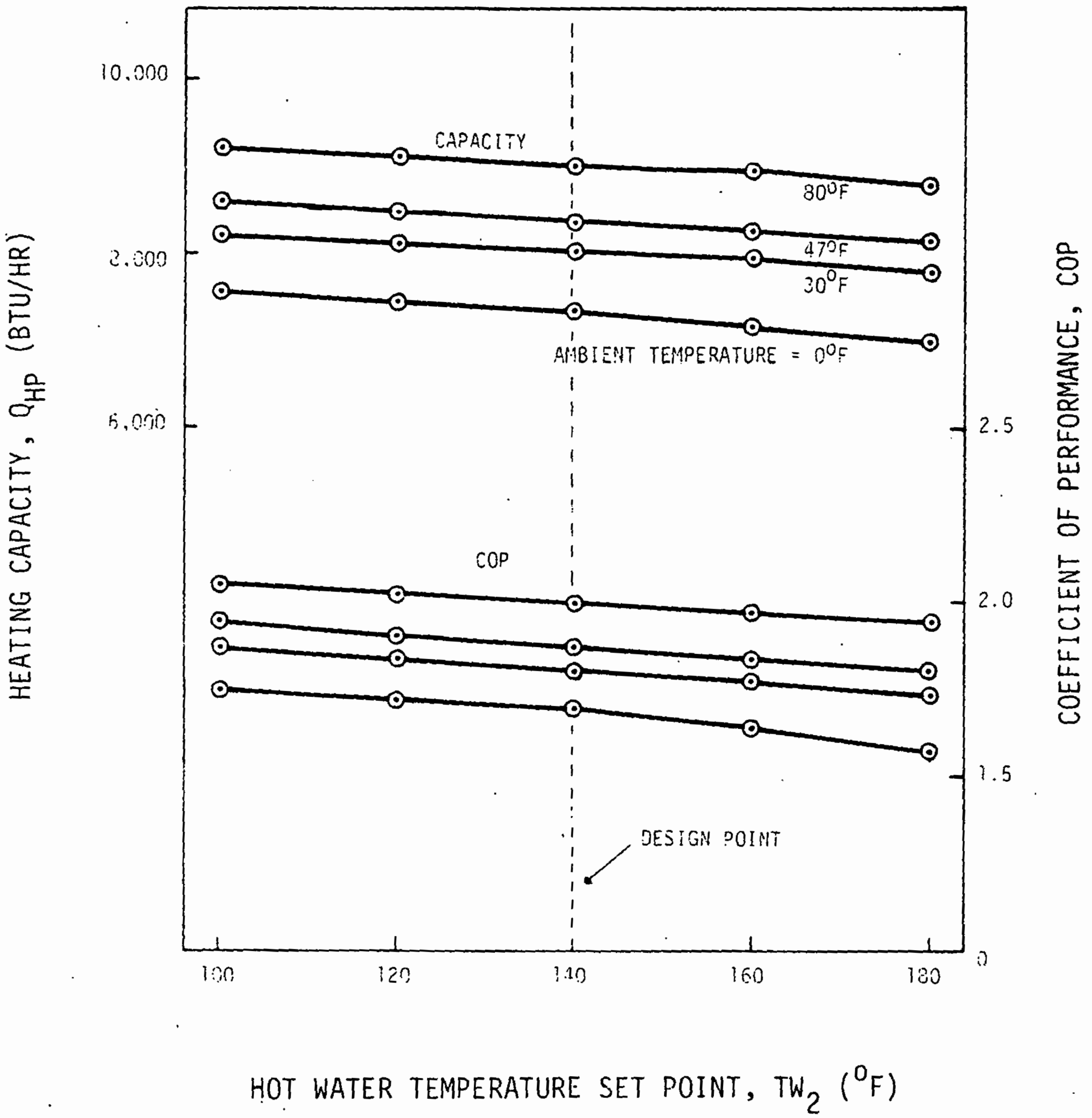


Figure 3-27. Variation of COP and Heating Capacity in Prototype Water Heater with Changes in Hot Water Temperature

4. THERMODYNAMIC ANALYSIS OF THE COMPRESSOR/EXPANDER

4.1 Selection of Compressor/Expander Type

The coupled functions of compression and expansion can be accomplished with turbo-machinery, rotary-volumetric machinery, or reciprocating machinery. Typical equipment of all types for the low-flow requirements of the present application, i.e., 30 to 50 scfm, have comparable isentropic efficiencies on the order of 70 to 85 percent. Within the present range of interest, for pressure ratios between 3.0 and 4.0, the usual design practice on the basis of specific speed requires a very high speed turbo machine, a 3600 rpm rotary machine, or 1200 rpm reciprocating machine (the latter speeds are determined for direct 50 Hz electrical drive) [1].

Turbo machines were eliminated early from the selection process because precision speed increasers or frequency converters appear prohibitively expensive for the current application. In addition, low noise and sufficiently high efficiency appear to be nearly impossible goals.

Rotary compressors and expanders have shown advantages in small size, simplicity, low cost, vibration and noise. Some have shown very high adiabatic efficiencies although the through-flow appears to be more nearly isothermal [2]. However, for the present application, heat is the real objective, not pressure, so high efficiency <u>and</u> adiabatic compression temperature are desired.

The reciprocating type of machine was selected for the first demonstration of the air-cycle water-heating application based upon two important factors; the time and risk associated with he development of this type machine is small, and the probability of achieving very high efficiency is high.

The methods that lead to high efficiency with a practical design are discussed in the following sections.

4.2 Compressor/Expander Analysis

As was shown in Section 3.2.1.1, a high coefficient of performance for the reverse Brayton cycle water heater requires high isentropic efficiencies for the compression and expansion processes. The major categories of losses which lower the overall efficiencies of a compressor or expander may be grouped as mechanical losses due to the bearings, seals, ring-friction, etc., and aerodynamic/thermodynamic losses due to pressure losses in the flow path, heat transfer, valve losses, etc. In order to optimize the aerodynamic performance of the air compression and expansion processes, subject to the constraints of finite valve opening time and limited flow areas for inlet and exhaust valves, the governing equations were developed and programmed for solution on a digital computer.

4.2.1 Derivation of Governing Equations

Because the aerodynamic losses in the valves depend on valve opening, the governing equations were developed in differential form in terms of the independent differential time variable, dt, and integrated numerically.

The basic equations required for the aerodynamic/thermodynamic analysis are obtained from conservation of mass, the first law of thermodynamics, gas property relations and flow-pressure drop relations for the valves. The physical restriction imposed by finite valve opening time results in reversed flow through inlet and exhaust valves during certain short time intervals during the cycle. Thus internal energy and work terms must be evaluated carefully. The pressure drop across the valves may be large in certain portions of cycle and therefore compressible flow equations are used to calculate the flow rates. Because all of processes vary with time in the compressor and expander, the governing equations are developed in differential form in terms of the independent differential time variable, dt, and subsequently integrated numerically. A schematic for the thermodynamic modelling of the compression and expansion is shown in Figure 4-1. The thermodynamic "system" chosen for analysis during the time interval dt is defined as the mass of gas, m, which is in the cylinder at the end of time interval dt.

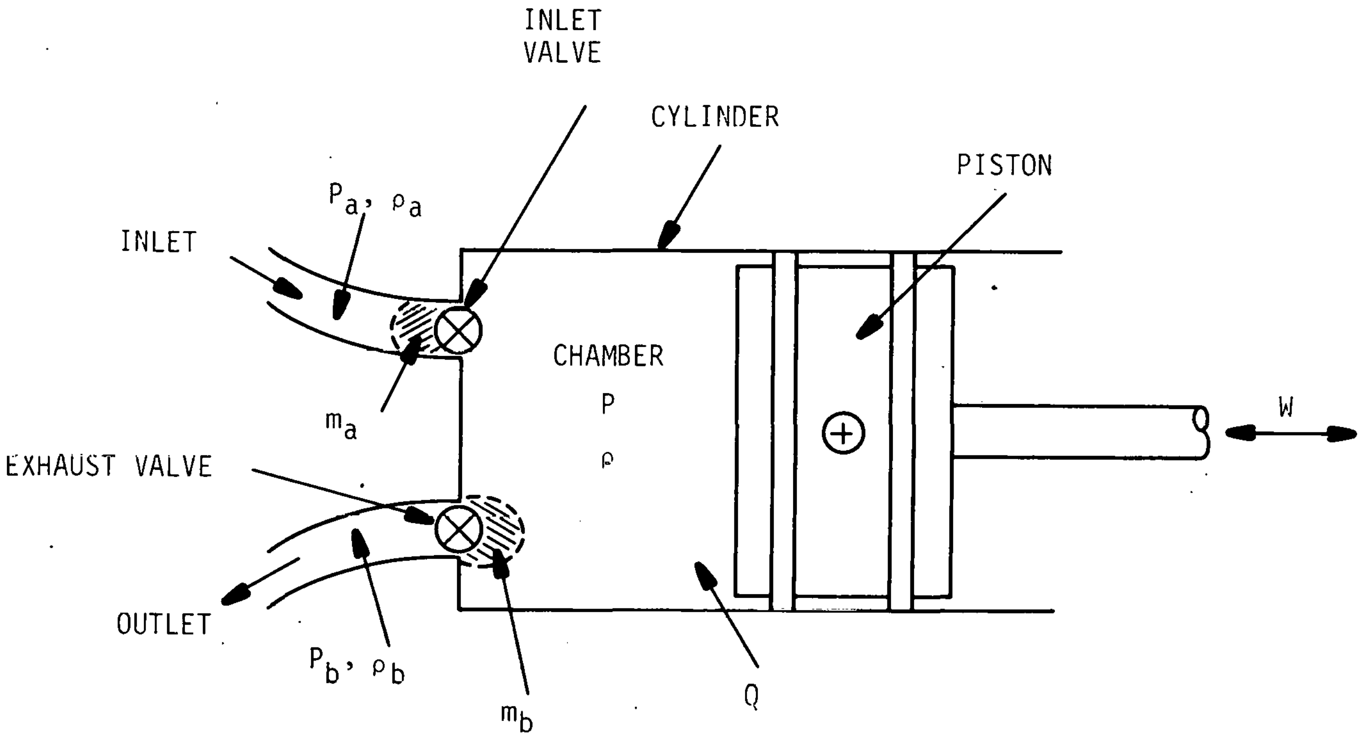


Figure 4-1. Schematic of Thermodynamic System Modeling the Compression/Expansion Processes

Conservation of mass then requires that the mass of gas, m_o, in the cylinder at time t at the beginning of the time interval, dt, is related to the mass m by

$$m = m_o + m_a - m_b \tag{4.1}$$

where

m_a = mass of gas entering the cylinder through valve a in time dt

m_b = mass of gas leaving the cylinder through valve b in time dt

m_o = mass of gas in the cylinder at time t

m = mass of gas in the cylinder at time t + dt

Usually the gas flow through the inlet valve is into the cylinder ($m_a > 0$) and the gas flow through the exhaust valve is out of the cylinder ($m_b > 0$).

The first law of thermodynamics applied to the system during the time dt yields the following:

$$Q - W = E_2 - E_1 \tag{4.2}$$

where

Q = heat added to the system in time dt
W = work done by the system in time dt
E_1 = initial internal energy of the system
E_2 = final internal energy of the system

The total work performed is given by the sum of work done on the piston plus work done on the boundaries of the system where gas masses m_a and m_b are flowing.

$$W = W_p + W_a + W_b \tag{4.3}$$

The work done on the piston W_p is given by

$$W_p = P\,(V_2 - V_1) \tag{4.4}$$

where

P = gas pressure in the cylinder
V_1 = cylinder volume at t
V_2 = cylinder volume at t + dt

The expressions for the flow work, W_a and W_b, depend on the signs of m_a and m_b.

$$W_a = \begin{cases} -P_a V_a = -P_a m_a/\rho_a & (m_a > 0) \\ -P m_a/\rho_b & (m_a < 0) \end{cases} \tag{4.5}$$

$$W_b = \begin{cases} P\ m_b/\rho & (m_b > 0) \\ P_b\ m_b/\rho_b & (m_b < 0) \end{cases} \tag{4.6}$$

W_a = work done by system at boundary a
W_b = work done by system at boudary b
P_a = gas pressure outside of valve a
ρ_a = gas pressure outside of valve a
ρ_b = gas density outside of valve b
ρ = gas density in the cylinder

It is assumed that the contents of the sytem are completely "mixed" at the end of each time interval dt. Then the internal energies are given by

$$E_2 = m\, u_2 \tag{4.7}$$

$$E_1 = m\, u_1 + E_a - E_b \tag{4.8}$$

$$E_a = \begin{cases} m_a\ (u_a - u_1) & (m_a > 0) \\ 0 & (m_a < 0) \end{cases} \tag{4.9}$$

$$E_b = \begin{cases} 0 & (m_b > 0) \\ m_b\ (u_b - u_1) & (m_b < 0) \end{cases} \tag{4.10}$$

where

u_a = specific internal energy of gas outside of valve a (inlet)

u_b = specific internal energy of gas outside of valve b (exhaust)

u_1 = specific internal energy of gas in the cylinder at time t

u_2 = specific internal energy of the gas in the cylinder at time t + dt

ρ = gas density

Using perfect gas relationships provides

$$P = \rho\, R\, T \tag{4.11}$$

$$u = c_v T \tag{4.12}$$

Isentropic flow of gas through a valve yields the expression for flow rate w

$$w = (\rho)((V_e l)A = A\sqrt{\frac{k}{R}}\ \frac{P_u}{T_u}\ \frac{M}{\left[1 + \frac{k-1}{2} M^2\right]^{\frac{k+1}{2(k-1)}}} \qquad (4.13)$$

and

$$\frac{P_u}{P_d} = \left[1 + \frac{k-1}{2} \quad M^2\right]^{\frac{k}{k-1}} \qquad (4.14)$$

where

c_v = specific heat at constant volume
R = gas constant
w = mass flow rate
k = ratio of specific heats
M = Mach number
P_u = upstream pressure
P_d = downstream pressure
A = valve discharge area
Vel = velocity

If the values of P_u and P_d result in $M > 1$ in Equation (4.14), then M is set equal to 1 in Equation (4.13) because the flow is choked at area A.

Then gas masses m_a and m_b are obtained from

$$m_a, m_b = w\ dt. \qquad (4.15)$$

The rate of heat transfer, q, to the gas from the piston and cylinder walls is given by

$$q = h\,A_w\,(T_w - T) \tag{4.16}$$

The heat transfer coefficient h is obtained from the Stanton number correlation [6].

$$St = \frac{h}{\rho V_s\, c_p} = \text{constant} \tag{4.17}$$

The heat transfer Q in the time internal is given by q dt. The terms in equations (4.16) and (4.17) are defined as follows:

h = heat transfer coefficient

ρ = cylinder gas density

c_p = specific heat at constant pressure

V_s = reference velocity (taken as maximum piston velocity x piston area/maximum valve area)

St = Stanton Number

A_w = wall area for heat transfer

T_w = wall temperature.

The Stanton number physically characterizes the ratio of heat transferred to the fluid to the thermal capacity of the fluid. The value of the Stanton number is specified as one of the parameters required for a numerical calculation.

In addition to the gas dynamic-thermodynamic equations modelling the gas flow through the valves, it is necessary to determine the valve opening versus time equation. There are two general methods of valve operation; mechanically driven valves such as cam driven valves or rotary valves, and pressure activated valves such as reed valves or diaphragm valves. For the mechanically driven valve, the opening is determined by the crank angle, whereas for the pressure activated valve, the valve opening is dependent upon the time-integrated differential pressure force, spring force, and the mass of the valve. Either pressure driven valves or crank driven valves may be used for the compressor; however, only crank driven valves may be used for the expander because the opening and closing of the expander valves are in the opposite direction of the resultant pressure forces.

4.2.2 Calculational Procedure

The above equations are used to determine the properties of interest at the end of the time interval dt based on the known values of all variables at time t. The steps in the calculation are as follows:

1. Given the upstream and downstream pressure across each valve and the valve area from the arbitrarily specified valve timing, calculate the mass flow rate through each valve and the values of m_a and m_b. [Equations (4.13), (4.14), and (4.15).]

2. Calculate the mass m and the gas temperature T from the continuity and energy equations (using work and heat transfer expressions). [Equations (4.1), (4.2), (4.3), (4.4), (4.5), (4.6), (4.7), (4.8), (4.9), (4.10), (4.12) and (4.16)].

3. Knowing the change in cylinder volume (from the stroke and engine speed), calculate the density from $\rho = m/Vol$.

4. Knowing the temperature T and density ρ, the pressure P is calculated from the ideal gas law, $p = \rho R T$.

All properties of interest are now known at the end of the time interval dt and the changes occurring in the next interval can be calculated.

In order to obtain higher accuracy in the numerical stepwise procedure described above, an interactive procedure is used. After the properties at the end of the time step have been calculated, the procedure is repeated using the average values of appropriate variables during the time interval. Thus, average values are used for w, q and P in Equations (4.4), (4.5), (4.6), (4.15) and (4.17).

The governing equations have been programmed for solution on a digital computer following the procedure described above. The specified parameters for each calculation are:

- inlet pressure
- exhaust pressure
- cylinder bore
- piston stroke
- clearance volume
- inlet temperature
- wall temperature
- Stanton Number
- compressor/expander speed
- maximum flow area through valves
- valve timing as a function of a crank angle.

4.2.3 Calculated results

Performance calculations for compression of air have been carried out. Table 4-1 lists examples of the calculated performance for compression of air as a function of the major compressor design variables. The parameters held constant in the table are the compressor inlet pressure and temperature, the compressor discharge pressure, the cylinder wall temperature and the clearance volume.

The isentropic efficiency for compression, η_c, shown in Table 4-1 is defined as:

$$\eta_c = \frac{W_{rev_c}}{W_c} \tag{4.18}$$

where W_c is the actual cycle compression work and W_{rev_c} is the work required for the reversible isentropic compression of air. The W_{rev_c} is defined as follows:

$$W_{rev_c} = m\, c_p\, T_{in} \left[1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{k-1}{k}}\right] \tag{4.19}$$

where m is the mass of compressed gas delivered per cycle, T_{in} is the compressor inlet temperature, c_p is the specific heat of air at constant pressure, P_{out} is the compressor discharge pressure, P_{in} is the compressor inlet pressure, and k is the ratio of the specific heats, c_p/c_v.

The isentropic efficiency of the compression process is strongly influenced by the engine speed, stroke, and the valve area to piston area ratio as is shown in the tabulated results. Specific results based on the computer simulation of the compressor are: as the engine speed increases, the isentropic efficiency decreases; as the stroke increases, the isentropic efficiency decreases; and as the valve area to piston area ratio increases the isentropic efficiency increases. Heat transfer from the gas to the cylinder and piston as characterized by the Stanton number changes the isentropic efficiency slightly, usually a reduction of a few percent or less. Other calculations in which the linear relationships for valve opening and closing are replaced with a sinusoidal or cycloidal variation changed

Table 4-1. Examples of Compressor Performance

P_{in} = 15 psia P_{out} = 45 psia T_{in} = 70°F T_{wall} = 170°F   Clearance Volume = 5 percent of the swept volume

Bore (in)	Stroke (in)	Compressor Speed (rpm)	Stanton Number	Valve Area / Piston Area	Valve Timing	Compressor Isentropic Efficiency (%)	Lbm Air Per Rev
4	3	1000	0	0.1	Linear, 30° of crank angle for opening/ closing	96.8	0.56×10^{-2}
4	3	4000	0	0.1	"	80.9	0.146×10^{-2}
4	3	3600	0	0.15	"	91.2	0.155×10^{-2}
6	3	1800	0	0.15	"	96.2	0.351×10^{-2}
6	3	1800	0.003	0.15	"	95.5	0.349×10^{-2}
6	3	1800	0	0.0575	"	86.9	0.343×10^{-2}
6	1.4	1800	0	0.0575	"	93.6	0.160×10^{-2}
6	2	1150	0.003	0.0424	Reed Valves	92.7	0.226×10^{-2}
6	2	1150	0	0.0424	Reed Valves	94.9	0.230×10^{-2}
Independent Variables						Dependent Variables	

the calculations very little. Changes in cylinder bore changes the air displaced per revolution by the square of the diameter ratio as expected.

Figure 4-2 shows, qualitatively, the P-V diagram for "ideal" compression and expansion strokes, the effects of finite valve flow area, and the effects of finite valve flow areas plus finite valve opening/closing times. Calculations have also been carried out using reed valves in the compressor. In Figure 4-3, the "ideal" valve timing is shown (for five percent clearance volume) as well as valve timing with finite opening/closing times.

Performance calculations for the expansion of air have also been carried out. Table 4-2 lists examples of the calculated performance for expansion of air as a function of the

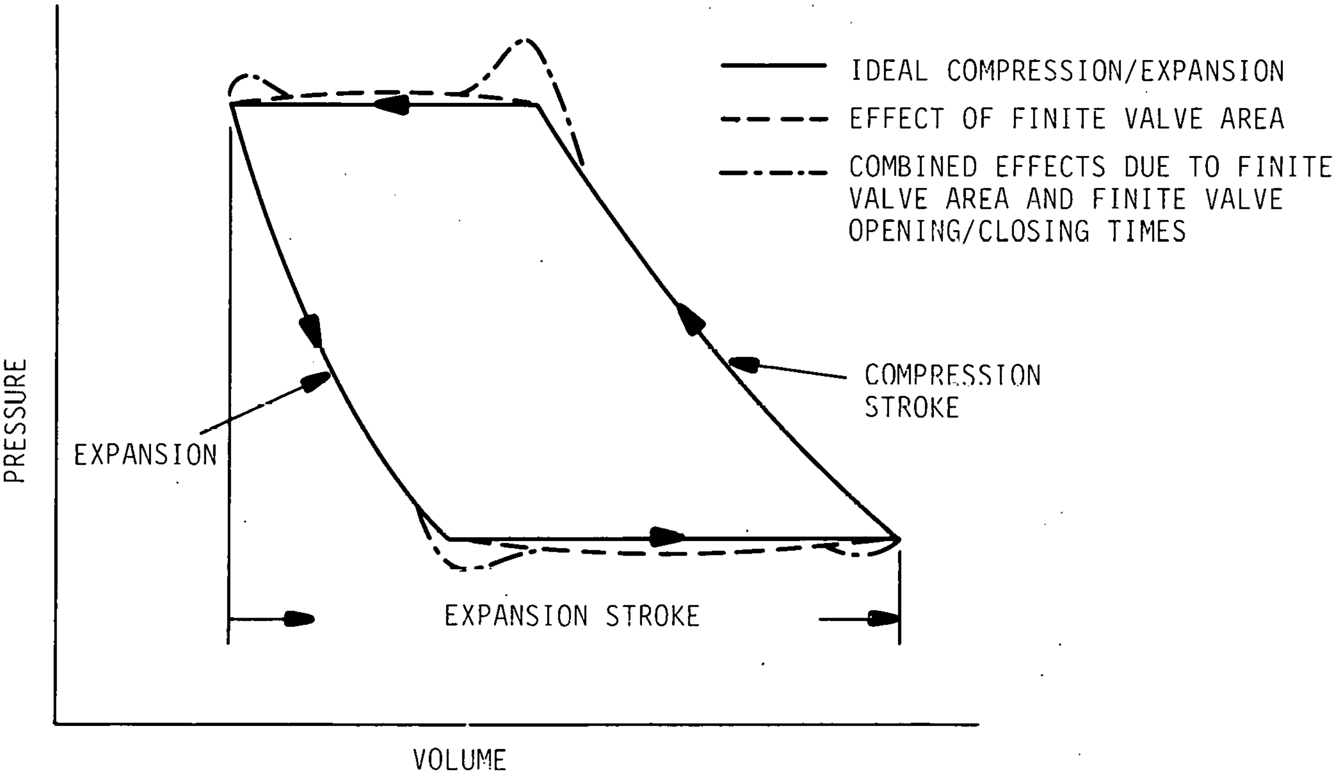


Figure 4-2. P-V Diagram of the Compression and Expansion Processes

Table 4-2. Examples of Expander Peformance

P_{in} = 45 psia P_{out} = 15 psia T_{in} = 70°F T_{wall} = 10°F   Clearance Volume = 5 percent of the swept volume

Bore (in)	Stroke (in)	Expander Speed (rpm)	Stanton Number	Valve Area / Piston Area	Valve Timing	Expander Isentropic Efficiency (%)	Lbm Air Per Rev
6	3	1800	0	0.15	Linear, 30° of crank angle for opening/ closing	95.1	0.347×10^{-2}
6	1.4	1800	0	0.0622	Cycloidal, opening/ closing in 45°	94.7	0.167×10^{-2}
6	2	1150	0	0.0622	"	95.4	0.224×10^{-2}
Independent Variables						Dependent Variables	

major expander design variables. The parameters held constant in the table are the expansion inlet pressure and temperature, the expansion discharge pressure, the cylinder wall temperature, and the clearance volume.

The isentropic efficiency for expansion, η_e, shown in Table 4-2 is defined as:

$$\eta_e = \frac{W_e}{W_{rev_e}} \tag{4.20}$$

where W_e is the actual expansion work per cycle and W_{rev_e} is the work delivered for the reversible isentropic expansion of air. W_{rev_e} is given by

$$W_{rev_e} = m\, c_p\, T_{in} \left[1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{k-1}{k}}\right] \tag{4.21}$$

Where T_{in} is the expander inlet temperature, P_{in} is the expander inlet pressure, P_{out} is the expander discharge pressure and m is the expander mass flow per cycle.

As was the case for the compressor, the isentropic efficiency of the expander is strongly influenced by the engine speed, stroke, and the valve area to piston area ratio as is shown in the tabulated results.

Other losses such as mechanical losses and heat transfer losses were also investigated. Several mechanical design configurations of the compressor/expander employing different valve types, valve drive mechanisms, crosshead mechanisms, etc. were analyzed. A summary of the range of values for the mechanical and thermodynamic losses occurring in the compressor/expander system for various mechanical design configurations is shown in Table 4-3. It is evident that the major contributors to total losses are the valves and valve-drive mechanism. Depending on the design, the isentropic efficiency of the compressor/expander system may be expected to fall between 78 and 92 percent.

The selected compressor/expander configuration for the system is summarized in Table 4-4.

Table 4-3. Summary of Thermodynamic and Mechanical Losses Occurring in the Compressor/Expander

Type	Percent Loss
Valve Losses	
• Valve Area	1.6
• Time of Opening	2.2
• Valve Leakage	1.0 to 5.0
Internal Heat Transfer Loss	1.0
Mechanical Losses	
• Piston Ring	0.4 to 1.6
• Crosshead Drive Mechanism	0.5 to 1.0
• Rod Bearings	0.1 to 1.0
• Main Bearings	0.1 to 1.0
• Valve Drive Mechanism	0.75 to 7.5
Total Percent Losses	7.65 to 21.9

Table 4-4. Summary of the Selected Characteristics for the Compressor/Expander

Variable	Value
Suction Pressure	15 psia
Discharge Pressure	45 psia
Engine Speed	1150 rpm
Piston Diameter	6 inches
Piston Stroke	2 inches
Valve Diameter	2 inches
Valve Lift	0.3 inches
Clearance Volume	5 percent

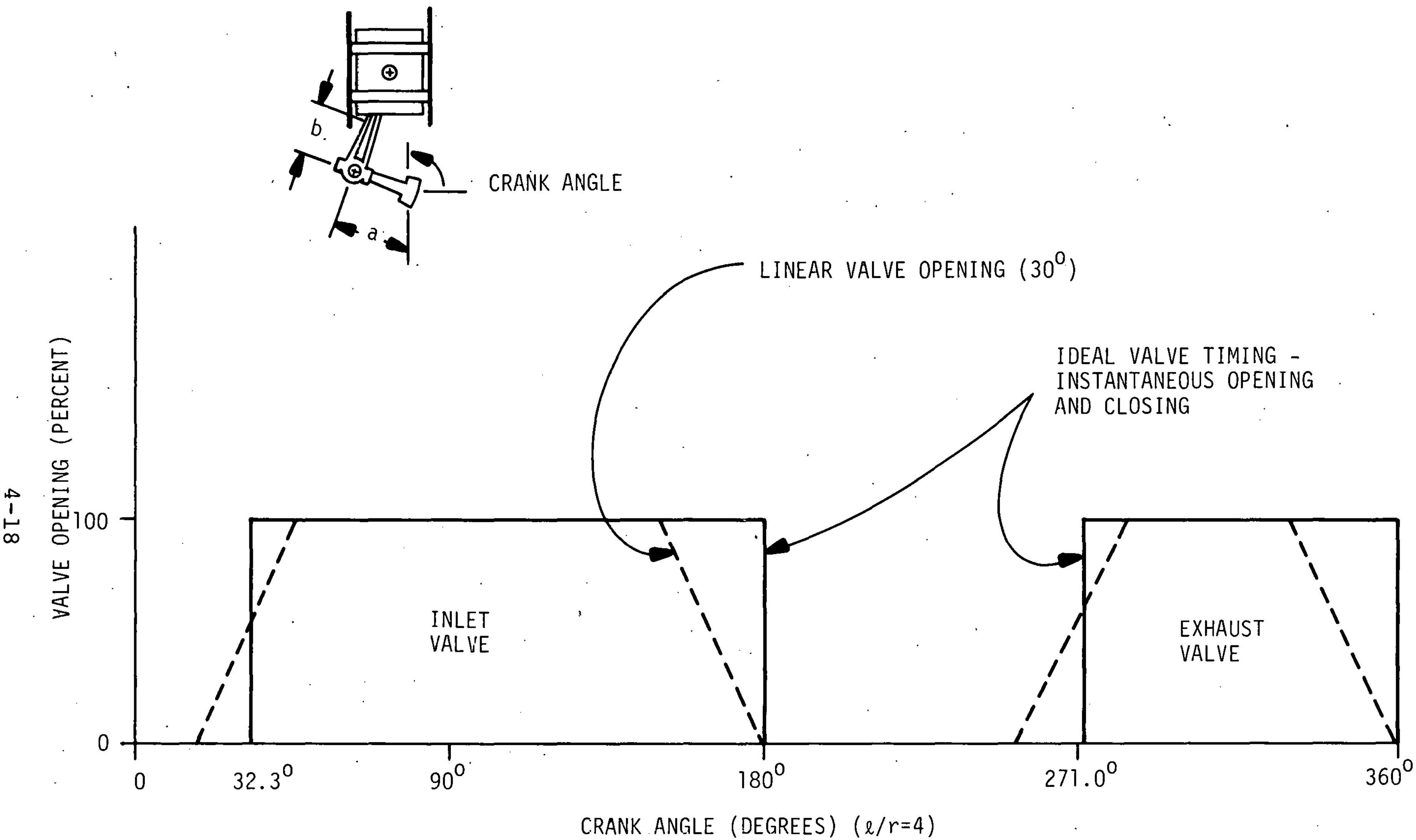


Figure 4-3. Compressor Valve Timing

5. COMPRESSOR/EXPANDER MECHANICAL DESIGN

5.1 Conceptual Mechanical Design

From the parametric analyses, achieving an attractive system COP requires a compressor/expander having extraordinarily high isentropic efficiencies, especially in relation to its low flow capacity. Equally important, in order for the system to suit the residential market, the compressor/expander must have low cost, noise, and maintenance and small size. Consequently, the design problem is to select the simplest configuration that will satisfy the aerodynamic and thermodynamic requirements and will transmit the power with very low friction.

5.1.1 Aerodynamic Design

Results of the thermodynamic analysis for a piston-type machine show that in order to attain high efficiency, very large valve areas are required relative to the airflow, and the valves must move quite rapidly relative to their total period of operation. The most critical operations are the opening of the compressor discharge valve and the closing of the expander input valve, because the piston is at mid-stroke and flow velocity is high.

Several types of valves were studied in preliminary designs, such as reed, poppet, sleeve, rotary, and plate valves. (Note that pressure operated reed valves are suitable only for the compressor; the expander must have mechanically driven valves.) Each valve type has its favorable characteristics but presents serious risk for this application. Not suprisingly, the "tried-and-true" reed and poppet valves were selected for the best balance of performance and risk. Their disadvantages are possible limited life and high noise characteristics. At the other extreme, the non-contacting rotary-valve appeared excellent in all respects except one. Very small clearance must be maintained (about 0.001 inch) between the port and valve spool in order to minimize leakage.

Sufficient valve area for airflow is made possible by using a large piston diameter of approximately 6 inches. Rapid valve operation is met with poppet valves that translate from open to shut or vice-versa in 30 degrees of crank angle, effectively, and with reed valves that are quite light and of advanced design in spring and damping characteristics [7].

Results of thermodynamic analysis also show that leakage past valves and the piston must be held to a very low fraction of the through-flow. Classical analyses of gas flow in narrow clearances and labyrinth configurations for prelimlinary designs indicate that valve leakage should be negligible, that piston rings are required for the large diameter, and that practical clearances for the piston rod and valve stems offer acceptably low leakage. Leakage is estimated to be 3 percent of the total flow, primarily past the piston rings.

5.1.2 Thermal Design

Design implementation of the specifications such as clearance volume fraction and valve timing is accomplished without difficulty or novelty. Temperatures are moderate and pose no problems for material selection, even uncooled. Only the compressor head may require some external insulation for personnel protection.

Transient heat flux occurs in and out of the gas and internal surfaces, while there is relatively steady flux through the machine walls and members. Results of the computer modeling and order-of-magnitude estimates show that although performance loss due to heat flow is small, the most significant amount of heat transfer occurs across the piston. Here the temperature range is greatest (300°F to -70°F extreme air temperatures). For this reason (as well as for low reciprocating mass) the piston is a sandwich of thin metal and insulating composite discs, approximately 1 inch thick.

5.1.3 Machine Configurations

A schematic view of the compressor/expander is given in Figure 5.1. Inside is the simplest possible arrangement and number of essential parts: a double-acting piston, two expander poppet valves under this piston, a piston crosshead, and a simple crank. The expander exhaust valve is driven by a push-lever; the input valve has a reverse-seat and is driven by a pull-lever; both levers operate directly by cam lobes on the crankshaft. The machine is direct-driven in order to reduce cost, noise, and maintenance. Ultimately, the electrical motor will be integrated with the flywheel and crank case.

The double-acting crosshead configuration not only offers few parts and a compact machine, but it permits oil-free compression, which makes permanent crank case lubrication possible and precludes oil contamination of the air-side of the heat exchanger.

The large bore of 6 inches is needed to provide enough valve area (without resorting to complex, multiple valves for each function.) The state-of-the-art for reed and poppet valves limits the maximum (electrical synchronous) speed to 1800 rpm, at which speed the required stroke for the design airflow of 33 scfm is only about 1.4 inches. Note that higher speed would require even shorter stroke and greater difficulty in achieving low clearance volume fraction. This is an extreme bore-stroke ratio compared to usual practice, but the present use of a cross-head design avoids the typical disadvantages of a heavy piston design with high friction, and offers other advantages.

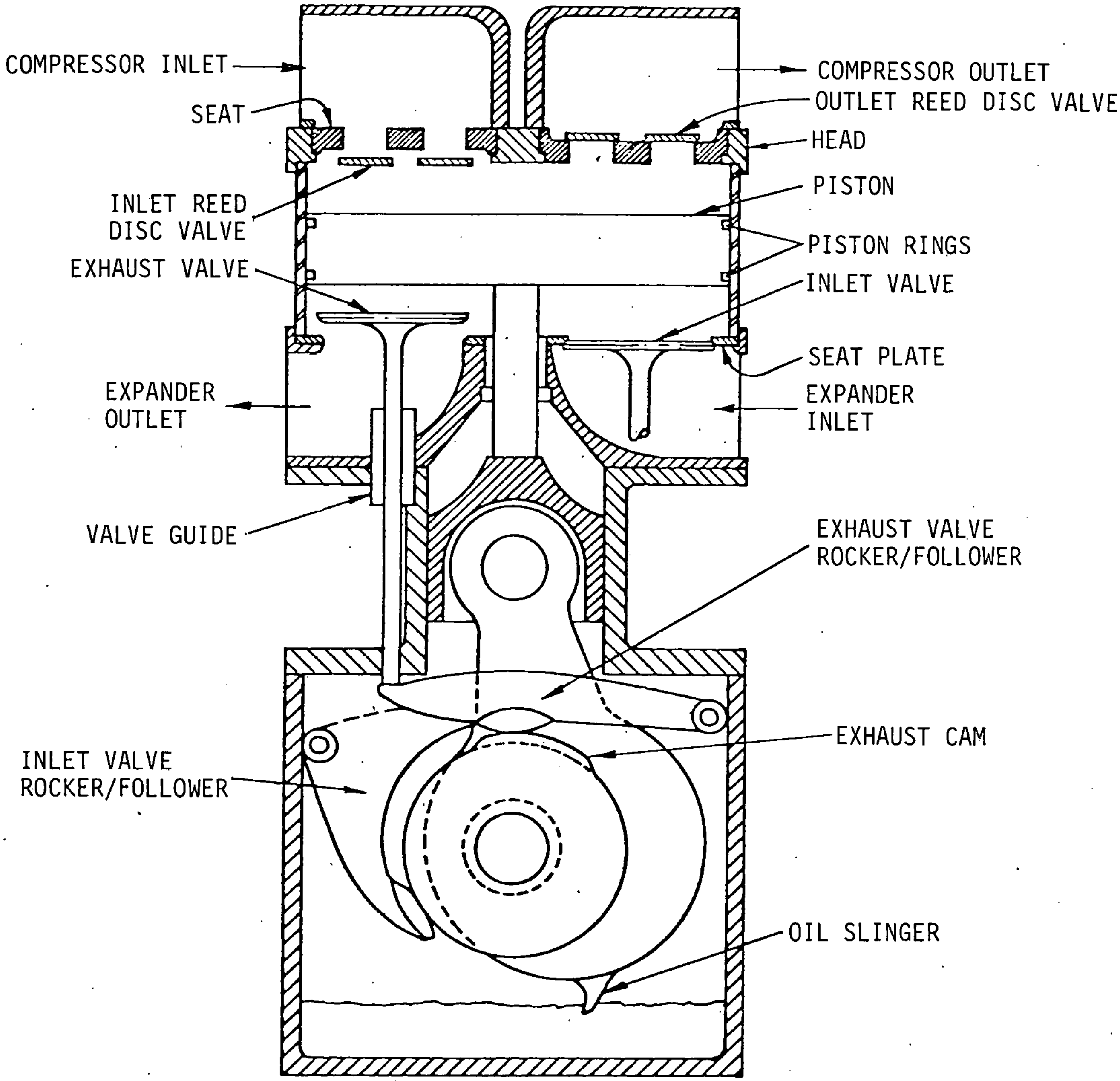


Figure 5-1. Schematic of the Compressor/Expander System

For example, the large bore to stroke ratio makes it possible to minimize the piston ring friction force compared to the piston working force (ring perimeter area versus piston area). Short stroke reduces centrifugal forces on the bearings, and reduces windage and oil splash power losses while a thick oil film can be maintained on cam and slider surfaces. A short stroke, of course, contributes strongly to obtaining a compact "squat" machine suitable for integration into the water heater.

Initially, the compressor/expander will be mounted with the bore vertical and a separate electrical motor. Later, with an integrated electric motor, the unit can be mounted horizontally under the water storage tank. In both configurations, the machine will be suspended on vibration isolation mounts. The major portion of the vibration is due to the fluctuating torque caused by alternating energy exchange between the air and the flywheel. Unbalanced vibrations due to translational inertia are minor, but they will also be isolated by the mounting.

The compressor/expander unit must include silencing means and flexible couplings for the air ducts. Hoses can be used for intake and exhaust, and for the low gas pressure (30 psig), flexible couplings to and from the heat exchanger are available. Silencing of the intake and exhaust is required for residential use, because they would generate a sound level of roughly 80 decibels at 3 feet, unsilenced.

The compressor/expander operation can be utilized advantageously for silencing - the intake pulse occurs at the same time as the exhaust pulse, and no flow occurs through the valves during the other half revolution of the crank. One common muffler volume divided by a flexible membrane between the intake side and the exhaust side can be used. During an intake-exhaust pulse, the diaphram translates and there is only a small net unsteady flow component superimposed on the steady flow in the and exhaust lines. This muffler can be designed using proven methods [3] considering the muffler as capacitance, the diaphram as a "loud-speaker", and the intake-exhaust hoses as chokes. However, the concept remains to be proven.

5.2 Final Design

After the preliminary design is selected, it is necessary to develop an integrated, detailed design which includes a quantitative evaluation of all components. For the present case the integration was not straightforward, but involved several iterations, primarily related to valve design. Following is a discussion of component design with minimal reference to the iterative process.

5.2.1 Compressor Valves

Existing spring-loaded disk valves were selected for the compressor on the basis of convenience, cost and proven reliability in non-lubricated service. Large single valves, one for exhaust and one for intake, provide the maximum flow area possible while remaining within the limits imposed by the cylinder bore and a flat head. Larger valves would require overhang beyond the bore or a highly-domed head. They are not desirable because they involve greater clearance volume, heat transfer area, and mechanical complexity, still without providing the desired flow performance.

The valves selected do not quite provide the performance desired according to our analysis based upon the manufacturer's quoted flow coefficient. If low performance does occur (compressor isentropic efficiency less than 84 percent), the stock valves will be modified. This will involve grinding of the valve disk to reduce its mass, and milling of the valve slots to provide more flow area. Computer-predicted dynamic response and aerodynam__ performance of the modified valves are adequate (94 percent predicted aerodynamic efficiency which is the measurement of pressure drop losses). If valve life or performance still proves to be inadequate, a novel head utilizing annular diaphragm valves and springs will be designed and fabricated. The preliminary design is in hand, but detail design of this head is postponed until after initial testing of the purchased valves.

5.2.2 Piston

The piston is in the form of a short, laminated disk to provide lightness and low thermal conductivity. Shortness is acceptable because the piston is guided by the piston rod and no side-load occurs other than the small weight of the piston if the bore is horizontal.

The piston is constructed of an aluminum alloy disk sandwiched between disks of fiber reinforced plastic. The metal disk carries the gas load, the piston rings, and the rod attachment. The piston rings (one or two) are available in several varieties of self-lubricating, filled plastics or bronze. Tests must be performed to judge the best performance as a combination of low friction and long life. (Estimates of the several friction lossess in the compressor/expander are summarized later.)

5.2.3 Crosshead

The crosshead takes the lateral forces of the connecting rod and provides alignment of the piston rod in combination with the rod bushing. Because the force reverses direction and the rod requires guidance in both lateral directions, a piston-type crosshead is used. This is the simplest construction, compared to linkages or flat guides.

The maximum lateral force is about one-quarter of the piston force, so sliding friction is important. Metal-to-metal sliding with boundary lubrication (non-hydrodynamic) is probably satisfactory, but the design includes two measures that should reduce friction further: first, the guiding surface is coated with a low-friction material - Teflon, and second, the crosshead is shaped to provide the maximum hydrodynamic load capability. Bearing pressures and velocities are moderate compared to usual practice, and the slider surface is provided with positive spray lubrication. A step-pocket bearing design is used [4].

5.2.4 Connecting Rod

The design of the connecting rod is consistent with the following factors: a large gas-pressure load, low rotational speed, a small dynamic load, and the desire for very low friction and long life. A needle roller bearing is used for the wrist end, and a spherical roller bearing for the crank end, both having long life (at least 10 years on duty cycle) and negligible friction.

The rod bearing selection maximizes the performance and minimizes the risk for the prototype. Eventually a split roller or journal crank-end bearing combined with a larger, one-piece crank might be the most cost-effective configuration. A well-developed journal bearing could handle the load conditions with a friction loss less than 3% of the net compressor/expander power.

5.2.5 Crank

The prototype crank is made in two pieces with two roller bearings per piece. The main piece provides for all load transmission from connecting rod to flywheel and motor-coupling, as well as supporting one cam. The follower-crank piece supports the other cam. This design provides low friction as well as convenience for the prototype. Ultimately a one-piece crank with integral cams would be the more economical design.

The crank is counter-balanced including the mass of the connecting rod end and bearing. The reciprocating mass (piston, etc.) is partially counter-balanced, resulting in reduced linear vibration in the piston axis direction but imposed linear vibration in the transverse direction. This distribution of vibration is believed to be favorable for the vibration-isolation mounting, but the counter-balance is demountable, so the distribution can be modified during testing. Of course, a production model would have integral balance mass.

The crank supports two cam plates, one on either side of the rod. The cams are demountable and adjustable in timing angle for the prototype, but they also would be integral in the production model.

5.2.6 Expander Valves

The expander valves must be cam driven, and their timing, flow area, and power absorption are critically inter-related for the expander performance. Several features offer good performance.

First, the cams are cut to the most rapid profile consistent with low noise and minimal surge "harmonic" excitation. The 45 degree cycloidal schedule provides gradual change of acceleration and 90 percent of the stroke within 30 degrees. (A modified trapezoidal schedule is similar in performance.) The exhaust valve has a long dwell at maximum opening; the input valve has practically no dwell. The profile requires a curved or roller follower for practical dimensions.

Secondly, the valves have maximum diameter consistent with the large bore and simple, direct drive from the crank. The seats are ground to 30 degrees for more "curtain" areas than the usual 45 degrees, and the block porting is smooth and generous.

Thirdly, measures are taken to minimize dynamic, friction, spring, and gas pressure forces. The valves are hollow and light. The high pressure intake valve incorporates a balance-pressure piston communicating with the cylinder space. The springs are minimal, in pre-load and spring rate, to just overcome inertial requirements. And finally, the cams and follower contact buttons have diameters that are optimized for minimum contact stress, conducive for boundary lubrication without galling. (A roller follower is not possible, considering the load, speed, and life requirements.)

Ultimately, valve gear might be designed that is simpler, lighter and lower in friction, using hairpin or torsion springs and a reverse-seat or pull-type input valve without a balance piston, but this involves too high a development risk for the test model.

5.3 System Assembly and Ancillaries

The compressor/expander consists of three main sections, the compressor cylinder and head, the expander valve block, and the crankcase. Assembly drawings are shown in Section 7.

The crankcase is split across the crank axis along the piston rod axis. This allows insertion of the crank and rod set with bearings, cams, follwers, and end caps. The valve block is assembled fully with valves, springs, seals, and cross-head slider tube, after which it is mated to the rod end of the crankcase, over the piston rod. The piston, cylinder, and compressor head are then assembled on the expander valve block with four tie bolts to the crankcase. Peripheral assembly of air ducts, mounts, flywheel and coupling is straightforward.

Positive spray lubrication is provided for the cam-follower contacts and the cross-head slider tube, with residual splash and drift wetting the bearings and other elements. Initially a separate oil pump will be used; ultimately small Gerotor (internal gear) pump parts can be installed directly on the free end of the crankshaft. No oil cooling, filtration, or changing are needed, thanks to the sealed, benign environment. A magnetic plug can be used to collect wear particles.

During start-up acceleration to design speed, the expander valve timing will be mis-matched with the compressor output, so more input torque will be required compared to design torque. Because the motor was sized for optimum efficiency, and because the heat exchanger volume requires time cycles to pressurize, the system may accelerate "over the hump" to full speed. If not, a decompression valve or bypass will be provided and switched appropriately.

The compressor/expander requires a muffler for the intake and exhaust, combined in one unit as described in Section 5.1. The volume of the muffler and the length of intake and exhaust hoses provides for a very low resonant frequency and, hence, excellent sound containment (an insertion loss of about 23 decibels or more for audible frequencies above 40 Hz). This provides for low direct radiated noise outside the house, and for greatly reduced inside noise due to structural transmission and re-radiation. High frequency machine-radiated noise will be contained and absorbed by acoustic material lining the unit enclosure.

5.4 Estimates of Mechanical Losses

As mentioned before, design efforts concentrated on reducing mechanical losses in the compressor/expander. In simplest terms, this is important because this unit has internal forces and power flux that are much larger than the net power indicates. For example, the compressor absorbs about 3 hp and the expander provides about 2 hp, mostly but not entirely via the crank linkage to and from the flywheel. Thus a one percent frictional loss from the gross power flux results in almost five percent increase in net power input required.

It is not particularly useful or even possible to assign the losses in this machine to either the compressor or the expander functions alone. They have been estimated according to the dynamics of individual machine components, and the results are presented in Table 5-1.

Table 5-1. Component Estimates of Frictional Horsepower Losses

Source	Loss (Hp)
• Piston Ring (1 or 2)	0.02 to 0.11
• Crosshead	0.02 to 0.08
• Bearings (4)	0.05 to 0.10
• Cams (2)	0.04 to 0.11
Total Mechanical Losses	0.13 to 0.40

The mechanical loss estimates vary by a large factor that significantly affects the predicted water heating COP. For example, assuming internal thermodynamic efficiencies of 90 percent for the compressor and expander, and 80 percent for the motor, the net COP varies from about 1.7 to 2.0 for the range of mechanical losses quoted.

The variation in loss estimates is chiefly due to uncertainty about the friction of sliding surfaces with filled materials (rings) or with boudary lubrication (cams, crosshead). We believe that the lower loss values are probably achievable, based upon maufacturers data on ring materials, partial hydrodynamic support of the crosshead, bearing manufacturer (SKF) predictions, and recent analyses of lubricant starvation and minimum film thickness in slider contacts [5,6]. Within the expected range of operation, the losses will remain nearly a constant fraction of net power.

6. AIR TO WATER HEAT EXCHANGER AND WATER CIRCULATION LOOP DESIGN

The air-water heat exchanger and water circulation loop portion of the Brayton cycle water heater is illustrated schematically in Figure 6-1. Included are the heat exchanger, a water circulation pump, the compressor/expander motor cooling jacket, a thermostat, a condensate drain on the air side, and shut-off valves allowing isolation of the water tank from the water circulation loop. The heat exchanger provides for transfer of available heat from the hot compressed air stream to the water to be heated. The water circulation pump forces the water through the water circulation loop and heat exchanger. The thermostat maintains a constant outlet temperature by varying the water flow through the loop in response to variable inlet temperatures of the hot compressed air and cold water streams.

This portion of the system and the individual components have been designed to meet certain overall performance requirements and to accomodate several other design constraints. They are:

1. Minimum heat exchanger effectiveness of 80 percent based on the air side

2. Maximum heat exchanger pressure loss of 1/2 psi on the air side

3. Minimal consumption of power to drive water circulation pump

4. Hot water outlet temperature control to within 5°F of the desired set temperature

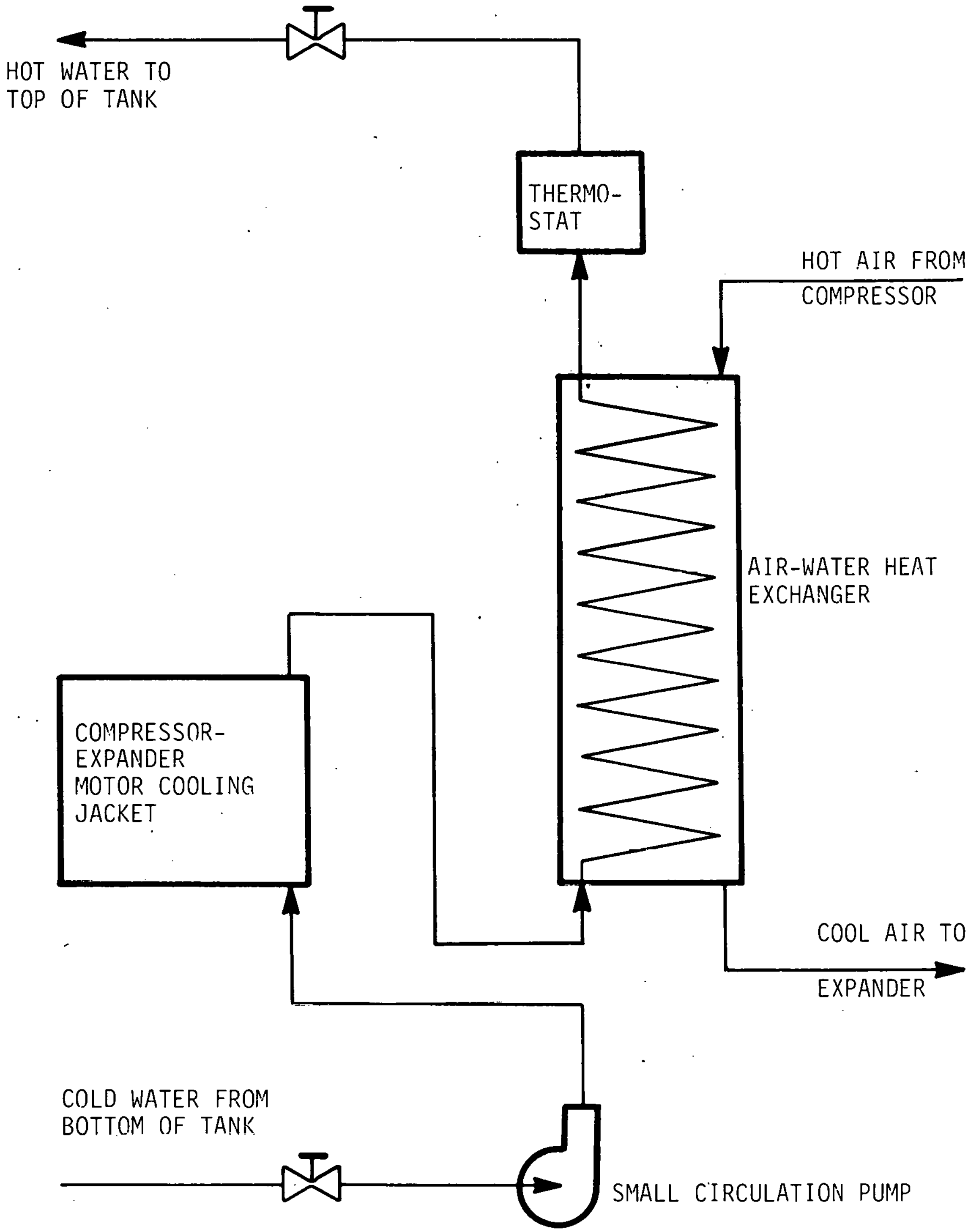


Figure 6-1. Air-Water Heat Exchanger and Water Circulation Loop

5. No corrosion-prone or toxic materials in contact with the water stream

6. Provision of adequate volume in the heat exchanger and air inlet and outlet tube to moderate the compressor discharge and expander inlet pressure and flow patterns

7. Water side working pressure of 150 psig

8. Air side working pressure of 50 psig

9. Ability to tolerate some water side scaling and fouling

10. Components sized compactly

11. Incorporation of standards parts and fittings

12. Minimal heat loss to the surroundings

13. 25,000 hour (greater than 10 years at 25 percent duty cycle) life of all of the above components.

The heat exchanger performance specifications (1, 2) were established in the system design study. They both represent the points at which heat exchanger performance improvements start to become quite costly, while offering only small improvements in the overall heating COP and capacity of the water heater. Reduction of the severity of air side pressure and flow pulsations (requirement 6) improves the performance of both the compressor/expander and the heat exchanger. Working pressures are based on the industry standard maximum water heater design pressure (7) and the maximum compressor discharge pressure (8). Component sizing (10) is critical because of the desirability of marketing the entire water heater as a single packaged unit.

6.1 Air-Water Heat Exchanger Design

The air-water heat exchanger is the second most important component in the Brayton Cycle Water Heater. Except for the compressor/expander, it is the most costly component in the system. Except for the compressor/expander, the performance of the heat exchanger has the most substantial effect on the COP and heating capacity of the water heater.

The air-water heat exchanger has been designed by the following method:

1. Identify all requirements including performance requirements established by the system design study

2. Develop heat exchanger design concepts, based both on existing heat transfer equipment and a fundamental examination of the requirements of this program

3. On the basis of a preliminary analysis, narrow the concept choice to two or three approaches

4. Perform the necessary analysis to approach an optimum heat transfer design and to estimate the manufacturing cost of the remaining heat exchanger concepts

5. Select the most attractive concept on the basis of projected cost, dimensional considerations, and other factors

6. Prepare a cost optimized design of the selected heat exchanger concept.

6.1.1 Heat Exchanger Design Requirements

The heat exchanger has been designed to meet the following requirements:

1. Minimum heat transfer effectiveness at design conditions of 85 percent

2. Maximum air side pressure loss at design conditions of 1/2 psi

3. Maximum water side pressure loss at design conditions of 1/4 psi

4. Minimum life of 25,000 hours (greater than 10 years at a 25 percent duty cycle)

5. Ability to tolerate a certain amount of fouling and scale build up

6. Adequate volume to absorb compressor/expander pulsations

7. Provision to be made for condensate drainage on the air side

8. Air side working pressure of 50 psig

9. Water side working pressure of 150 psig

10. Insulation to minimize heat loss to the environment

11. Minimum size and manufactured cost.

6.1.2 Heat Exchanger Design Conditions and Effect of Off-Design Conditions

The design conditions applicable to the heat exchanger, listed in Table 6-1, are based on the predicted performance of the optimum system under average Northern United States ambient conditions, determined in the system design study. The estimated average ambient conditions are listed in Table 6-2.

Due to varying ambient temperatures and varying water temperatures, the heat exchanger will often operate at conditions different than the design conditions listed in Table 6-2. Three effects account for variance from the design point conditions.

1. Due to variations in ambient temperature and barometric pressure, small variations in the air mass flow rate through the heat exchanger and modest variations in the air inlet temperature to the heat exchanger will occur.

Table 6-1. Average Northern United States Ambient Conditions

Parameter	Value
Ambient Temperature	$40^{o}F$
Ambient Relative Humidity	50 Percent
Ambient Barometric Pressure	14.7 psia
Cold Tap Water Temperature	$50^{o}F$

Table 6-2. Design Conditions For Heat Exchanger

Parameter	Value
Air Inlet Temperature	240°F
Air Inlet Pressure	45 psia
Air Inlet Humidity Ratio	0.005
Air Flow Rate	2.80 lbm/min
Water Inlet Temperature	50°F
Water Flow Rate	0.145 gpm (72 lb/hr)
Environment Temperature (For estimating heat losses)	70°F

Neither of these variations is expected to have a significant effect on the performance of the heat exchanger

2. Due to annual variations in surface and ground water temperatures, the water inlet temperature to the heat exchanger will be variable. The water flow rate through the heat exchanger is adjusted by the thermostat to maintain constant outlet temperature and will result in the water flow rate being varied modestly. Again, no substantial effect on the heat exchanger performance is expected

3. The water heater operates in two modes - normal heating of cold tap water; and reheating water to make up standby losses. The system design conditions are specified for the heat up from cold case. During reheating, the flow controller opens much wider, allowing a much greater water flow rate through the exchanger. This has the net effect of increasing the heat exchanger effectivensss, but not the amount of heat recovered because the water inlet temperature to the heat exchanger is much higher. The water pressure drop also increases and this must be accounted for in the design of the water circulator.

The net effect of operating off design, then, is either negligible or an increase in heat exchanger effectiveness. The specified design conditions lead to a conservative design of the heat exchanger.

6.1.3 Heat Exchanger Materials

The material properties of greatest importance to the functioning of the heat exchanger are

1. Strength in pressure containing parts

2. Thermal conductivity of the heat exchanger core material

3. Corrosion resistance, particularly the water containing parts

4. Cost.

The materials most appropriate for the heat exchanger core are copper (particularly for those parts containing the water, also for fins) and aluminum (also suitable for fins). For pressure containing parts on the air side various metals are suitable and possibly fiberglass reinforced plastic could be used.

The combination of materials selected for a given design concept depends greatly on the geometry and the function of each part. Material selection is discussed in greater detail with the description of the selected design concept.

6.1.4 Heat Exchanger Concepts Considered

Several heat exchanger concepts which meet the design requirements were identified, both at the time the proposal for this project was prepared and during Phase I of the project. They are:

1. Finned tube coil around blank core, Figure 6-2

2. Serpentine finned tube and shell, Figure 6-3

3. Internally finned tube and shell, the standard geometry of an American Standard Compressed air after cooler, Figure 6-4

4. Externally finned tube and shell heat exchanger, Figure 6-5

5. Folded heat transfer surface heat exchanger, Figure 6-6

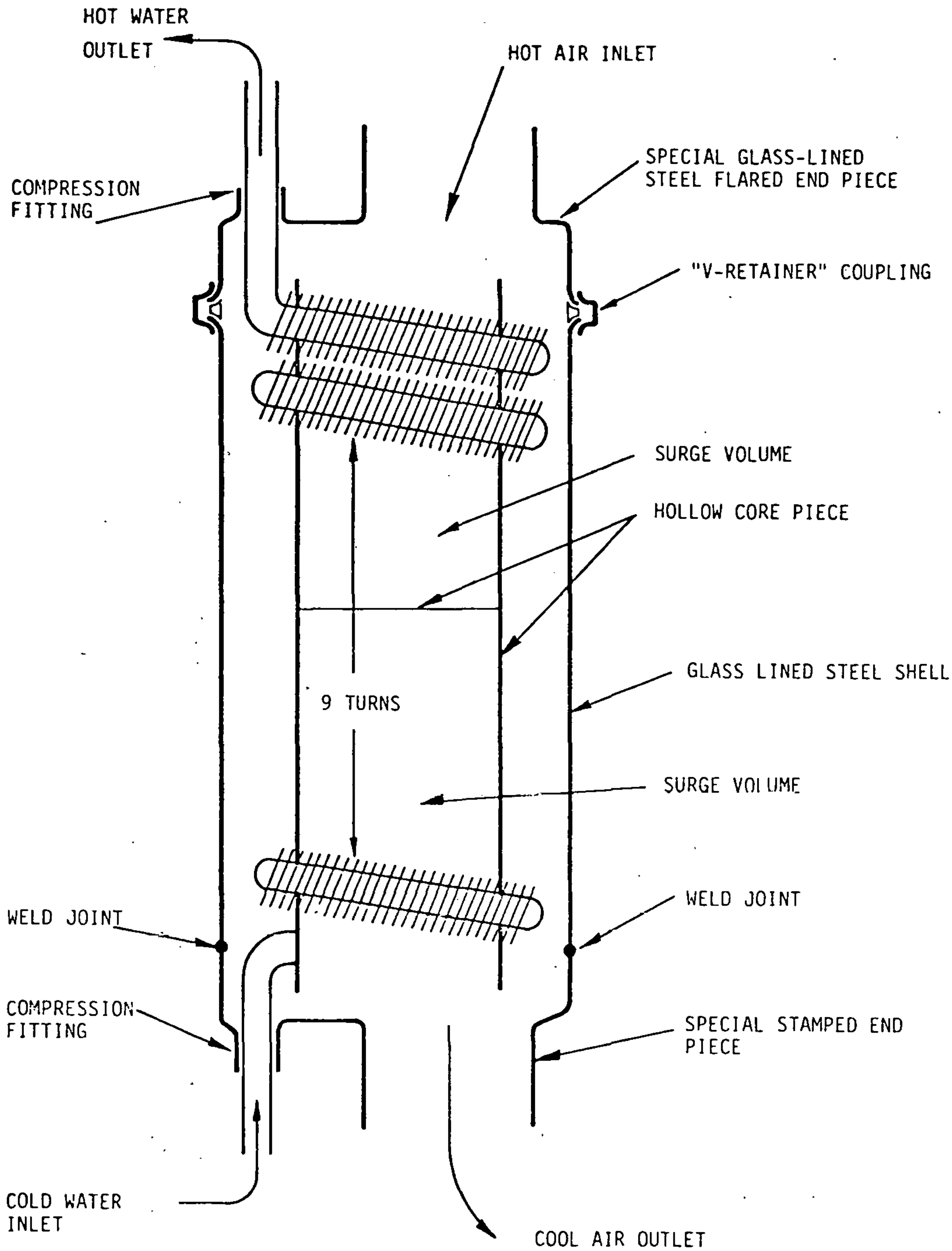


Figure 6-2. Finned Tube Coil Around Blank Core

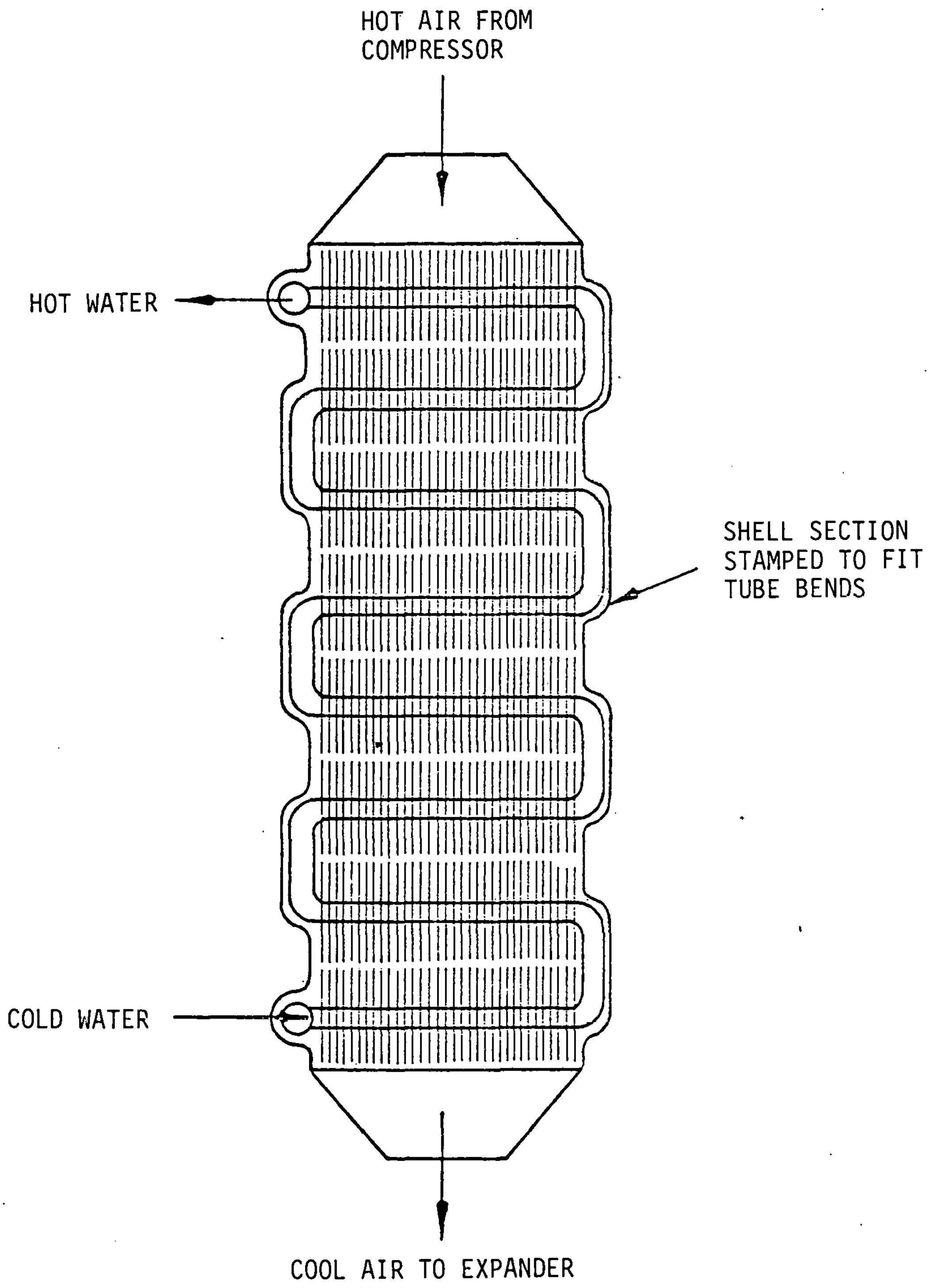


Figure 6-3. Serpentine Finned Tube and Shell

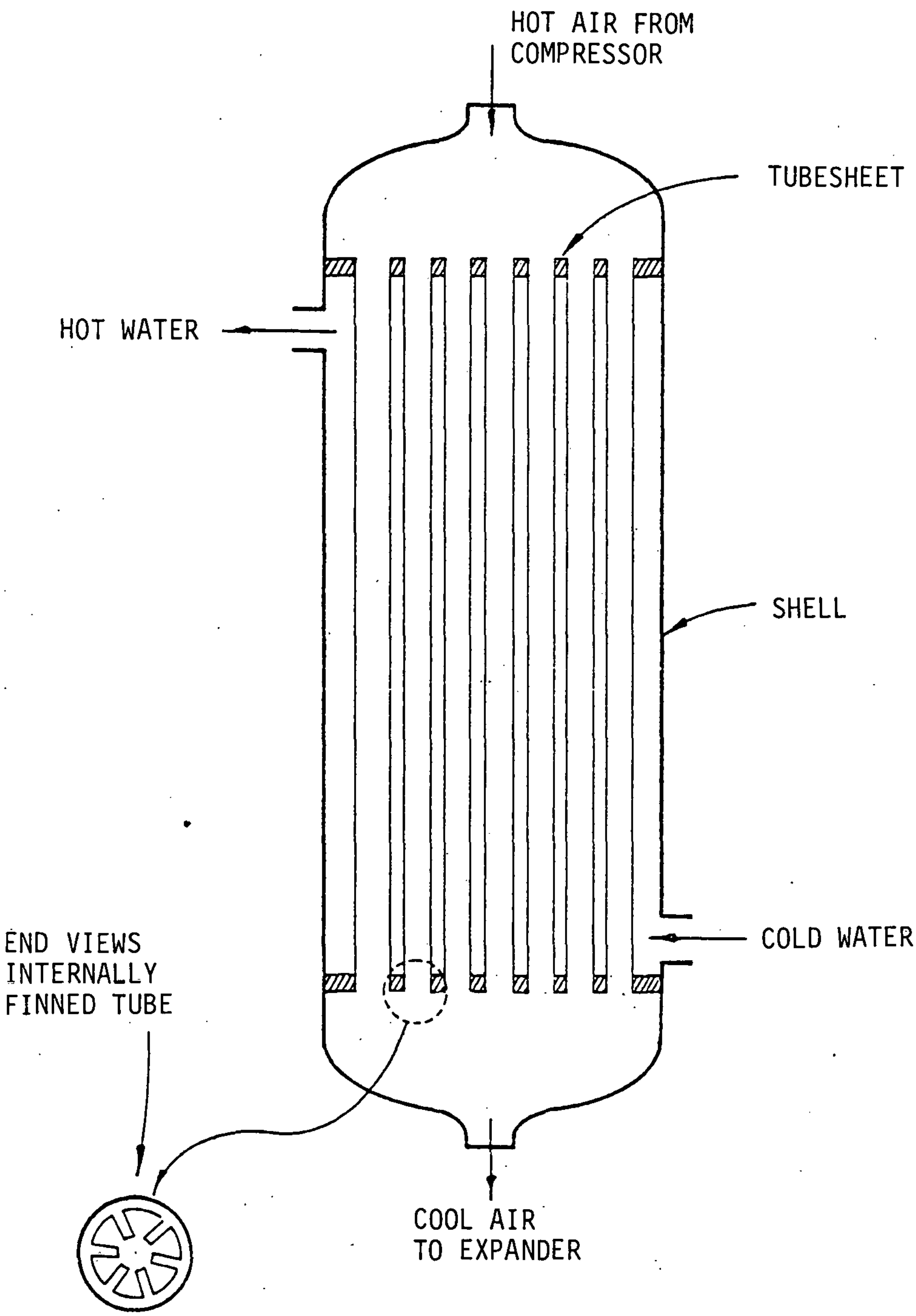


Figure 6-4. Internally Finned Tube and Shell Heat Exchanger

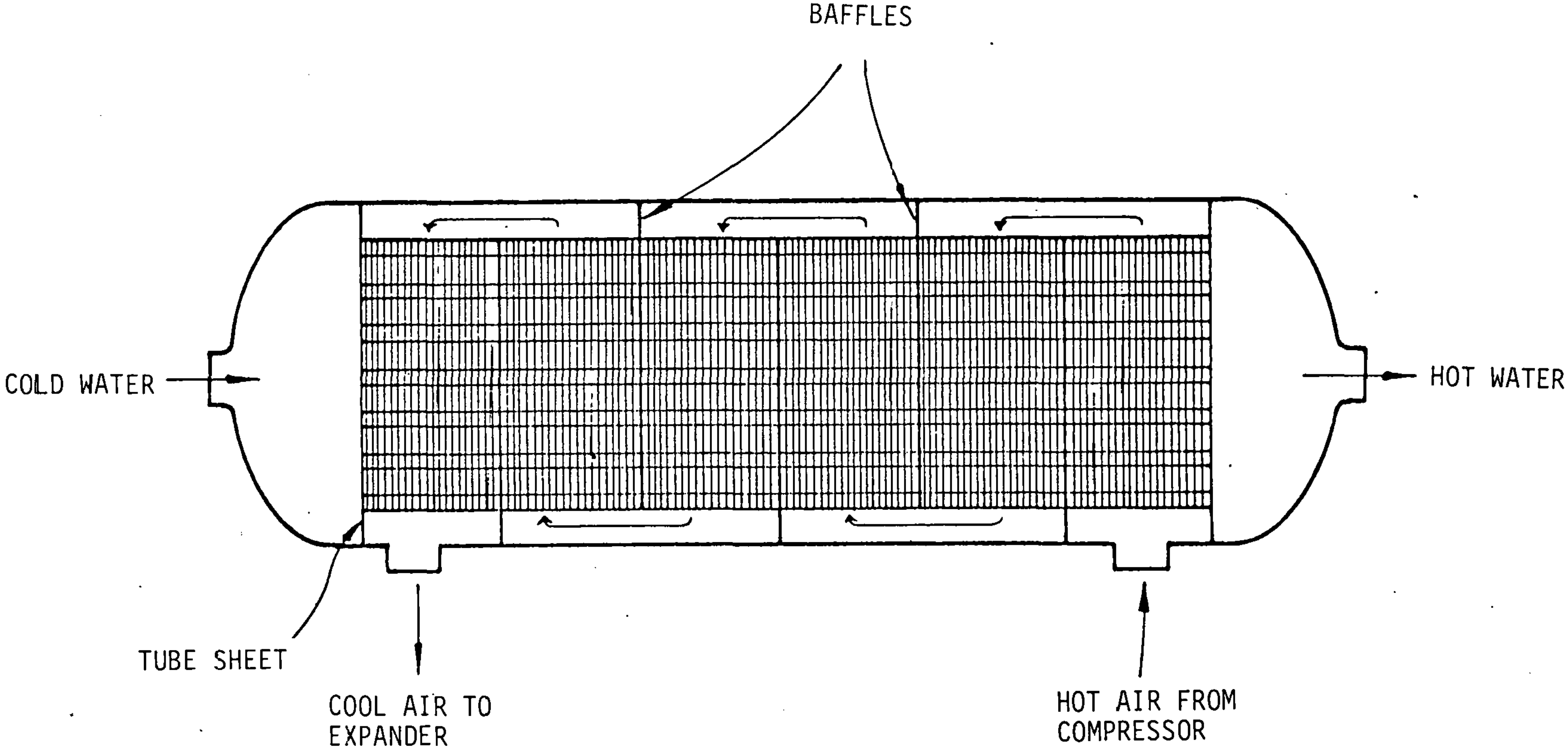


Figure 6-5. Externally Finned Tube and Shell Heat Exchanger

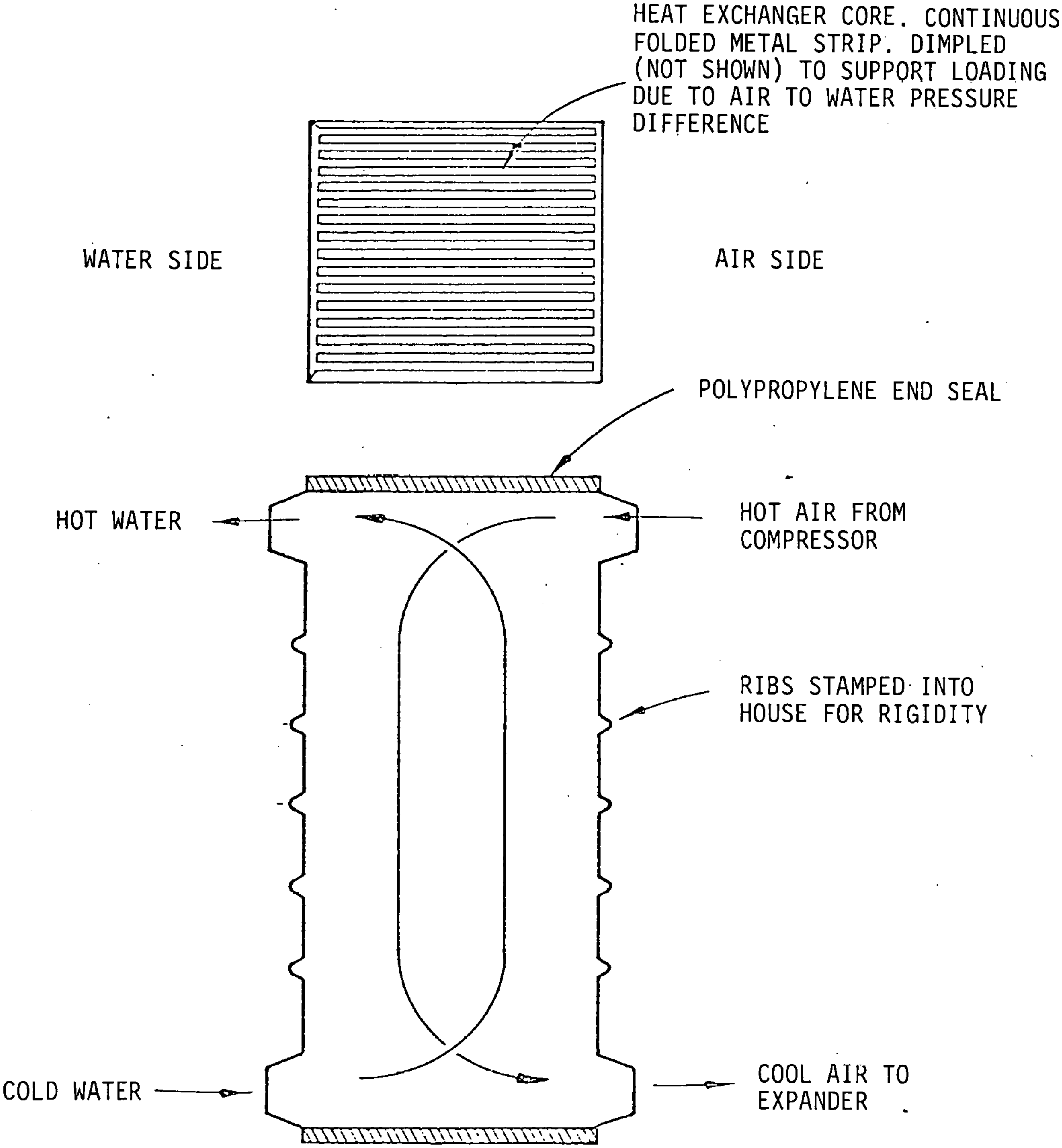


Figure 6-6. Folded Heat Transfer Surface

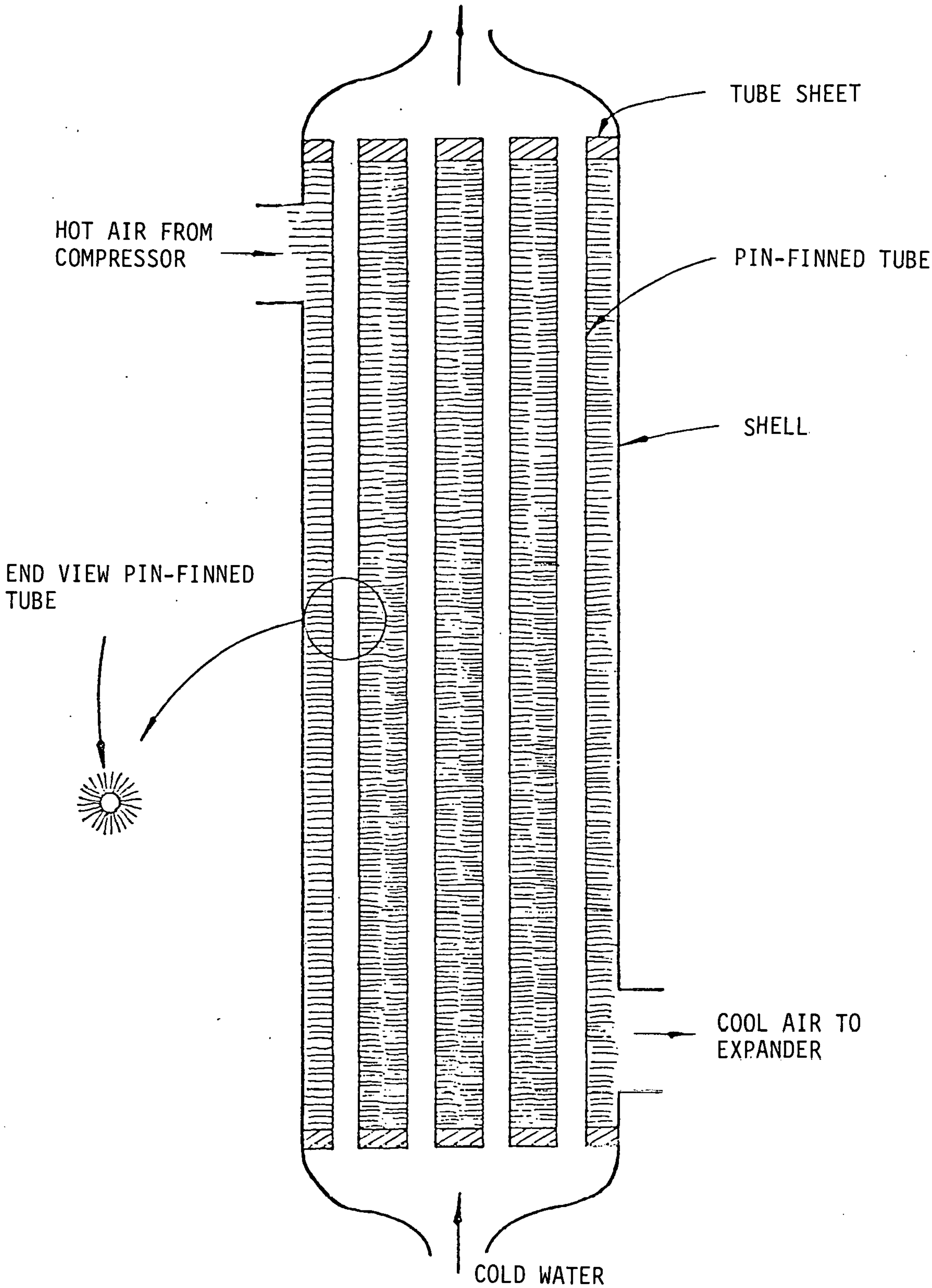
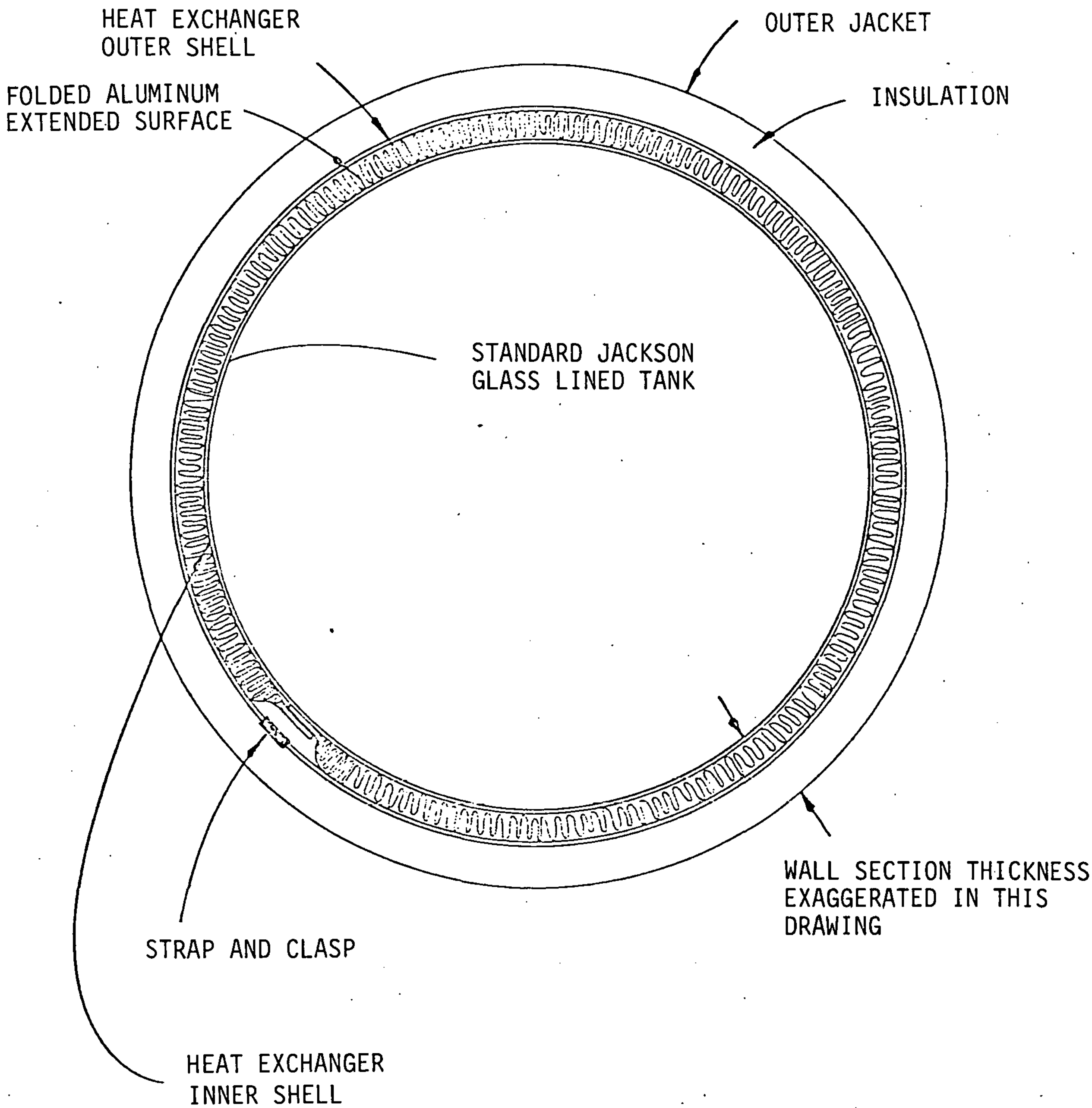


Figure 6-7. Pin-Finned Tube and Shell Heat Exchanger



NOTE: COVERS ENTIRE TANK VERTICAL SURFACES. HOT COMPRESSED AIR ENTERS HEADER AT TOP, COOL AIR DISCHARGE FROM HEADER AT BOTTOM

Figure 6-8. Annular Shell Strapped Around Standard Glass Lined Tank

6. Pin-finned tube and shell heat exchanger, Figure 6-7

7. Annular shell strapped around tank, Figure 6-8.

Of these concepts, the "finned tube coil around blank core" concept is the best one for the requirements of the water heater. At least one prototype of this concept will be fabricated and tested. The design, material specifications, and analysis of this concept are presented in Sections 6.1.1 and 6.1.2. Concepts 2. and 3. above were suitable for the water heater but were less attractive for a variety of reasons.

The serpentine finned tube and shell core concept is almost as attractive as the preferred concept. Preliminary estimates indicate that it will be approximately 50 percent more costly to manufacture. The increased manufacturing cost is attributable to several factors.

- The cost of the dead space baffles between the square core cross section and the round shell
- The cost of mechanically bonding or metallurgically bonding the plate fins to the tubing
- The cost of soldering multiple tubing return bends and the inlet and outlet water manifolds.

The internally finned tube and shell heat exchanger can be purchased on an OEM basis from the American Standard Corporation Heat Transfer Division. The quoted OEM price for a heat exchanger meeting the specifications was approximately 3 times the estimated manufacturing cost of the selected air-water heat exchanger.

The externally finned tube and shell heat exchanger was rejected after preliminary study indicating high manufactured cost when designed to meet the performance requirements.

The folded heat transfer surface heat exchanger is susceptible to water side fouling and would be costly to manufacture.

The pin-finned tube and shell heat exchanger, was rejected after preliminary study indicating that its manufactured cost would be high.

The annular tank heat exchanger, could not be designed to have the required heat exchanger effectiveness, nor does this type of heat exchanger allow control of the water flow rate over its surface. It would also be expensive to manufacture and needlessly add to the tank diameter. In addition, water temperatature gradients would occur in the tank.

6.1.5 Heat Exchanger Core Design

The air-water heat exchanger geometry selected for the prototype water heater is shown in Figure 6-2. It consists of a single length of helically finned tubing wrapped into a helical coil. The coil is placed inside an air side pressure carrying shell and the space inside the coil is blocked by a light gauge sheet metal can which is coated on the inside with polyurethane foam. Compressed air enters and leaves the shell through fittings located on the side of the shell at either end, facilitating the connection of the heat exchanger with the compressor outlet and expander inlet. The water is heated as it passes through the finned tube coil, counter flow to the air. The ends of the finned tube are oriented axially and pass through fittings on the ends of the shell.

The finned tube is manufactured by the Modine Manufacturing Company, Racine, Wisconsin. It consists of a wavy aluminum fin that is helically wrapped around a type 304 stainless steel tube. The aluminum fin is metallurgically bonded to the stainless steel tube by a proprietary bonding process. The dimensions of the finned tube are summarized in Table 6-3.

This geometry was selected over the others under consideration because of the following inherent advantages:

- Least costly heat transfer surface to produce
- Simple geometry of the complete heat exchanger geometry is inexpensive to produce and assemble the component parts
- High degree of design flexibility is inherent in this geometry so that various performance requirements can be met
- The empty space inside the annular coil provides the needed volume to absorb compressor/expander pulsations.

The heat exchanger core was conservatively sized to meet the heat transfer and flow requirements within the systems dimensional constraints by using the Number of Transfer Units (NTU) design procedure [8] in conjunction with published heat transfer data for the finned tube surface used in the heat exchanger. The NTU design method provides a highly efficient means of determining the necessary size of a heat exchanger core, given the geometry of the heat exchanger and the performance requirements. The core was sized to meet the following requirements.

Table 6-3. Dimensions of Finned Tubing Used for Air-Water Heat Exchanger

Parameter	Value
Tube O.D.	3/8 inch
Tube Wall Thickness	0.028 inch
Fin Height	1/4 inch
Fin Spacing	10 per inch
Finned Tube O.D.	7/8 inch

a. Heat transfer effectiveness of at least 85 percent at the design conditions in Table 6-2

b. Maximum air-side pressure loss through the heat exchanger of 1/2 psi at design conditions

c. Maximum water side pressure loss of 1/4 psi at design conditions

d. Minimum internal volume of 500 in.^{3} (approximately 10 times the compressor displacement) to absorb compressor discharge and expander inlet pulsations.

The dimensions of the heat exchanger core are summarized in Table 6-4. The basic performance features that result from this sizing are:

- In providing sufficient internal volume for absorbing compressor/expander pulsations, the air flow cross sectional area is sufficiently increased so that an air side pressure loss of less than 1/4 psi is expected

Table 6-4. Air-Water Heat Exchanger Core Dimensions

Parameter	Value
Number of Rows	6
Coil O.D.	7 7/8 inch
Coil I.D.	6 1/8 inch
Length of Coil	5 1/2 inch

- The heat recovery effectiveness will be in excess of 85 percent

- Water side pressure loss will be less than 1/10 psi at design flow, approximately 2 psi when making up stand-by losses.

6.1.6 Final Design of Heat Exchanger

The prototype air-water heat exchanger assembly is illustrated in Figure 6-9. It consists of a pressurized stainless steel shell, the finned tube coil heat exchanger core described in 6.1.1, and an insulated can inside the finned tube coil. The insulated can forces the air flow through the finned tube coil and minimizes the heat conduction loss between the hot inlet air and the cool exhaust air. The shell is built in two halves, each with an air inlet or outlet welded in. A circumferential weld joins the two halves when the heat exchanger core is assembled. The open space between adjacent rows of finned tube on the inner and outer walls is packed with cotton sash cord. This serves to force the air flow over a greater proportion of the fin area, increasing the heat transfer capacity of the heat exchanger. The air inlet and outlet are oriented

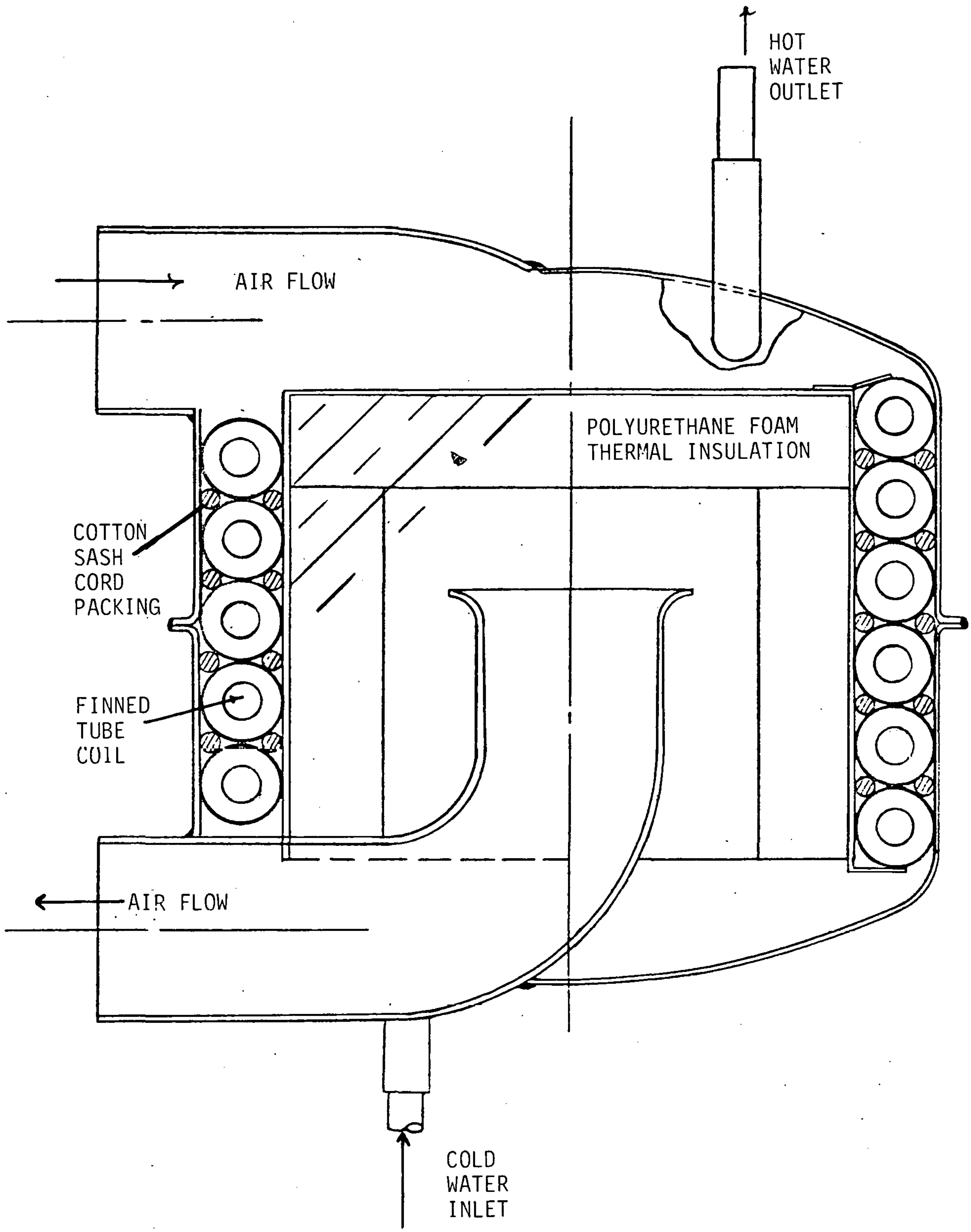


Figure 6-9. Prototype Air-Water Heat Exchanger Assembly

radially and are aligned with the compressor air outlet and the expander air inlet. The water inlet and outlet are axially aligned, primarily to accomodate assembly of the two shell halves. The water inlet and outlet tubes are welded to the shell after assembly of the shell halves to seal between each tube and the hole in the shell that the tube passes through.

This method of assembly proved to be the most expedient for construction of the laboratory prototype. Without changing the geometry of the heat exchanger, the following changes will be considered for the field test model and subsequent production model.

a. Internally glass-lined, externally painted carbon steel shell (instead of stainless steel) to reduce materials cost

b. Sealing of the water inlet and outlet tubes to the shell with compression fittings (instead of welding), reducing assembly cost and allowing disassembly of the heat exchanger

c. Joining of the two shell halves by a V-retainer clamp (instead of welding) the joint would now be gasketed, but the use of this type of joint allows diassambly

d. Use of thinner wall stainless steel tubing for the heat exchanger core, saving material cost

6.2 Thermostat Design

The thermostat regulates the water flow rate through the air-water heat exchanger so that a constant outlet temperature is maintained under vairable conditions. As the water inlet temperature or the air inlet temperature to the heat exchanger falls, the water flow rate is reduced to compensate. As the water inlet temperature or air inlet temperature rises, the water flow rate is increased to compensate.

The control method selected must meet several requirements

1. Provide stable and reasonably accurate control (± 5^{O}F) of the outlet water temperature from the heat exchanger

2. No toxic or corrosion prone material should be in contact with the water

3. 150 psig working pressure

4. Low cost, subject to the above requirements.

A conventional hot water mixing valve meets all of these requirements. This type of valve is illustrated in Figure 6-7. It is a three port valve - a hot water inlet, a cold water inlet, and a controlled temperature water outlet. A bi-metallic element positions two pressure balanced valve seats, one controlling the inlet cold water flow, the other controlling the inlet hot water flow. To serve the normal hot-cold mixing function the cold inlet is throttled on falling outlet temperature. Thus, as used in the hot water heater, the heated water from the air-water heat exhanger enters the cold water port, the hot water port is capped, and water discharges through the outlet port and is returned to the tank.

The temperature setting is adjustable by means of a dial that sets a bias position of the bi-metallic element.

For the production model, a cost saving is possible by redesigning the valve to a two port configuration. This possibility will be discussed with mixing valve manufacturers in preparation for the Phase II field testing.

6.3 Water Circulation Pump Design

The water circulation pump forces cool water from the bottom of the storage tank through the motor cooling jacket, the air-water heat exchanger, and the thermostat. The performance requirements of the circulation pump are summarized in Table 6-5.

A small centrifugal pump will meet these requirements. A model designed for high efficiency in this flow and head range is desireable to minimize electric power consumption. Cost and power consumption are minimized by direct driving the pump from the compressor/expander drive motor.

Table 6-5. Water Circulation Pump Performance Requirements

Parameter	Value
Flow Rate	0.15 gpm - 1 gpm
Head	10 feet
Maximum Operating Temperature	$180^{o}F$
Maximum Operating Pressure	150 psig

For the laboratory system an off-the-shelf motor driven centrifugal circulation pump meeting these requirements was purchased.

The use of a pump with a seprate drive motor for laboratory testing is an expedient, allowing effort to be concentrated on the development of the compressor/expander.

For the production water heater, a pump directly driven by the compressor/expander motor or crank shaft will be selected and designed into the system. Manufacturing design decisions will be made with regard to the following options:

a. Pump driven by second motor shaft or shaft from compressor/expander

b. Pump close coupled to motor or compressor/expander or driven by flexible shaft for vibration isolation

c. Seal-less magnetic drive or mechanical shaft seal for pump

d. Separate pump impeller housing or pump housing integral with compressor/expander crank case.

The basic consideration in making these choices is providing a sufficient level of reliability at minimum cost. For example, close coupling the pump to the compressor/expander drive, but a mechanical shaft seal may not survive the vibration levels expected.

6.4 Motor Cooling Jacket Design

The motor cooling jacket recovers the waste heat from the compressor/expander drive motor, increasing the COP of the heat pump. On the average, the water cooled motor will run cooler than an air cooled motor, increasing motor life. The heat load is both small (approximately 700 BTU/hr) and easy to handle because the water heat transfer coefficient is high. The primary requirement of the cooling jacket is that the potable hot water not be contaminated when passing through the jacket.

The simplest cooling jacket consists of a helical coil of 3/8 inch diameter soft, flattened, copper tubing wrapped around the motor housing and soldered to provide a mechanical and thermal bond. Two loops of tubing are sufficient to cool the motor. This arrangements is illustrated in Figure 6-10.

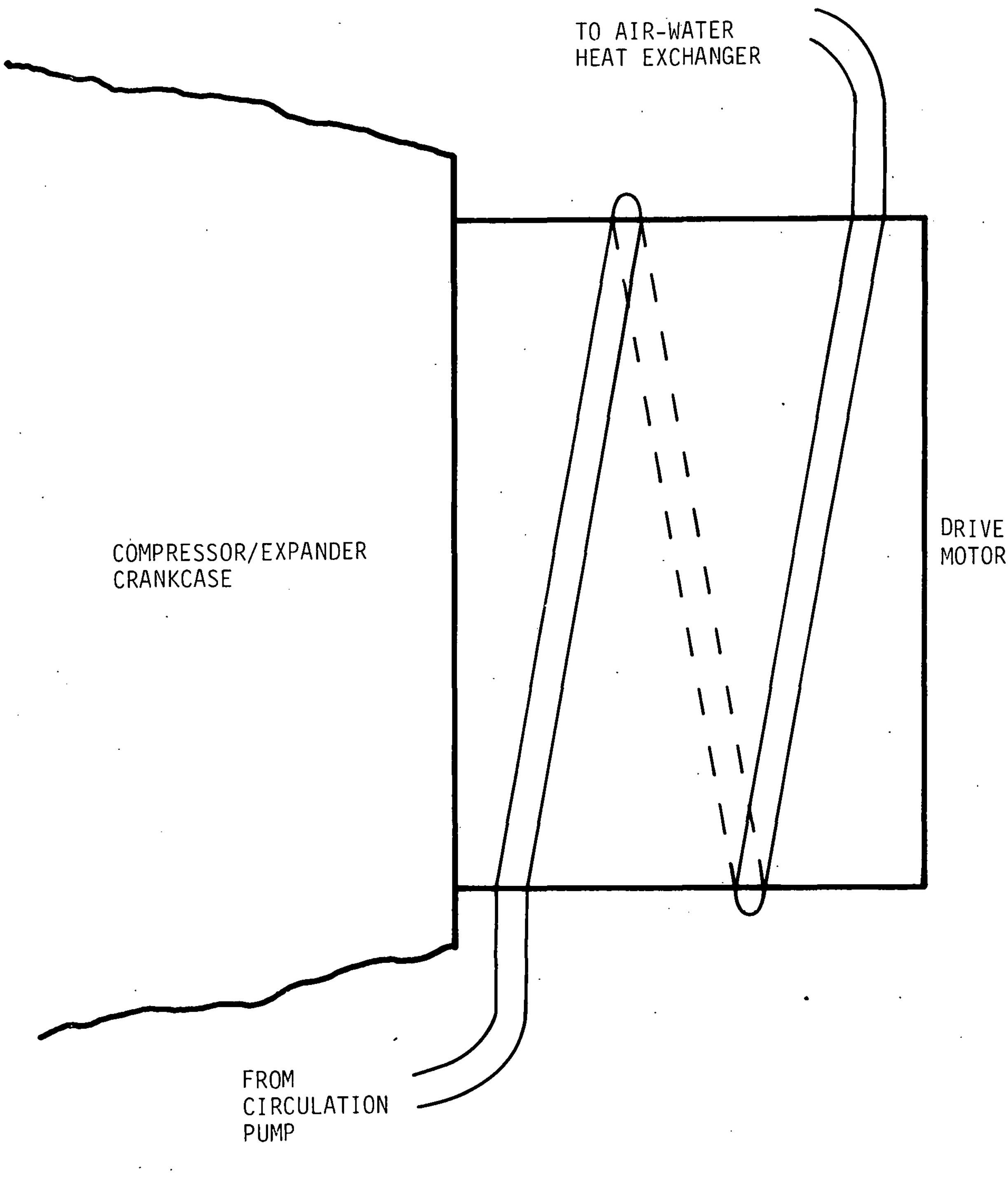


Figure 6-10. Motor Cooling Jacket

7. SYSTEM LAYOUT AND SPECIFICATION

7.1 Final System Layout

In Figure 7-1 is presented a detailed drawing of the compressor/expander system. In Figure 7-2 is presented the detailed drawing of the heat exchanger and in Figure 7-3 is presented the detailed drawing of the complete high efficiency water heater system.

7.2 Detailed Engineering Drawings

Detailed drawings of all the parts comprising the compressor/expander, heat exchanger, and fluid management piping and controls have been prepared.

7.3 Parts List

In Table 7-1 is presented the parts list for the Compressor/Expander Assembly and in Table 7-2 is presented the parts list for the heat exchanger assembly and support frame.

7.4 Final List of Projected Performance Calculations

Final Projected Performance calculations for the high efficiency water heater were determined. It was predicted that the compressor/expander isentropic efficiency will be 87 percent, the heat exchanger effectiveness will be 85 percent and the motor efficiency will be 80 percent. The resulting system COP is predicted to be 1.7. A summary of the heat pump system losses are presented in Table 7-3. The largest single contribution to the total loss is the compressor/expander losses which account for 53 to 65 percent of the total loss. The electrical motor losses account for 22 to 30 percent of the total loss.

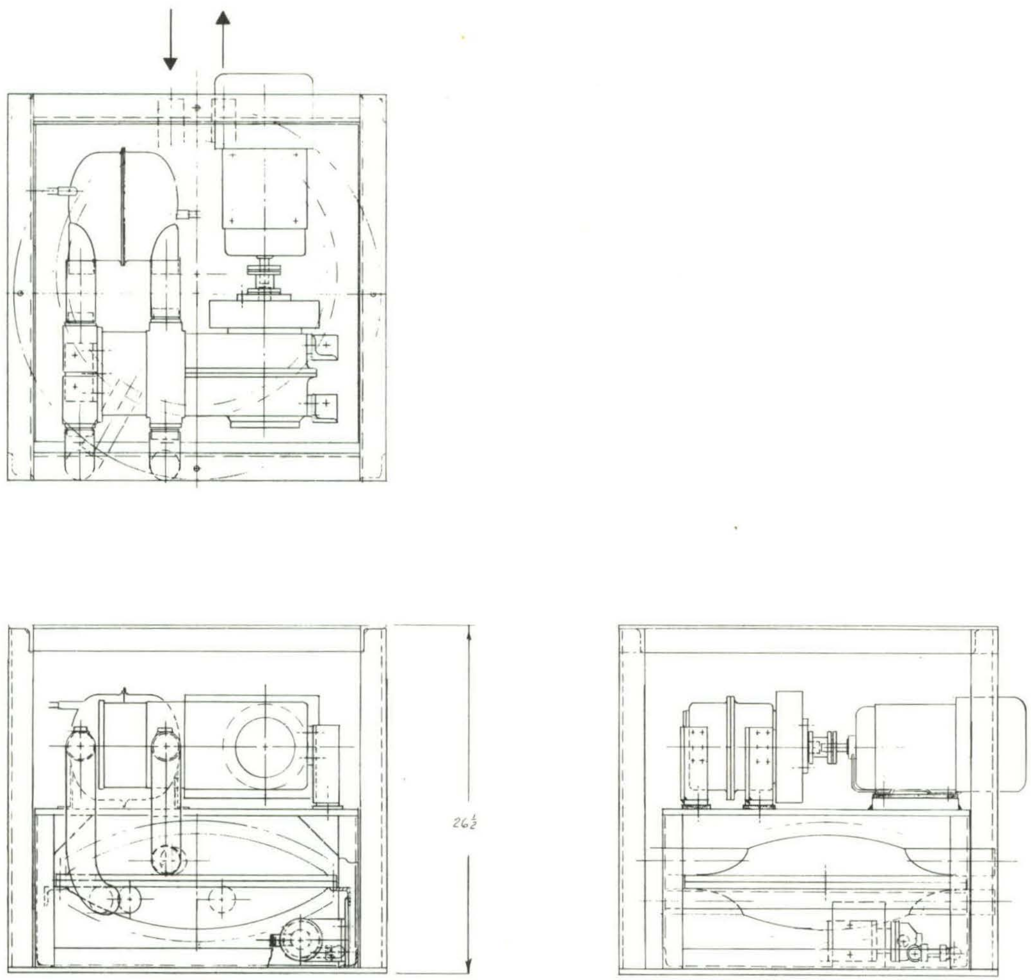


Figure 7-1. Detailed Drawing of the Compressor/Expander System

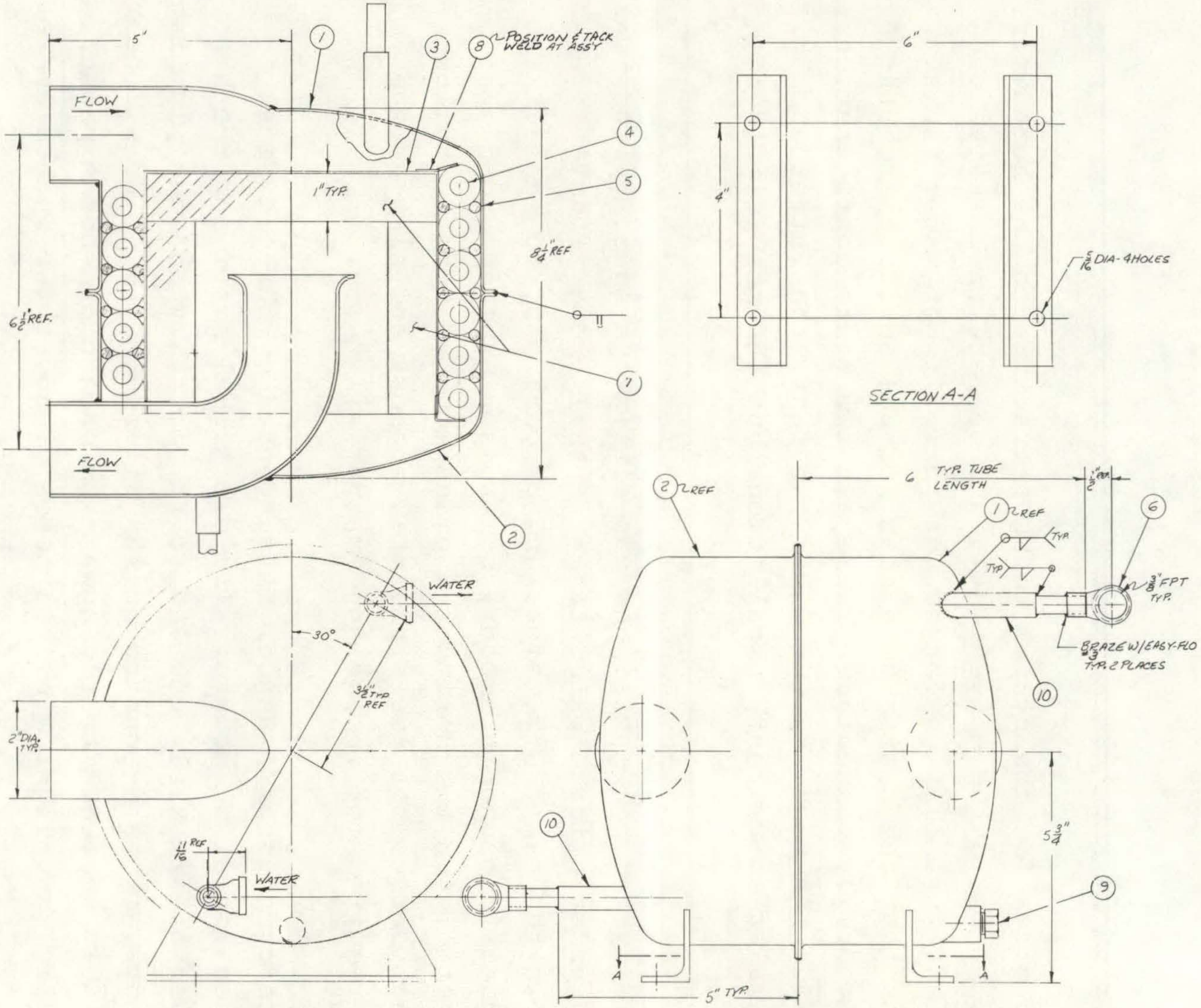


Figure 7-2. Detailed Drawing of the Heat Exchanger

The ideal/calculated systems operating conditions for the case of an ambient temperature of 70°F are presented in Table 7-4.

The predicted PV Diagrams for the compressor and expander are shown in Figures 7-4 and 7-5, respectively.

7.5 Packaging and Installation of the System

In order for the system to be attractive to the homeowner, it must be installed conveniently and operate unobtrusively. Convenient installation for new or retrofit hot water tanks dictates that the system be a neat package separate from the tank but easily connected. There are two basic configurations: one beside the tank and one under the tank, the latter appearing most similar to current installations but the most difficult to design.

The system now envisioned can fit into a squat cylindrical package under a large water tank of usual proportions. Figure 7-6 shows the basic dimensions and proportions, to scale. The three major components, motor, compressor/expander, and heat exchanger lie in the horizontal plane mounted upon a shallow drum that serves both as structural support and also as the volume required for intake and exhaust silencing.

The compressor/expander is suspended by three typical vibration isolation mounts. The electrical motor and the heat exchanger are flexibly coupled to it, but they and the plumbing components are hard-mounted on the silencer/base. The silencer is flexibly ducted directly upward to the compressor/expander. Externally, two light flexible hoses of about two inch diameter connect the system to and from the outside air.

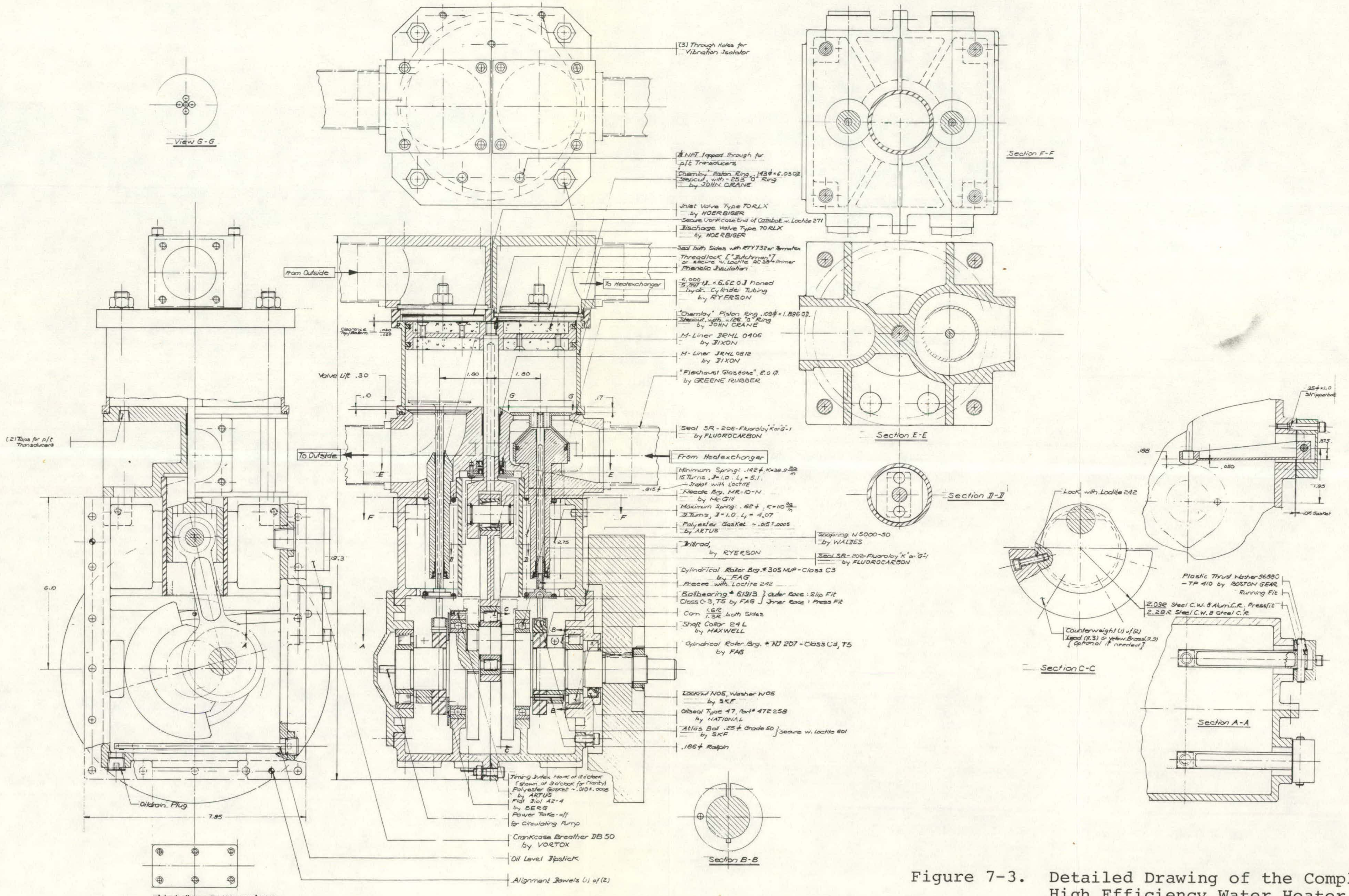


Figure 7-3. Detailed Drawing of the Complete High Efficiency Water Heater System

Table 7-1. Parts List for the Compressor/Expander Assembly

Item	Part No.	Description	Quantity
D	7714-01001	Casting-Crankcase Housing	1
D	7714-01002	Machining-Crankcase Housing	2
D	7714-01003	Casting-Valve Housing	1
D	7714-01003	Machining-Valve Housing	1
C	7714-01005	Crankshaft-(Flywheel Side)	1
C	7714-01006	Crankshaft (Left)	1
C	7714-01007	Flywheel	1
A	7714-01008	Bushing-Crankshaft	1
A	7714-01009	Shaft Collar, Modified	1
A	7714-01010	Dial	1
C	7714-01011	Bearing End Cap Flywheel Side	1
C	7714-01012	Bearing End Cap - Left	1
B	7714-01013	Counterweight	2
A	7714-01014	Blank Cam	2
A	7714-01015	Inlet Cam	1
C	7714-01016	Valve Head, Solid	1
A	7714-01017	Exhaust Cam	1
A	7714-01018	Cam Follower	2
B	7714-01019	Cam Follower Arm	2
B	7714-01020	Cam Follower Pivot Block	2
A	7714-01021	Push Rod Assembly	2
A	7714-01022	Dipstick-Oil Level	1
A	7714-01023	Piston Rod Seal Retainer	1
A	7714-01024	Piston Rod Seal Plate	1
A	7714-01025	Pin Connecting Rod	1
B	7714-01026	Valve Stem (Inlet)	1
B	7714-01027	Valve Head (Inlet)	1
B	7714-01028	Valve Exhaust	1
B	7714-01029	Valve Guide-Exhaust	1
B	7714-01030	Pushrod Pivot Plate	1
C	7714-01031	Valve Guide (Inlet)	1

Table 7-1. Continued

Item	Part No.	Description	Quantity
B	7714-01032	Piston Sleeve	1
C	7714-01033	Connecting Rod	1
C	7714-01034	Air Port (Inlet)	1
C	7714-01035	Air Port (Outlet)	1
C	7714-01036	Piston Head	1
C	7714-01037	Cross Head	1
C	7714-01038	Piston Cylinder	1
C	7714-01039	Piston Phenolic Insulation	2
B	7714-01040	Crosshead-Etch	1
B	7714-01041	Spring - Max.	1
B	7714-01042	Spring - Min.	1
B	7714-01043	Cover-Cam Assembly	2
B	7714-01044	Inlet Valve Assembly	1
B	7714-01045	Inlet Valve Grinding	1
B	7714-01046	Exhaust Valve Grinding	1
B	7714-01047	Gasket-Crankcase Housing	1
C	7714-01048	Clamping Plate	1
B	7714-01049	Hose Adapter Flange	2
A	7714-01050	Timing Point	1
B	7714-01051	Tie Bolt	4
A	7714-01052	Washer Spring, Retainer	2
A	7714-01053	Stud Spring, Retainer	2
D	7714-01054	Welded, Crankcase	2

Table 7-2. Parts List for the Heat Exchanger Assembly and the Support Assembly

Item	Part No.	Description	Quantity
D	7714-02000	Heat Exchanger Assembly	1
C	7714-02001	Coil Detail	2
C	7714-02002	Core Detail	1
D	7714-02003	Shell - Inlet Side	1
D	7714-02004	Shell - Exhaust Side	1
C	7714-02005	Bracket Detail	2
B	7714-02006	Spunhead Detail	2
B	7714-02007	Inlet Nozzle	1
B	7714-02008	Exhaust Nozzle	1
D	7714-03000	Heat Pump Assembly	1
D	7714-03001	Tank Support Frame	1
B	7714-03003	Bracket Detail Comp. Mounting	1
C	7714-03004	Bracket Detail Comp. Mounting	2

Table 7-3. Summary of Heat Pump System Losses

Component/Type	Watts	Percent of Total
Heat Exchanger (ϵ = 0.85)		
Pressure Drop Losses	4	
Temperature Differences Losses	100	
Heat Leak Losses	0	
Subtotal	104	17.5 to 13.1
Compressor/Expander (η_s = 87%)		
Valve Losses		
• Valve Area	59.6	
• Time of Opening	82.0	
• Valve Leakage	37.3	
Internal Head Transfer Loss	37.3	
Mechanical Losses		
• Piston Ring(s)	14.9 to 82.0	
• Crosshead Drive Mechanism	14.9 to 59.6	
• Rod and Main Bearings	37.3 to 74.6	
• Valve Drive Mechanism	29.8 to 82.0	
Subtotal	313.1 to 514.4	52.6 to 64.6
Motor (η = 80%)		
Friction and Windage Losses	14.8	
Core Losses	24.5	
Load Losses	33.7	
I^2R Losses	104.7	
Subtotal	177.7	29.9 to 22.3
Total Heat Pump Losses	594.8 to 796 watts	100%

Table 7-4. Ideal Operating Conditions

Compressor	
Inlet Pressure	14.7 psia
Inlet Temperature	70°F
Exit Pressure	45 psia
Exit Temperature	300°F
Air Mass Flow Rate	37 scfm
Speed	1150 rpm
Expander	
Inlet Pressure	45 psia
Inlet Temperature	70°F
Exit Pressure	14.7 psia
Exit Temperature	-69.5°F
Heat Exchanger	
Air Inlet Temperature	300°F
Air Inlet Pressure	45 psia
Pressure Drop	Less than 0.1 psia
Air Outlet Temperature	70°F
Water Inlet Temperature	40°F
Water Inlet Pressure	70 psia
Pressure Drop	Less than 2 psia
Water Outlet Temperature	140°F
Water Mass Flow Rate	0.17 gal/min
Electrical Motor	
Current	6.78 amps
Voltage	230 volts
Power Factor	0.871
Speed	1150

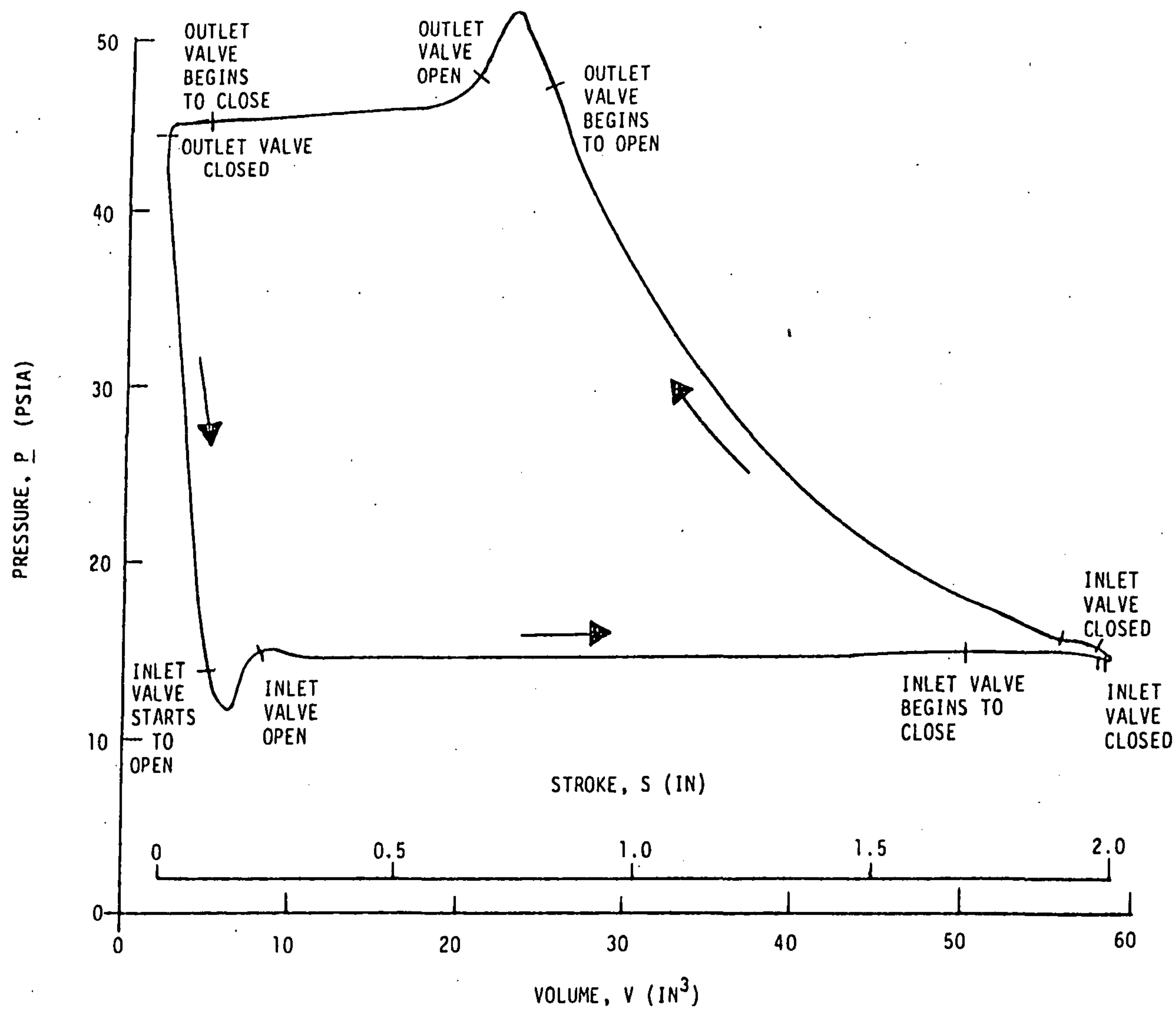


Figure 7-4. Predicted Compressor P-V Diagram

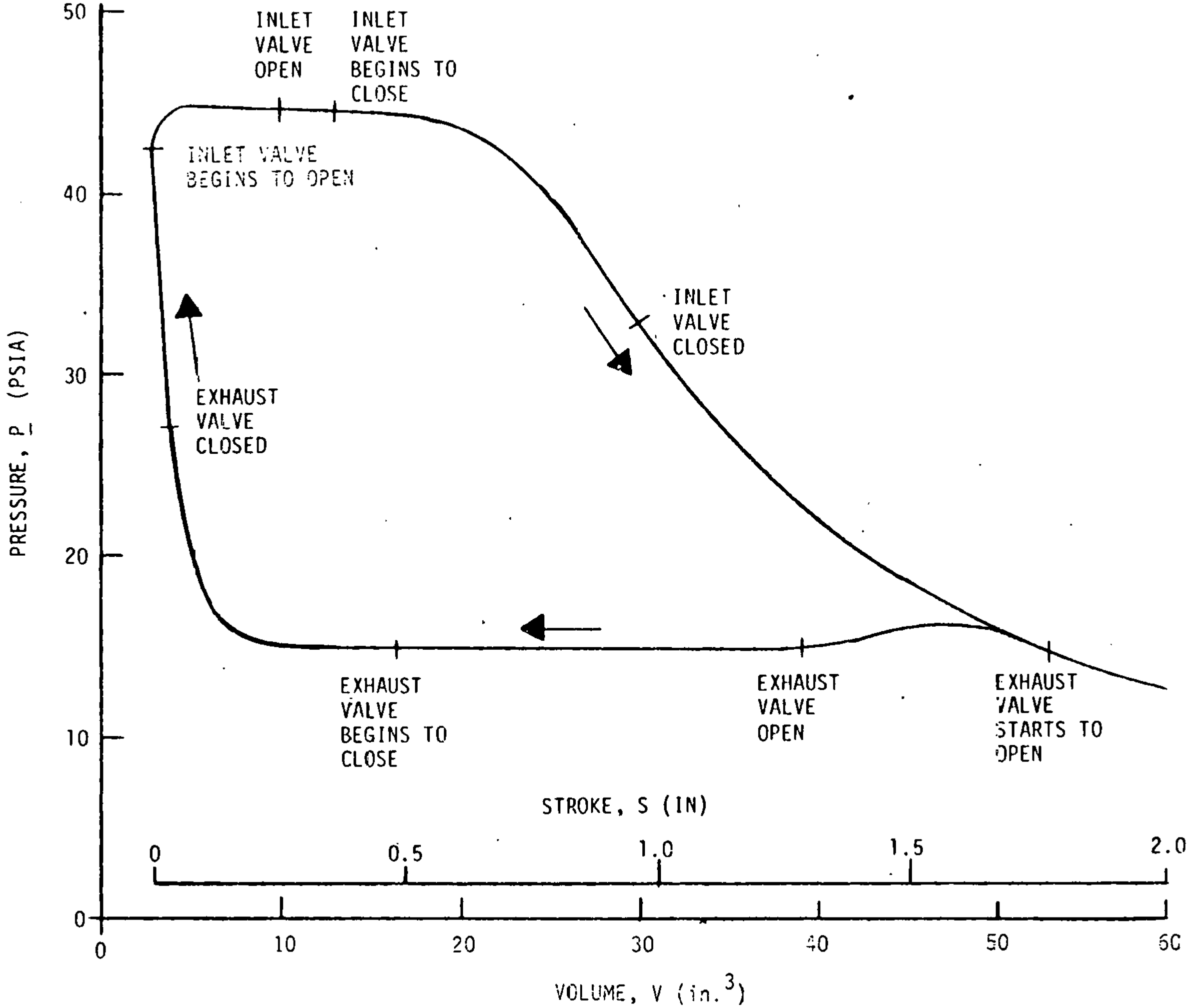


Figure 7-5. Predicted Expander P-V Diagram

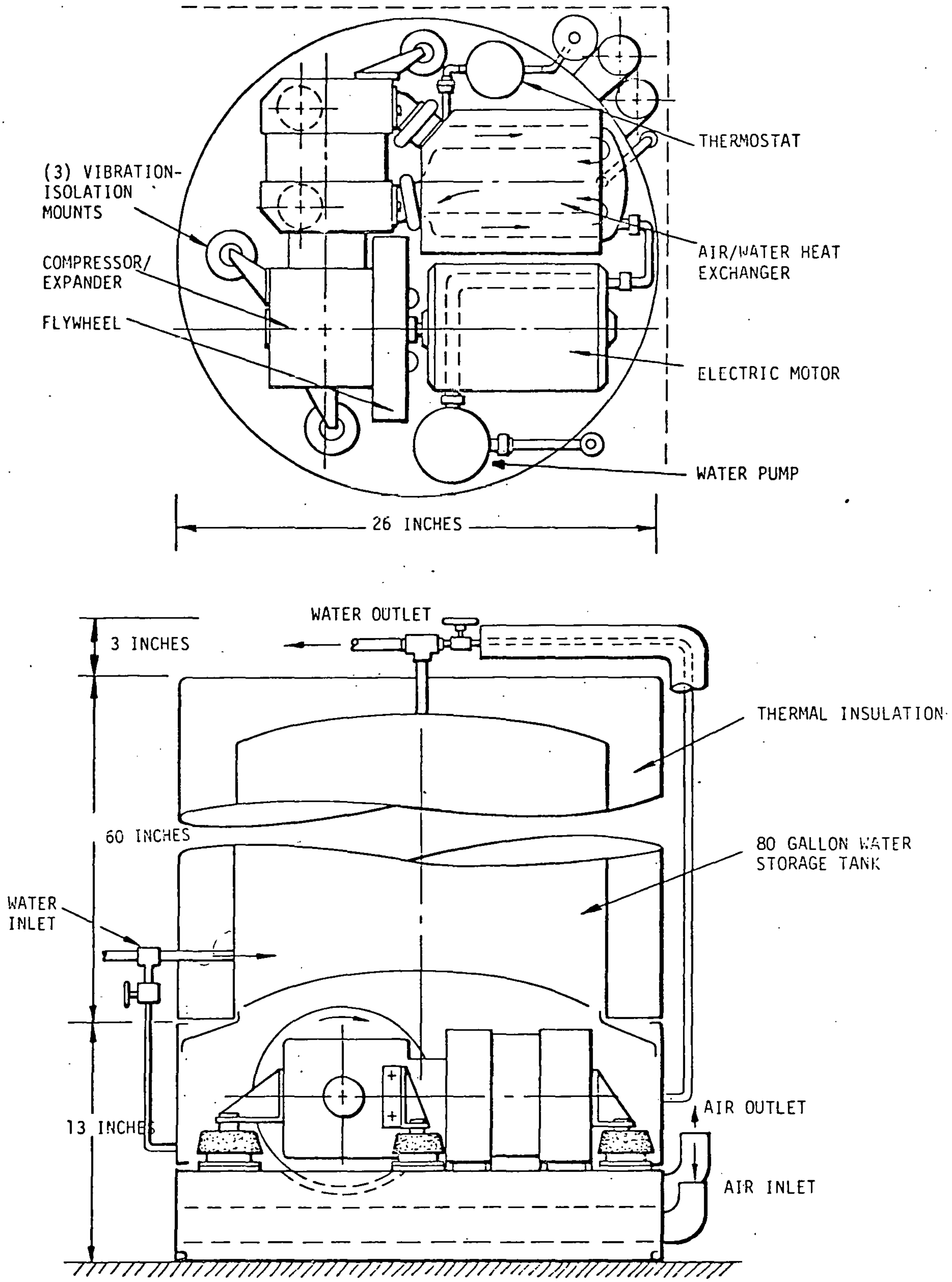


Figure 7-6. Schematic of the Brayton Cycle Heat Pump Water Heater System

Plumbing of the system is simple and fail-safe, with shut-off valves to isolate the heat pump unit if necessary. Supply water flows directly to the tank and out of the tank upon demand, and there is resistance heating in the top zone for rapid recovery, or use during heat pump shutdown. The heat pump loop is in parallel with demand flow. Water is drawn from the bottom of the tank by a very small, sealed or potted, circulating pump (fractional gpm). From the pump the water goes through the heat exchanger through a peak-temperature-regulating thermostat and then up to the top of the tank through an external, insulated pipe. Ultimately, if desired, a special motor can be plumbed so that its thermal losses serve to heat the circulating water.

On-off thermostats mounted in the tank, near top and near bottom, are wired to controls in the base. One last connection is required: the water condensate drain from the cool end of the heat exchanger. The water is under air pressure, so a small float-trap is needed, but the drain line can be very small in diameter and led up to any convenient drain or outside.

8. TEST PLAN

In order to refine and document the performance of the components and the system as a whole, extensive testing is planned. The thermodynamic, thermal, electrical and mechanical behavior of the components and system will be determined and compared with predicted behavior. This will require measurements of mass flows, temperatures, pressures, voltages, currents, and mechanical tolerances and parameters at key points in the components and detailed analysis of these measurements. The tests will be performed in three stages.

- Component development testing
- System development testing
- System endurance testing.

8.1 Component Development Tests

The main purpose of the component level tests is to debug the components by comparing measurements with theoretically predicted values. In addition, the actual behavior of the components can be documented to assist in understanding the system level tests. The test plans for the components are broken down into tests for the heat exchanger, compressor/expander and motor.

8.1.1 Heat Exchanger

8.1.1.1 Purpose

Heat exchanger tests will be conducted under steady-state flow conditions in stand-alone mode to determine the correlations between experimental and design values, pressure drops, heat transfer coefficients and heat exchanger effectiveness. In the event that heat exchanger performance is worse than predicted a combination of analysis and detailed local temperature measurements will be used to determine the correct course of action to correct the performance. The exchanger will then be mated to the compressor/expander and the tests will be repeated to verify the non-steady-state flow performance.

8.1.1.2 Parameters to be Measured

Figure 8-1 presents a schematic of the heat exchanger and pressure, temperature, and mass flow measurements for both the air and water sides of the heat exchanger. On the air side the parameters to be measured are the inlet pressure, inlet temperature, and relative humidity, mass flow, differential inlet/exit pressure, and exit temperature. On the water side, inlet temperature, mass flow, differential inlet/exit pressure and exit temperature will be measured.

8.1.1.3 Instrumentation and Test Equipment

The basic instrumentation for this state of testing includes four thermocouples with recorder, a 50 scfm Rotameter for air, a 0.5 gal/min Rotameter for water, a 50 psig pressure gauge for inlet air measurement, and two differential pressure gauges.

In addition, special test equipment is required to simulate actual operating conditions and to perform separate air and water side tests. When the heat exchanger is mated to the compressor/expander, the air entering the heat exchanger will have a pressure of about 45 psia and temperatures between 170°F and 300°F. To reproduce these conditions in the stand alone tests, a compressed air source and electric heaters will

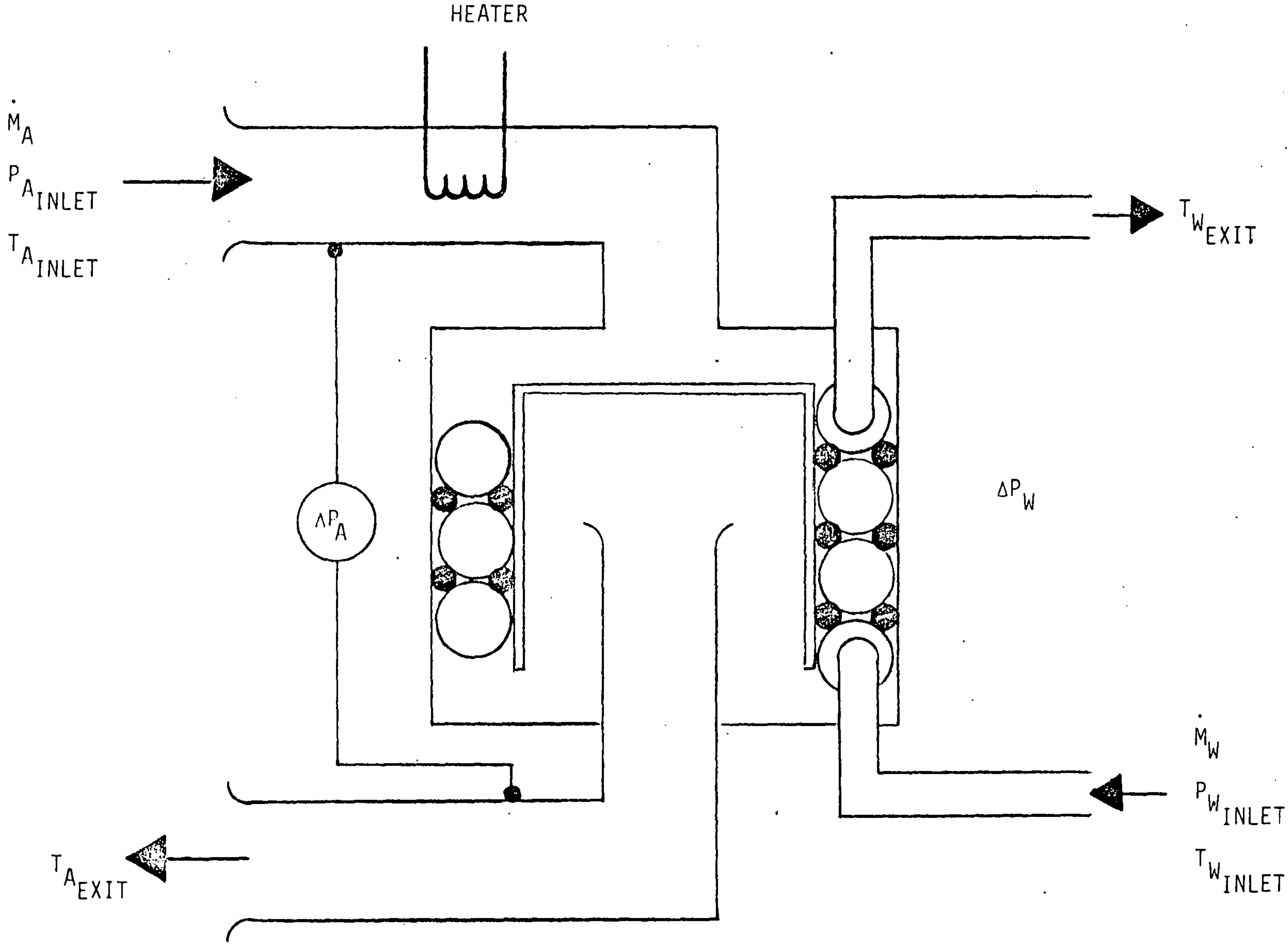


Figure 8-1. Heat Exchanger Tests

be used. When the heat exchanger is coupled with the compressor/ expander, the compressed air source and heaters will not be needed.

8.1.1.4 Testing Protocol

Stand Alone Testing

The air inlet side of the heat exchanger will be attached to a source of compressed air that is electrically heated. Tests will be conducted at design point air and water flows to determine the overall steady-state flow rating and at high water flow rates to determine the air side rating.

The overall rating will be performed using temperatures and mass flows representative of actual heater operating conditions. The 0.5 gal/min Rotameter will be used to measure water flow. Flow rates, pressures, and inlet heater rates will be adjusted to the desired settings with the inlet pressure equal to 45 psia and flow set at outlet valve. The equipment will be allowed to reach equilibrium state with the heated water being discharged. Measurements of temperatures, flow rates, and pressures will then be recorded. This experiment will be repeated to cover the range of actual heater operating conditions. Proposed testing conditions are shown in Table 8-1.

The air side rating tests will require high water flow rates, with the 5 gal/min Rotameter replacing the 0.5 gal/min unit. Otherwise, the test will be performed as described in the previous paragraph.

Table 8-1. Proposed Testing Conditions

Test Number	T_{air} (^{o}F)	T_w (^{o}F)	$\dot{M}_a$ (scfm)	$\dot{M}_w$ (gal/min)
1	300	50	40	0.5
2	300	50	30	0.5
3	250	40	40	0.5
4	250	40	30	0.5
5	200	30	45	0.5
6	200	30	35	0.5

Compressor/Expander Coupled Testing

These tests will be conducted to determine the effects of unsteady air flow in capacity, heat transfer, and pressure drops. The heat exchanger will be mated to the compressor/expander and tests will be run according to the procedure described above.

8.1.1.5 Data Analysis

The heat exchanger data will be analyzed to determine pressure drops, heat transfer coefficients, Number of Transfer Units (NTU), and heat exchanger effectiveness. The pressure drops for a set of operating conditions can be read directly from the differential pressure gauges. These results can be scaled to other operating conditions through the relation

$$\Delta p \propto f\rho V_{el}^{2} \tag{8.1}$$

where f, the friction factor, is a function of the Reynold's number.

The overall heat transfer coefficient, U, can be determined from the relation

$$U = Q/A\Delta T_m \tag{8.2}$$

where

$$\Delta T_m = \frac{(T_{ah} - T_{wh}) - (T_{ac} - T_{wc})}{\text{In}\left(\dfrac{T_{ah} - T_{wh}}{T_{ac} - T_{wc}}\right)} \tag{8.3}$$

Q is the overall heat rate, A is a consistently defined surface area for heat transfer and ΔT_m is the logarithmic mean temperature difference. By conducting experiments at significantly different water flow rates, the air side and water flow rates can be separated by

$$UA = \frac{1}{1/h_a A_a + 1/h_w A_w} \tag{8.4}$$

The heat transfer conduction path through the tube wall is assumed to be a negligible temperature drop and thus was eliminated from the above equation.

The heat exchanger effectiveness, ε, and the number of exchange heat transfer units (NTU's) are related. For this heat exchanger

$$NTU = UA/(MC_p)_{air} \tag{8.5}$$

The relation for complex geometries is easily read from tabulated results such as those presented in Kays and London [11]. These tabulations will be used in the ultimate determination of ε and NTU.

8.1.2 Compressor/Expander

8.1.2.1 Purpose

Detailed measurements over a variety of operating conditions for air and water temperatures and speed will be taken to debug the unit and to quantify the loss mechanisms and performance. A heat exchanger will remove compressor heating. In particular, the following aspects of the compressor/expander operation and performance will be examined.

- Mechanical losses
- Flow losses (pressure losses)
- Heat transfer effects
- Isentropic and volumetric efficiencies
- Mechanical input and shaft power consumption
- Noise.

These measurements will be compared with computer predictions, as described in Section 4, to evaluate the performance. The prototype compressor/expander design will be sufficiently flexible to allow necessary changes to be made if initial performance is not adequate. Thorough instrumentation will serve to provide adequate information to readily identify the areas where changes are identified.

8.1.2.2 Parameters to be Measured

Figure 8-2 presents a schematic of the compressor/expander with its instrumentation and associated test equipment. The thermodynamic and aerodynamic parameters to be measured for the compressor and the expander are inlet and exit pressure and temperature, inlet relative humidity, pressure characteristics of valve operation, instantaneous cylinder pressures, and mass

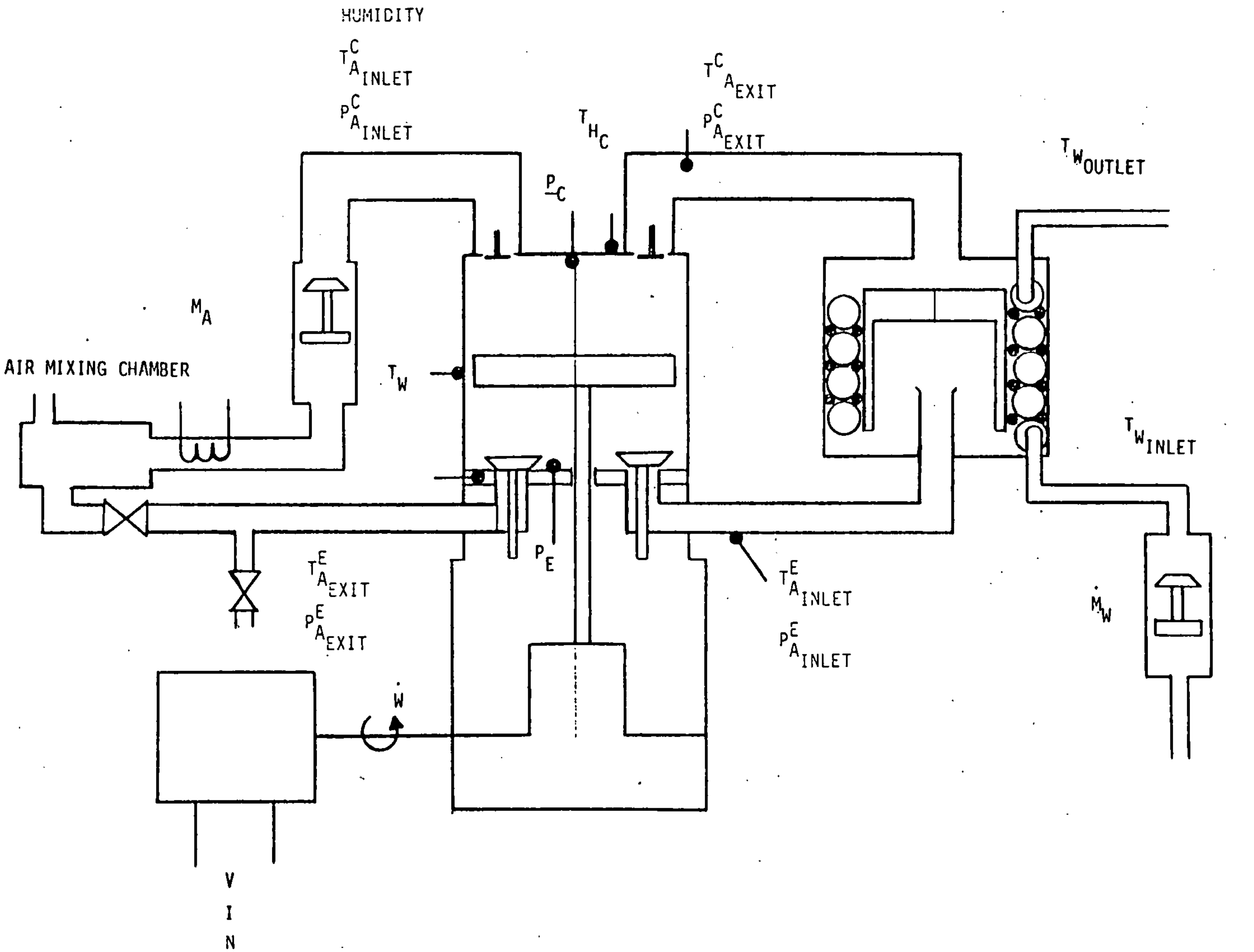


Figure 8-2. Schematic of Equipment and Instrumentation for Compressor/Expander Test

flow. Other temperature measurements will be taken at the cylinder heads and walls to determine the thermal performance of the unit. Mechanical losses, valve timing, and compressor/expander speed will be measured. Finally, noise measurements will be taken unmuffled to provide baseline information that will be used to determine muffler effectiveness.

8.1.2.3 Instrumentation and Test Equipment

Instantaneous cylinder pressures will be measured with two 50 psig, 4 kHz pressure transducers. The instantaneous pressure characteristics of valve operation will be measured by placing another transducer on the other side of the valve being studied, either 50 psig for the high pressure side or 5 psig for the ambient pressure side. The pressure traces will be photographed from an oscilliscope. Thermocouples will be placed in the heads and on the cylinder wall to determine the effects of heat transfer on compressor/expander performance. Other thermocouples will be placed in the inlet and exit manifolds to measure the amount of heat delivered and will be used to determine capacity and COP. A twelve channel chart recorder will record the temperatures. Compressor inlet relative humidity will be measured. A 50 scfm Rotameter will measure inlet mass flow. The drive motor will be calibrated so that torque can be determined from ammeter and tachometer readings (see Section 8.1.3). Noise will be measured with a dB(A) meter.

Other equipment will be required to test the unit over a range of expected operational ambient temperatures. For simulated ambient temperatures higher than test room temperature, electric resistance heaters will be used. Water injection will be used to simulate high humidity conditions. For simulated ambient temperatures colder than test room temperature, some of the expander exhaust air will be injected into the mixing chamber that will be constructed for the tests.

If the heat exchanger is not ready when the compressor/expander is ready for testing, a simple heat exchanger will be assembled to remove heat before the expansion cycle. This heat removal is necessary to determine how thermal behavior affects the unit performance.

8.1.2.4 Testing Protocol

The compressor/expander will be assembled with the drive motor, a heat exchanger, and the necessary instrumentation and test equipment. The initial debugging tests will be performed using ambient room air. The first tests will measure the instantaneous cyclinder pressures by photographing the oscilliscope traces. A p-V curve for the cylinder can be produced from the pressure readings (see Section 8.1.2.6). By comparing this p-V curve with computer projections, problems related to ring or valve leakage and valve operation can be determined. If one or more valves are operating poorly, detailed pressure drop measurements will be made across the offending valve(s). This information will then be used to improve valve operation and/or cylinder sealing to the extent practical. When pressure measurements indicate that the rings and valves are operating acceptably, more extensive debugging testing will begin. These tests will also be conducted at ambient room temperature, but temperatures and mass flows will be recorded. This data will be used to determine the effects of heat transfer in the compressor/expander and the values of isentropic and volumetric efficiencies. If necessary, insulation will be used to block unwanted heat transfer to the extent practical. The final debugging test will be to measure the frictional losses with the valves in place, but with air allowed to enter and leave the cylinder through the pressure transducer ports. While this is not a realistic measurement of actual operating friction losses, experience has shown that the approximation is a fairly good one [9]. If the friction losses are abnormally high, steps will be taken to determine the cause of the problem.

The next stage of the development testing will involve measuring temperatures and mass flows over the range of conditions expected in actual operation. This will require varying air inlet temperature and relative humidity and water inlet temperature. All temperature measurement points shown in Figure 8-2 will be recorded, as will mass flows and the average pressure in the air side of the heat exchanger. The measurements from these tests will also be used in determining heat exchanger performance. The effect of machine speed will also be tested. Table 8-2 lists the test conditions that will be used in this stage of testing. The "ambient" temperature will be set by adjusting valves in the mixing chamber or by adjusting the amount of electric heating. The relative humidity will be adjusted by varying the amount of the water injected.

Table 8-2. Compressor/Expander Tests

Test Number	T_{amb} (°F)	RH_{amb} (approx) (%)	T_{H_2O} (°F)	Speed (rpm)
1	0	20	40	1200
2	0	80	40	1200
3	20	20	40	1200
4	20	80	40	1200
5	40	20	50	1200
6	40	20	50	1800
7	40	80	50	1200
8	60	20	50	1200
9	60	80	50	1200
10	80	20	60	1200
11	80	20	60	1800
12	80	80	60	1200

Noise measurements will be taken during some tests. This data will be used later to refine muffler design if necessary.

8.1.2.5 Analysis

The oscilloscope photograph will provide a trace of pressure versus time. The volume versus time for the cylinder is known from geometrical analysis. The two will be combined to form the cylinder p-V diagram. This diagram will be inspected for anomalies.

Volumetric efficiency is the ratio of air drawn into the cylinder to the swept volume of the cylinder. That is,

$$\eta_v = \frac{\dot{M}_a}{NV_c \, \rho_a} \tag{8.7}$$

This value of η_v will be compared with the computer prediction.

The isentropic efficiencies will first be determined assuming η_c equals η_e and the frictional losses are divided between the compressor and the expander in proportion to the work done by each. For the compressor,

$$W_{in} = W_{sc}/\eta_c + 0.6 \, \ell_f \tag{8.8}$$

For the expander

$$W_{out} = W_{se} \, \eta_e - 0.4 \, \ell_f \tag{8.9}$$

where ℓ_f is the total functional losses.

The net work is then

$$W_{net} = W_{in} - W_{out} \tag{8.10}$$

$$W_{net} = W_{sc}/\eta_c - \eta_e W_{se} + \ell_f \tag{8.11}$$

By assuming $\eta_c = \eta_e = \bar{\eta}$, the expression for average isentropic efficiency is

$$\bar{\eta} = \frac{\left(-W_{net} - \ell_f\right) + \sqrt{\left(W_{net} - \ell_f\right)^2 + 4\, W_{se}\, W_{sc}}}{2\, W_{se}}$$

The computer program described in Section 4 predicts deviations from isentropic operation for the compressor and expander individually. Thus, by combining measured data and the computer calculated values, η_c and η_e can be estimated if it is necessary.

The friction loss will be estimated by comparing the motor speed recorded during the friction test with the motor's speed vs torque curves.

The coefficient of performance for the unit is the ratio of heat delivered to the energy supplied at the motor terminals. For a system COP_{system} based on heat exchanger water outlet,

$$COP_{system} = \frac{M_w c_p \, \Delta T_w}{\dot{E}} \tag{8.12}$$

where ΔT_w is measured between the inlet to the water pump and the heat exchanger water outlet and $\dot{E}$ is the electric energy used. The compressor/expander/heat exchanger COP will be

$$COP_{c/e/he} = COP_{system}/\eta_m \qquad (8.13)$$

The heat pumping capacity will be

$$\dot{Q}_{HP} = \dot{M}_w \; c_p \; \Delta T_w = \dot{M}_w \; c_p \; (TW_4 - TW_1)$$

8.1.3 Motor

8.1.3.1 Purpose

The motor manufacturer will provide curves of torque efficiency and current versus speed. The motor will be run at different loads and the current and speed recorded in order to verify the performance. The motor power factor will be determined.

8.1.3.2 Parameters to be Measured

The current, power, and power factor will be measured at the terminals. Motor speed will be measured at the shaft. The motor temperature will also be monitored.

8.1.3.3 Instrumentation

An ammeter (50A rating) and power factor indicator will be attached at the terminals. The speed will be measured by tachometer. A thermocouple will be placed on the motor to measure operating temperature. A watt-hour meter will be installed for later use.

8.1.3.4 Testing Protocol

The motor will be run at no load and at operating load. The current, power factor, and speed recorded for comparison with the manufacturer's curves. The operating temperature will be recorded for later use.

8.1.3.5 Analysis

As slip in an induction motor increases past the point of maximum torque, the power factor drops and the current increases. By measuring speed, current and power factor, the torque can be inferred based on manufacturer's data. If the value of current at tested speeds agrees with provided data, the motor will be assumed to be operating properly.

8.2 System Development Tests

8.2.1 Purpose

The purpose of the system development tests is threefold. First, any problems with the performance of the system caused by interactions among the components will be identified and corrected. Second, the performance of the system over a variety of operating conditions will be documented. Third, the operation of the control system will be verified. Important aspects of performance to be determined in this phase include, but are not limited to, COPs, heat delivery capacity, efficiencies, power consumption, heat leakage rates, system recovery rate, recovery efficiency, tank thermal stratification, and losses in inlet and exhaust lines.

8.2.2 Parameters to be Measured

A schematic of the system is shown in Figure 8-3, with measurement points identified. The measurements to be taken will be the same as those described in the component test section, with the following exceptions:

- Detailed pressure measurements of the compressor/expander will not be made unless necessary to identify an unexpected problem

- Detailed temperature measurements of the compressor/expander will be eliminated except as above

- Temperature measurements in the water tank will be taken to determine stratification and heat leakage

- Mass flow through the water tank will be instrumented along with inlet and outlet temperatures.

If performance at the system level is worse than predicted from the component tests, more detailed measurements will be taken.

8.2.3 Instrumentation and Test Equipment

The instrumentation described in earlier sections will be used with the exception of the additions and deletions related to the points described in Section 8.2.2. In terms of specific hardware, changes will be:

- Removal of pressure transducers and oscilloscope

- Transfer of thermocouples from compressor to water tank

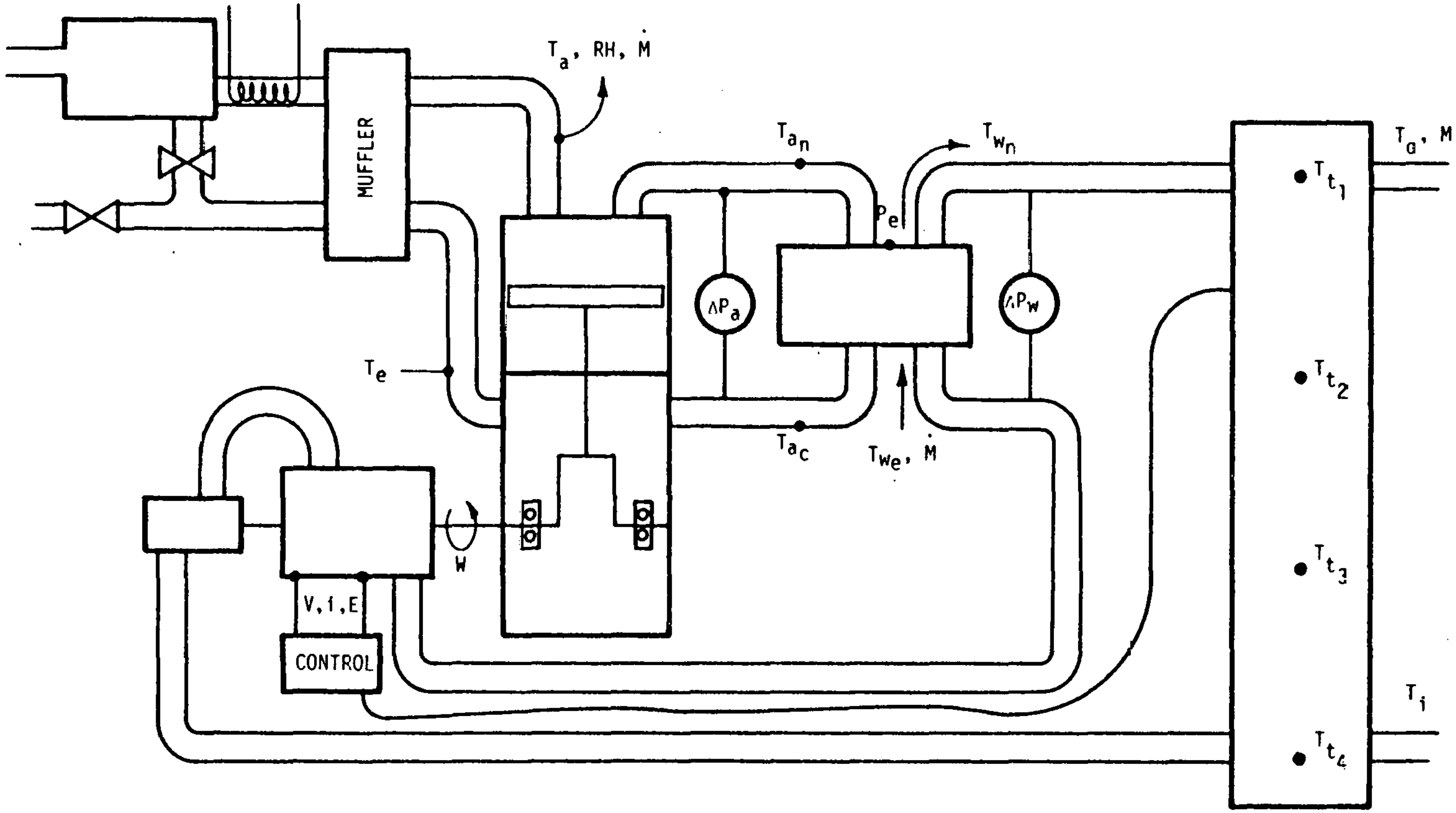


Figure 8-3. Schematic of Instrumentation for System Testing

- Installation of a 5 gal/min Rotameter into the water tank line.

Any instrumentation removed will remain available should problems arise during this state of testing.

8.2.4 Testing Protocol

After assembly of the complete system, it will be operated at ambient room temperature and the results will be compared with ambient room temperature test results taken during the compressor/expander development tests. Any obvious problems caused by system integration will be solved before proceeding with the test plan.

The system will then be operated over a range of ambient conditions to document system capacity, COPs, and efficiencies. The test procedure for setting inlet conditions and taking measurements will be the same as that used during the compressor/expander tests, with the addition of the tank and its instrumentation. Hot water produced during each run will be discharged. The system will be operated at the conditions described in Table 8-2. These tests will provide the measurements necessary to determine COP, capacity, isentropic and volumetric efficiencies, and inlet and exhaust line losses.

The standby loss test will be conducted next. Conditions will be initialized by running the system to cutout, zeroing the watt-hour meters, and recording initial tank centerline and ambient temperature. The system will be allowed to run for at least 48 hours, ending when the system reaches a cutout point. At that time, the following data will be taken:

- Duration of test in hours, t
- Total energy used, E
- Final centerline and ambient temperatures.

The tests to determine recovery efficiency will then be run. With the compressor/expander off, the tank will be filled with water and water will be run through the tank until the mean centerline temperature is about 70^{o}F. The system will then be turned on and run at ambient room temperature to cutout. Total energy used, E, and centerline temperatures will be recorded.

Data taken during the above tests will also be used to calculate recovery rate.

8.2.5 Analysis

The analysis required to determine capacity, COP, and isentropic and volumetric efficiency will be the same as that described in Section 8.1.2.5.

8.3 Endurance Testing

8.3.1 Purpose

The endurance testing phase will be conducted to determine what, if any, parts are subject to premature wear or failure. The endurance testing will simulate approximately one year of use during which time performance will be monitored. Any outright failure or performance degradation due to ring or valve wear, heat exchanger fouling or other causes will be analyzed.

8.3.2 Parameters to be Measured

Before initiation of this testing phase detailed dimensional and tolerance measurements will be made. Heat exchanger air side pressure and inlet temperature, water exit temperature and pressure drops through the heat exchanger will be monitored through the test to indicate performance degradation.

At the middle and end of the test, the system will be disassembled and measurements of parts made. Critical components such as valves will be examined for such failures as cracks.

8.3.3 Instrumentation and Test Equipment

Pressure and temperature measurements will be made with equipment described in earlier sections. Detailed dimensional data will be taken with micrometers and clearance gauges.

To perform the endurance tests, the tank emptying function will be automated. This will require a temperature actuated relay in the water tank and a solenoid valve on the tank outlet. These are shown in Figure 8-4.

8.3.4 Testing Protocol

After the system is checked out and reassembled, it will be started and will be able to operated without attention. Daily checks will be made to ascertain the unit is functioning properly.

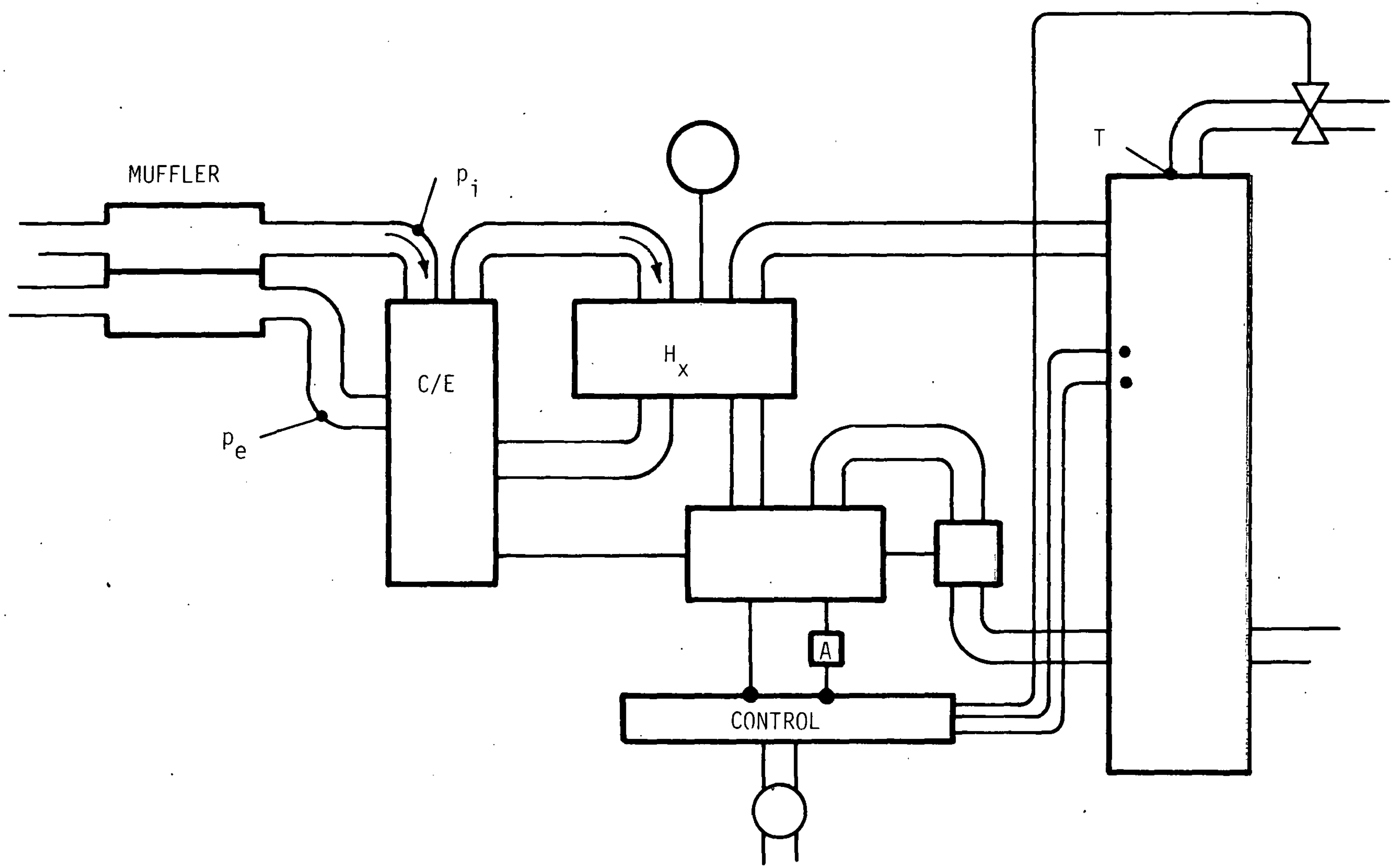


Figure 8-4. Instrumentation and Test Equipment For Endurance Testing

9. CONCLUSIONS

A reverse Brayton air cycle heat pump water heater has been designed for residential applications. The system consists of a compressor/expander, an air-water heat exchanger, an electric motor, a water circulation pump, a thermostat, and fluid management controls.

A thermodynamic parametric analysis was performed on the water heater system. The effects of ambient temperature, air-water heat exchanger effectiveness, pressure ratio, and compressor/expander isentropic efficiency on the coefficient of performance of the complete system were investigated. It was determined that to obtain a coefficient of performance of 1.7 the isentropic efficiency of the compressor/expander system must be at least 85 percent.

An analysis was performed on the compressor/expander in order to quantify the valve losses, heat transfer losses, and mechanical losses. It was determined that the major sources of irreversibility are due to the valves and valve drive mechanism. The valve losses are comprised of throttling losses and lift response time losses. Depending on the mechanical design configuration, the compressor/expander isentropic efficiency may vary between 78 and 92 percent. The mechanical design configuration was optimized in order to minimize the thermodynamic losses occurring in the compressor/expander system. The piston diameter and stroke are 6 inches and 2 inches, respectively, and the valve diameter and lift are 2 inches and 0.3 inches, respectively. The engine speed is 1150 rpm.

A mechanical design study was performed in order to reduce the losses in the compressor/expander. Various valve types, valve drive mechanisms and crosshead mechanisms were investigated. The selected mechanical configuration is a reciprocating compressor/expander system with reed valves for the compressor and poppet valves for the expander. The reed valves are pressure actuated and the poppet valves are operated by levers which are directly coupled to cam lobes located on the crankshaft. The crosshead drive mechanism consists of a simple crank driving a double-acting piston through a crosshead.

The air-water heat exchanger has been designed for a counterflow heat exchanger effectiveness of at least 85 percent. The heat exchanger configuration is a helical coil of fin tube wrapped around a hollow, blocked center core.

A system package has been designed. The heat pump fit beneath the water storage tank, the overall dimensions are a diameter of 25 in. and a height of 73 in.

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RESEARCH AND DEVELOPMENT OF AN AIR-CYCLE HEAT-PUMP WATER HEATER

DEVELOPMENT TESTING

October 1, 1979

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Development Testing

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1. INTRODUCTION

A development testing program was conducted for the Brayton cycle water heater system. A photograph of the system is shown in Figure 1.1. The test program consisted of the following:

- Performance measurements of the electrical motor
- Heat pump system preliminary performance tests
- Reed and poppet valve tests
- Compressor tests
- Expander tests
- Heat pump system final performance tests.

From the results of the preliminary tests, it was discovered that major losses were occurring in the compressor/expander. It was inititally believed that either flow leakage losses or heat transfer losses were the major sources of irreversibility. However, the results of the final tests revealed that both flow leakage losses and heat transfer losses were occurring simultaneously. The accumulation of these losses resulted in a system COP of 1.3 and less, which is below the design goal of 1.7. It was concluded that prior to designing a second generation heat pump system, the following should be performed:

- A rotary valve development
- Transient heat transfer phenomena investigation.

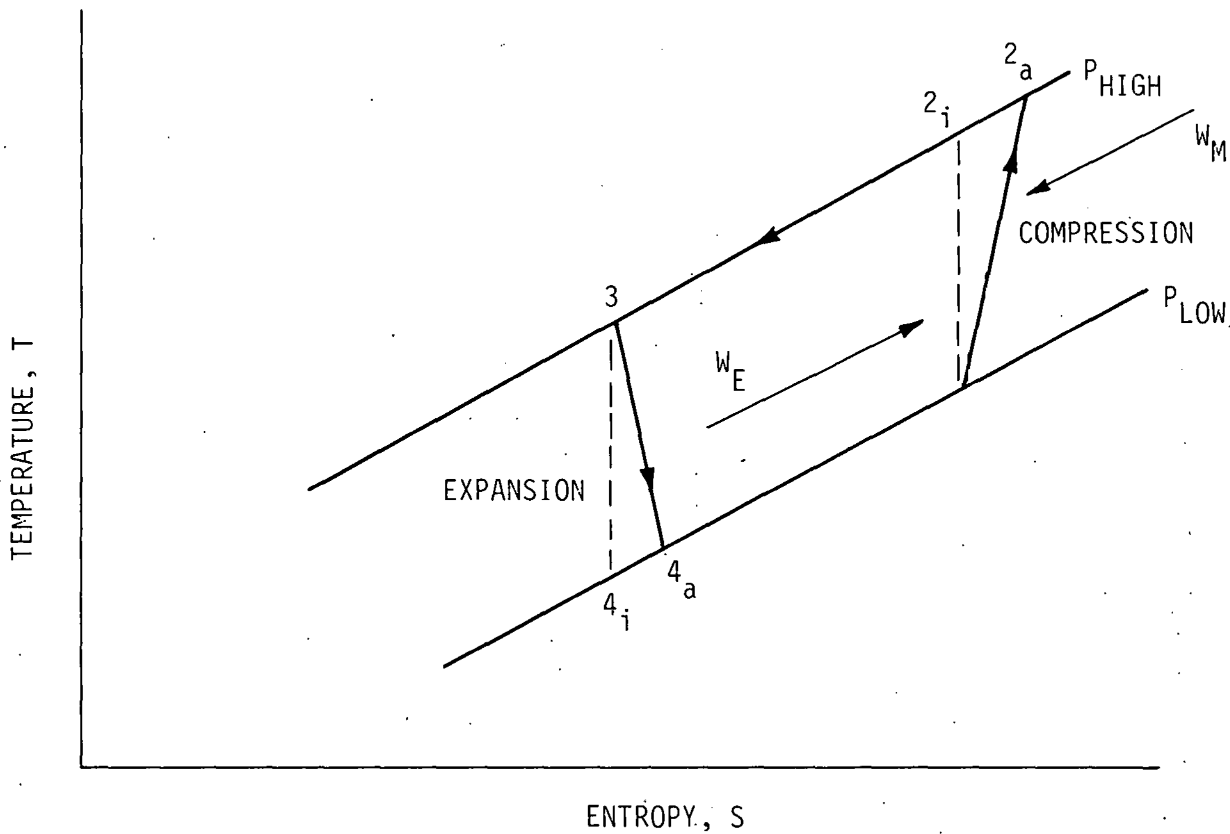


Figure 1-1. T-S Diagram Depicting the Reverse Brayton Air Cycle

2. REVIEW OF TESTING PROTOCOL

The series of tests performed on the Brayton Cycle water heater are divided into five major tests:

- Preliminary System Tests
- Valve Tests
- Compressor Tests
- Expander Tests
- Final System Tests.

2.1 Preliminary System Tests

The system test was performed on the high efficiency water heater system. The system includes the compressor/expander, heat exchanger, water storage tank, electrical motor, and water pump. A schematic of the system test is shown in Figure 2.1. The instrumentation and test equipment are discussed in the Design Report.

2.2 Valve Tests

Prior to performing the compressor and expander tests, it was thought to be useful to determine the pressure drops across the reed and poppet valves in the non-operational modes. Pressure sensors were placed across each valve and the gas flows were adjusted and measured.

2.3 Compressor Test

The compressor was isolated from the expander and a series of tests were performed. A schematic illustrating the compressor

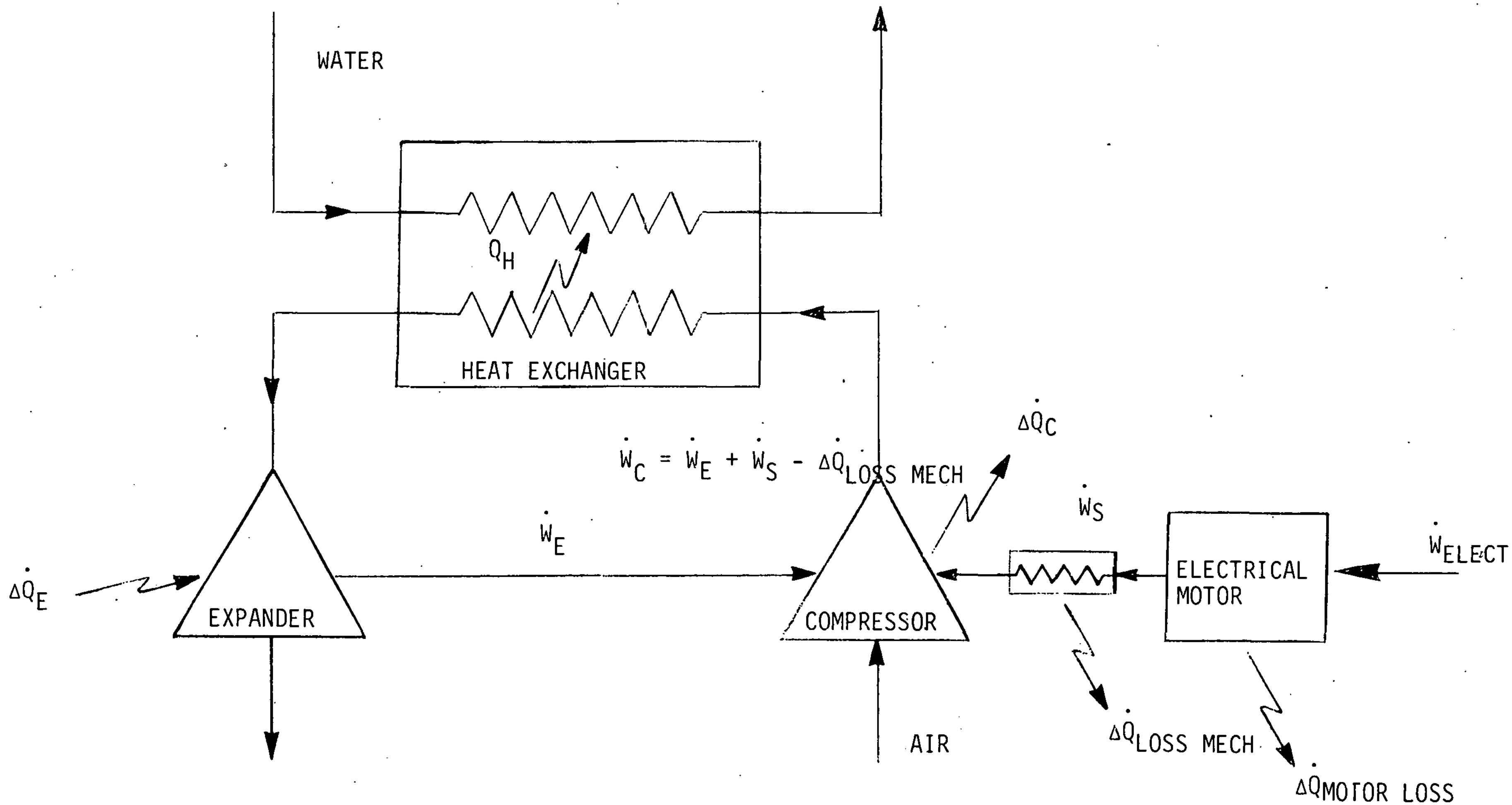


Figure 2-1. System Test

test arrangement is shown in Figure 2-2. Instrumentation was provided to measure the pressures and temperatures of the air flow. Also the power input to the electrical motor was measured and recorded.

2.4 Expander Test

The expander was isolated from the compressor and a series of tests were performed. A schematic illustrating the expander test arrangement is shown in Figure 2-3. An external compressor was required to provide a high pressure gas source. Instrumentation was employed to measure the pressure and temperature of the gas stream. The expander work output was directed to an electrical generator. The electrical energy developed by the electrical generation was dissipated in an electrical resistance bank.

2.5 Final System Tests

After the valve, compressor, and expander tests were completed, a final system test was performed in the same manner as for the preliminary system test.

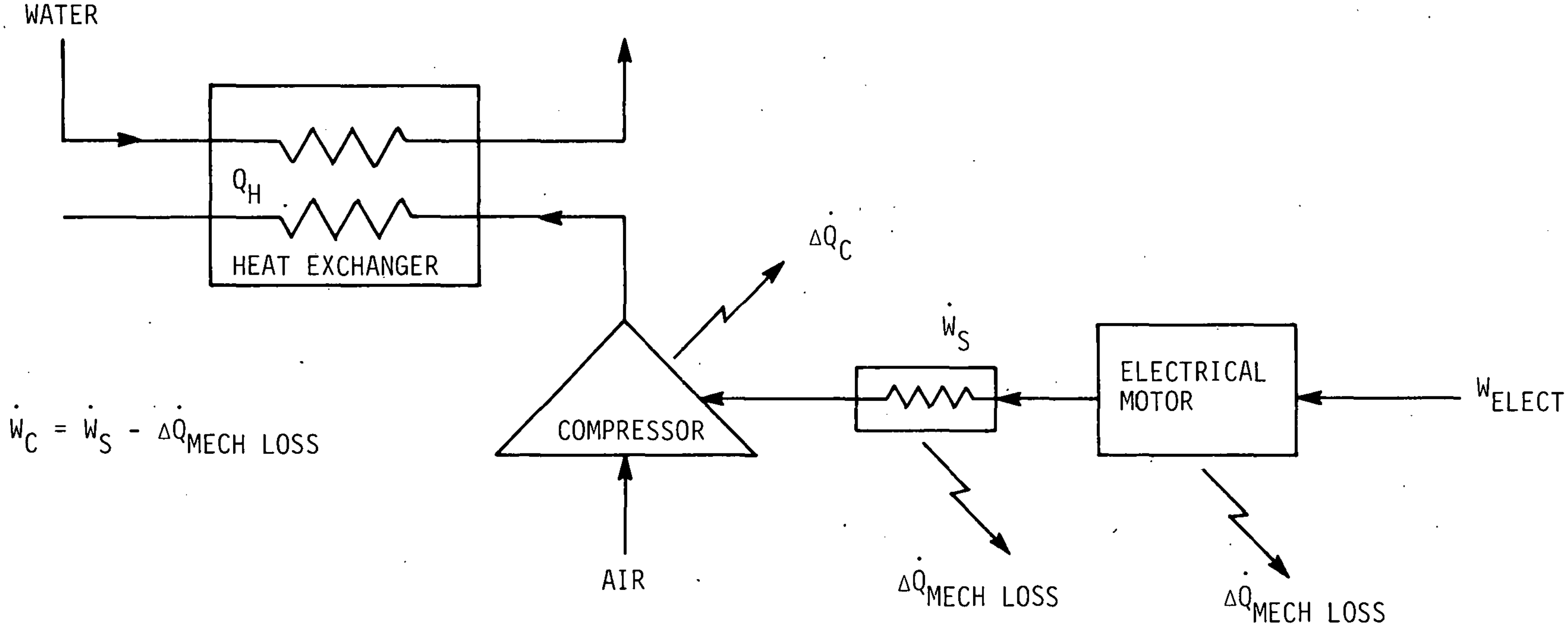


Figure 2-2. Compressor Test

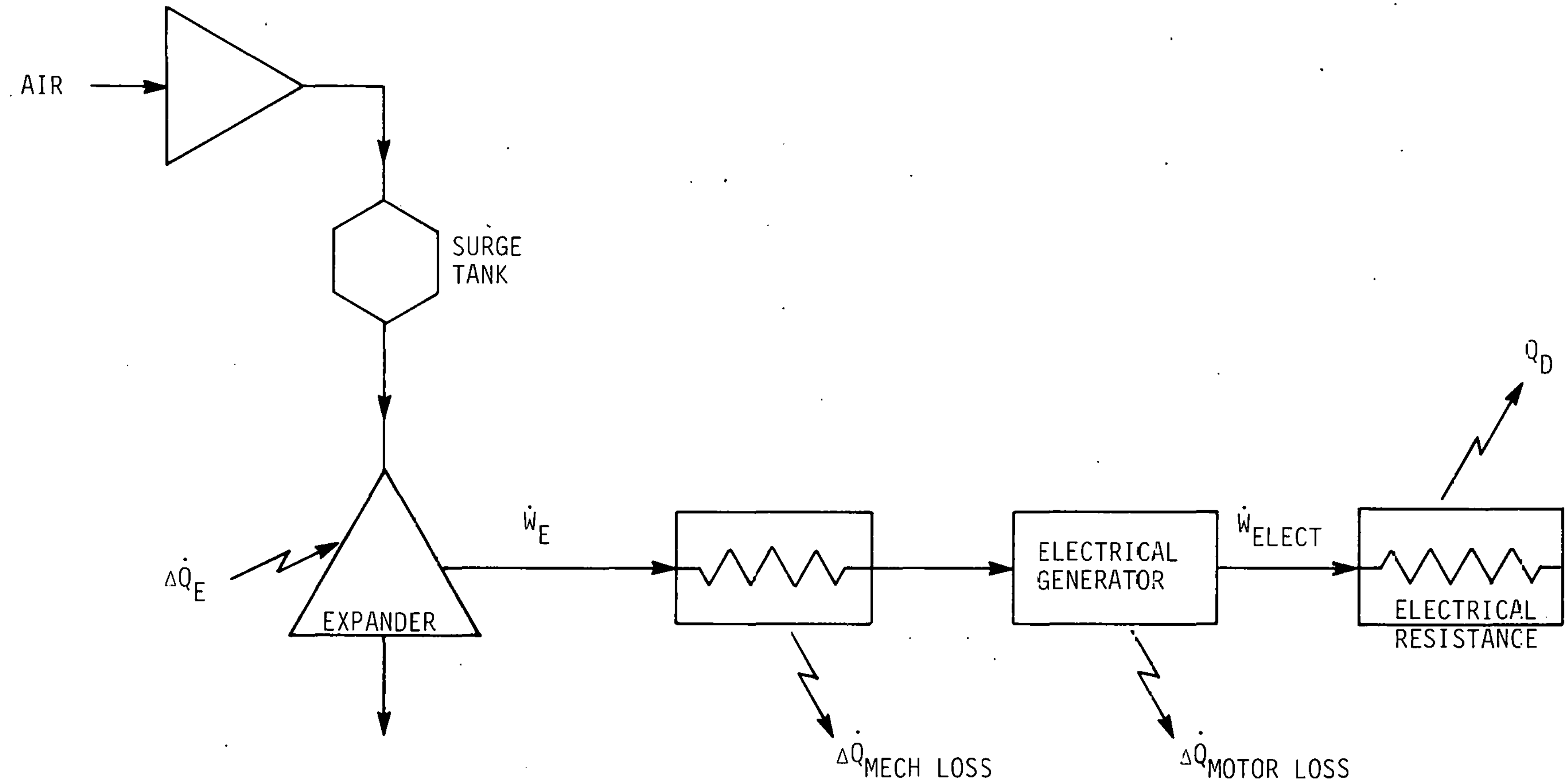


Figure 2-3. Expander Test

3. TEST RESULTS

The test results of the Brayton Cycle water heater are summarized for the following tests and system components:

- Motor test results
- Preliminary test results
- Reed and poppet valve test results
- Compressor test results
- Expander test results.

3.1 Motor Test Results

An electrical motor was purchased from Gould, Inc. The motor was modified to include proprietary high efficiency features. A summary of the motor specifications is presented in Table 3-1.

A performance test was performed by Gould engineers and a summary of the test results is included in Figures 3-1 and 3-2. Figure 3-1 illustrates the motor curves for motor horsepower efficiency, power factor and motor speed as a function of the motor torque. The solid portions of the curves represent experimental results and the dotted portions of the curves represent extrapolated results predicted by Gould. One will note that the maximum motor efficiency is approximately 78 percent at a motor speed of 1140 rpm and a motor capacity of approximately 1.3 hp. Figure 3-2 presents the motor speed and motor current as a function of the torque.

3.2 Preliminary System Test Results

A series of preliminary tests was performed on the Brayton cycle water heater system. A photograph of the experimental system is shown in Figure 3-3. A summary of the preliminary

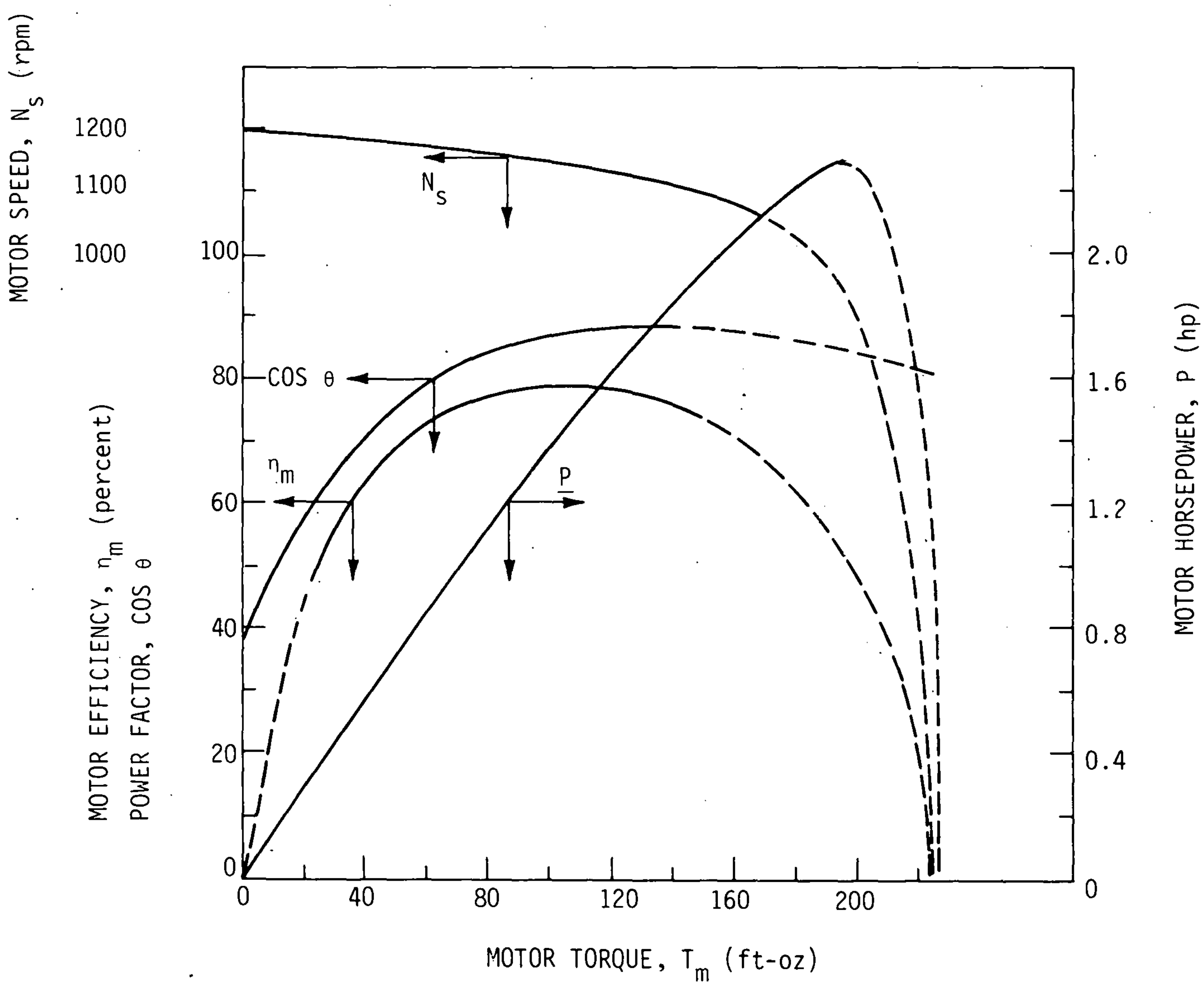


Figure 3-1. Motor Efficiency, Power Factor, Speed, and Power Characteristics of the Gould Motor

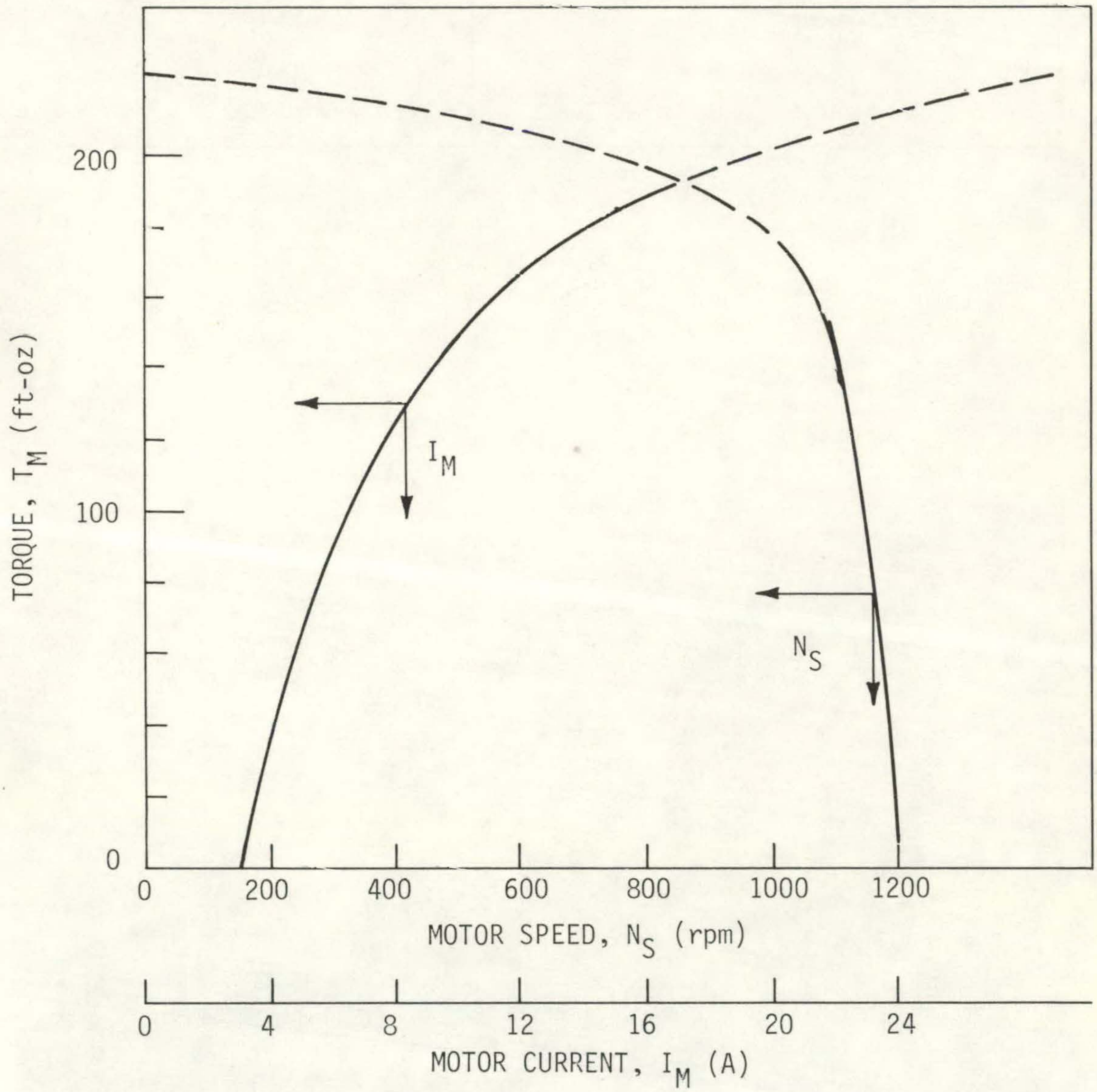


Figure 3-2. Motor Current and Speed Characteristics of the Gould Motor

Table 3-1. Electrical Motor Specifications

Specification	Value
Horsepower Rating	1.25 hp
Frequency	60 Hz
Speed	1140 rpm
Voltage	230 volts

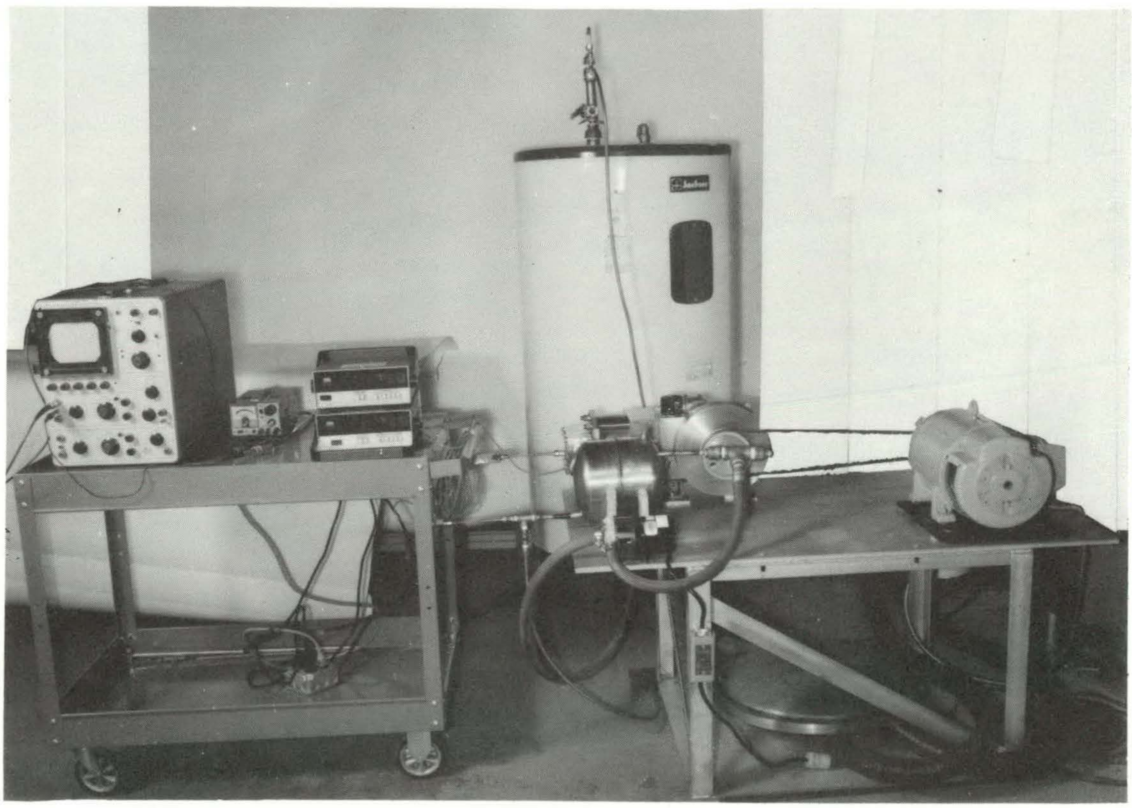
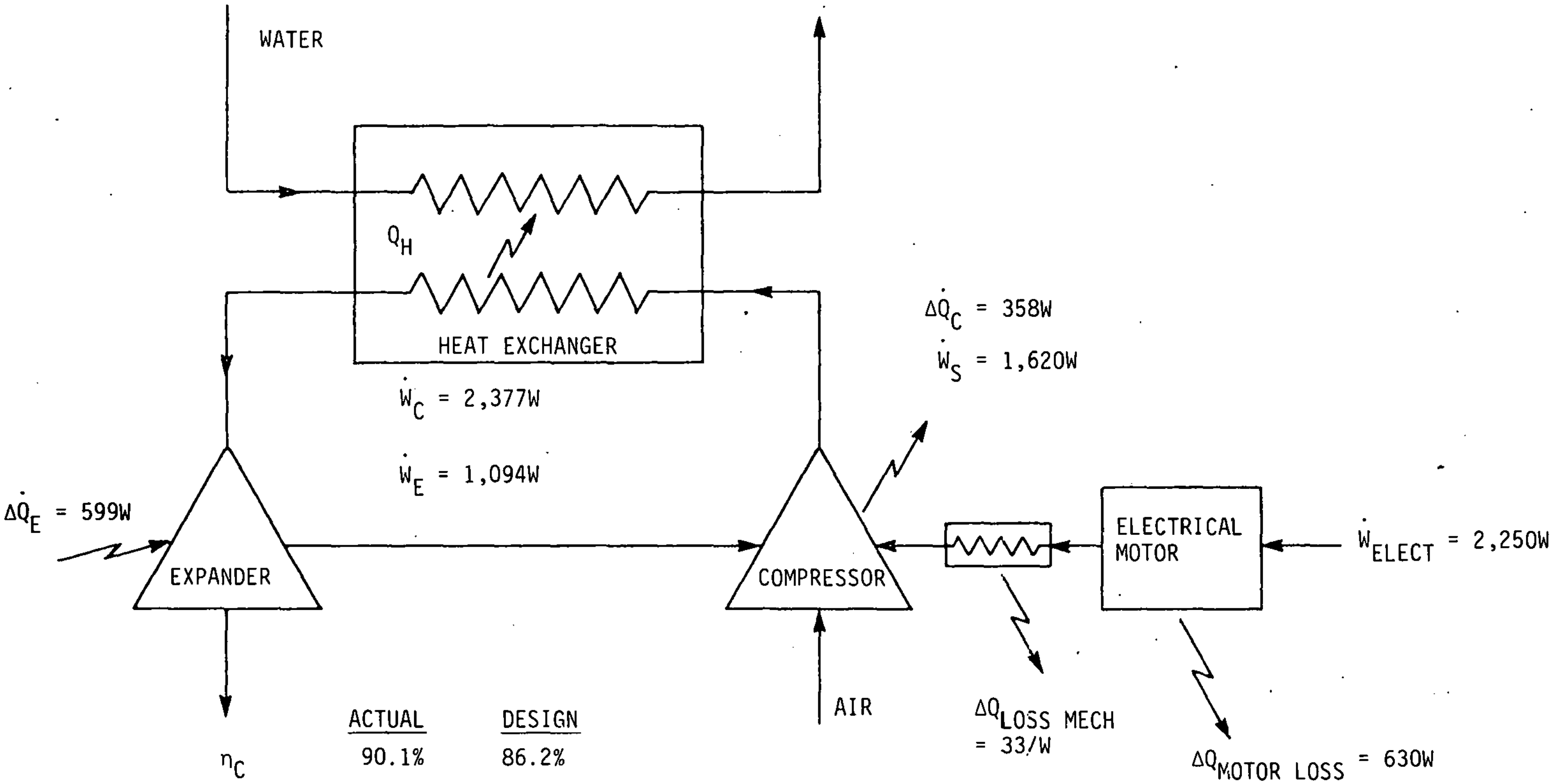


Figure 3-3. Prototype Reverse Brayton Cycle Heat Pump During Test

test results is shown in Figure 3-4 and in Table 3-2. The actual compressor and expander isentropic efficiencies based on gas temperatures are 90.10 percent and 39.81 percent, respectively. The compressor indicated power, $\dot{W}_C$, is greater than the energy transferred to the gas, $\dot{W}_{CG}$. This suggests that either a heat transfer loss or a flow leakage loss is occurring in the compressor; the total loss in the compressor is designated as $\Delta\dot{Q}_C$. Alternatively, the expander indicated power is significantly less than the energy extracted from the gas; the total loss in the expander is designated as $\Delta\dot{Q}_E$. The results suggest that the compressor thermodynamic performance is excellent at the expense of the expander thermodynamic performance. In Figure 3-3, the terms η_C, η_E, η_M, and ε are compressor isentropic efficiency, expander isentropic efficiency, motor efficiency, and heat exchanger effectiveness, respectively, and $\dot{Q}_H'$ and $\dot{Q}_H$ are the heat transfer rates to the water without the motor losses and with the motor losses, respectively. For the case of a reversible motor (η_M = 100 percent) the actual coefficient of performance, COP_S, is 1.170. The system coefficient of performance, COP, is 1.222 compared to the design COP of 1.62. A summary of the performance parameters is presented in Table 3-2. The difference between columns labelled Design (Stock Valves) and Design (Hot Valves No HTX) is that stock valves refer to standard reed valves and the hot valves refer to reed valves machined to increase flow area. Note that the hot valves predict a COP of 2.0 whereas the stock valve predict a COP of 1.62.

Figure 3-5 presents the experimental PV and Pθ diagrams for both the compressor and expander. Figure 3-6 presents a comparison of the expander PV diagram with the predicted PV diagram from the computer program described in the Design Study. From the results, it appears that a significant pressure drop is occurring across the inlet poppet valve. As a check to determine if significant heat transfer is occurring across the piston/cylinder walls and the gas, the gas specific heat ratios are

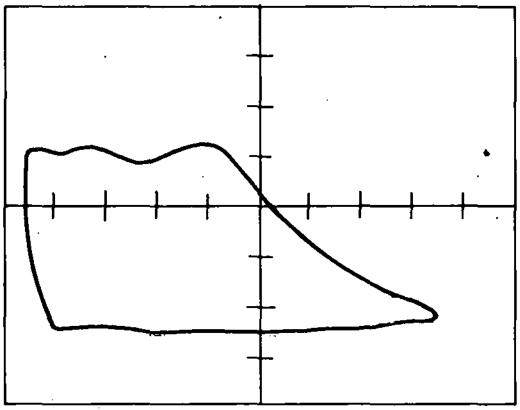


	ACTUAL	DESIGN
η_C	90.1%	86.2%
η_E	39.8%	88.8%
η_m	72.0%	80.0%
ε	91.8%	87.0%
$\dot{Q}_H$	1895W	2303W
$\dot{Q}_H$	2525W	2626W
COP	1.122	1.62
COP_S	1.170	1.776

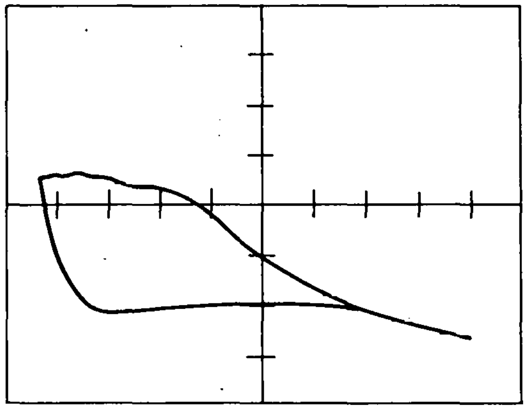
Figure 3-4. Summary of Preliminary Test Results

Table 3-2. Preliminary Test Results

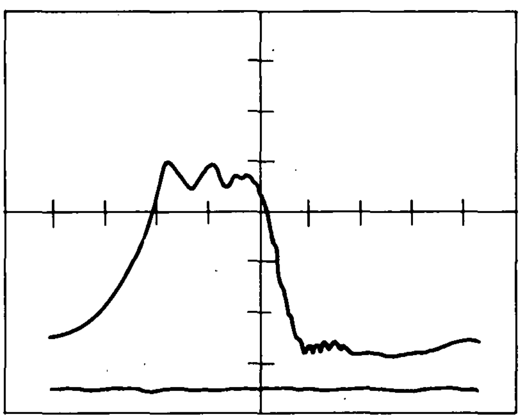
Compressor	Design (Hot Valves No HTX)	Design (Stock Valves)	Actual
P_{IN}	15.0 psia	15.0 psia	14.7 psia
T_{IN}	$70^{o}F$	$70^{o}F$	$72^{o}F$
P_{OUT}	45 psia	45 psia	46.2 psia
T_{OUT}	$273^{o}F$	$280^{o}F$	$300.6^{o}F$
PR	3.0	3.0	3.143
Q_{A}	34.95 scfm	34.76 scfm	28 scfm
N_{S}	1150 rpm	1150 rpm	950 rpm
η_{V}	92.92%	92.41%	91.05%
$\dot{W}_{C}$	2.302 kW	2.489 kW	2.377 kW
η_{C}	93.71%	86.2%	90.10%
$\dot{W}_{CG}$	2.237 kW	2302 kW	2.019 kW
$\Delta\dot{Q}_{C}$	65 Watts	187 Watts	358 Watts
Expander			
P_{IN}	45 psia	45 psia	41.7 psia
T_{IN}	$70^{o}F$	$70^{o}F$	$86^{o}F$
P_{OUT}	15 psia	15 psia	14.7 psia
T_{OUT}	$-68.5^{o}F$	$-56^{o}F$	$30^{o}F$
PR	3.0	3.0	2.837
$\dot{W}_{E}$	1.507 kW	1.392 kW	1.094 kW
η_{E}	95.58%	88.84%	39.81%
$\dot{W}_{EG}$	1.527 kW	1.381 kW	0.495 kW
$\Delta\dot{Q}_{E}$	-20 Watts	11 Watts	599 Watts
Summary of Losses			
$\Delta\dot{Q}_{MECH\ LOSS}$	200 Watts	200 Watts	337 Watts
$\Delta\dot{Q}_{MOTOR\ LOSS}$	249 Watts	324 Watts	630 Watts
System Performance			
η_{m}	80%	80%	72%
$\dot{Q}_{H}$	2.486 kW	2.626 kW	2.525 kW
COP	1.998	1.62	1.122
ε	87%	87%	91.8%



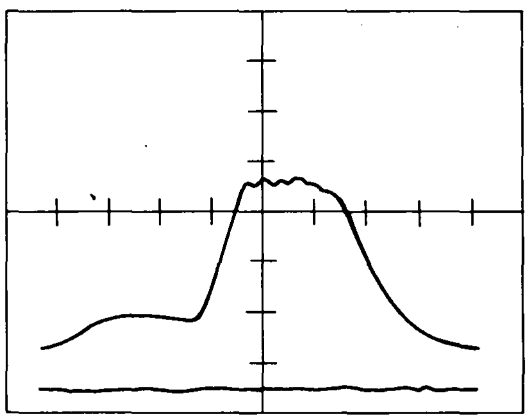
COMPRESSOR PV DIAGRAM



EXPANDER PV DIAGRAM



COMPRESSOR P-Θ DIAGRAM



EXPANDER P-Θ DIAGRAM

Figure 3-5. Compressor and Expander PV and Pθ Diagrams

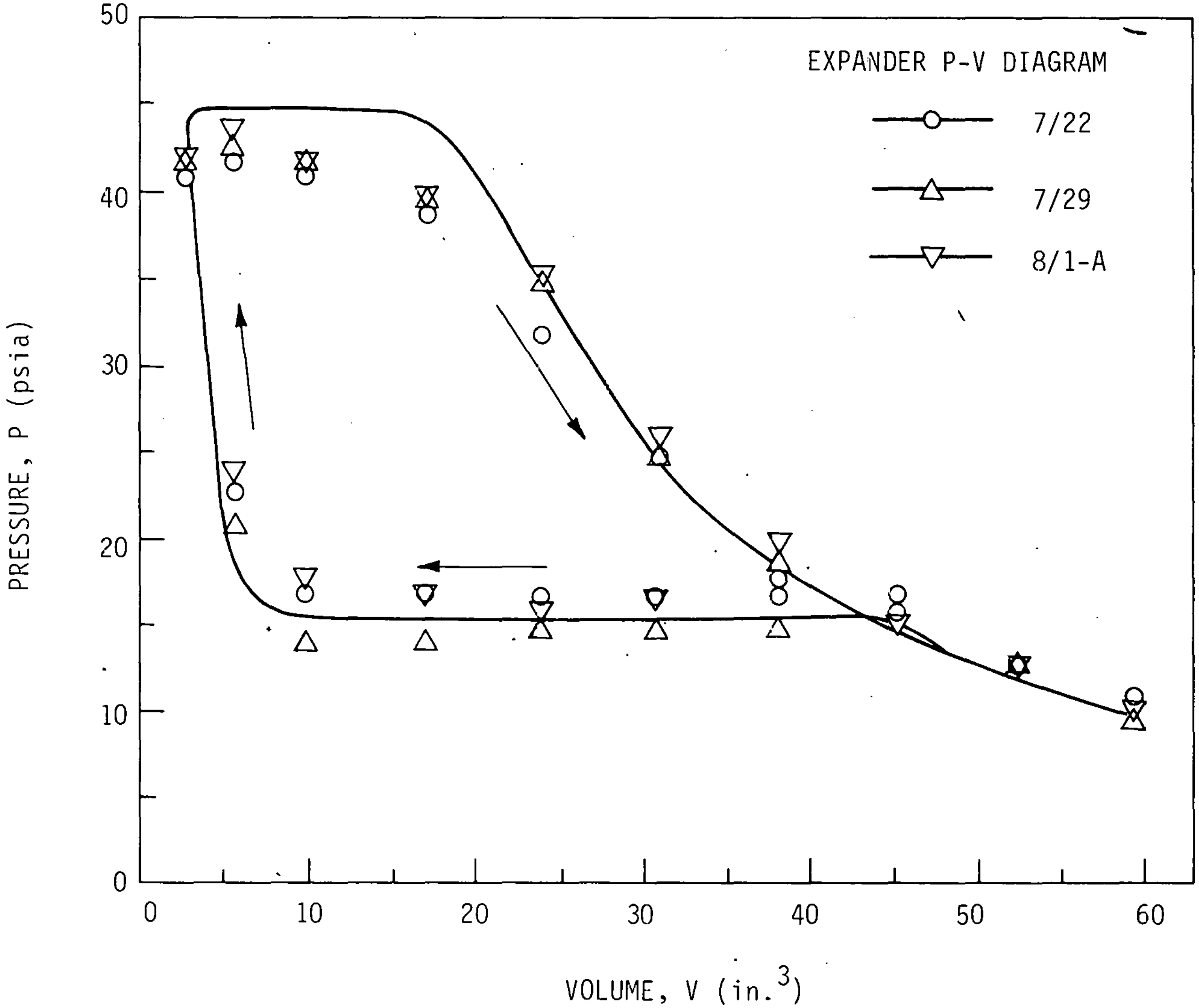


Figure 3-6. Comparison of the Expander Experimental and Analytical PV Diagrams

compared with the actual gas expansion and compression processes as is shown in Figure 3-7. One will note that the actual expansion process departs from the isentropic case at the higher pressure levels.

It is possible that either a heat transfer or flow leakage loss mechanism is occurring in the compressor and expander. The total heat transfer includes a steady-state heat transfer contribution and a transient heat transfer contribution. The steady-state heat transfer may occur along the cylinder walls or across the piston. The transient heat transfer is due to the fact that the piston and cylinder walls are not able to respond rapidly to the temperature fluctuations of the gas in both the expander and compressor. Superimposed on the steady-state heat transfer flux is the unsteady-state heat transfer contribution. The amplitude of the oscillating temperautre of the piston and cylinder walls adjacent to the gas is smaller than the oscillating temperature of the gas in both the expander and compressor. As a result of the large difference in the temperature amplitudes, there occurs an unsteady-state heat transfer between the gas and the piston/cylinder walls. The consequence of the unsteady-state heat transfer is the presence of local transient heat transfer coefficients which may be 2 to 10 times larger than the steady-state heat transfer coefficients [1]. In addition, flow leakage may occur across the valves as well as the piston. As is shown in Figure 3-8, flow leakage may occur in either direction across the piston, i.e., for certain piston positions, the gas pressure in the compressor may be either larger or smaller than the gas pressure in the expander.

Consequently, to improve the Brayton Cycle water heater performance, modifications were made. A summary of the modification is presented in Table 3-3.

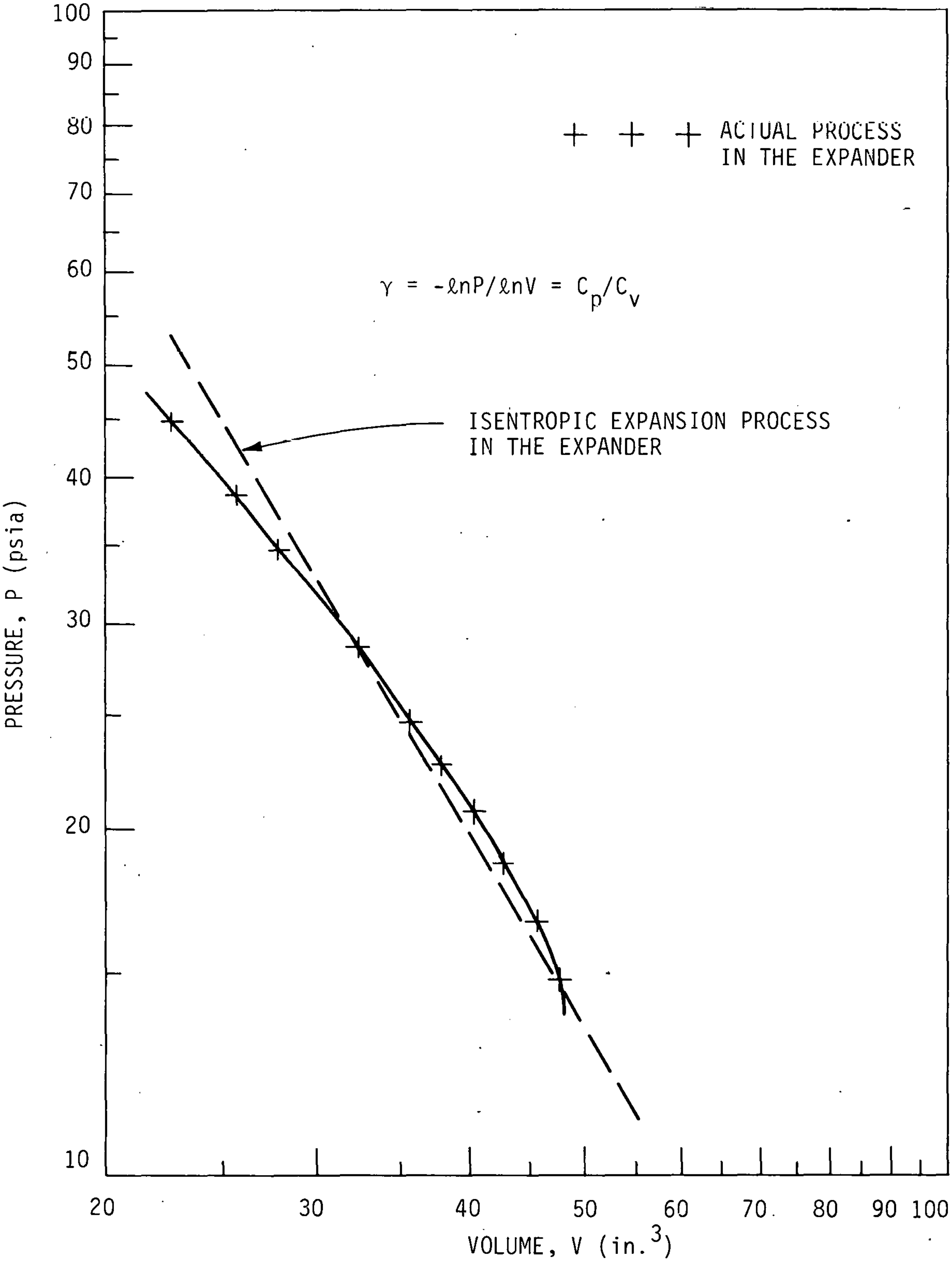


Figure 3-7. Comparison of the Experimental and Analytical Specific Heat Ratios for the Expander

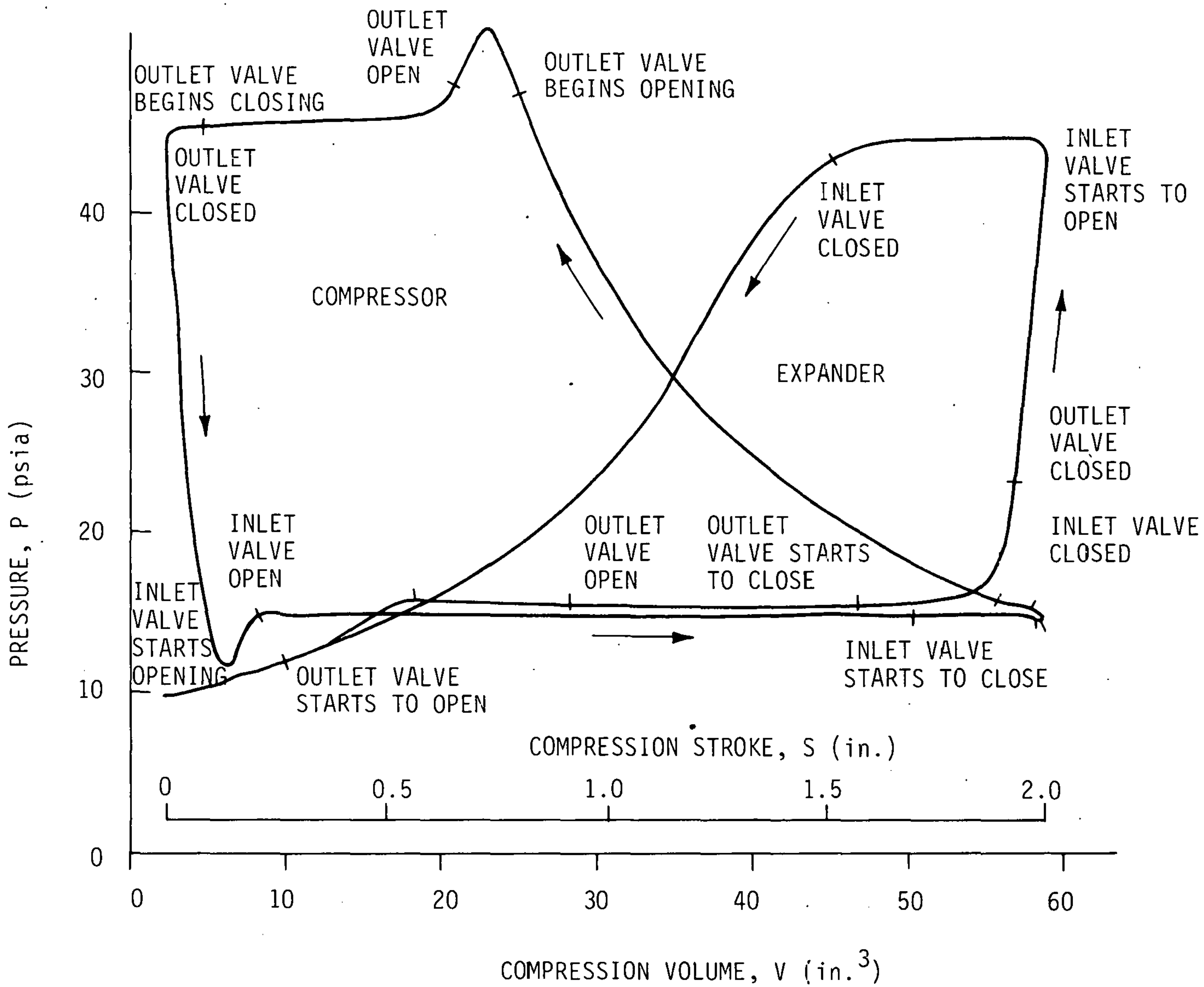
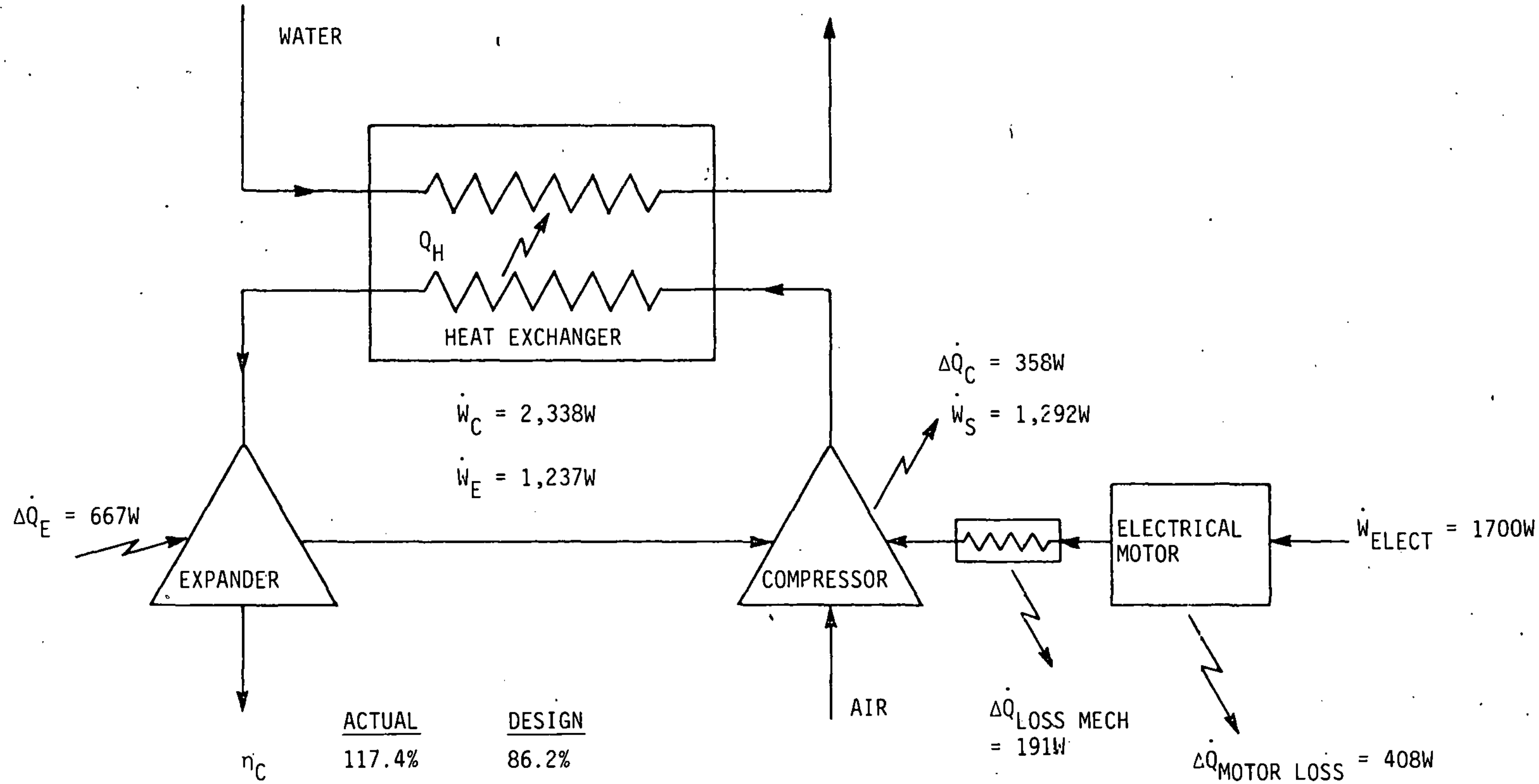


Figure 3-8. Compressor and Expander PV Diagrams

Figure 3-3. First Generation Design Modifications

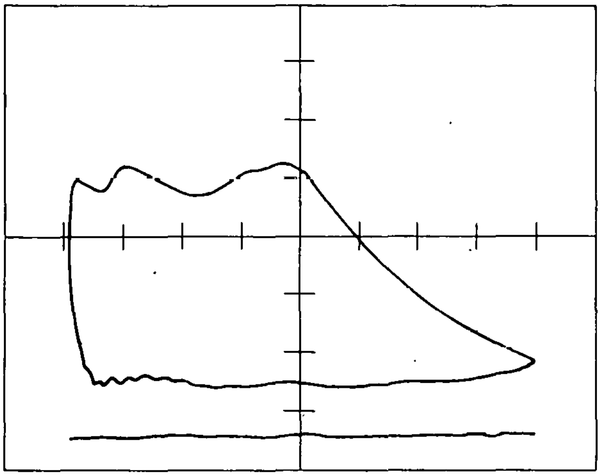
Modification	Objective
• Increased Size of Porting of Expander Inlet Valve	• Reduce pressure drop across the expander inlet valve
• Place Softer Ring Seal on Expander Inlet Valve	• Reduce mechanical losses
• Reduced Cylinder Wall Thickness	• Reduce heat transfer between gas and cylinder wall due to axial heat transfer in the walls

Additional preliminary tests were performed on the Brayton Cycle water heater to determine the effect of the minor design improvements. A summary of the test results is presented in Figure 3-9 and Table 3-4. The actual compressor and expander isentropic efficiencies based on gas temperatures are 117.4 percent and 46.0 percent, respectively. The compressor isentropic efficiency is greater than 100 percent because of a heat transfer loss or a flow leakage loss. Again it appears that the compressor thermodynamic performance is excellent at the expense of the expander thermodynamic performance. The system COP is 1.26 as compared to the design COP of 1.62. The corresponding PV and Pθ diagrams for both the compressor and expander are presented in Figure 3-10. A plot of the experimental data for the expander PV diagram is presented in Figure 3-11. The data very nearly correlates to the computer predictions. Also, the experimental specific heat ratios for both the expansion and compression processes are presented in Figure 3-12. The compression process is very nearly isentropic whereas the expansion process departs from the isentropic case.

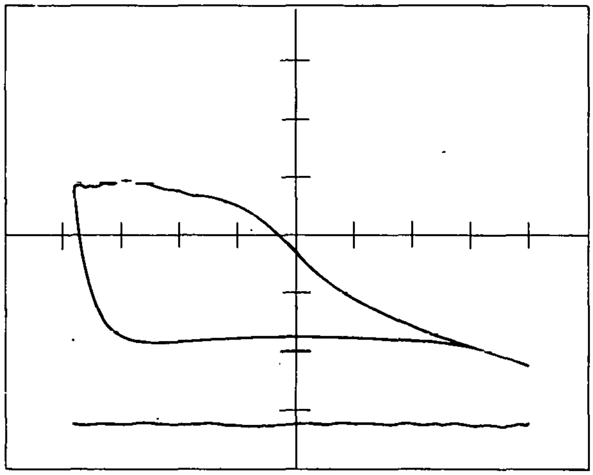


	ACTUAL	DESIGN
η_C	117.4%	86.2%
η_E	46.0%	88.8%
η_m	76.0%	80.8%
ε	90.8%	87.0%
$\dot{Q}_H$	1,733W	2,302W
$\dot{Q}_{H/}$	2,141W	2,626W
COP	1.26	1.62
COP_S	1.34	1.776

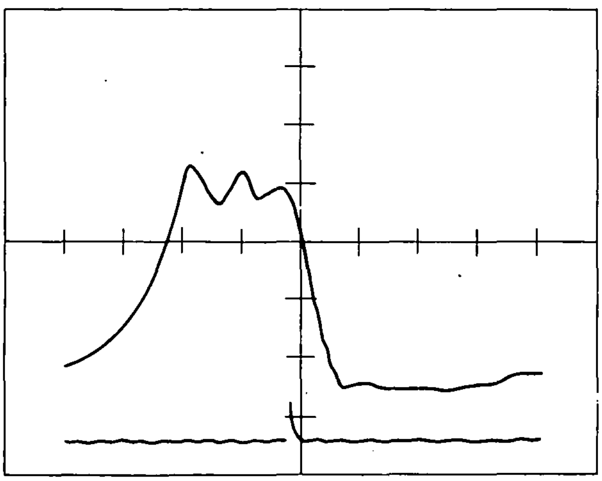
Figure 3-9. Preliminary Test Results



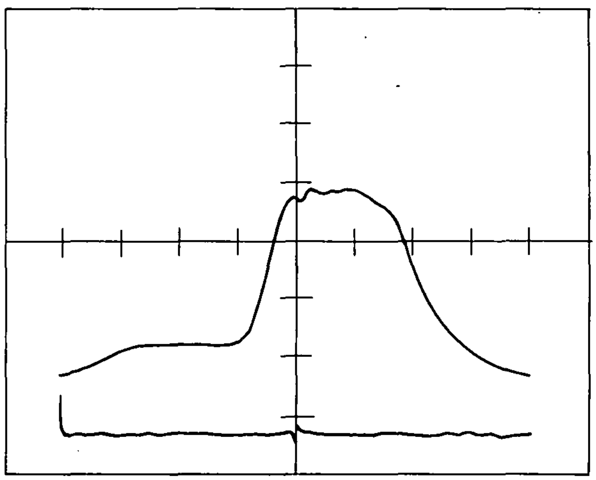
COMPRESSOR PV DIAGRAM



EXPANDER PV DIAGRAM



COMPRESSOR P- DIAGRAM



EXPANDER P-Θ DIAGRAM

Figure 3-10. Compressor and Expander PV and Pθ Diagrams

Table 3-4. Summary of Test Results

Compressor	Design (Hot Valves No HTX)	Design (Stock Valves)	Actual
P_{IN}	15.0 psia	15.0 psia	14.7 psia
T_{IN}	70°F	70°F	76°F
P_{OUT}	45 psia	45 psia	45.2 psia
T_{OUT}	273°F	280°F	308.5°F
PR	3.0	3.0	3.075
Q_A	34.95 scfm	34.76 scfm	27.0 scfm
N_S	1150 rpm	1150 rpm	1000 rpm
η_V	92.92%	92.41%	88.8%
$\dot{W}_C$	2.302 kW	2.489 kW	2.338 kW
η_C	93.71%	86.2%	117.4%
$\dot{W}_{CG}$	2.237 kW	2.303 kW	1.980 kW
$\Delta\dot{Q}_C$	65 Watts	187 Watts	358 Watts
Expander			
P_{IN}	45 psia	45 psia	41.7 psia
T_{IN}	70°F	70°F	105°F
P_{OUT}	15 psia	15 psia	14.7 psia
T_{OUT}	-68.5°F	-56°F	38°F
PR	3.0	3.0	2.837
$\dot{W}_E$	1.507 kW	1.392 kW	1.237 kW
η_E	95.58%	88.84%	46.0%
$\dot{W}_{EG}$	1.527 kW	1.381 kW	0.571 kW
$\Delta\dot{Q}_{EG}$	-20 Watts	11 Watts	667 Watts
Summary of Losses			
$\Delta\dot{Q}_{MECH\ LOSS}$	200 Watts	200 Watts	191 Watts
$\Delta\dot{Q}_{MOTOR\ LOSS}$	249 Watts	324 Watts	408 Watts
System Performance			
η_{m_1}	80%	80%	76%
$\dot{Q}_H$	2.486 kW	2.626 kW	2.141 kW
COP	1.998	1.62	1.26
ε	87%	87%	90.8%

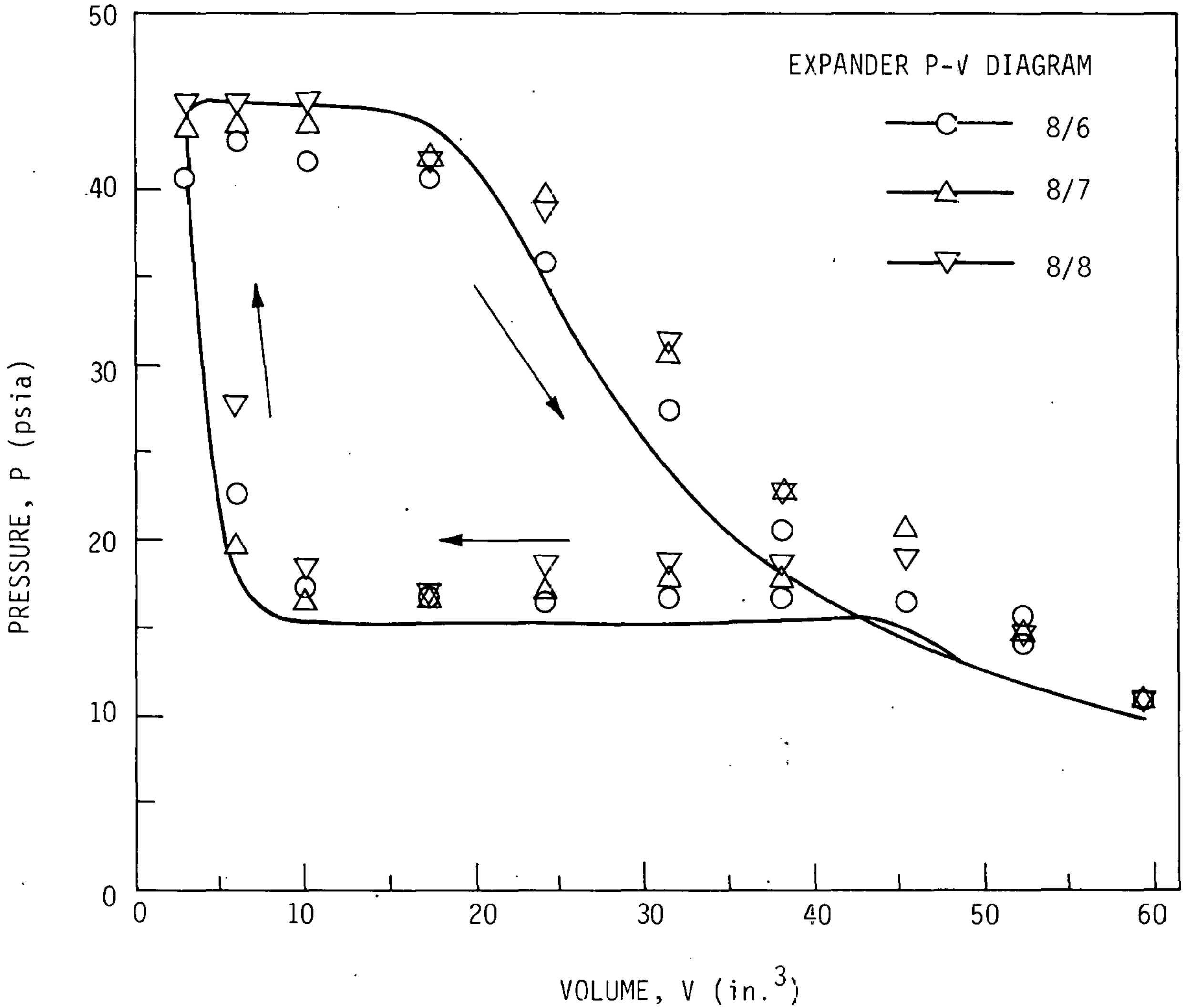


Figure 3-11. Comparison of the Experimental and Analytical Expander PV Diagram

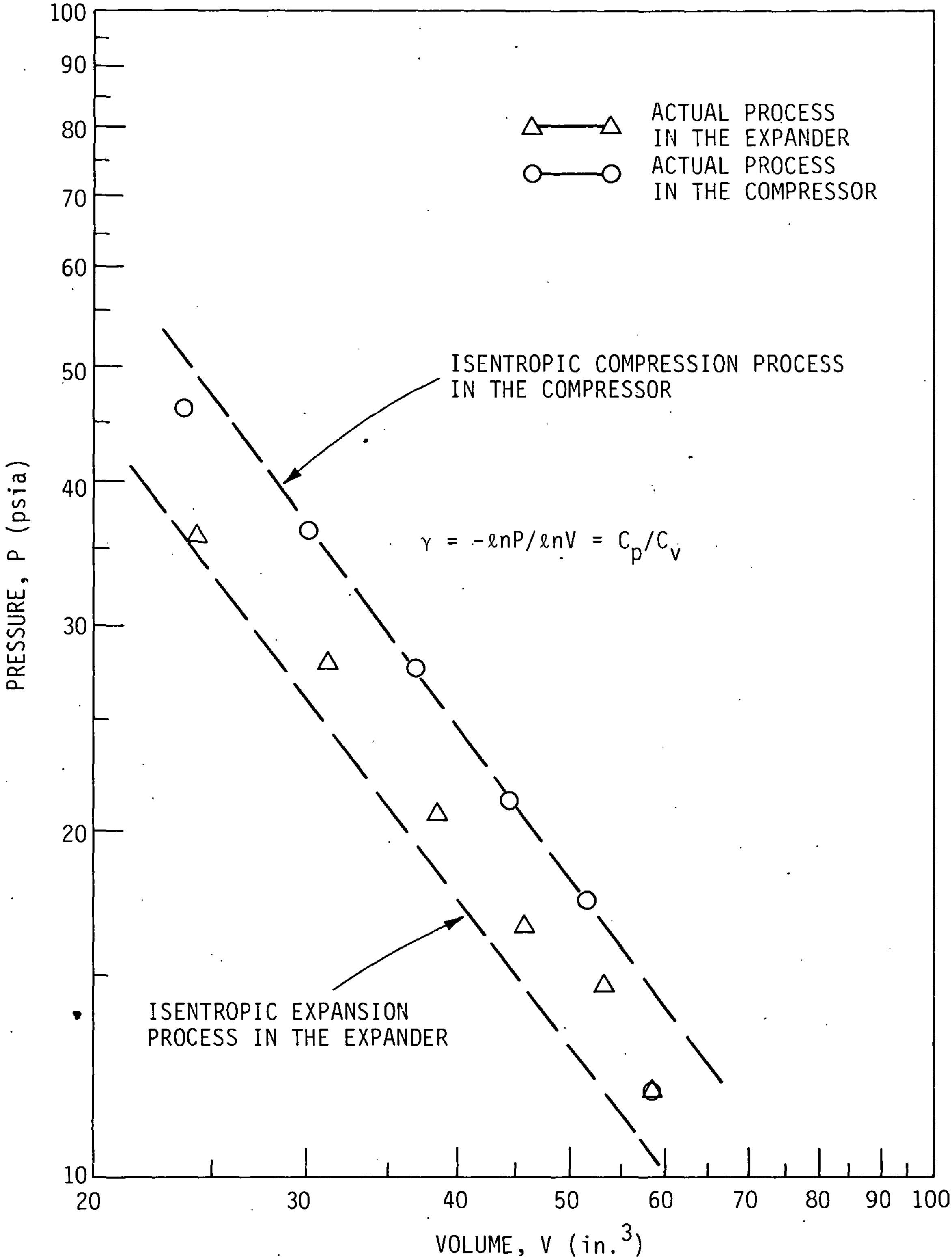


Figure 3-12. Comparison of the Experimental and Analytical Specific Heat Ratios for Both the Compressor and Expander

From the preliminary test results, it was thought that either a flow leakage or a heat transfer was occurring in the compressor/expander. A series of additional tests was performed and the results of these tests are discussed in the following sections.

3.3 Valve Test Results

A series of valve tests were performed to determine the pressure drop losses with steady flow. Figure 3-13 presents the results of the compression suction valve tests. The pressure drop is presented as a function of the volume flow rate across the inlet valve for the case of an inlet pressure of 15 psia. The slope of the curve reveals that ΔP is proportional to V^2. The significance of this curve is that the magnitude of the pressure drop is on the order of 0.5 psia for a flow rate of 30.5 scfm. The corresponding experimental flow discharge coefficient is larger than the flow discharge coefficient used for estimating the pressure drop across the compressor inlet valve.

A similar test was performed for the compressor discharge valve. Figure 3-14 presents the compressor discharge valve pressure drop as a function flow rate for an inlet pressure of 38.7 psia. As for the case of the compressor inlet valve, the magnitude of the pressure drop is approximately 0.6 psia for a volume flow rate of 30 scfm. Again the experimental flow discharge coefficient is larger than the flow discharge coefficient used for estimating the pressure drop across the compressor discharge valve.

A test for the pressure drop across the expander poppet inlet valve was performed. Figure 3-14 presents the expander inlet valve pressure drop versus crank angle and the valve lift versus crank angle for a flow rate of 37.5 scfm and a valve inlet pressure of 38.7 psia. The results of the test show that the pressure drop is approximately 0.02 psi. The corresponding

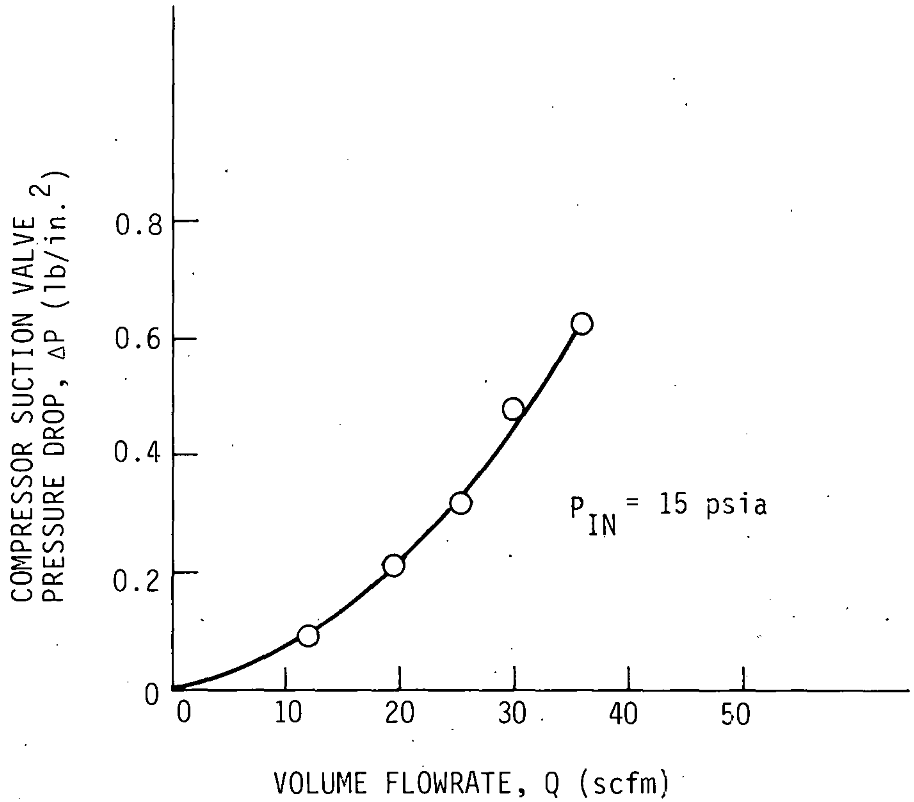


Figure 3-13. Pressure Drop Characteristics of the Compressor Inlet Valve

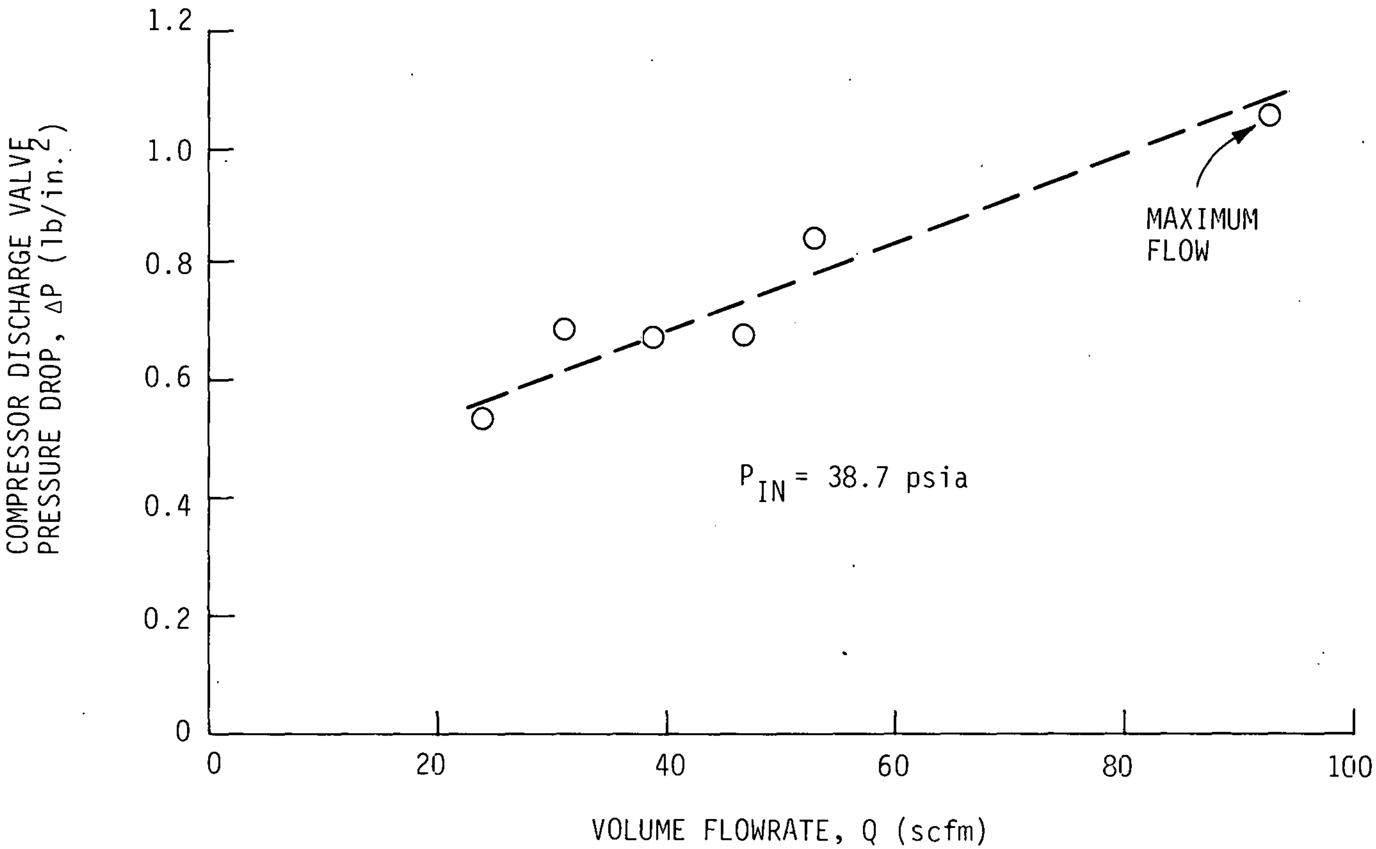


Figure 3-14. Pressure Drop Characteristics of the Compressor Discharge Valve

experimental flow discharge coefficient is significantly larger than the flow discharge coefficient used for estimating the pressure drop across the expander inlet valve. In a similar fashion, the expander discharge valve pressure drop and valve lift versus crank angle are presented in Figure 3-15. The volume flow rate is 34 scfm and the valve inlet pressure is 15.4 psia. The pressure drop across the expander discharge valve is of the order of 0.3 to 0.37 psi. An anomaly does appear in the lower portion of the curve; an explanation for this has not been formulated. Again the corresponding experimental flow discharge coefficient is larger than the flow discharge coefficient used for estimating the pressure drop across the expander discharge valve.

3.4 Compressor Test Results

A summary of the compressor test results is presented in Table 3-5. Eight different experimental runs are presented. A larger 3 hp motor was employed and a new plastic piston was installed. It is worth mentioning the variations in the physical test. Half of the tests were performed with metal backup piston ring band. Tests were performed at two different speeds at approximately 1100 rpm and also at 600 to 700 rpm. Finally, for each engine speed and metal backup configurations, the compressor was operated with external water cooling of the compressor cylinder head.

The salient results of the compressor test may be summarized as follows:

- The flow leakage past the piston rings is approximately 1 scfm or 3 percent of the total compressor capacity

- The compressor isentropic efficiency based on gas temperatures is approximately 95 to 119 percent and the volumetric efficiency is 85 percent. The absence

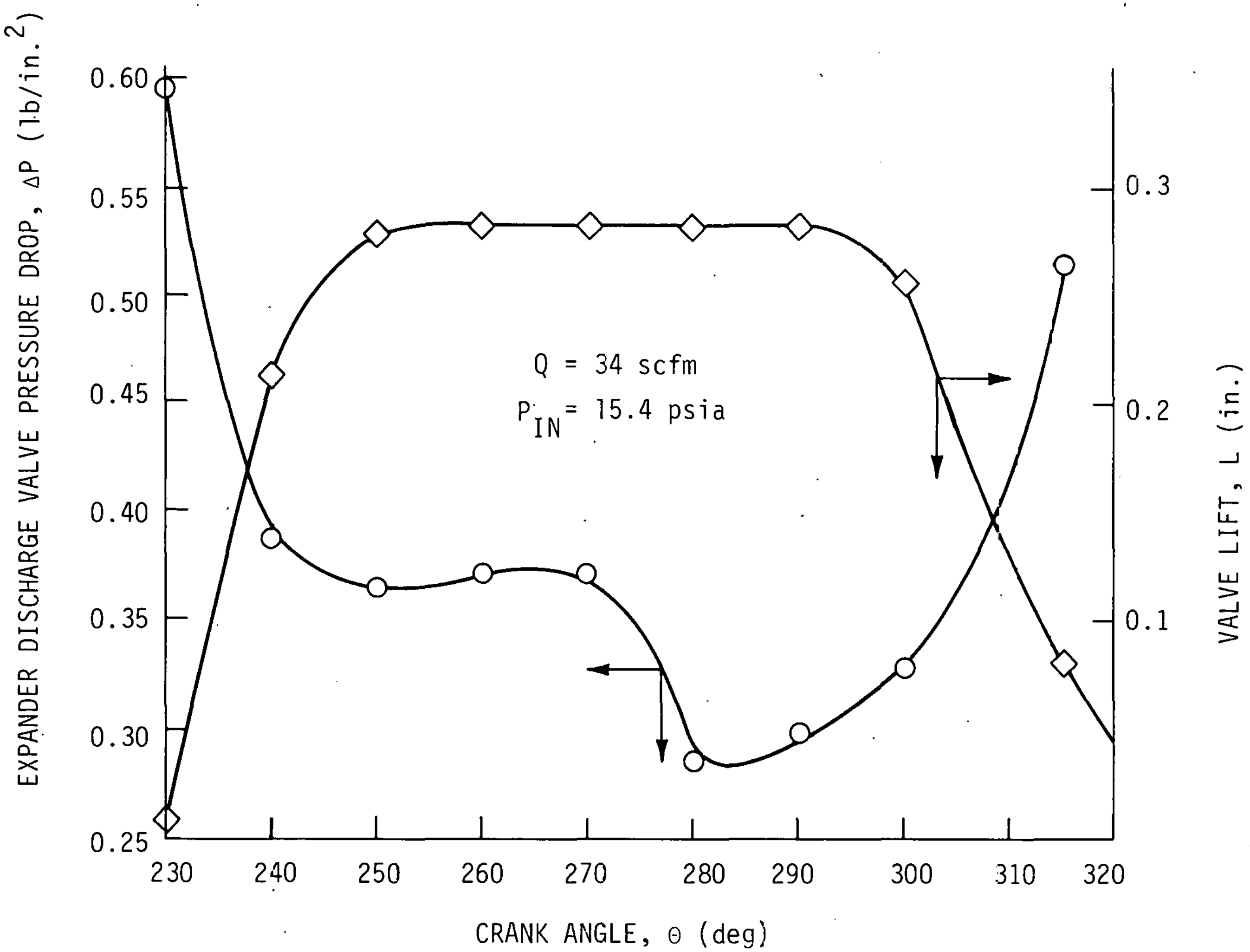


Figure 3-15. Pressure Drop Versus Crankangle for the Expander Discharge Valve

Table 3-5. Summary of Compressor Test Results

RUN NUMBER	1-18-I	1-18-II	1-18-III	1-18-IV	1-29-I	1-29-II	1-29-III	1-29-IV
Comments								
• Plastic Piston • 3 HP Motor • No Insulation	← METAL BACKUP BAND →				← NO METAL BACKUP BAND →			
• External H_2O Cooling	No	Yes	No	Yes	No	Yes	No	Yes
Compressor								
Speed (RPM)	1150	1110	683	683	1100	1100	600	600
Gas Flow In (SCFM)	31.0	30.0	18.0	18.0	27.4	28.2	14.7	15.0
Gas Flow Out (SCFM)	30.0	28.7	17.9	17.1	26.6	27.1	12.8	12.8
Total Gas Flow Leakage (SCFM)	1.0	1.3	0.1	0.9	0.8	1.1	1.0	2.2
Gas Internal Energy (kW)	2.245	1.917	1.155	1.001	1.906	1.676	0.828	0.684
Indicated Power (kW)	2.072	1.877	0.967	0.963	2.089	2.072	0.986	0.986
Isentropic Efficiency								
Gas (%)	94.3	105.3	104.7	118.8	94.4	104.4	95.4	116.8
Power (%)	102.2	107.5	125.0	123.5	85.9	84.2	80.0	80.9
Volumetric Efficiency (%)	85.8	85.1	86.2	82.4	79.6	81.1	70.2	70.2
Heat Exchanger								
$\dot{Q}_{AIR}$ (kW)	2.12	1.53	1.089	0.790	1.483	1.112	0.676	0.488
$\dot{Q}_{WATER}$ (kW)	1.98	1.23	1.122	0.875	1.705	1.289	0.883	0.666
$\dot{Q}_{COMP\ HD}$ (kW)	0	0.62	0	0.581	0	0.539	0	0.383
ΔQ_{HX} (kW)	0.14	(0.32)	(0.03)	(0.666)	(0.22)	(0.716)	(0.207)	(0.561)
Electrical Motor								
Motor Power Input (kW)	4.4	4.45	2.9	3.3	3.85	3.90	2.1	2.125
Summary of Losses								
Mass Leakage (kW)	0.074	0.085	0.006	0.051	0.057	0.067	0.114	0.108
Heat Transfer (kW)	0.173	0.04	0.188	0.038	0.183	0.396	0.158	0.302
$\Delta\dot{Q}_C$ (kW)	0.247	0.125	0.194	0.089	0.240	0.463	0.272	0.410

of the metal backup ring reduces both the volumetric efficiency and the required motor input power

- For all cases, the presence of water cooling of the compressor cylinder head, improves the isentropic efficiency because of an increase in the heat transfer

- At slower speeds the compressor isentropic and volumetric efficiencies are reduced because the ratio of the losses to compressor capacity is larger

- For most of the experimental runs, the heat transfer losses are greater than the flow leakage losses; however, both loss mechanisms contribute to the total compressor losses.

3.5 Expander Test Results

A summary of the expander test results is presented in Table 3-6. Five different experimental runs are presented. A 3 hp motor was used for the brake and the plastic piston with metal backup rings was employed. Again there were variations in the physical tests and it is worth discussing these variations. The first three tests were performed without thermal insulation and the latter two were performed with thermal insulation. Again the tests were performed at two different speeds; 1140 rpm and 840 rpm. Finally, one test was performed with the compressor port closed to determine if gas leaked past the piston rings.

The salient results of the expander test may be summarized as follows:

The flow leakage past the poppet valves during operation is approximately 3.5 to 4.3 scfm which represents 15 to 17 percent of the total expander capacity.

Table 3-6. Summary of Expander Test Results

	4-13-I	4-13-II	4-13-III Compressor Ports Closed	4-17-I Expander Insulated	4-17-II Expander Insulated
<u>EXPANDER</u>					
Speed (RPM)	1100	840	1105	1105	842
Gas Flow In (SCFM)	34.80	26.49	34.74	34.63	26.43
Gas Flow Out (SCFM)	30.84	22.99	30.46	30.57	22.59
Total Gas Flow Leakage (SCFM)	3.96	3.5	4.28	4.06	3.84
Gas Internal Energy (kW)	0.690	0.550	0.668	0.732	0.563
Indicated Power (kW)	1.001	0.657	0.794	0.838	0.565
Isentropic Efficiency					
Gas (%)	58.3	58.4	53.4	57.5	60.6
Power (%)	84.6	69.8	63.5	65.8	60.8
Volumetric Efficiency (%)	93.3	91.1	91.8	92.1	89.3
<u>ELECTRICAL MOTOR</u>					
Motor Power Ouput (kW)	0.635	0.580	0.645	0.640	0.575
Motor Efficiency (%)					
<u>SUMMARY OF LOSSES</u>					
Mass Leakage (kW)	0.090	0.084	0.094	0.097	0.096
Heat Transfer (kW)	<u>0.311</u>	<u>0.107</u>	<u>0.126</u>	<u>0.106</u>	<u>0.098</u>
$\Delta\dot{Q}_E$ (kW)	0.401	0.191	0.22	0.203	0.194

- Plastic Piston plus new ring with metal back-up bond
- 3 HP Motor for Brake

- The expander isentropic efficiency based on gas temperatures is approximately 58 percent and the volumetric efficiency is approximately 92 percent

- A significant heat transfer loss occurs for the expander. Approximately 16 percent of the refrigeration (extracted work) developed for the expander is lost to the ambient

- The heat transfer losses are of the same order of magnitude as the flow leakage losses.

3.6 Final System Test Results

After the compressor and expander tests were completed, a series of final system tests were performed. A summary of the system test results is presented in Table 3-7. The system operated with the plastic piston, thermal insulation and 3 hp motor. The system was operated at two different speeds; 1100 rpm and 55 rpm. Also, the system was operated at both speeds for the case of external water cooling of the compressor cylinder wall.

The results of the final system tests are summarized as follows:

- Compressor isentropic efficiency based on gas temperatures is 95 percent

- Expander isentropic efficient based on gas temperatures is 70 percent

- The estimated COP is 1.2 to 1.4

- The heating capacity is approximately 2 kW.

Table 3-7. Summary of System Test Results

	5-1-I	5-1-II	5-1-III	5-1-III
Comments				
• Plastic piston				
• 3 hp motor				
• Insulation				
• New Reed valves				
• External H_2O Cooling	No	Yes	No	Yes
Compressor				
Speed (rpm)	1100	1100	550	550
Gas flow in (SCFM)	31.2	31.9	20.2	20.1
Gas internal energy (kW)	2.468	2.160	1.074	0.902
Indicated power (kW)	2.767	2.984	0.996	1.029
Isentropic efficiency:				
Gas (%)	96.8	114.6	94.3	122.9
Power (%)	86.3	83.0	101.7	107.7
Volumetric efficiency (%)	92.7	94.2	85.7	87.9
Expander				
Gas flow out (SCFM)	31.0	31.5	14.33	14.7
Gas internal energy (kW)	1.100	1.232	0.510	0.548
Indicated power (kW)	1.055	1.064	0.426	0.426
Isentropic efficiency:				
Gas (%)	70.5	74.8	64.5	66.95
Power (%)	67.6	64.6	53.9	52.0
Volumetric efficiency (%)	93.8	95.3	86.7	89.0
Heat exchanger				
$\dot{Q}_{AIR}$ (kW)	1.937	1.497	0.928	0.717
$\dot{Q}_{WATER}$ (kW)	1.950	1.458	0.715	0.527
$\dot{Q}_{COMP\ HEAD}$ (kW)	0	0.614	0	0.286
$\Delta\dot{Q}_{HX}$ (kW)	(0.013)	(0.575)	0.213	(0.096)
Electrical motor				
Motor power input (kW)	2.42	2.45	0.980	0.990
Motor efficiency (%)				
Summary of Losses				
Total gas flow leakage (SCFM)	0.2	0.4	5.87	5.4
$\Delta\dot{Q}_C$ (kW)	0.299	0.824	(0.078)	0.127
$\Delta\dot{Q}_E$ (kW)	0.045	0.168	0.084	0.122
$\Delta\dot{Q}_{HX}$ (kW)	0.013	0.575	0.213	0.096
Net Loss	0.241	0.081	0.375	0.091
System Performance				
QH (kW)	1.950	2.072	0.715	0.813

4. SOURCES OF THERMODYNAMIC IRREVERSIBILITIES

4.1 Summary of Heat Pump System Losses

As was shown in the Design Report, the Brayton Cycle water heater was designed to achieve an overall COP of 1.7 or greater. To satisfy this performance goal, it was necessary to design a compressor and expander with an isentropic efficiency of 87 percent or greater. Hence, the major potential sources of irreversibility were identified and the system was designed to minimize the losses. Table 4-1 presents a summary of the predicted sources and values of the thermodynamic design losses occurring in the heat pump system. As one will observe, the compressor/expander losses contribute the most to the total system loss, comprising 53 to 65 percent. The electrical motor losses account for 22 to 30 percent of the total system loss.

The compressor/expander losses consist of the valve losses, mechanical losses and the internal heat transfer losses. The valve losses are a function of the valve flow area, the valve time of opening/closing or response time, and valve gas leakage. The mechanical losses are due to the piston ring friction and leakage, crosshead drive mechanism friction, rod and main bearings friction, and the valve drive mechanism friction. Finally, the internal heat transfer loss consists of a steady-state heat transfer loss and a transient or unsteady-state heat transfer loss. The steady-state heat transfer loss represents the heat transfer along the compressor/expander cylinder walls, across the piston, and along the valve and piston rods. The unsteady-state heat transfer loss represents the transient heat transfer between the gas and the compressor/expander cylinder walls and piston. Because of the transient behavior, the local heat transfer coefficients may be 2 to 10 times greater than the

Table 4-1. Summary of Heat Pump System Design Losses

Component/Type	Watts	Percent of Total
Heat Exchanger (ξ = 0.85)		
Pressure Drop Losses	4	
Temperature Differences Losses	100	
Heat Leak Losses	0	
Subtotal	104	17.5 to 13.1
Compressor/Expander (η_s = 87%)		
Valve Losses		
• Valve Area	59.6	
• Time of Opening/Closing	82.0	
• Valve Leakage	37.3	
Internal Heat Transfer Loss	37.3	
Mechanical Losses		
• Piston Ring(s)	14.9 to 82.0	
• Crosshead Drive Mechanism	14.9 to 59.6	
• Rod and Main Bearings	37.3 to 74.6	
• Valve Drive Mechanism	29.8 to 82.0	
Subtotal	313.1 to 514.4	52.6 to 64.6
Motor (η = 80%)		
Friction and Windage Losses	14.8	
Core Losses	24.5	
Load Losses	33.7	
I^2R Losses	104.7	
Subtotal	177.7	29.9 to 22.3
Total Heat Pump Losses	594.8 to 796.1 watts	100%

steady-state heat transfer coefficient [1]. Hence, unlike the steady-state case for which the gas heat transfer coefficient is the limiting heat transfer term, the unsteady-state heat transfer coefficient may become the dominant heat transfer term. There has not been enough experimental data present in the technical literature to substantiate quantitatively the magnitude of the unsteady-state heat transfer coefficient even though this transient phenomena has been observed in other heat pumps/engine applications [2].

Except for the electrical motor, the major losses for the heat pump system have been measured and are presented in Table 4-2. The first column represents the predicted or designed heat pump system losses which were presented in Table 4-1. The second column represents the experimentally measured/calculated heat pump system losses. Because of the difficulty in determining consistent electrical motor efficiency values, the motor losses were estimated to be 458 watts. As can be observed from the total heat pump losses, the actual total losses exceed the design total losses by 500 to 700 watts. The reason for the large discrepancy is that both the actual compressor/expander losses and motor losses are more than double their respective estimated losses.

4.2 Predicted Thermodynamic Performance

A summary of the heat pump system performance characteristics are presented in Table 4-3. The data for July 1, 1978 represents the initial preliminary test results which are also presented in Figure 3-4. The data for August 7, 1978 represents the initial preliminary test results that were obtained after the first generation design modifications listed in Table 3-3 were made. The results for August 7, 1978 are also listed in Figure 3-9. Finally, the data for May 21, 1979 represents the final system test results which are also presented in Table 3-7. The data reveals that the system COP has initially increased with time, i.e.,

Table 4-2. Summary of Heat Pump System Losses

Component/Type	Design (watts)	Actual (watts)
Heat Exchanger	(ε = 85%)	(ε = 90%)
Subtotal	104	25
Compressor/Expander	($\eta_E = \eta_C$ = 87%)	η_E = 70.5% η_E = 96.8%
Valve Losses	178.9	312
Internal Heat Transfer Losses	37.3	254
Mechanical Losses	96.9 - 298.2	250
Subtotal	313.1 - 541.4	816
Motor	(η_M = 80%)	(η_M = 60%)+
Subtotal	177.7	458*
Total Heat Pump Losses	594.8 - 796.1	1300

*Estimated
+ Actual efficient of the large 3 hp motor

from 1.12 to 1.26. The low value of COP for the more recent data is due to the influence of the additional piston friction due to the presence of the metal backup ring on the piston.

In Table 4-4 is presented a list of the thermodynamic performance projections. It is projected that with a new valve design and piston ring design that the expander isentropic efficiency may be increased to 75 percent which will result in a projected system COP of 1.57. The value is still less than the design goal of 1.7, which according to market report studies, represents a viable COP for a payback period of three years or less.

Table 4-3. Summary of Heat Pump System Performance

	Design	Actual 7/1/78	Actual 8/1/78	Actual 5/21/79
η_C	86.2%	76.5%	72.1%	96.8%
η_E	88.8%	88.0%	99.8%	70.5%
η_M	80.0%	72.0%	76.0%	81.1%
ε	87.0%	91.8%	90.8%	91.0%
$\dot{Q}_H$	2,302W	1,895W	1,733W	1,950W
$\dot{Q}_H$	2,626W	2,525W	2,141W	2,408W
COP	1.62	1.12	1.26	1.0
COP_S	1.78	1.17	1.34	

Table 4-4. Thermodynamic Performance Projections

Thermodynamic Performance Variable	Design	Actual	Projected
η_E	88.8%	65%	75%
η_C	86.2%	85%	85%
η_M	80%	76%	76%
ε	87%	91%	91%
$\dot{Q}_H$	2303W	1950W	2150W
$\dot{Q}._{H}$	2626W	2408W	2666W
COP	1.62	1.0	1.57
COP_S	1.78		1.66

5. CONCLUSIONS AND RECOMMENDATIONS

The conclusions derived from the development testing may be summarized as follows:

- The electrical motor maximum efficiency is 78 percent at a motor speed of 1140 rpm and a motor capacity of approximately 1.3 hp.

- The experimental heat pump system COP is 1.26 which is less than the design goal of 1.7 or greater.

- The results of the valve tests indicate that the experimental compressor and expander valve pressure drops for steady flow are less than the predicted valve pressure drops.

- The compressor isentropic efficiency is approximately 95 to 119 percent and the volumetric efficiency is 85 percent.

- The expander isentropic efficiency is approximately 58 percent and the volumetric efficiency is 92 percent.

- The flow leakage past the poppet valves during operation is approximately 3.5 to 4.3 scfm which represents 15 to 17 percent of the total expander capacity.

- A significant heat transfer loss occurred for the expander; approximately 16 percent of the refrigeration developed for the expander (extracted work) was lost to the ambient.

- For the expander the heat transfer losses are of the same order of magnitude as the flow leakage losses.

- From the results of the development testing, it is projected that with new valve designs, a system COP of approximately 1.57 may be achieved.

The recommendations for future work are the following:

- A small scale valve development be initiated to develop rotary valves which should reduce significantly the valve losses in the expander and mechanical losses in the crank case. In addition, the initial cost of the heat pump system would be reduced.

- An investigation of transient heat transfer phenomena should be undertaken. The results will provide heat pump/engine designers better information to predict quantitatively the effects of transient heat transfer on the system overall thermodynamic performance.

- In the event the rotary valve development proves to be successful, a second generation design of the heat pump system should be contemplated. The decision should be based on the updated projected heat pump system COP, revised market study results, and the energy situation in the United States.

REFERENCES

1. K.P. Lee and J.L. Smith, Jr. "Influence of Cyclic Wall-to-Gas Heat Transfer in the Cylinder of the Valved Hot-Gas Engine", Proceedings of the 13th Intersociety Energy Conversion Engineering Conference, Society of Automotive Engineers, Inc., Warrendale, PA, 1978, pp. 1798-1804.

2. Proceedings of the 1976 Purdue Compression Technology Conference, Purdue Research Foundation, 1976.