ENERGY CONSERVATION

ORNL/Sub/79-24610/3

Development of a Residential Gas Fired Absorption Heat Pump

Final Report

Component Development and Field Trial Program

Report Prepared by

Chemical Sector
ALLIED CORPORATION
Columbia Road
Morristown, New Jersey 07960

under
Subcontract 86X-24610C

for

OAK RIDGE NATIONAL LABORATORY
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the
U.S. DEPARTMENT OF ENERGY

OFFICE OF BUILDINGS ENERGY
RESEARCH AND DEVELOPMENT

and under
Subcontract 5014-341-0113

for

GAS RESEARCH INSTITUTE

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
NTIS price codes—Printed Copy: A11; Microfiche A01

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August 1985

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GAS RESEARCH INSTITUTE
Chicago, Illinois 60631

REPORT DOCUMENTATION PAGE	1. REPORT NO. ORNL/Sub/79-24610/3	2.	3. Recipient's Accession No.

4. Title and Subtitle

Development of a Residential Gas Fired Absorption Heat Pump. Component Development, Field Trial Program

5. Report Date May, 1983

6.

7. Author(s)

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8. Performing Organization Rept. No.

9. Performing Organization Name and Address

Phillips Engineering Company
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10. Project/Task/Work Unit No.

11. Contract(C) or Grant(G) No.

(C) 86X-24610C (ORNL)
(G) 5014-341-0113 (GRI)

12. Sponsoring Organization Name and Address

Oak Ridge National Laboratory
Martin Marietta Energy Systems, Inc.
P. O. Box Y
Oak Ridge, TN 37831

Gas Research Institute
8600 W. Bryn·Mawr Avenue
Chicago, IL 60631

13. Type of Report & Period Covered

14.

15. Supplementary Notes

This work was performed under a subcontract from Allied Corporation, Morristown, NJ operating under subcontract 86X-24610C from ORNL and 5014-0113 from GRI.

16. Abstract (Limit: 200 words)

This is the third report of the series on the development of absorption type gas fired heat pumps using organic working fluids. The residential heat pumps were to provide the full heating and cooling requirements of a house down to 0°F outdoor temperature. The subject matter of this volume concerns the design, construction, development and testing of early units for an outdoor field trial under both residential use and in continuous run operation outdoors. Simultaneously, those components of the heat pump system which had not attained full performance levels, i.e. components of the sealed absorption system and auxiliary components, were developed individually on breadboard test units to attain the design requirements for full performance. Six absorption heat pumps using R-133a and ETFE as the working fluids were designed, constructed, developed and tested as part of the field trial. Five units were operated outdoors during most of two winters and two summers, two on residences. Three were applied to the laboratory building, two of them operating on continuous run and the third on a continuous on/off cycle. The sixth unit was used for development of cycling operation to high efficiencies. Problems of operation in a northern climate were investigated during the outdoor operation of the five units during a total of 34,263 hours of operation.

17. Document Analysis a. Descriptors

Absorption
Heat Pump

b. Identifiers/Open-Ended Terms

c. COSATI Field/Group

18. Availability Statement	19. Security Class (This Report) Unclassified	21. No. of Pages
	20. Security Class (This Page) Unclassified	22. Price

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PREFACE

This report covers work performed by Phillips Engineering Co., St. Joseph, Michigan under subcontract to Allied Corporation. The authors of this report are:

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Mr. Kevin Murphy was the Program Manager for Allied Corporation.

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ABSTRACT

This is the third report of the series on the development of absorption type gas fired heat pumps using organic working fluids. The residential heat pumps were to provide the full heating and cooling requirements of a house down to 0°F outdoor temperature. The subject matter of this volume concerns the design, construction, development and testing of early units for an outdoor field trial under both residential use and in continuous run operation outdoors. Simultaneously, those components of the heat pump system which had not attained full performance levels, i.e. components of the sealed absorption system and auxiliary components, were developed individually on breadboard test units to attain the design requirements for full performance. Six absorption heat pumps using R-133a and ETFE as the working fluids were designed, constructed, developed and tested as part of the field trial. Five units were operated outdoors during most of two winters and two summers, two on residences. Three were applied to the laboratory building, two of them operating on continuous run and the third on a continuous on/off cycle. The sixth unit was used for development of cycling operation to high efficiencies. Problems of operation in a northern climate were investigated during the outdoor operation of the five units during a total of 34,263 hours of operation.

Executive Summary

EXECUTIVE SUMMARY

This report covers the development of those components of the R133a/ETFE heat pump which had not reached full performance at the start of the program, and also the construction and testing of six preliminary field trial units. The purpose of the field trial units was to expose research models to northern weather conditions year-round, in order to determine how well the existing designs stood up to the weather and to learn what unexpected problems had yet to be solved or encountered. In order to provide maximum exposure, early installation was given first priority at the expense of COP's or complete development of the components.

Six field trial units were constructed. Five were operated outdoors for most of two winters and two summers. Two were operated in residences, providing all the heating and cooling requirements to outdoor temperatures below 0°F. The other three were operated on the roof of the laboratory building, being used to heat and cool the building. Two of the units on the laboratory were run continuously, being changed either manually or automatically between cooling mode and heating mode as the indoor loads required. The third laboratory unit was operated similiarly but was cycled on and off on a fixed cycle. The sixth unit was used for testing in a controlled temperature room to develop the means of obtaining high efficiency under cycling operation.

The five units were operated outdoors for a total of 34,263 hours. The maximum operating time for one unit was 13,614 hours. The second continuous run unit was removed from test at 7,913 hours, disassembled, and cut apart for inspection. The aluminum surfaces of all components were found to be unaffected on both the organic fluid sides and the permanent antifreeze/water sides. Evidence of overheating was found only on two small spots in the boiler. The

overheating was due to the specific design at those spots. The design can be changed to eliminate the condition without affecting the overall design. Due to leaks and other construction difficulties, the units at various times operated with air and humidity in the systems. Such conditions have also occurred at times in laboratory testing. In no case was any harm indicated other than a temporatory effect on performance. The residential units provided the desired comfort conditioning during winter and summer except during a period of unusual storms which will be discussed later. The test program also served its primary purpose by uncovering a number of problem areas which required further development work. The more common of the difficulties are listed below:

Problems with the purchased gas valve/ignition system. They included power loss from the high voltage ignition cable, the proof-of-pilot sensor, gas valve operation and transformer failures.

Eight way valve leaks and internal friction at 0°F temperatures.

Non-uniform movement or "oil canning" of the power element diaphragm of the thermostatic expansion valve, as well as insufficient temperature range without resetting.

Movement of the pump motor mount, which allowed the magnet to touch the stainless steel housing.

Variability of operation of the generator level sensors.

Other difficulties related to fabrication, in welding and leaks. The majority of the difficulties were related to auxiliary components or fabrication problems rather than to performance of the absorption sealed system.

During the second winter the units were exposed to unusual storms with winds to 70 miles an hour at temperatures from 0°F to -10°F. At that temperature the water vapor picked up across Lake Michigan was in the form of a fine ice fog. The ice particles were blown through every crevice in the unit as well as through the fins of the outdoor coil. The ice packed around all components and controls. The units all shut off under those conditions, in some cases apparently while attempting to defrost.

Methods of improving the cycling efficiency to over 90% of steady state efficiency were developed. Part of those means were applied to the residential field trial unit, resulting in operation at 85% of steady state efficiency, which was considered to confirm the laboratory results obtained with more sophisticated controls.

The components for the heat pump that had not reached full capacity or performance were developed to their full specifications, using commercially available aluminum materials, i.e. round tubing, etc. The component performance and the matching required to achieve the heat pump objectives were determined.

The regenerative turbine pump was chosen as the most desireable for pumping the R133a/ETFE heat pump fluids. The efficiency of that pump system was increased, reducing the wattage to the motor to 5% below the objective of 400 watts. Continuous run operation of two regenerative turbine pumps was continued to the 50,000 hour objective, equivalent to 15 to 20 years of operation. Two other pumps were run on start-stop operation for over 230,000 cycles, also representative of over 20 years of operation. No indications of approaching failure were evident on any of the pumps.

INTRODUCTION

INTRODUCTION

Background

The working fluids that have been practical for broad use in absorption refrigeration systems have been limited to the two inorganic pairs, ammonia/water and water/lithium bromide. Yet, many investigations and trials of organic fluids in operating units have been made during a period of over 100 years. (1) The most extensive were those of Zellhoffer, including the construction of limited quantities of the Williams Air-O-Matic water-cooled air conditioners in the 1930's and 1940's. No commercially successful products had been possible with organic fluids.

The investigations of this project were related to the development of air-cooled absorption (heat-actuated) heat pumps using new organic fluid pairs developed by the Allied Chemical Corporation. The major part of the program has been related to systems using R133a, 1-1-1 trifluorochloroethane, as the refrigerant and ethyltetrahydrofurfuryl ether, ETFE, as the absorbent. Limited investigations were also made with the lower-pressure refrigerants R123, (CF_3 $CHCl_2$) and R123a, (CF_2Cl $CHFCl$).

This report concerns only the design, construction and development of test heat pumps and their components using R133a/ETFE.

Scope

The development of the hardware of the absorption heat pump based on the R-133a/ETFE fluids, under contract 86X-24610C, "Advanced Development of Residential Gas-Fired Absorption Heat Pumps", for the U.S. Department of Energy and the Gas Research Institute could logically be divided into the following three parts.

The first part consisted of the development of components and systems for field trial, using commercially-available aluminum shapes (primarily round tubing) for the sealed absorption system. The second part included the development of more compact components by means of special aluminum extrusion shapes. Using those more compact components, total heat pumps were then developed to fit into cubes approximately 40 inches on a side, referred to as the M^3 (meter cube) prototypes. The third part of this program was the investigation, and limited testing of lower-pressure refrigerants, R-123 and R-123a in operating units.

The following report refers to the first part. It includes the development of those heat pump components which had not reached full performance, and to the investigations and development of six residential, heating and cooling, heat pumps for field trial. The Field Trial units were used for life testing, under continuous and cycling operation, and for field testing in residences. Investigations of means of reducing the heating/cooling losses that occur in cycling were also conducted.

Due to the use of available materials of construction, the Field Trial units of this first outdoor test program were expected to be too large and too tall to be marketable heat pumps. The purpose of the Field Trial was to expose the units to outdoor operation over as long a period as possible in residential operation and in continuous-run life test, with the objective of encountering and investigating the conditions and problems they would be subjected to in winter and summer use in a northern climate.

The schedule requirements, and the need to put the Field Trial units in operation as soon as possible, prevented the proper sequencing of component development prior to the construction and testing of the Field Trial units.

To obtain the field and life test operating hours desired, the design and construction of the Field Trial units had to be started at the beginning of the project. The general system sizing was intended for three tons of cooling and 90,000 Btuh heat output. However, because most of the system components had not reached the full capacity or efficiency, the desired capacity and coefficients of performance of the Field Trial units were subordinated to the need for early exposure to outdoor weather conditions. Thus, no hard-and-fast performance specifications were placed on the Field Trial units. It was primarily intended that they operate reliably so that they be able to experience outdoor conditions and weather variations for a minimum of a year, and preferably longer.

The development of those components which had not been brought to target performance during the preceding project was scheduled to continue so that the capacity and coefficient of performance objectives of 1.25 in heating mode and .5 in cooling mode might be met at rating conditions. Component performance improvements achieved in time for inclusion in Field Trial units were to be utilized as they occured.

Objectives

The objectives for the work reported herein were:

1. To improve to full performance those absorption sealed system components which had not reached system requirements for capacity, efficiency or size. The components included the generator, absorber, solution pump, refrigerant and weak solution flow controls and concentration controls. The evaporator and precooler were added to these when the project started.

2. To develop those auxiliary components which had not attained the specifications for operation in the heat pump system. Those components included the eight-way valve, the outdoor finned coil and fan, an indoor coil and blower, and the heat pump electronic controls.

3. To design, construct, test and install six absorption heat pumps for outdoor operation, to be operated for a period of at least one year. Two were to be operated continuously as a life test, and the other four were to be operated under start-stop cycling conditions, two of the four in residential use.

This report is divided into the following four sections: Absorption Components, Auxiliary Components, Solution Pump, and Field Trial Units.

1.0 ABSORPTION COMPONENTS

1.0 GENERATOR

INTRODUCTION

In the organic absorption fluid pair, 1,1,1, trifluorochloroethane and ethyltetrahydrofurfuryl ether (R-133a/ETFE) the absorbent, ETFE, has a small vapor pressure relative to R-133a. Yet, with a boiling point of 315°F, that vapor pressure is significant at generator temperatures. Hence, instead of merely boiling the refrigerant out of solution, it is necessary to distill the refrigerant to obtain the refrigerant purity required in other parts of the system. The generator must therefore include a distillation column along with the boiler. The distillation column is normally divided into a stripping section ("analyzer") below the feed point and a rectifier and reflux condenser above the feed. In order to conserve energy and space in this design, the rectifier coil was made to serve not only for vapor-liquid contact but also as a partial condenser.

Since the "absorber-heat exchanger" cycle was to be used in this project, recuperation of heat from the hot weak solution leaving the boiler was also required. Sensible heat from the weak liquid was to be transferred by counterflow heat exchange to the solution flowing down the analyzer. The analyzer was thus composed of finned coils which served as packing and as the heat transfer tubing for the weak liquid from the boiler. A diagram of the generator design is shown in Figure 1-1.

As an absorption heat pump system intended to operate winter and summer, the system required a concentration control subsystem to adjust the relative concentrations of refrigerant and absorbent for best operation over a range of evaporator temperatures from 0°F to 45°F, or more. One component of the

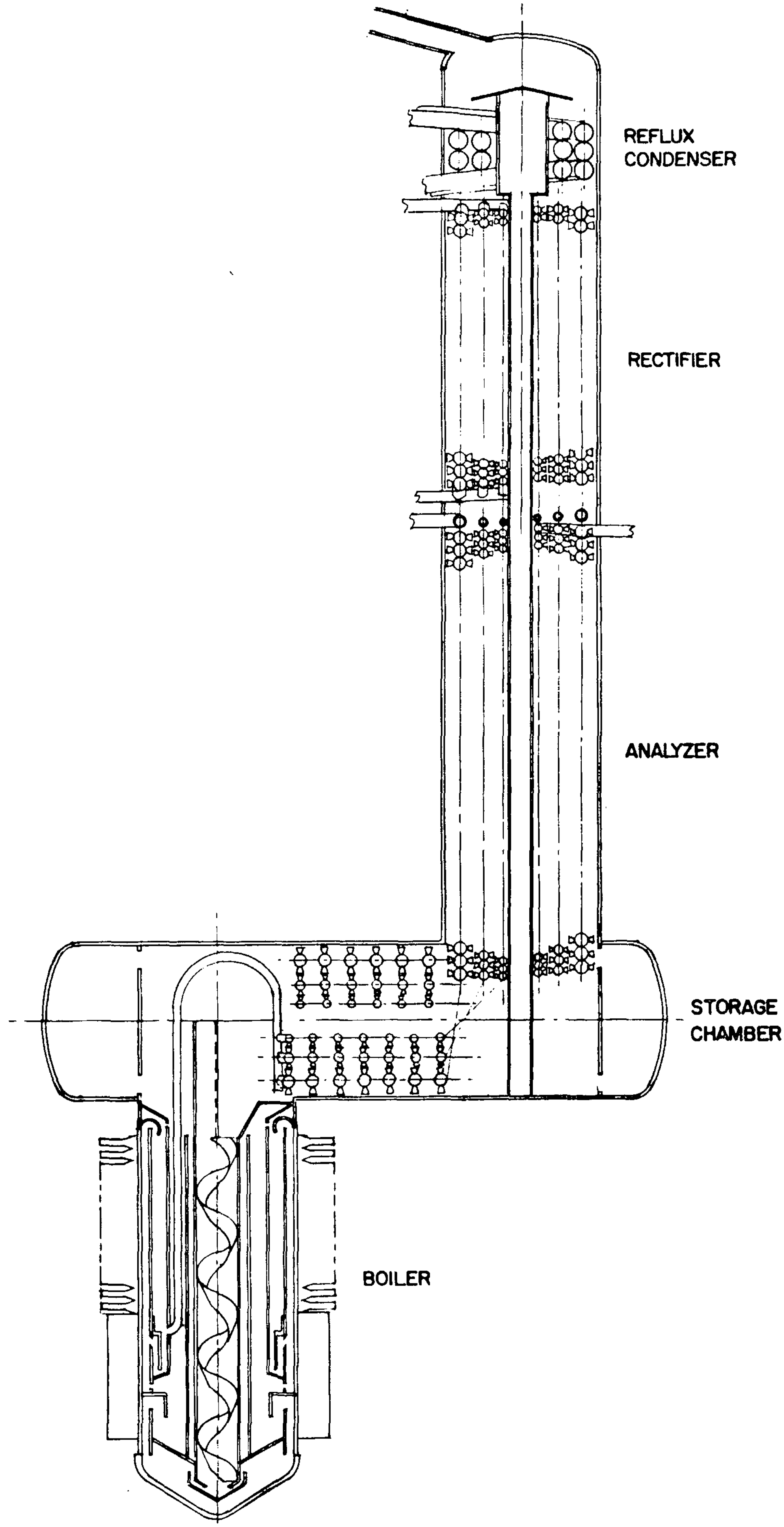


Figure 1-1 Field Trial Unit Generator Design

concentration control was a solution storage chamber which was placed between the analyzer and the boiler. The storage chamber was in the form of a horizontal tube which also housed additional heat recuperation surfaces.

Another generator requirement was that the working fluids not be overheated in the boiler as a result of inadequate heat transfer rates or localized hot spots. Of special concern was the possibility that the boiler wall surface might be at an excessive temperature even though the liquid itself was not.

Most of the generator development had been accomplished in preceding projects. However, the refrigerant purity specification had not been reached at full heat input, nor had the boiler wall temperature been brought to the design objective. This project was to complete that development for construction of generators for the Field Trial heat pumps.

OBJECTIVES

The primary performance objective of the generator development was to increase its refrigerant output to 475 lb/hr of R-133a vapor at a 99% purity, with a heat input of 72,000 Btuh. The generator was also to be capable of producing over 500 lb/hr of refrigerant vapor without flooding of the six-inch diameter distillation column. The heat transfer into the boiler was to be achieved with wall temperatures no higher than 20°F above the boiling point of the R-133a/ETFE weak solution at the operating pressure.

STATUS AT BEGINNING OF PROGRAM

The generator had been developed to close to acceptable status. (2) It had performed satisfactorily at reduced heat inputs. However, when operated at the specified 72,000 Btuh input, the generator was lacking in one or more of the following aspects: ability to carry the counterflow vapor and liquid through the packing without flooding, purity of the refrigerant, and heat transfer and proper separation of the fluids in the boiler. Concepts intended to solve these problems had been developed. Realizing that the absorber requirements would increase the height of the Field Trial units, a generator as in Figure 1-1 had been designed and built, with an overall height of 61 inches. The packing and heat exchange section of the analyser, or stripping section, had been increased from a height of 10 inches to 20 inches. The analyzer was constructed of three concentric coils of finned tubing made by modifying shredded-fin tubing, Part No's 450-1, 450-2 and 450-3, manufactured by Brazeway Inc., Adrian, MI, to the design shown in Figure 1-2. The rectifier section, formed of the same coils, was 15 inches tall. The reflux condenser coil, of plain tubing, was three inches high.

Due to flooding of the column when the fins of the Brazeway tubing had been set at a 45° angle to the tube axis in the preceding generator, a new analyzer and rectifier had been built with fins at 90° to the tube, i.e. essentialy vertical and aligned with the upward flow of the vapor and downward flow of the liquid. The purpose of that trial design had been to determine whether three concentric coils of that tubing in a 6" tube, might have vapor and liquid flow limits below the 500 lb/hr specification. That test generator had operated at 500 lb/hr of R-133a vapor without flooding. However, the refrigerant purity had dropped to the 98% - 98.5% range. To improve the design toward 99% purity, analyzer and rectifier coils with the fins bent to a 60% angle to the tube were to be built and tested as this program began.

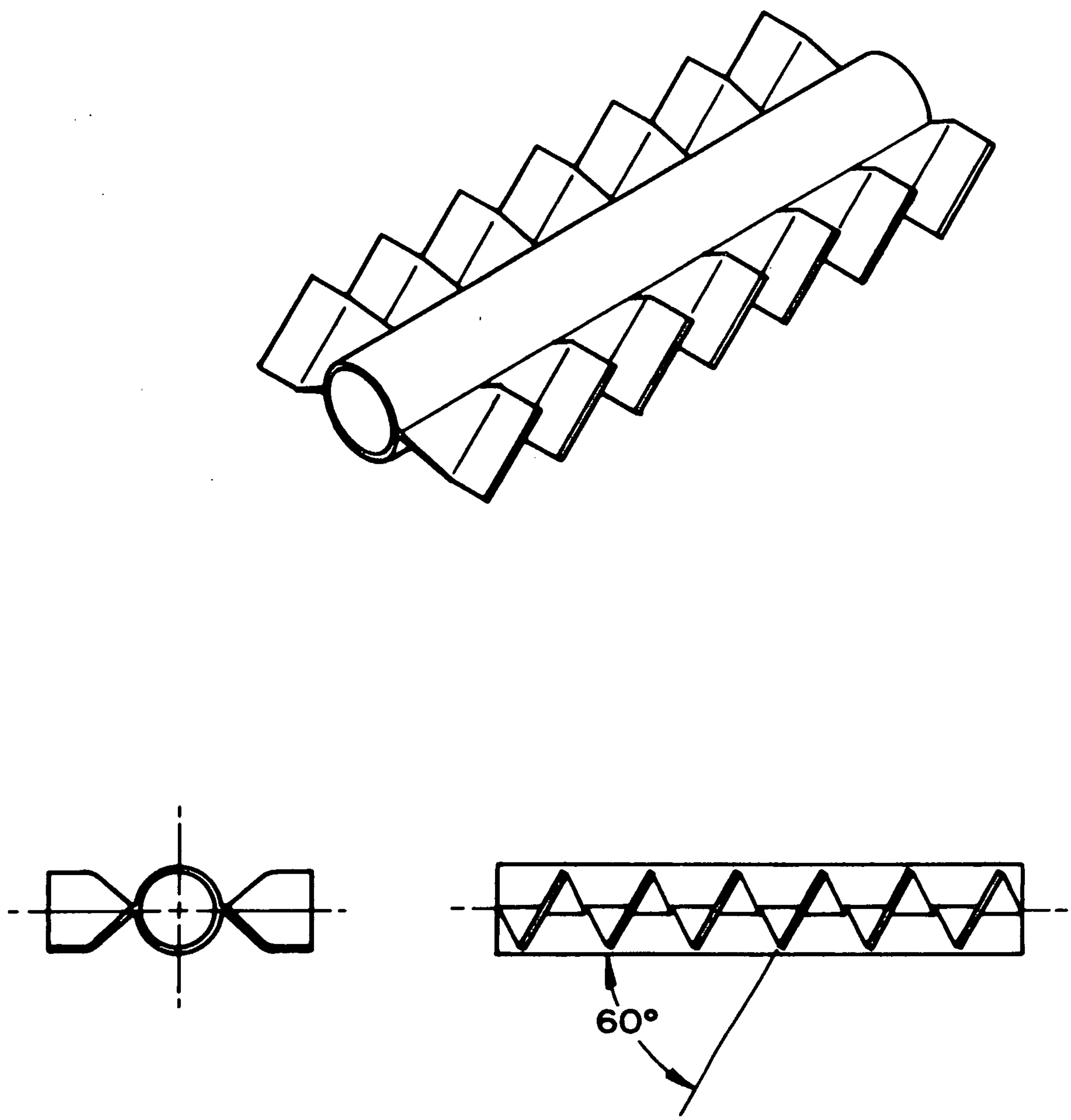


Figure I-2 Finned Tubing to Serve for Heat Transfer and as Packing in Distillation Column

The boiler, of the type shown in Figure 1-1, had longitudinal holes, 5/16" in diameter, gun-drilled into the boiler wall to form vapor lift tubes for enhancing the boiling heat transfer coefficients. Two types of difficulties had been encountered. The wall temperatures measured were somewhat above the specified limit of 20°F above the peak boiling temperature, and the outlet temperatures from the vapor lift tubes were appreciably higher than the boiling point of the weak solution flowing to the absorber. The concentrations of the weak solutions, as analyzed at the Buffalo Laboratory, indicated that after reaching their minimum concentrations at the peak temperature at the outlet of the pump tubes the weak solutions had been enriched, either by mixing with richer solution or by absorption of vapor. The boiler had then been redesigned to drawing No. 422-1, equivalent to Figure 1-1. By means of that redesign, the weak solution concentration had been brought to closer correspondence with the peak boiler temperature. The wall temperatures had not been measured as this project started.

INVESTIGATION PLAN

The plan was to continue the distillation column development by adjusting the angle of the Brazeway coil fins to attempt to reach the specified refrigerant output and purity without flooding in the existing column design. Preceding results indicated those steps should be successful, but if not, other column designs were to be utilized. The testing of the newly-constructed boiler was to be continued, measuring weak liquid concentrations and boiler wall temperatures to determine whether the new design had brought the boiler operation to the specified objectives. If negative results were obtained, a decision on the generator design for the Field Trial units was to be made within two months of the start of this program.

REFRIGERANT PURITY AND OUTPUT

A distillation column was constructed with the analyzer and rectifier, Figure 1-1, made of three concentric Brazeway coils having all the fins set at a 60° angle with the horizontal.

Testing of the new column demonstrated that flooding was not a problem with fins at a 60° angle. The following table lists representative results.

TABLE 1-1

DISTILLATION TEST DATA

Gas Input, Btuh	72,144	74,679	77,905	72,210
Evaporator, °F	42.1	40.2	40.7	39.7
Refrigerant Rate, Lbs/Hr	475	500	523	495
Refrigerant Conc., %	98.7	98.5	98.3	98.8
Cooling Capacity, Btuh	35,703	37,445	38,606	37,880

As shown, refrigerant flow rates up to 523 lb/hr had been obtained without flooding. Since the 500 lb/hr target was exceeded, no attempt was made to determine the point at which flooding would begin. The refrigerant purity had been improved, but had not reached the objective, being up to 98.7% at a refrigerant rate of 475 lb/hr. In the last test shown in Table No. 1-1, adjustment of charge and flow rate had resulted in purity of 98.8% at a refrigerant rate of 495 lb/hr.

Analysis of column operation indicated that the flooding problems previously encountered would have initiated in the analyzer section below the feed, and not in the upper sections. Therefore, rather than attempting to adjust the fin angles in the analyzer to operation at just above a flooding condition, it was decided to keep the 60° angle in the analyzer, and to use the fin angle in the rectifier as the means of increasing refrigerant purity. Heat transfer rates previously obtained on rectifiers at various fin angles provided the basis for expecting significant improvement by that means. The following table lists those heat transfer data.

TABLE 1-2

RECTIFIER HEAT TRANSFER

HEAT INPUT TO BOILER, Btuh	FIN ANGLE	UA, RECT., Btuh/°F
72,830	90°	195
72,290	60°	304
72,070	45°	512

The UA product, from the usual heat transfer equation, $Q=UA\Delta t$, represents the heat transferred per degree Fahrenheit log mean temperature difference. The large change in UA with fin angle may represent not only better contacting and turbulence but also prevention of channeling of vapor through the fins. From that viewpoint, if flooding does not occur, the use of the lower fin angles would appear to be a necessity for improved vapor-liquid equilibrium. The next column was therefore built with the fin angles of the analyzer coil at 60° and those of the rectifier at 45°, as in Figure 1-3.

Due to other testing programs on Test Unit 1, this new distillation column was installed on Test Unit 2. The tests are therefore not completely comparable. The components of Test Unit 2, other than the generator, were more limited in performance than those on Test Unit 1; therefore, the system operation as a whole was not equal to operation under the previous test conditions. However, the distillation performance reached the objectives, as listed in the following test data:

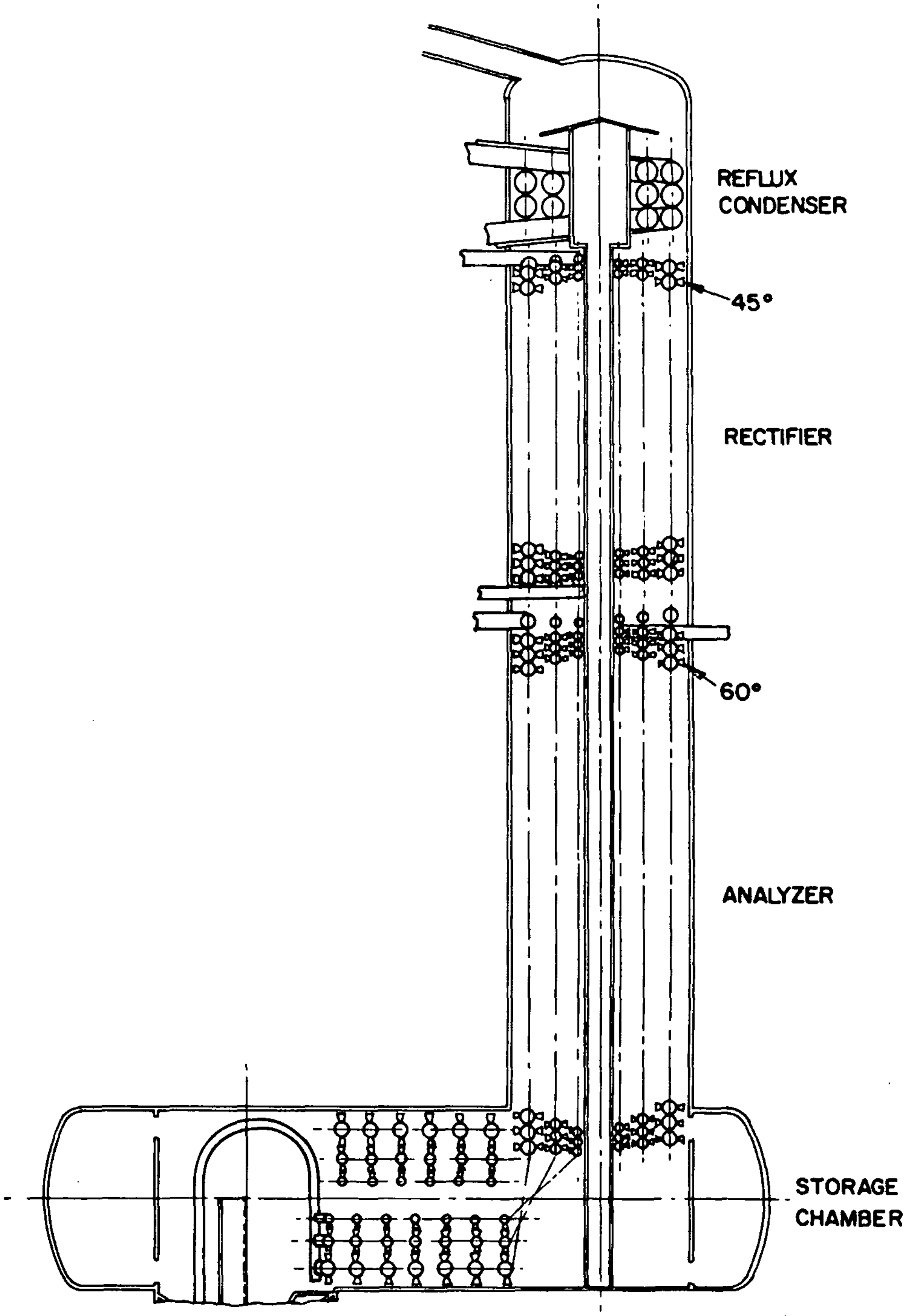


Figure I-3 PT Unit Generator Design

TABLE 1-3

DISTILLATION COLUMN PERFORMANCE

Gas Input, Btuh	72,650	72,730
Refrig. Rate, lb/hr	450	490
Refrig. Concentration, %	99.1	98.9
Cooling Capacity, Btuh	32,730	35,650

The gain in refrigerant purity was small, considering the UA rise expected, but the average was equivalent to 99% purity at 475 lb/hr of refrigerant. No further improvement was attempted, partly because other developments had reduced the need for peak refrigerant purity slightly.

Generators with distillation columns having 60° fin angles in the analyzer and 45° fin angles in the rectifier were installed on the Field Trial units of the field and life testing program. Fewer measurements could be made on those units than on the Test Units, but the data available on the first Field Trial units indicated the refrigerant purities to be in the 98.9% to 99.0% range. No flooding difficulties were encountered. Recent testing of a similar generator on Test Unit 1 has also confirmed the 99% operation.

Improvements to the evaporator and precooler had reduced the need for 99% purity at the warmer ambients. The refrigerant concentration required for good system performance at 47°F ambient conditions had dropped to the 98.5% level. However, the purer refrigerant eases the operational requirements of the thermostatic expansion valve, and improves system performance at low ambient temperatures. Therefore the 99% purity remains a design objective, but is less critical.

BOILER OPERATION

In the preceding program, the boiler section of the generator had been redesigned per drawing #422-1 and Figure 1-1, to incorporate detail improvements in order to reduce the possibility of enrichment of the weak liquid within the boiler and also to improve heat transfer on the solution side so that the wall temperatures would be no more than 20°F above the boiling point of the weak solution. In prior work, (2) wall temperatures, measured with thermocouples on the outer surface just above and below the fins, had been brought down close to 20°F above the peak liquid temperatures. In a few cases, the 20°F figure had been reached. The redesign (per drawing 422-1) was made to assure consistent operation below that limit.

Enrichment of Weak Liquid

The possibility that the weak liquid was being enriched within the boiler had surfaced in tests during 1979 when gas chromatography analysis of the weak solutions consistently gave concentrations up to 3% higher than those determined from the temperature and pressure at the outlet of the boiler tubes. Those results indicated a possible enrichment of the weak liquid downstream from the boiler tube outlets. If true, that enrichment would probably have occurred in the vapor separator chamber.

Preliminary testing of the redesigned boiler just prior to the start of this program included a temperature measurement in a well at the liquid outlet from the separator chamber. The temperature was found to be higher than the temperature at the outlet of the boiler tubes, providing an indication that enrichment was no longer occurring in the separator chamber, but rather that some slight additional boiling might have resulted.

Further testing under this contract has confirmed those results and enlarged on them. The following table presents data from the redesigned boiler.

TABLE 1-4

WEAK LIQUID CONCENTRATION IN THE BOILER

High Side Pressure, psia	59.5	61.5	62.4	61.4
Boiler Tube Outlet Temp., °F	305	339	335	350
Corresponding W.L.* Conc., % R-133a	12.7	8.6	9.3	7.3
Separator Outlet Temp., °F	306	345	340	353
Corresponding W.L.* Conc., % R-133a	12.5	7.9	8.7	7.0

* Weak Liquid

It can be seen that the solution temperature at the separator outlet was in all cases higher than at the boiler tube outlets, averaging almost 4°F higher. A 4°F increase represents about 1/2% reduction in the weak liquid concentration. The solution concentrations listed result from solving the following vapor pressure (PTX) equation from page 71 of the 1979 report to the Gas Research Institute (2).

$$P = 10^A + 10^B x + 10^C x^2$$

$$A = \sum_{m=0}^{2} a_m T^{-m}$$

$$B = \sum_{m=0}^{2} b_m T^{-m}$$

$$C = \sum_{m=0}^{2} c_m T^{-m}$$

Where x = Wt. fraction R-133a
T = Temperature, °R
P = Pressure in PSIA

In a separate test, gas chromatography analyses of weak solution samples were made. Those analytical results compare to the pressure-temperature determinations as follows:

TABLE 1-5

COMPARISON OF WEAK LIQUID CONCENTRATIONS

High Side Pressure, psia	61.4
Boiler Tube Outlet Temp °F	346
Corresponding W.L. Conc., % (PTX)	7.8
Separator Outlet Temp. °F	350
Corresponding W.L. Conc., % (PTX)	7.3
Analytical Conc., Avg. of 4, %	6.1

Both the analytical concentration and the concentration at the separator outlet can be seen to be lower than the concentration at the boiler tube outlet. That result was a major improvement over the analytical concentrations that had been obtained in prior boilers, (up to 3% richer than the PTX concentrations) and confirms the conclusion that the enrichment problem was eliminated by the new design. The 1.2% discrepancy between the analytical and the separator outlet concentrations is larger than desired but can fall within the limits of experimental error and uncertainties in the PTX equation at that concentration.

Reduction of Boiler Wall Temperatures

The boiler wall temperatures were measured in other tests. Thermocouples were embedded in the aluminum wall at diametrically-opposed locations both above and below the fins. The following data from two tests show comparisons of those wall temperatures with internal readings of the weak liquid temperatures when operating at the 72,000 Btuh heat input range.

TABLE 1-6

BOILER WALL TEMPERATURES

Weak Liquid Temp., Boiler Tubes, °F	353	335
Weak Liquid Temp., Separator Outlet, °F	357	340
Wall Temperatures Above Fins, °F *	338, 338	323, 324
Wall Temperatures Below Fins, °F *	356, 347	320, 330

* Wall temperature readings taken at diametrically-opposed locations.

As a result of the improved boiler design, the wall temperatures had been expected to drop to within the specified 20°F differential, but, as seen in Table No. 1-6, those wall temperatures were found to have dropped to below the weak liquid boiling point. A reduction that large was unexpected and required investigation.

Figure 1-4 is a drawing of a cross-section of the boiler wall and related parts. Previous wall temperatures had been measured with thermocouples in the aluminum wall at the top and bottom of the fins as shown at Points A. The thermocouples had been fiberglass-shielded. That fiberglass shielding of the bottom thermocouples had at times melted in the flue gases. The new measurements were made with the thermocouple leads encased in ceramic shielding of larger diameter which was not affected by the flame. The new measurements were, therefore, less likely to read high; the wall was the coolest element in the thermocouple's environment. Since the new readings had changed in the proper direction and were unlikely to be lower than the actual wall temperature, an understanding of the reasons for wall temperatures below the peak boiler temperature was sought.

For the new measurements, due to the diameter of the ceramic insulators, the thermocouples had to be implanted at Points B in the boiler wall, about 3/8" from the root of the fin, up to 1/4" further than before. The temperature readings could therefore be expected to be lower, due both to the better boiler design and to the thermocouple location.

Considering first the bottom end of the fins, the solution being boiled there was relatively rich, compared to the weakest liquid at the top of the pump tubes. It would have had boiling temperatures 50°F to 75°F lower than the peak boiling point of the weak solution at the top. The wall temperatures

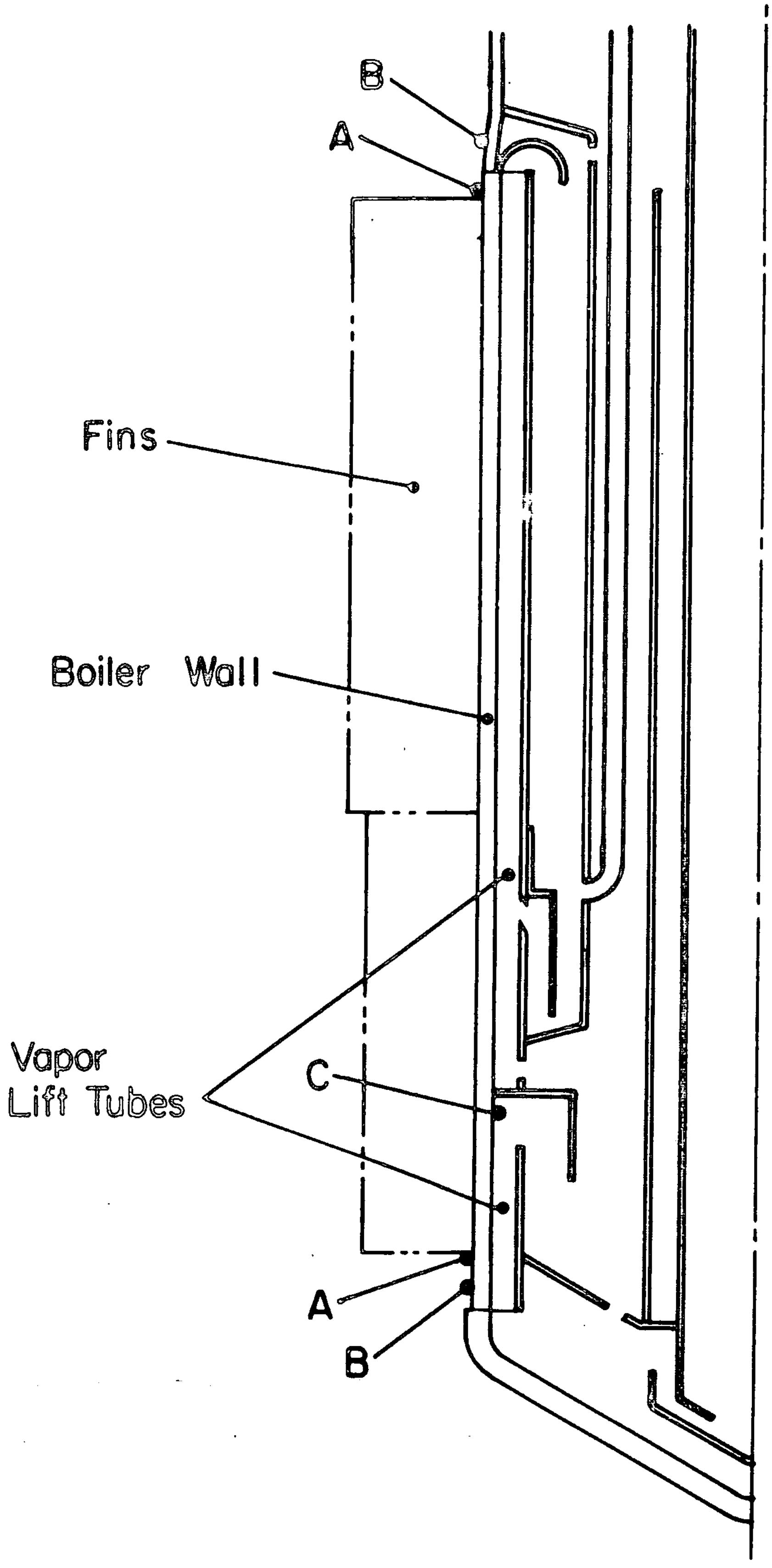


Figure 1-4 **Boiler Wall Cross Section**

measured ranged from 3°F above to 15°F below the temperature at the outlet of the vapor lift tubes, averaging 6°F below. Calculation of the boiling heat flux at the inlet section of the vapor lift tubes and on the boiler end cap determined that temperature drops of 20°F to 30°F along the aluminum wall, from the root of the fins to the measurement points B, were easily possible. The temperatures read were therefore not unreasonable at that location.

At the upper end of the fins, the situation was not as straightforward. At that level, the liquid being heated was at its peak temperatures. The wall temperature need not have been much above the liquid boiling point because the heat flux would have been at its minimum (flue gases only 50°-75°F above the weak liquid boiling point) and the heat transfer coefficient on the liquid side at its maximum. Nevertheless, the wall temperature would have been expected to be equal to or slightly hotter than the liquid. A review of a cross-section drawing of the boiler wall just above the fins, provided the explanation. It showed that immediately above the point where the thermocouple was implanted there was a horizontal partition separating the boiler section from richer solution. That solution would be expected to be at a boiling point 75°F or more below the temperature at the boiler tube outlet. Longitudinal heat transfer along the wall to that liquid could therefore lower the temperature at the new thermocouple location. Calculation of heat transfer for that temperature difference along the aluminum wall again indicated that the temperature read was of the right magnitude. The temperatures read, while unexpected, thus appeared to be proper. Based on those temperatures, the wall temperature goals, as written, had been met. The past basis for concern about wall temperatures had been corrected by the new boiler design, better measurement technique, or both.

Those unexpectedly low temperatures were also helpful with the question of whether the boiler wall might be above the 20°FΔt specification at inaccessible locations such as at a midpoint along the fins. The boiler had been designed with the concept of placing the liquid having the lowest boiling point at the location of highest flux, i.e. at the entry of the combustion products at the bottom of the fins. A high boiling Δt (up to 70°-90°F) would be available there to prevent exceeding the wall temperature specification. As boiling occured up the boiler tubes, the boiling point of the liquid would rise toward the peak. The Δt to the wall would decrease but the liquid side heat transfer coefficient would be higher and the heat flux would be lower as the flue gases cooled. Thus it was expected that if the wall temperatures at the top and bottom were within the specified limit, the others in between would be also.

These latest wall temperature measurements, with temperatures below the peak boiling point so close to the fins, indicate high boiling heat transfer rates for these organic fluids and reduce concern about a hot section in the middle of the finned length. The low probability of a hot section has also been confirmed by the appearance and composition of solution samples taken of Field Trial unit # PT-2 at intervals of about 2500 hours during 10,000 hours of outdoor operation. That data is listed in the Field Trial section of this report. A final confirmation resulted from dissecting Field Trial unit # PT-1 after 7,913 hours of operation which is also discussed more completely in the Field Trial Section. The boiler surface showed no evidence of overheating, except at the spot labeled C in Figure 1-4. That location had been designed as a vapor space into which the two-phase fluid from the lower sections of the boiler tubes might exit without having to displace liquid. A brown film was found at point C in the PT-1 boiler. A hot area was evidently created at that vapor space, presumably because the vapor space did not have the cooling ability of boiling liquid. In future designs, that vapor space should therefore be eliminated, even though the decomposition effect was minor. That brown discoloration was also an indication that similar brown spots would have been found elsewhere if there had been hot spots on the boiler surface. No others were found. The use of vapor lift tubes to increase the boiling heat transfer coefficient, combined with placing the lowest boiling liquid at the location of peak heat flux are indicated to have kept the wall temperatures below the decomposition limits everywhere but at that vapor lift discontinuity. Therefore, it appears from the temperature measurements and the life tests that the wall temperatures can be kept close to the peak boiler temperature, and the peak boiler temperature can, in turn, be used to judge the decomposition potential in relation to laboratory stability tests.

2.0 EVAPORATOR

INTRODUCTION

The evaporator and precooler had not been included in the original scope of work as sealed system components requiring improvement. But by the start of the investigation, experimental results had shown that higher evaporation capacities would be required at close approaches of the chilled water temperature to the evaporation temperature. Improved precooler performance was needed for efficient use of the evaporator overflow. The two components were added to the list of components to be further developed.

OBJECTIVES

The performance objective for the evaporator had been defined as:

Refrigeration Effect	36,000 Btuh
Evaporation Temperature	40°F
Chilled Water Temperature In	60°F
Chilled Water Temperature Out	50°F

The chilled water was to be a 35-40% ethylene glycol-water antifreeze solution, to allow for operation to 0°F. The above specification was established on the basis of three-ton air conditioning operation using all the large house coil needed for winter heating. That coil was to have a minimum of 72,000 Btuh heat output in heating mode operation. In cooling mode use, it was calculated to be capable of 25% latent removal at the 50°F chilled water temperature. That calculation had been confirmed by psychrometric room tests of a trial A-coil in an earlier program.

Although the above objective had been developed on the basis of the air conditioning load, more detailed analysis has since indicated that an appreciably better evaporator is needed. The above specification represents a UA product of 2500 Btuh/°F. Calculation of the Δt's available in heating mode operation in a 47°F ambient versus a 36°F evaporation temperature has determined that the design UA product should be increased to close to 5000 Btuh/°F. As with the other components, a part of the objective was to arrive quickly at an evaporator design suitable for use in the Field Trial units.

STATUS AT BEGINNING OF PROGRAM

During 1979, in efforts to provide large evaporation surface areas in a small volume by means of available extrusions, evaporators had been tested having up to five coils of finned aluminum tubing of the type shown in Figure 1-2, the tube diameters ranging from 1/4 in. O.D. to 1/2 in. O.D. The tubes were formed into concentric helical coils with a vertical axis. The fins extended horizontally, and the tubes were in close-wound contact. These evaporators were 31 inches tall and had surface areas up to 80 square feet on the refrigerant side. Operating at about 80% of specified capacity with 99.2% refrigerant, the approach of the outlet chilled water temperature to the evaporation temperature had reached a 7.8°F minimum. Extrapolation to the specified conditions showed the performance to be about 90% of requirements. Because only small diameter tubes were available, the water-side pressure drop through the coils was very high. Overall heat transfer coefficients of the order of 50 Btuh/Sq Ft/°F were calculated, at a UA product of 5000 Btuh/°F.

PROJECT INVESTIGATIONS

Evaporator

A helical coil evaporator made of finned tubing of the Figure 1-2 type was on the Test Unit at the beginning of this project. It was a four concentric coil evaporator with tubes ranging in size from 5/16 inches on the inner coil to 1/2 inch O.D. on the outer. The fins were twisted at a 45° angle to the tube axis. The coils were 29 inches tall and had a total outside surface area of 59 square feet. Flow had been measured through the individual coils, and restrictions added when necessary to adjust the flow in each coil to match the inside heat transfer surface of the coil.

This evaporator was used on the Test Unit while other components were being tested. The following table of test results is listed here only to illustrate the surface usage factor that must be considered in judging evaporator performance.

TABLE 1-7

EVAPORATOR PERFORMANCE - FINNED COIL

Internal Area 27 Square Feet, External Area 59 Square Feet

Evaporator Temp., °F	45.4	38.9	36.5	34.9	40.2	37.0
Evaporator Load, Btuh	29490	26270	23790	17980	22060	14200
Water Temp., In °F	72.2	60.4	55.1	43.0	54.4	46.4
Water Temp., Out °F	60.6	49.7	45.1	34.9	44.9	40.0
LMDT *, °F	20.5	15.5	13.0	9.1	8.6	5.6
Approach, °F	15.2	10.8	8.6	5.6	4.7	3.0
U**, Btuh/Sq.Ft/°F	53	63	68	73	95	94
UA, Btuh/°F	1440	1695	1830	1975	2565	2535

* Log mean temperature difference
** U based on internal area

The test results listed represent a range of temperature differences between the evaporating temperature and the chilled water temperature. The U values in this table are based on the internal, or chilled water side, surface area. The UA products of this and subsequent evaporators are plotted in Figure 1-5. They are seen to be inverse functions of the temperature differential and of the approach. This inverse effect represents the extent to which the evaporator surface is wetted by refrigerant rather than true evaporator capacity. At the higher water temperature, and hence higher temperature differentials, the refrigerant was evaporated before it covered all the evaporater surface. The heat transfer coefficient, being calculated on the basis of the total area of the evaporator, was therefore low under those partial usage conditions.

Another aspect of evaporator performance concerns the purity of the refrigerant being used, i.e. the amount of absorbent in the refrigerant liquid. The effective evaporation area is the area covered by relatively pure refrigerant. For example, if the refrigerant enters at a 98% purity, evaporation will have reduced the purity of the unevaporated liquid to 90% when it is 80% evaporated, and to 80% when 90% evaporated. This reduction in the purity will increase the evaporating temperature as shown by the curves of Figure 1-6. The rapid rise in evaporating temperature greatly reduces the effectiveness of the surface on which the last 10% or 20% of the refrigerant is being evaporated if refrigerant purities below 99% are used. Since the useful span of chilled water temperature is limited, the use of the precooler as a means of utilizing that impure refrigerant liquid is very important.

An evaporator's capacity in a specific system can therefore be characterized only when all the surface is wetted with refrigerant and the refrigerant purity is sufficiently high. In Table No. 1-7, the coefficients based on the water-side surface area are seen to have reached fairly good levels. The UA products have reached the original project goal of 2500 Btuh/°F

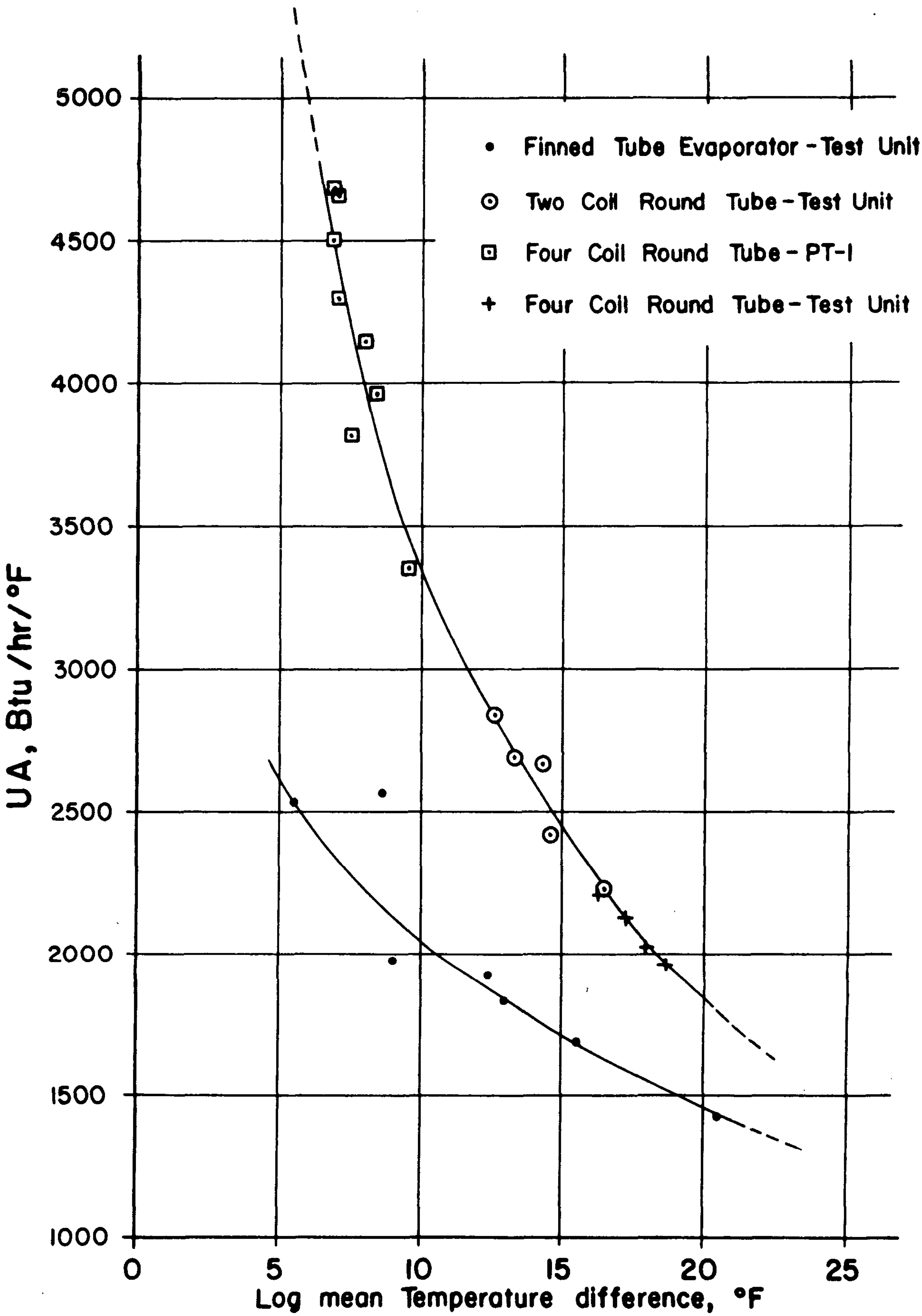


Figure I-5 Evaporator Performance, UA Product

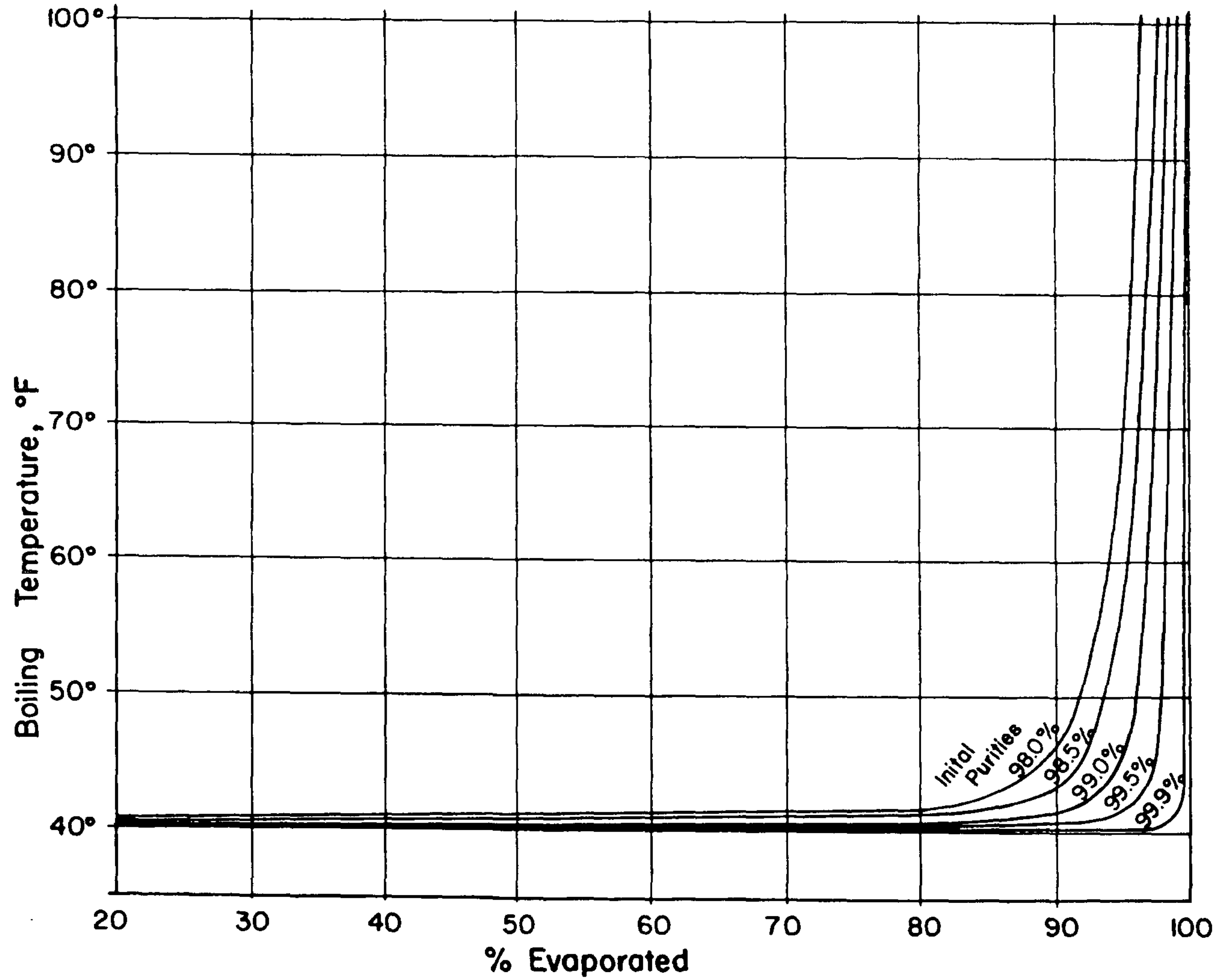


Figure 1 - 6 Change in Refrigerant Boiling Point with Evaporation, R133a/ETFE, 40°F

at the lowest temperature differences. As indicated earlier, that objective has now been raised.

To investigate the refrigerant flow over the surfaces, simple bench tests were made. A series of 5/8 inch diameter tubes aligned vertically on one-inch centers had been tested with a 15% solution of R-133a/ETFE distributed over the top tube by orifices spaced 5/8 inches apart horizontally. It was noted that the liquid tended to flow down from tube to tube in streams under the orifices, the areas between the streams being wet but relatively stagnant. To check the evaporator coil, straight finned tubes were mounted vertically one above the other and in contact with each other as in the evaporator. The fins made the streaming effect more difficult to distinguish, and probably helped the spreading, but film streams definitely occurred over only part of the surface. It was also noted that, in some locations, the liquid dripped off the tips of the fins rather than remaining in contact with the tube surfaces. It was obvious that good distribution over finned tubes of this type would be difficult to achieve consistently. These visual results supported the conclusion that this type evaporator had limitations that would prevent its reaching the new target.

Other design concepts were under consideration. A sample of a 3/8 O.D. aluminum tube having longitudinal grooves of capillary size as shown in Figure 1-7 had been obtained. A quick trial of the evaporation rate was made on the 2-foot sample in a vertical position to obtain a first-order indication of the potential evaporation rate. Warm water was circulated upward through the tube, and refrigerant flowed from the top of the tube downward onto the tube's grooved exterior. The water flow rate through the tube and the entering and leaving water temperatures were measured, as was the refrigerant temperature. From

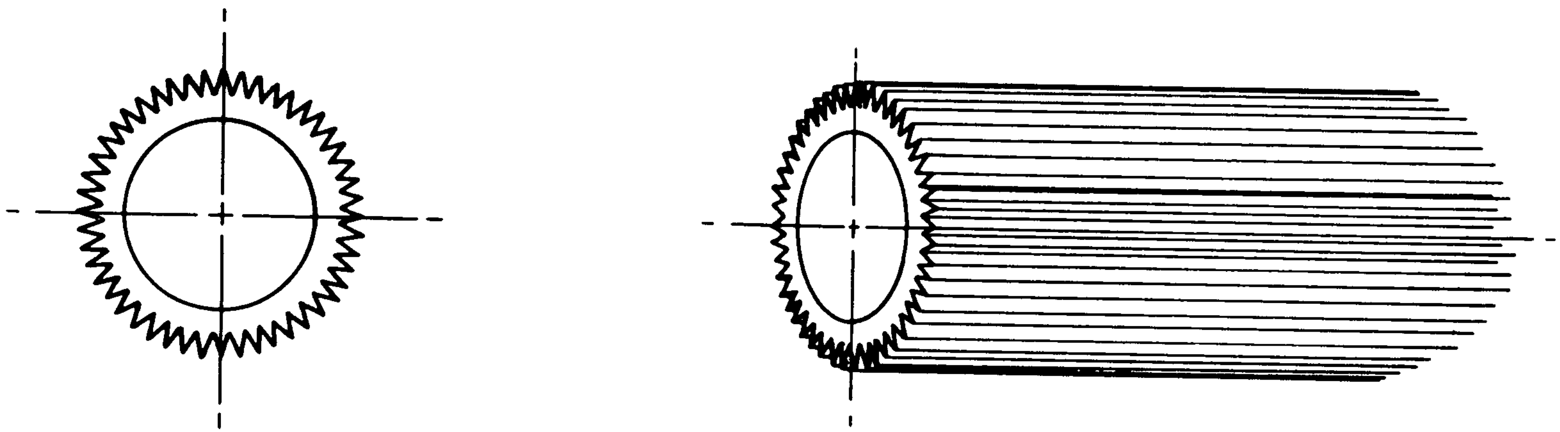


Figure 1-7 $^{3}/_{8}$ in. O.D. Serrated Tube those data, an overall heat transfer coefficient of 370 Btuh/Sq Ft/°F was calculated on the basis of the internal area.

This relatively high heat transfer coefficient would be partly due to the high evaporation temperatures and to flowing water through the tube instead of glycol/water solutions. Making corrections to convert to 40% glycol/water and to lower the evaporation temperature from 90°F to 40°F reduced the overall U value to 215 Btuh/Sq Ft/°F. Other corrections were probably required, but a significant net gain appeared likely. The indicated gain would be partly due to the increase in surface area on the refrigerant side and partly to the more uniform wetting produced by the capillary surface. Capillary surface, which has been very effective in ammonia water systems, was thus indicated to also be valuable to this organic fluid system. Unfortunately, no practical design using the 3/8 inch diameter tube could be devised for the three-ton evaporators of the Field Trial units, and leak-tight welding of the small serrated tubes was found to be very difficult. The development of evaporators using such grooving was left for the prototype M^3 unit program. A preliminary design of a grooved tube of larger diameter, shown in Figure 1-8, was drawn up, however. As can be seen, this design concept includes the use of capillary surface on the refrigerant side of the tube, plus internal fins to increase the glycol-water surface area.

The continued evaporator development for the Field Trial units was directed toward existing or more conventional shapes. Tests were made of coils that had previously been used as absorber surface. One was a flat extrusion coil similar to Figure 1-12. It had a large surface area, 80 square feet on the refrigerant side and 140 square feet on the glycol side. Modified for evaporator use, it did not operate well. There was loss of unevaporated

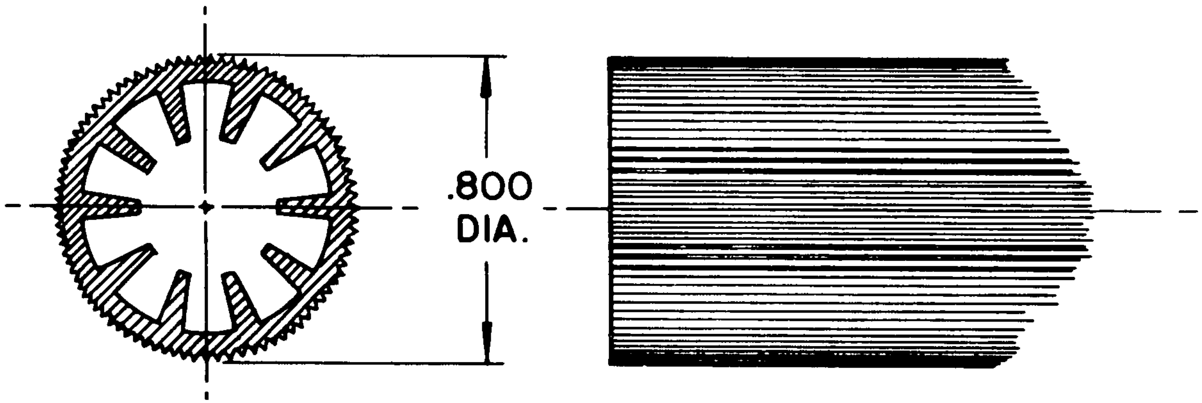


Figure 1-8 Evaporator Tube Concept

refrigerant to the precooler, and on to the absorber. The refrigerant could be seen to be dripping off the bottom of the coil at two locations rather than uniformly from the whole coil. It appeared that the refrigerant tended to flow in streams down the flat extrusion surfaces. Consideration of the time involved in construction of flat extrusion coils also led to a decision to find other designs.

Another available coil was the 59 inch high absorber coil consisting of two concentric helical coils made of 3/4 inch tubing. This coil was tried because it was thought that the larger round tubes would provide an improvement in refrigerant distribution and uniformity of wetting. The greater pitch of the coils, by promoting flow in the grooves between the tubes, might help spread out the supply of refrigerant over the surface. In addition, the coil had performed well as an absorber.

The coil was modified for evaporator operation and installed on the Test Unit. It performed well. No problems were encountered with refrigerant flow or wetting of the surfaces. Table No. 1-8 lists test results obtained with that coil.

TABLE 1-8

EVAPORATOR PERFORMANCE - ROUND TUBE, TWO COIL

Internal Area 33 Square Feet, External Area 40 Square Feet

Evaporator Temp., °F	38.5	38.2	39.7	40.2	42.1
Evaporator Load, Btuh	36870	35410	37880	35280	35700
Water Temp., In °F	62.1	59.7	61.5	60.4	61.8
Water Temp., Out °F	49.5	47.6	48.4	48.2	49.5
LMDT, °F	16.5	14.6	14.3	13.2	12.6
Approach, °F	11	9.4	8.7	8.0	7.4
U*, Btuh/Sq. Ft/°F	68	73	80	81	86
UA, Btuh/°F	2235	2425	2650	2675	2835

* U based on internal area

The temperature differences (LMDT's) in these tests were higher than those in Table 1-7 for unrelated reasons. At over the roughly-comparable LMDT range, the UA products are 30% to 50% higher than those of the finned coil of Table 1-7. The heat transfer coefficients, based on internal surface area, are 10% to 20% greater. If the external surface areas had been used as the bases for calculation, the relative gain in the U would appear much greater. The UA products, plotted in Figure 1-5, are seen to be not only improved but to represent an important trend to much higher UA's at the lower Δt's.

On the basis of those results it was decided to design the evaporators for the Field Trial units as concentric coils of round tubing. Limiting the design to the 45 inch height that would fit in the Field Trial unit, four concentric coils that were essentially equivalent to those in the absorber were utilized. The tubing ranged in diameters from 1/2 inch to 7/8 inch. The evaporator had 52.5 square feet of external area. The 40% glycol-water solution to be chilled was circuited to flow in parallel through all four coils from the bottom up, in counter-flow to the refrigerant. This evaporator, Drawing #410-4, shown in Figure 1-9, was put on the first Field Trial unit, No. PT-1. Test results on that unit are given in Table No. 1-9.

TABLE 1-9

EVAPORATOR PERFORMANCE
FOUR CONCENTRIC COIL DESIGN ON PT-1
Internal Area 43 Square Feet, External Area 52.5 Square Feet

Evaporator Temp., °F	45.2	46.2	46.5	51.8	49.5
Evaporator Load, Btuh	33270	32875	30100	30440	31660
Water Temp., In °F	58.5	59.0	58.0	63.0	60.9
Water Temp, Out °F	50.1	50.7	50.4	55.5	53.1
LMDT, °F	8.4	7.9	7.0	6.8	6.8
Approach, °F	4.9	4.5	3.9	3.7	3.6
U*, Btuh/Sq. Ft./°F	92	96	100	105	109
UA, Btuh/°F	3955	4140	4280	4495	4680

* U based on internal area.

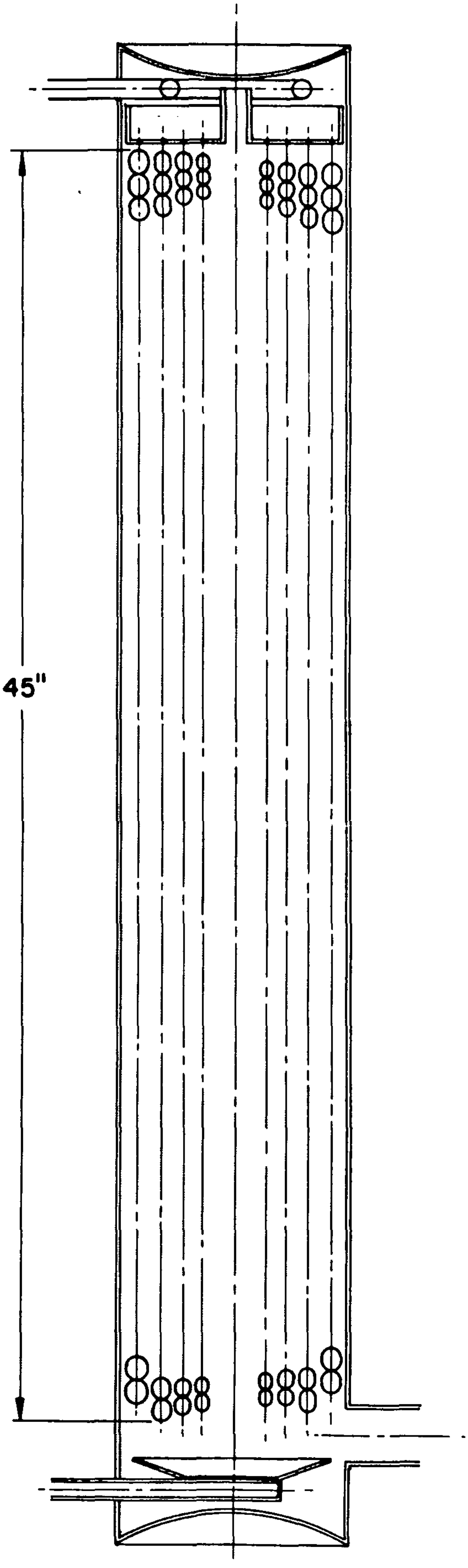


Figure 1-9 Evaporator - Round Tube, Four Coil Design

These Field Trial unit data were taken at lower LMDT's than the preceding two-coil evaporator. Higher heat transfer coefficients were obtained as a result of being able to operate to lower temperature differentials. Those U values, combined with the increased area of the four coils, resulted in the rise of the UA products to satisfactory levels. As can be seen on the plot of Figure 1-5, the UA product was increasing at a very high slope as the Δt was reduced. Extending the curve to the endpoint of the finned tube curve increases the UA product to well over the 5000 Btuh/°F goal set informally after the project started. The upper limit of the UA is not determinable from these data. The need to install the units for outdoor operation prevented further testing.

A four-coil evaporator of this design was installed later on the Test Unit. It was used on that unit primarily to load other components, however. As a result, tests were run at higher Δt's to allow full-load operation of the evaporator without careful control of the refrigerant flow rate. The UA data for a few of those tests are plotted on Figure 1-5. They can be seen to be essentially an extension of the curve for the tall, two-coil evaporator. Overall, the UA data for the round tube evaporators with both two and four concentric coils are seen to form one curve.

The evaporator development for the Field Trial units were considered completed by the results of the four-coil design. That evaporator was used in all six Field Trial units and performed without difficulty on all units. The information gained in the study was subsequently applied in the development of evaporator extrusions.

3.0 PRECOOLER

The precooler's function is to recover the cooling capacity of the cold vapor and liquid exiting the evaporator. It does so by counterflow heat exchange with the condensate flowing from the condenser to the evaporator. Normally, the cooling capacity of the vapor and liquid represents only a fraction of the sensible heat in the condensate. However, the difficulties in obtaining high refrigerant purity with short generators, and the need to operate with a minimum refrigerant temperature rise in the evaporator increased the quantity of unevaporated refrigerant exiting from the evaporator. That made it necessary to improve the precooler.

OBJECTIVE

A specific quantitative objective was difficult to set for the precooler because it would depend on the ultimate performances of the generator and evaporator. A general objective was to have the precooler raise the temperatures of the evaporator outlet fluids to within 30°F of the condenser outlet. A broader intent was to adjust the development of the precooler, along with those of the generator and evaporator, to meet the system performance requirements as a group.

STATUS AT BEGINNING OF PROGRAM

The precooler which had been in use is outlined in Figure 1-10. It consisted of a 2 1/4 inch OD tube for the fluids exiting the evaporator, which was concentric with a 2 1/2 inch OD tube with .083 wall. The two tubes provided an annular passage with a radial dimension of .042 inches for condensate flow.

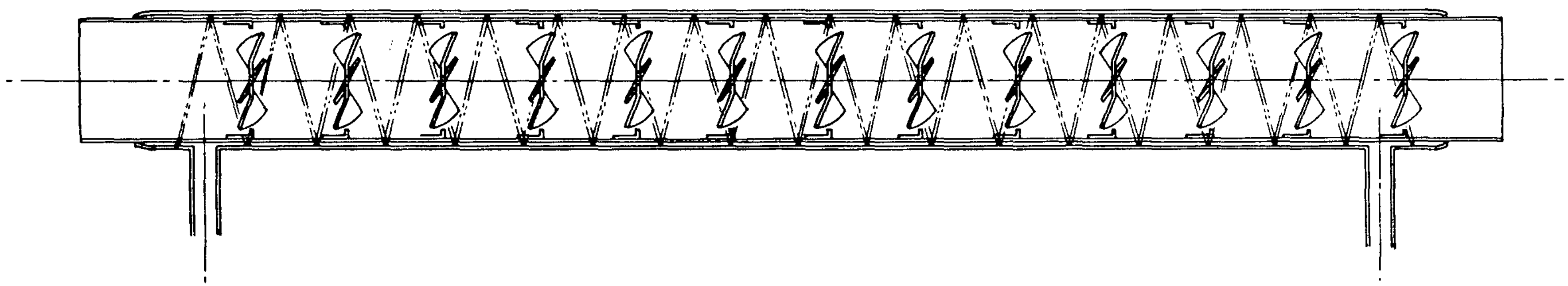


Figure 1-10 Precooler, Pre 1980 Model

The inner tube contained propeller disks to produce mixing and turbulence of the vapor. An aluminum wire helix was wound at a 1 1/4 inch pitch in the annular passage so that the path of the condensate would be partly helical over a cross-section of about 1 1/4" X .042". The purpose was to increase the velocity of the liquid refrigerant and to produce turbulence wherever the liquid crossed over the wire.

DEVELOPMENTS DURING PROGRAM

Improvement of the existing precooler design would require increasing the vapor side heat transfer area and the liquid evaporation area. Various concepts were investigated, but when the serrated tube of 3/8 inch diameter shown in Figure 1-7 became available, it made possible the simple design of drawing # 470-13, shown in Figure 1-11. The vapor side heat transfer area was essentially quadrupled because the normal surface of a 3/8 inch tube, coiled as shown, was double the inner surface of the original 2 1/4 inch tube and because the serrations in turn doubled the normal 3/8 inch tube area.

For better liquid heat transfer, a 3/8 inch dam was placed at the outlet end of the vapor/liquid passage so liquid level would cover the bottom of the tube turns. The coils were wound at 1/2 inch pitch so there would be space between the coils for both liquid and vapor. Some extension of the evaporating liquid surface by capillary flow up the grooves on the tube coils was expected. The relative value of that capillary action is unknown, since the absorbent ETFE will remain in the grooves as the R-133a evaporates. ETFE is lower density than the R-133a, so gravity will not help it to remix. Most of the grooved surface should be dry, however, and be good vapor transfer surface.

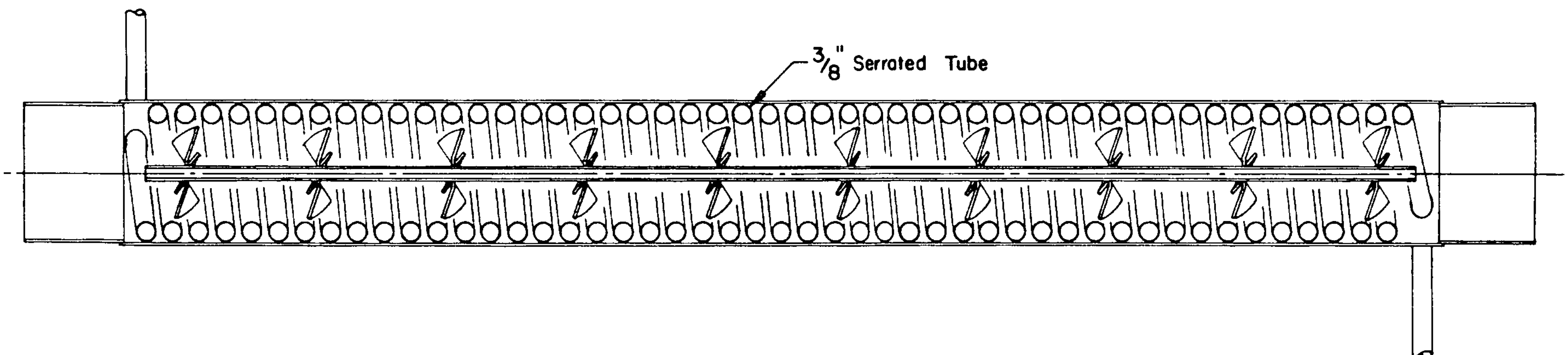


Figure 1-11 Precooler, Serrated Tube Coil

When built and tested, this precooler design was found to be a major improvement over the preceding one. In normal operation, this precooler design raised the temperature of the evaporator exit fluids to within 20°F of the condensate inlet temperature, and appeared to evaporate the overflow R-133a well. An abrupt temperature change could be detected some two-thirds of the way from the evaporator to the absorber. This was taken to indicate that the evaporation of the overflow liquid refrigerant had been essentially completed at that point. By producing abnormal overflow conditions, the condensate could be cooled to within 2°F of the evaporator outlet temperature with no more than a 42°F differential at the warm end of the precooler. Heat exchange capacities of 6700 Btuh were obtained under those conditions.

The performance of this precooler design was thus an important gain for the generator-evaporator-precooler combination. It relieved the requirements on the generator by reducing the refrigerant purity requirements a small, but significant, amount and made the evaporator performance less critical. Since the most a precooler can do is bring the refrigerant vapor temperature close to the condenser outlet temperature, the performance of this design was near a maximum. The development was therefore considered complete for this project. Lower cost methods of fabricating equivalent heat transfer surface are undoubtedly possible. Their overall cost advantage may depend on high production volumes, however.

4.0 ABSORBER

INTRODUCTION

During previous development of the sealed system components, it had become probable that the absorber, in a size to suit a 40-inch high heat pump, would be the most difficult and time-consuming component to bring to its specified performance. That consideration led to the conclusion that, being the component that would take longest to develop, the absorber's performance specifications should include meeting the total system performance rather than the absorber specifications alone. The original absorber requirements for this program were therefore changed to refer primarily to the system performance specifications.

As in other components, the absorber investigations were divided into two parts. The design and development of absorbers constructed of round tubing will be reported here. That work was partly directed toward producing absorbers for the Field Trial units, and partly toward achieving the program objectives of system efficiency and capacity. The second portion of the program, the development of new absorber surfaces to make possible a compact absorber for the M^3 prototype, will be part of a later report.

OBJECTIVES

The performance goals for absorption sealed system included the following:

<u>Heating Mode, 47°F ambient</u>

Heat Output	90,000 Btuh
Gas Input	72,000 Btuh
COP	1.25
Cooling Water Temperature	95°F in, 130°F out
Evaporator Temperature	37°F

<u>Cooling Mode, 95°F ambient</u>

Cooling Capacity	36,000 Btuh
Gas Input	72,000 Btuh
COP	0.5
Evaporator Temperature	45°F

More generally, the COP and heat output objectives were established by the line which passed through the 1.25 COP at 37°F, a 1.12 COP at 17°F and a 1.0 COP at 0°F. The cooling water temperatures listed were the original objectives, but consideration of the temperatures utilized by electric heat pumps led to a conclusion that the outlet water temperature objective had been set unnecessarily high. The water temperature specifications also affect the design and size of the outdoor and indoor finned coils; hence, rather than drop the outlet temperature to the temperature of electric heat pumps, an output temperature range of 120°F to 125°F was chosen as a compromise between the needs of the various system components.

The total quantity of heat to be transferred in the absorber was calculated to be close to 80,000 Btuh. It was expected that over 50,000 Btuh would be transferred to the cooling water (glycol-water brine) and the remainder to the rich solution which serves as the coolant in the absorber-heat-exchanger section. The exact heat quantity and the split between the two coolants would depend on the efficiency reached and the designs used. Using a heat transfer temperature differential of the order of the 20°F which commonly results in reasonable component costs, the UA product for the total absorber heat and mass transfer surface would be 4000 Btuh/°F. That was taken as a minimum. Due to the criticality of the absorber, it was also assumed that Δt's lower than 20°F might be necessary, and that a UA product as high as 5000 Btuh/°F might become necessary.

An immediate requirement of the absorber investigation was a decision in the first two months of the program concerning the absorber design to be used in the Field Trial units.

STATUS AT BEGINNING OF PROGRAM

During 1979, the best short absorber tested was made of flat extrusions, similar to Figure 1-12. It had an area of 51 square feet, was 28 1/8 inches high, and had been designed to fit in a 40-inch high cabinet. The absorber was divided vertically into four sections as a means of applying the coolants (the water/glycol and the rich solution) so as to maintain the maximum temperature differentials between the film of absorbent liquid and the coolants. The bottom section was water-cooled, the second section solution-cooled, and the third water-cooled and the fourth solution-cooled. (2) Between the first and third sections the cooling water was used to cool the condenser. That absorber had brought the absorption system to the required heating mode performance only when the cooling water entering the absorber had been lowered to 80°F, from the 95°F specification. The absorber sections and the individual extrusions making up the sections were difficult to align, leaving uncertainties about the completeness of the wetting of the entire

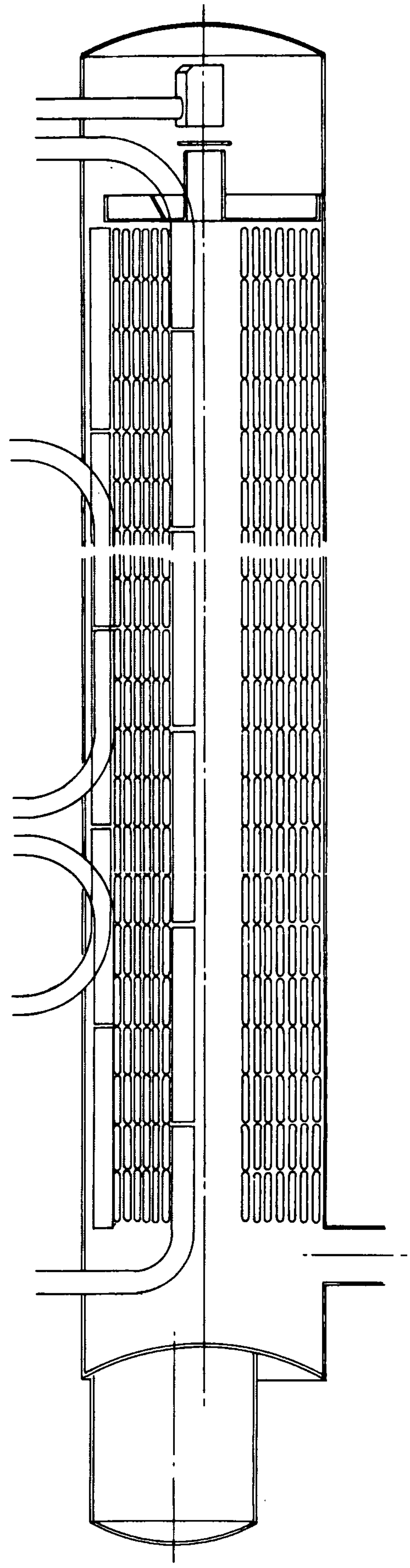


Figure 1-12 Flat Extrusion Absorber

surface by the absorbent. Calculations of the mass and heat transfer rates of each of the four sections of the absorber, using the In and Out temperatures of the fluids, but making assumptions concerning the internal temperatures between the sections, had resulted in UA products ranging from 1200 to 2500 Btuh/°F on a total absorber basis. The overall heat transfer coefficient U ranged from 24 to 50 Btuh/Sq Ft/°F.

Those overall coefficients incorporated the mass transfer coefficient as well as the heat transfer coefficients. Separately, on a heat transfer fixture, measurements had been made of heat transfer alone, without absorption, on an equivalent flat extrusion absorber. The heat transfer temperatures were equivalent to those in an absorber. The tests were not completely comparable to unit conditions because, at equal absorbent flow rates, the heat flux, consisting only of sensible heat, and the coolant rate were lower than in the absorber. Overall U's of 60 Btuh/Sq Ft/°F had been obtained, at a heat flux approximately half that of normal absorber operation. Due to the difference in conditions, the heat transfer coefficient in the absorber should be larger than 60 Btuh/Sq Ft/°F. Such heat transfer coefficients, being larger than the combined heat and mass transfer coefficients, were in general agreement with the overall absorber coefficients obtained during unit operation. However, attempts to calculate mass transfer coefficients from the heat transfer test data and absorber data from an operating unit would incorporate serious uncertainties due to the assumptions made about internal temperatures, the differences in heat flux, and the range of unit operating conditions. The relative magnitudes of the heat transfer coefficients and the overall coefficients in the operating absorber implied that the absorption coefficients were relatively high, and that heat transfer was controlling.

Those coefficients were all so low, however, that finding means of improving them was a prime consideration. Two approaches to increasing heat and mass transfer were proposed. One was the investigation of possible additives which might improve absorption and heat transfer coefficients by the Marangoni effect. (3) That study has been underway at Allied Chemical as an ex-scope activity. The other approach was to continue mechanical design attempts to enhance mass and heat transfer coefficients and to increase surface area within the space limitations.

Calculations based on the earlier results had determined that a mass and heat transfer area of 100 square feet or more would be required to produce the specified performance. Such an absorber would be prohibitively large if constructed of the flat extrusions. Therefore, attempts to enhance the coefficients by increasing turbulence and mixing in the absorption film had followed. A variety of other extrusions, finned and otherwise, had been tested, generally with less than satisfactory results. One clue to possible gains was an indication that high vapor velocities over the flat extrusions had increased the heat transfer coefficients. Another indication was that an absorber constructed of two, concentric helical coils of 3/4" tubing, having 40 square feet of area, but 70 inches high (60 inch coils) had provided heat and mass transfer coefficients ranging from 50 to 89 Btuh/Sq Ft/°F over the surface, almost double those of the flat extrusion. That absorber was too tall, but its performance seemed to indicate that round surfaces had advantages over flat surface, perhaps the shape factor, or due to the large number of crevices between tubes where mixing could take place. Height alone could be a factor, or the increased film thickness resulting from flow over only two coils. The latter could be positive, if the extra film thickness caused more mixing and turbulence, or negative if it did not.

The higher coefficients had resulted in a decision in 1979 to temporarily remove the height limitation from the absorber development program. Because much more surface could be put into an absorber by using flat extrusions instead of round tubes, a flat extrusion absorber had then been designed with narrower vapor passages and with 24 extrusion spirals stacked to a coil height of 45 inches. The absorber had 82.5 square feet of surface area. Many difficulties were encountered in building the coils with the narrow spacings and in aligning and fitting the large number of extrusions to each other. The first absorber built did not match the drawings and did not exhibit much improvement over earlier ones. An improved method of controlling the fit of the coils had been devised, and at the beginning of this program, the second absorber was under construction.

DEVELOPMENTS DURING PROGRAM

Flat Extrusion Absorber

The close-wrapped and more accurately assembled flat extrusion absorber of 45" height was completed and tested. That absorber, No. 870-8-2, similar to Figure 1-12 in general design, had 24 extrusions and a heat transfer surface of 82.5 square feet, the same as the preceding one, 870-8-1. The spacing for vapor flow between the windings was held to the 3/16 inch drawing specification, and the coils were accurately assembled to maintain equal absorbent flow on both sides of the extrusions from spiral to spiral. The following table shows the average heat transfer coefficients and UA products of the new absorber and of the 70-inch tall, helical coil absorber, as well as the preceding absorber of the same design.

TABLE 1-10

COMPARISION OF ABSORBER PERFORMANCE

	HELICAL COIL ROUND TUBE ABSORBER 870-6	FLAT EXTRUSION ABSORBER 870-8-1	FLAT EXTRUSION ABSORBER 870-8-2
Surface, Square Feet	40	82.5	82.5
Average U, Btuh/Sq/Ft/°F	73.4	39.4	61.1
Range of U, Btuh/Sq/Ft/°F	50-89	24-50	48-71
Average (UA) Btuh/°F	2936	3251	5041

This second flat extrusion absorber thus operated at a 61% higher average heat transfer coefficient than the preceding one. It did not quite reach the U's of the tall, helical coil absorber, but its large surface area resulted in a UA product that was 70% larger. The UA product of 5040 Btuh/°F met and exceeded the maximum end of the specification range. The results also demonstrated the importance of the time and care that had been spent in fabrication of this absorber.

The absorber should have produced very good unit performance, but unit test results were not up to the absorber performance.

TABLE 1-11

UNIT PERFORMANCE, ABSORBER DWG. #870-8-2

Heat Input, Btuh	71,640	72,140	76,110
Evaporation Temp., °F	40.2	40.0	40.7
Cooling Water In, °F	95.3	94.9	95.0
Cooling Water Out, °F	129.6	129.2	130.9
Absorber Solution Outlet, °F	106	107	106
Absorber Heat Exchanger Outlet, °F	172	168	172
Heat Output, Btuh	87,250	87,710	92,430
COP, Heating	1.22	1.22	1.21

In the above data, the outlet temperatures of the solution from the absorber and from the absorber-heat exchanger, and of the cooling water were representative of very good absorber performance. The system COP's were not up to that level, however. Investigation determined that the limited performance was due partly to other unit components, but also that the coolant circuiting had allowed the temperature differentials to drop to low levels over part of the absorber.

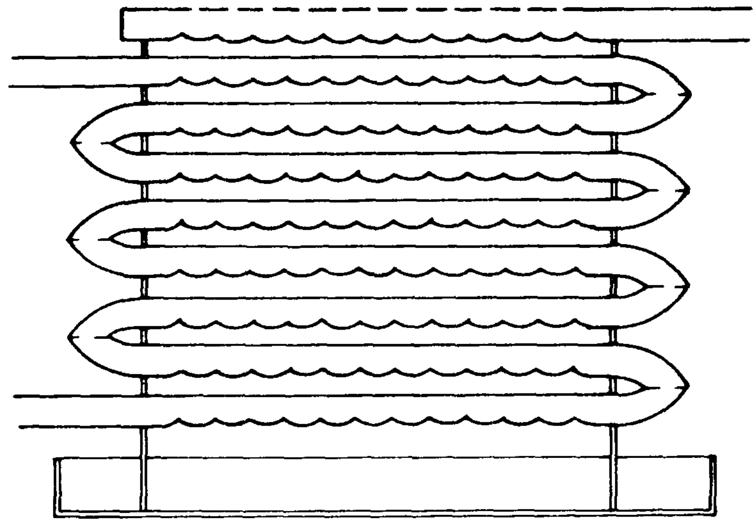
It was concluded that the 870-8 absorber design met the target requirements of the absorber and that coolant circuiting improvement would cause it, and the unit, to perform even better. However, the care and the time which would be consumed in building that design to its best performance prevented further work in that direction. One of the objectives had been to arrive at a design that could be used to construct six absorbers for the Field Trial units early in the program. The construction time required for this absorber eliminated it from that application.

It then became necessary to proceed in two paths. One was to choose a design for the Field Trial units that could be built accurately, consistently and quickly. The round-tube coils had exhibited the best heat transfer coefficients and were the easiest to construct uniformly. They were made the basis of the Field Trial absorbers. The second direction was to continue investigations toward an absorber that would meet the capacity and heat and mass transfer targets. A step in that direction was a trial of a horizontal design which had solved heat and mass transfer problems in lithium bromide absorbers.

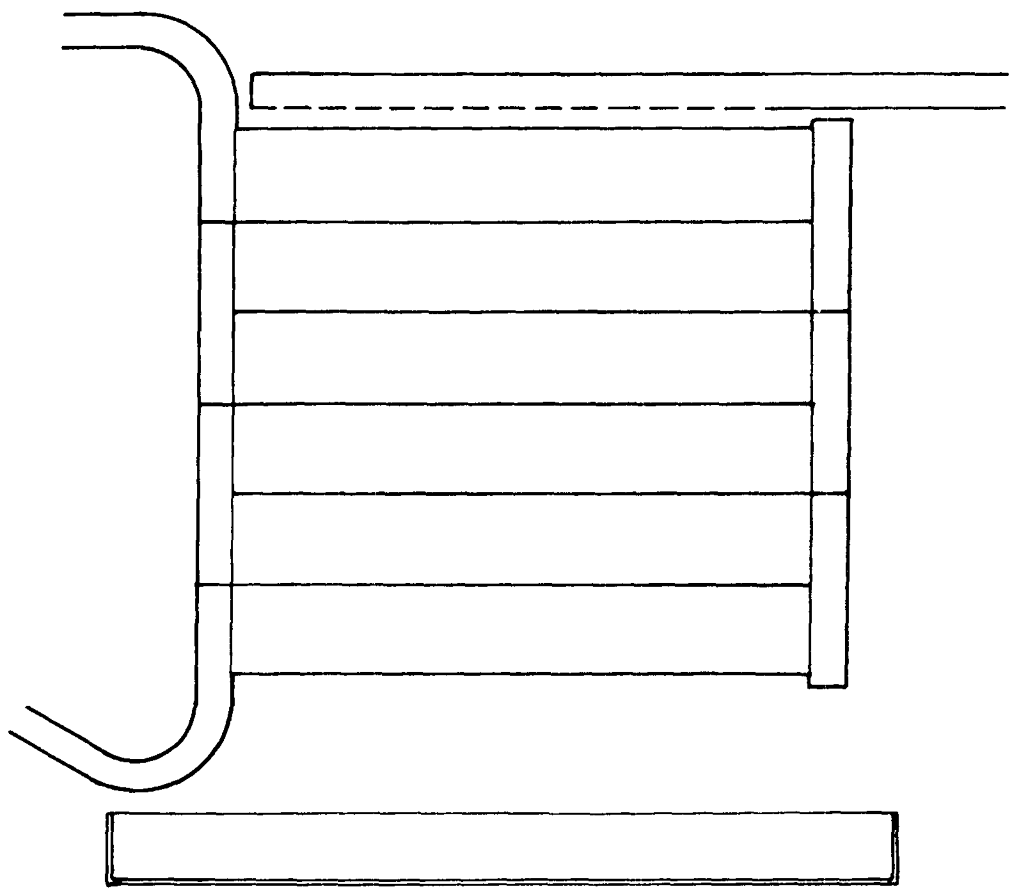
Horizontal Tube Absorber

The limited overall heat transfer coefficients that had been obtained on the flat extrusion absorbers were due partly to the naturally low coefficients of organic fluids, but probably also to laminar flow of the falling film on the coil surfaces. A similar problem had been faced in the lithium bromide/water units. It had been solved by designing the absorber with horizontal tubes spaced vertically so that the absorbent solution dripped, as droplets, from one tube to the next lower one. The dripping action had provided vapor-liquid contact, turbulence, mixing and scrubbing contact with the cooling surface of the tube. Significantly improved absorption and heat transfer had resulted.

To try this design for the organic fluids, a small bench scale test bank was made from 5/8 inch OD tubes, knurled on the outside to promote liquid spreading. The tubes were aligned vertically on one-inch centers and connected by return bends to form a serpentine coil, as in 13a of Figure 1-13. For this rough determination of heat transfer rates, the heat flow direction was reversed. Warm water was passed through the coil from the bottom up. The cold test fluid was dripped over the outside. In the first test, cold water was distributed over the top tube and allowed to drip from tube to tube to a pan on the bottom. The flow rates and temperatures of the inner and outer liquids were measured, providing the data for calculation of heat transfer coefficients. The tests were repeated with ETFE flowing over the outside of the coil. For comparision, an equivalent test was also made with the flat extrusions of the 870-8 absorbers, one directly above the other in straight lengths. That assembly is shown in 13b of Figure 1-13. The results of both sets of tests are given in the following table.



13a Test Bank - Knurled Round Tubes



13b Test Bank - Flat Extrusions

Figure 1-13 Absorber Tubing Test Banks

TABLE 1-12

HEAT TRANSFER COEFFICIENTS FROM BENCH SCALE EXPERIMENTS

	HORIZONTAL 5/8" TUBES	FLAT EXTRUSIONS
U, Water to Water	256	237
U, Water to ETFE	87	80
Ratio, $\frac{U\ ETFE}{U\ Water}$	0.34	0.34

U = Overall heat transfer coefficient, Btuh/Sq Ft/°F

Although these were rough tests made in the open air, the results were relatively consistent and provided significant information.

1. In both cases the overall heat transfer coefficients were reduced by over 60% when ETFE was substituted for water as the outside film. That implies a reduction of the external coefficient of 80% or more.

2. The ETFE side is very much the controlling factor in such exchangers.

3. The horizontal dripping tubes were better than the flat extrusions, but not to an important extent.

Those results pointed to heat transfer through the absorbing film as a major problem to be solved in designing a satisfactory absorber. The gain in heat transfer of the dripping tubes over the flat extrusions was much less than expected. It was noted in the tests of the dripping tube model that the ETFE did not drip from tube to tube in droplets as did the water; but rather flowed in small streams, thus reducing the vapor/liquid contacting, the turbulence and the mixing effects sought. The tendency to stream rather than to drip would be partly a function of the surface tension and viscosity of the ETFE. It was therefore possible that the action of R-133a/ETFE solutions in an absorber would be different, since their temperature and density would be higher, and the surface tension and viscosity lower. The concentration changes and heat generated by absorption might also act to change the streaming into drop-wise flow in an operating absorber.

Due to those potential differences between the R-133a/ETFE solutions and ETFE alone, a horizontal tube absorber designed for test on the Test Unit was completed and tested. It consisted of an assembly of 64 tubes in an 8 X 3 array, as shown in Figure 1-14. The vertical spacing was one inch center-to-center; and the horizontal spacing 7/8 inch center-to-center. The 36 inch long tubes were knurled to promote spreading of the liquid, and dimpled on the bottom to assure uniform distribution of the dripping along the tube length. The total absorbing surface was 30.5 square feet. The lower five tubes in each vertical column were connected in series as a serpentine and were water-cooled. The upper three tubes were similarly connected for solution cooling. The eight circuits of each coolant were then connected in parallel. The distributor for the weak absorbent solution consisted of a row of tubes above the top row of the absorber, with holes drilled at 3/4 inch intervals to provide uniform distribution of the absorbent solution over the coils.

Due to the small surface area of the coil, the absorber was found to be too small for full-load operation. Most of the tests were therefore run at reduced inputs. The heat transfer coefficients calculated for individual sections ranged from 45 to 75 Btuh/Sq Ft/°F, averaging just over 60 Btuh/Sq Ft°F, for a UA product of 1830. Some improvement in that performance was probably possible, but since the hoped-for major improvement in mass and heat transfer coefficients had not been obtained and the surface-to-volume ratio was basically low, work on this concept was stopped.

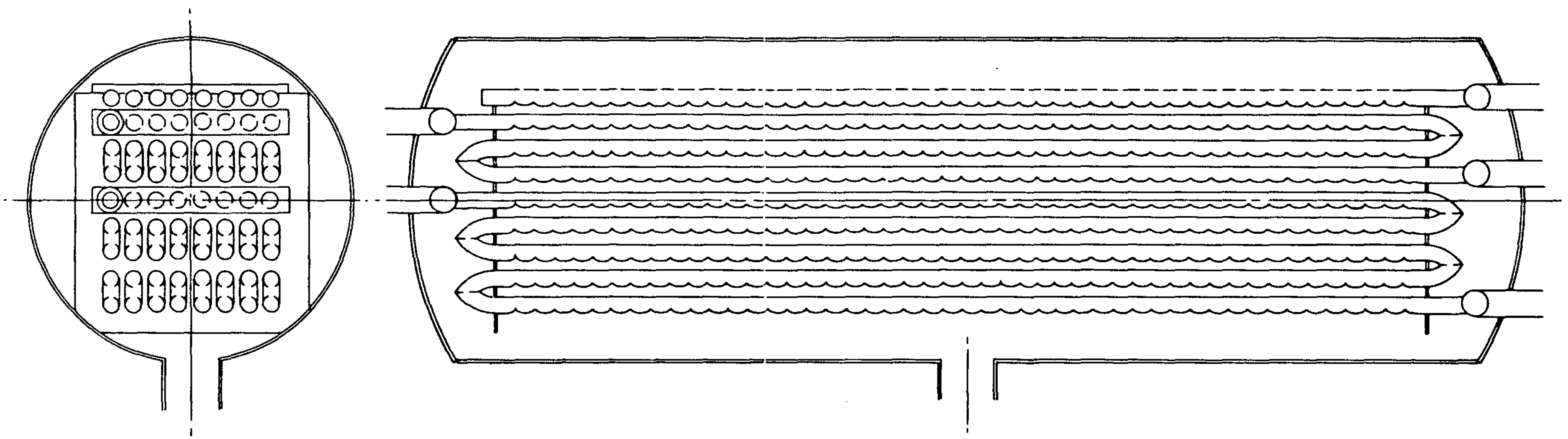


Figure 1-14 Horizontal Absorber

Round Tube, Helical Coil Absorbers

The round tube, helical coil absorber designed for the Field Trial units consisted of four concentric coils with a total height of 45 inches. The design, Drawing #411-13, is shown in Figure 1-15.

Using the U of 73 Btuh/Sq Ft/°F measured on the two-coil model, the absorber was designed for a total absorption surface area of 62.5 square feet for a UA product well in the 4000-5000 Btuh/°F range. As the coils were wound, however, the aluminum tubing flattened, and fewer turns could be accommodated within the 45-inch height than in the design for round tubing. The coil area thus became 52.3 square feet, with a potential UA of 3800. Longer coils could not be accomodated in the Field Trial units at that stage.

In the first configuration of the absorber, Figure 1-15, each of the four concentric coils was made full-length, as on the two-coil absorber. Counting from the center out, the second and fourth coils were connected in parallel to form the water-cooled section. The first and third coils were also in parallel and constituted the solution-cooled portion. The surface area was thus divided approximately in the 60/40 ratio that corresponded to the ratio of the heats to be transferred to the glycol-water solution and to the rich solution, respectively.

Three absorbers with this circuiting were built, two for Field Trial units and one for laboratory evaluation on the Test Unit. The performance of the Field Trial units is to be reported in Section 4. On the Test Unit tests, the overall heat transfer coefficients were found to range from 52 to 62 Btuh/Sq Ft/°F, with a weighted average of 59 Btuh/Sq Ft/°F. These heat transfer coefficients compared favorably with most of the previous absorbers, but did not equal those of the tall, helical-coil absorber which had served as the

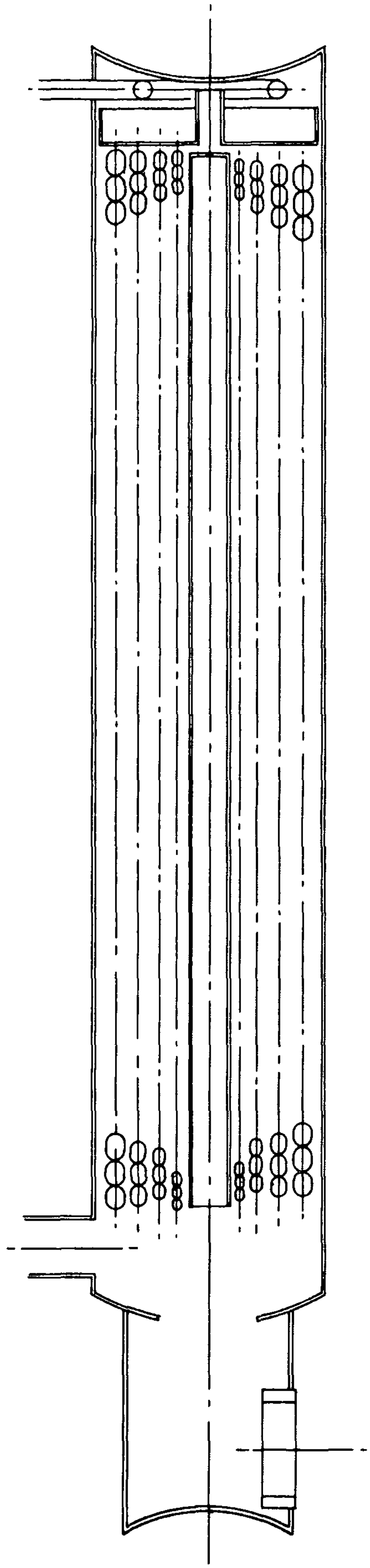


Figure 1-15 Helical Coil, Round Tube Absorber

model. A reason for the lower heat transfer coefficients, as compared to the two-coil absorber, may have been the formation of a thinner film of absorbent, when spread over four coils as compared to two, and a resulting reduction in turbulence.

In terms of unit operation, the performance of this absorber was found to be worse than could be accounted for by the limited heat transfer coefficients alone. Investigation determined that a greater difficulty was caused by the solution-cooled coils. Those coils were operating at higher temperatures than the water-cooled coils concentric to them. The situation is shown schematically in Figure 1-16 in which the solution-cooled coils are placed alongside, rather than concentric to, the water-cooled coils. In the data of Figure 1-16 taken from Test No. 813, the cooling solution is seen to have been 20.4°F warmer than the cooling water. The solution dripping from the solution-cooled coils was 22.5°F warmer than that dripping from the water-cooled coils. That higher temperature corresponds to an equilibrium rich solution from the solution-cooled coils of 31% R-133a, as compared to 41% for the water-cooled coils. The concentration of the resulting mixture, being perhaps 5% lower than had been possible with other designs, was the major limit to the system performance. In the tall absorber the solution temperature effect had not been serious enough to be noticeable. It is probable that the effect builds up cumulatively, in that the warm solution dripping from the solution-cooled coils warms the mixed absorber outlet solution, which then becomes a warmer coolant for the solution-cooled coils, in turn raising the temperature of the solution dripping from the coils, etc., until an equilibrium condition is reached. In such a case, small variations can swing the equilibrium in either direction. In the tall, two-coil absorber, only a small

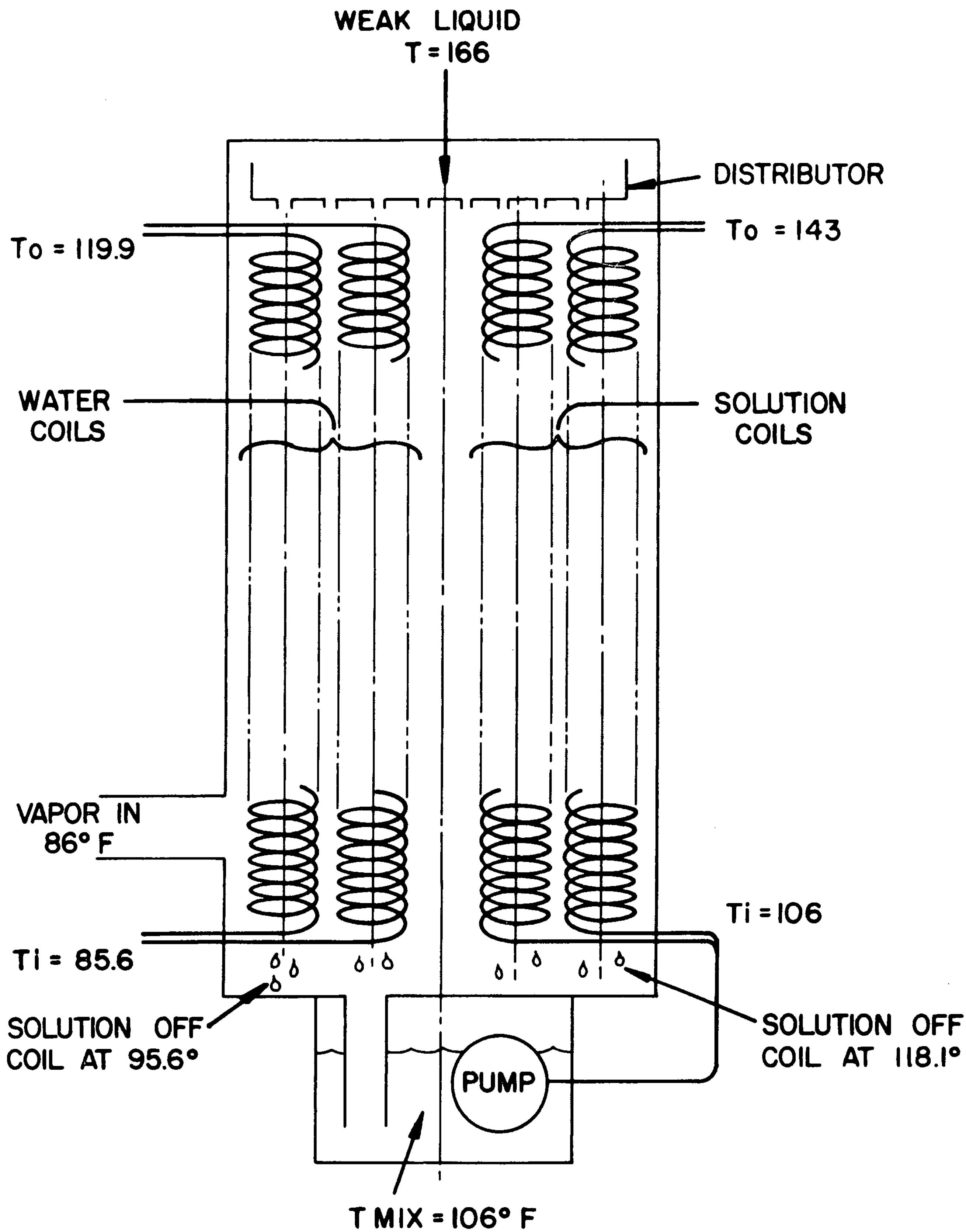


Figure 1-16 Test of Absorber #411-13
Test Unit Test #813

temperature rise had apparently occurred, perhaps due to the higher heat transfer coefficients in that absorber, or to the greater height of the coils.

To correct this coolant circuiting problem, the next four-coil absorbers were built with the coils divided into upper and lower sections of equal length. The coils of the upper section were solution-cooled, and those of the lower section were water-cooled. That circuiting and the even division of area were not expected to be ideal, but were used to prevent delays in the construction of the Field Trial units. The change in the ratio of solution-cooled to water-cooled surface was made because the overall coefficients of the solution-cooled sections had tended to be lower than those of the water-cooled coils, and the upper, solution-cooled section was expected to operate with lower Δt's.

Absorbers of this design were built for three Field Trial units, Nos. 3, 4, and 5. Following preliminary tests to establish the best circuiting of the coolants through other components, tests of those absorbers demonstrated substantial improvements in unit performance. The results are reported in Section 4 of this report. Since these were prototype units for outdoor installation, no instrumentation could be applied to evaluate the absorber alone. However, since the unit performance was significantly improved, it was decided to use this design in building a larger absorber for test on the Test Unit.

Assuming that the heat transfer coefficients would remain at the average of 59 Btuh obtained on the first four-coil assembly, the surface area was increased to 73.5 square feet. The absorber remained a four concentric-coil type, divided into a solution-cooled upper half and a water-cooled lower half, but each section was increased to a height of 33 3/4 inches.

The absorber was tested on the Test Unit, which in the interim had had an evaporator and a precooler of the final designs installed and the combustion system updated. The heat transfer coefficients and UA products obtained are shown in the following table.

TABLE 1-13

ABSORBER HEAT TRANSFER PERFORMANCE

Surface Area, Sq Ft	73.5
Average U, Btuh/Sq Ft/°F	60.0
Range of U, Btuh/Sq Ft/°F	39.7 - 78.5
Average UA, Btuh/°F	4412

The heat transfer coefficients were thus found to average the same as those on the first four-coil absorbers, but the increased area raised the UA product to the midrange of the absorber specifications.

The system performance that was achieved with that absorber are shown in Table No. 1-14, which lists data at evaporator temperatures that bracket the 37°F temperature used in the project performance specifications.

TABLE 1-14

PERFORMANCE OF 73 SQUARE FOOT ABSORBER

Heat Input, Btuh	72630	72580	71640	72360
Evaporator Temp, °F	41.0	39.2	36.6	34.1
Cooling Water In, °F	94.7	95.0	94.0	94.9
Cooling Water Out, °F	121.8	125.6	123.4	122.8
Rich Solution Out of Absorber, °F	108.4	109.4	108.1	108.1
Rich Solution Out of AHE, °F	175.2	178.5	173.7	179.9
Refrigeration Effect, Btuh	36480	35530	33480	33050
Heat Output, Btuh	93730	91820	91010	89660
COP, Heating	1.29	1.265	1.27	1.24

In the series of tests summarized in this table, the absorber performance met its specific goals. The rich solution outlet temperatures of 108°F - 109°F, and the heating of the rich solution to the 170°F - 180°F range in the absorber heat exchanger equalled or exceeded the absorber performance targets. The results also demonstrated that a UA product of 4400 Btuh/°F, calculated from the inlet and outlet temperatures, was sufficient to produce the required absorber performance.

Table 1-14 also shows that the combination of this absorber with the final evaporator and precooler designs, but not the best generator, produced Test Unit performance that met the project objectives for the system. The COP's listed all fall above the curve specifying the performance targets, i.e. a straight line from a COP of 1.0 at 0°F evaporator to 1.25 at 37°F. Similarly the heat outputs also fall above the objectives when adjusted to 72,000 Btuh input. The outlet water temperatures were in the revised 120°F - 125°F objective.

The design of this absorber, for construction from round tubing, was not suitable for compacting to the size of the projected M^3 prototype unit. Special absorber extrusions had been under development for an absorber to fit in the 40-inch high M^3 heat pump, and an absorber made of the new extrusion was ready for testing. Testing of this round tube absorber was therefore stopped at that point so the ensuing development could proceed.

The actual heat transfer differentials within the absorber, as differentiated from those calculated from inlet and outlet temperatures, had been under question, however. To perhaps improve the true average Δt in the four-coil absorber, the absorber for the sixth Field Trial unit was built with a further modification of the coolant circuits. Two of the four coils were water-cooled over their full height. The upper 80% of the other two were solution-cooled, and their bottom 20% were water-cooled. As in the other Field Trial unit absorbers, they had a surface area of 52.5 square feet. As in all complete units, it was impossible to install instrumentation for detailed measurements on the absorber. The unit performance was significantly improved, however. The unit COP's increased from the 1.16 - 1.17 level of Field Trial units No. 3, 4, and 5 to 1.22 for Unit No. 6. Those results are reported in more detail in Section 4 of this report.

Those results and similar indications from other tests imply that the true average temperature differential between the absorbent film and the coolant streams was less than the log mean temperature difference calculated from the inlet and outlet streams. If that was the case, the true overall heat transfer coefficients were higher than the value of about 60 Btuh/Sq. Ft/°F calculated from the inlet and outlet data. Measurements in the detail needed to determine those coefficients correctly were not part of the scope of this project, but would be important in further work.

FINDINGS AND CONCLUSIONS

Absorbers that met the performance requirements were developed from both flat extrusions and round tubes. Unfortunately, neither of those absorber designs could be applied to the Field Trial units, due to time limitations in one case and size in the other. Both of those designs were too large for use on the projected M^3 size prototypes.

The absorbers applied to the Field Trial units, though all of the round tube, four concentric coils type were improved as succeeding units or groups of units were built. The improvements were achieved by changing the circuiting of the coolant streams to improve the temperature differentials between the coolants and the absorbing liquid films over the total coil surface.

The test results throughout the program indicated that the absorption heat and mass transfer coefficients can be enhanced beyond those achieved in the best absorbers of these absorbers. Further investigations to develop higher heat and mass transfer rates are recommended. It is possible, however, that the geometry of the optimum designs will be incompatible with preferred configurations of outdoor heat pumps. Investigations to optimize coolant - absorbent relationships are also needed.

5.0 REFRIGERANT CONTROL VALVE

INTRODUCTION

In a compression refrigeration or heat pump system, the flow of condensed refrigerant from the condenser to the evaporator is normally controlled by a modulating control which restricts the flow of the liquid to the amount required by the load and the temperature of the evaporator. It also maintains the required pressure differential between the condenser and the evaporator, allowing only liquid to flow. The means that have been commonly used for automatic control of the flow of liquid refrigerant include the following:

Low-side float valve
High-side float valve
Automatic expansion valve
Thermostatic expansion valve
Capillary tube restrictor

The applications of these control techniques to the organic absorption heat pump are discussed below:

Low-Side Float Valve

The low-side float valve controls the flow of liquid refrigerant to the evaporator to maintain a fixed level of liquid in the evaporator. It is generally applied in the case of a coil submerged in a boiling refrigerant. The method is not readily applicable to this R-133a/ETFE absorption heat pump for the following reasons:

1. Any appreciable depth of liquid refrigerant will significantly raise the boiling point of a low-pressure refrigerant such as R-133a.

2. Since ETFE is somewhat volatile, the refrigerant contains traces of absorbent. Unless the absorbent is continually removed from the evaporator, the ETFE concentration will increase and raise the boiling point.

The low-side float valve is therefore not well suited to an absorption system using R-133a and ETFE as the working fluids.

High-Side Float Valve

The high-side float valve operates by limiting the flow of liquid refrigerant to the evaporator by maintaining a fixed liquid level at the exit of the condenser. It also maintains the pressure difference between the condenser and evaporator and delivers liquid refrigerant at the rate produced in the condenser. On its own, it makes no provision for controlling flow in proportion to the variable load in the evaporator of an absorption heat pump. For heat pump use, that would have to be accomplished by another component.

Automatic Expansion Valve

The automatic expansion valve is a pressure-sensitive device that controls the flow of liquid refrigerant into the evaporator at the rate required to maintain a fixed evaporator pressure, and hence a fixed temperature in the evaporator. It is not applicable to the absorption heat pump because the operation of the heat pump requires that the evaporator temperature be a function of the ambient temperature. The valve is described in greater detail in the weak liquid flow control report.

Capillary Tube Restriction

Being fixed restrictions, capillary tubes and multiple orifices control refrigerant flow over a limited range. The range is broadened by the effects of vapor, either flashing near the outlet or entering from the condenser. The capillary has been applied mainly in small systems operating at relatively constant temperatures, such as refrigerators, freezers and some air conditioners. It is less applicable to a heat pump with the broad temperature range planned for this absorption system, in which the refrigerant rate may vary by a factor of two or more.

Thermostatic Expansion Valve

The function of the thermostatic expansion valve is the controlling of the refrigerant flow rate to the evaporator in order to match the evaporator load by maintaining a pre-determined temperature difference between the refrigerant vapor leaving the evaporator and the evaporator temperature. In compression systems, for which it was developed, it operates by assuring that the liquid refrigerant is all evaporated within the evaporator coil. The last few percent of evaporator surface serves to superheat the vapor. As shown schematically in Figure 1-17, the device compares the evaporator pressure, through a diaphragm, to the pressure in a sensing bulb which is placed in the superheated vapor region at the evaporator outlet. The spring force establishes the desired superheat. If the actual superheat is too high, it will cause the valve to increase flow, and if too low, to decrease flow. Thus the valve serves to adjust the flow to keep the evaporator nearly full of refrigerant regardless of the load. Because it can, in principle, modulate flow to match a range of loads, it is well-suited to provide the control needed by absorption heat pump systems.

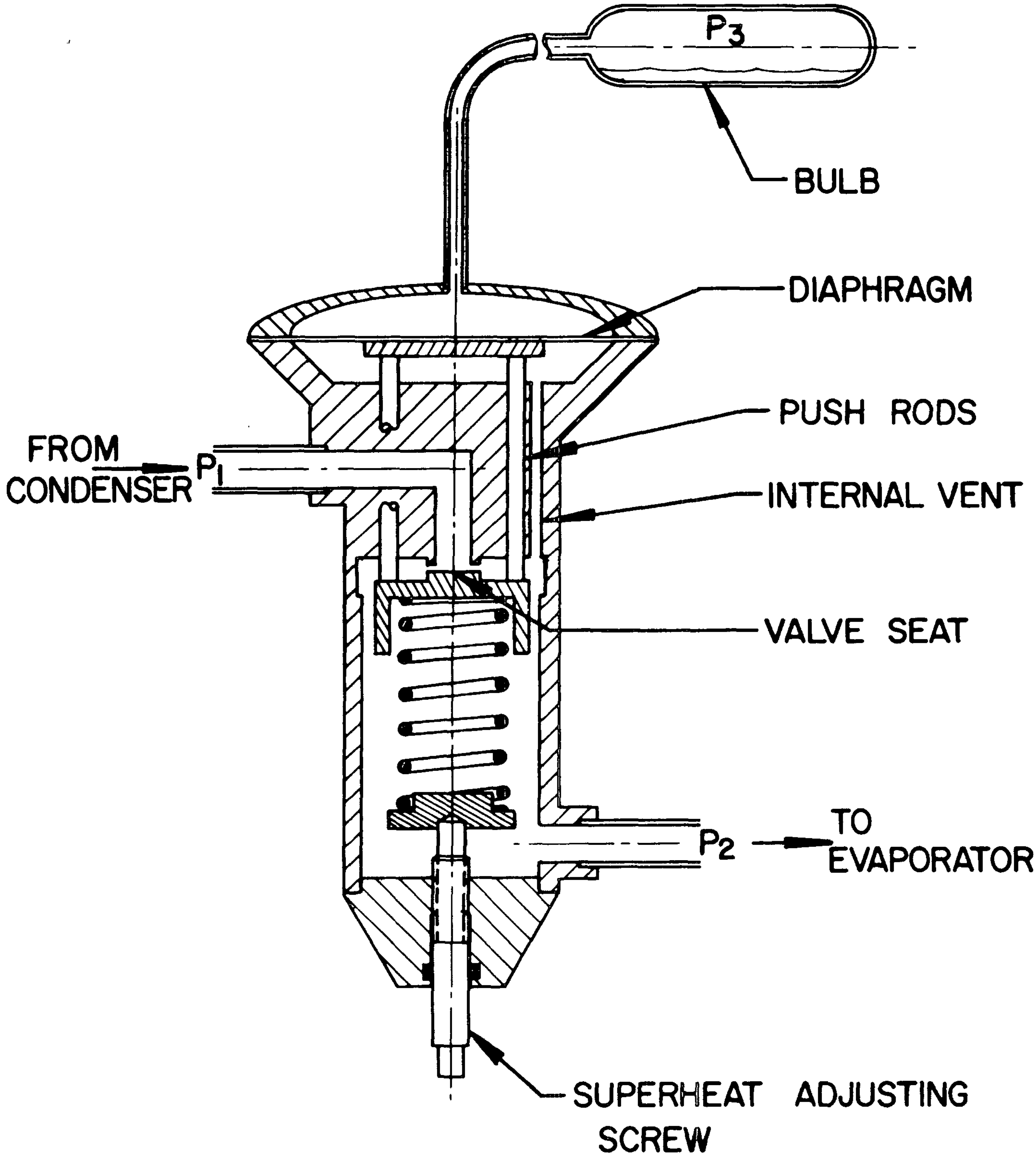


Figure 1-17 Thermostatic Expansion Valve

All of these control means were considered in early stages of this project. Capillary tubes were tried unsuccessfully. The two found to be best suited to the absorption heat pump application, the thermostatic expansion valve and the high-side float valve, were investigated and developed at some depth.

OBJECTIVE

A first objective was to select for further development either the thermostatic expansion valve or the high-side float valve as the one best suited for the organic fluid absorption heat pump, but more specifically, to select the one best suited for early use in the six Field Trial units to be operated under outdoor conditions. The primary objective was to develop the valve chosen to the point that it would control the flow of refrigerant to match the system requirements under all weather and load conditions. The flow rates were expected to range from 200 lb/hr to 475 lb/hr.

STATUS AT BEGINNING OF PROGRAM

At the beginning of the organic absorption heat pump studies, the thermostatic expansion valves (TEV's) had been investigated, using commercial valves from two suppliers as the starting point. Problems encountered included failure to maintain control when rapid changes in flow rate occurred, a change in superheat as a function of temperature which was the reverse of that required, good performance only at high superheats, stick-slip movement of the valve due to lack of lubricant in the refrigerant, and a limited range of operating temperatures. After developing them to the best performance, including Teflon coating the rubbing surfaces to correct the stick-slip problem, the valves had been installed on prototype units. They operated well over a part of the temperature range for a few months, but then the stick-slip problems appeared to recur.

By 1978 the group of problems had led to the conclusion that a high-side float valve might be better suited to the system. Superheat and bulb location would not be problems, and the valve should assure that all the refrigerant condensed would flow over the evaporator, keeping the evaporator surface wet to the extent possible at the rate of condensation. Hence, the development of the refrigerant flow control during 1979 concentrated on a float valve. The mechanical problems encountered included stick-slip problems, the relatively small buoyancy forces of the float, Bernoulli effects at the valve seat, apparent flashing of liquid refrigerant in the valve bearings, and hunting operation at low refrigerant flow rates. These problems were substantially solved and the valve developed to relatively good performance by October 1979. However, no fully satisfactory solutions to the related load matching and concentration control requirements had been devised. The performance of the float valve itself had reached the point that two valves were built for life testing, however.

The uncertainties about the extra component, or components, needed for total refrigerant control were sufficiently serious that consideration of the thermostatic expansion valve had been reinstituted, and testing of new samples received from Sporlan Valve had been started.

INVESTIGATIONS

As indicated above, two float valves had been completed for life test. The life test fixtures were completed as part of this project, and two float valves were placed on test in October, 1979. The tests were cycling tests in order to subject the valves to movement and wear of the valve stem bearings and valve seats. The tests continued without serious interruptions through July 27, 1981 when 81,063 cycles had been completed and "on" time was 8,192 hours. A brass solenoid valve in the refrigerant cycling circuit had deteriorated and failed. The life test was stopped at that time.

Consideration of the requirement for complete control of the refrigerant flow in the Field Trial units, the first of which were to be completed within six months of the start of this program, led to the conclusion that the probability of success would be much greater with the thermostatic expansion valves. Factors in the decision were the TEV's capability of performing all the required functions in refrigerant control, the more advanced status of the commercial products and the ability to obtain manufacturer support in solving the application problems.

In the earlier trials, the first thermostatic expansion valves tested had had standard commercial power elements charged for R-12 systems. With R-133a in the evaporator, as the evaporator temperature was reduced from 40°F to 0°F the operating superheat increased, due to the relative vapor pressures of the refrigerant and of the power element charge. Since the evaporator load would diminish as the evaporator temperature dropped, a reduction in superheat was required rather than an increase.

The first step taken to solve that problem had been to have the charge of the power element changed to refrigerants having vapor pressures closer to those of R-113a. R-114 and R-11 were tried, with limited improvement. The R-11 work led to the conclusion that the addition of inert gas to the R-11 charge should bring the total pressures in the power element to the proper level, while maintaining the rate of pressure change close to the vapor pressure of R-11. The first samples had not worked satisfactorily. As interest in the valve revived in mid-1979, a new sample valve with an R-11/air charge was received. After Teflon coating the rubbing surfaces, etc., it was brought to practical operating performance at the beginning of this project in October, 1979.

The data shown in Table No. 1-15 were taken over a range of evaporator temperatures with a constant setting of the superheat adjustment.

TABLE 1-15

THERMOSTATIC EXPANSION VALVE WITH R-11/AIR CHARGE

Heat Input, Btuh	72080	72300	72400	72400	72400	72200	70500
Evaporator Inlet Temp, °F	47	40	38	30	31	25	22
TEV Bulb Temp, °F	67	55	51	38	38	30	25
Superheat, °F	20	15	13	8	7	5	3
Precooler Outlet Vapor, °F	94	84	76	51	47	32	25

The data are plotted in Figure 1-18. As can be seen, the superheat dropped as the evaporator temperature was lowered. That was the direction of superheat change desired, but the rate of reduction was much greater than necessary. Instead of the superheat reduction obtained, from 20°F to 3°F on the evaporator temperature change from 47°F to 22°F, a reduction from 10°F to 5°F was desired over a 40°F evaporator temperature drop. As shown by the change in temperature of the precooler outlet, that high rate of change of the superheat resulted in serious overflow of refrigerant from the evaporator as the temperature dropped. The rate of superheat reduction obtained was thus too high for the bulb location at the outlet of the evaporator. Because similar results were

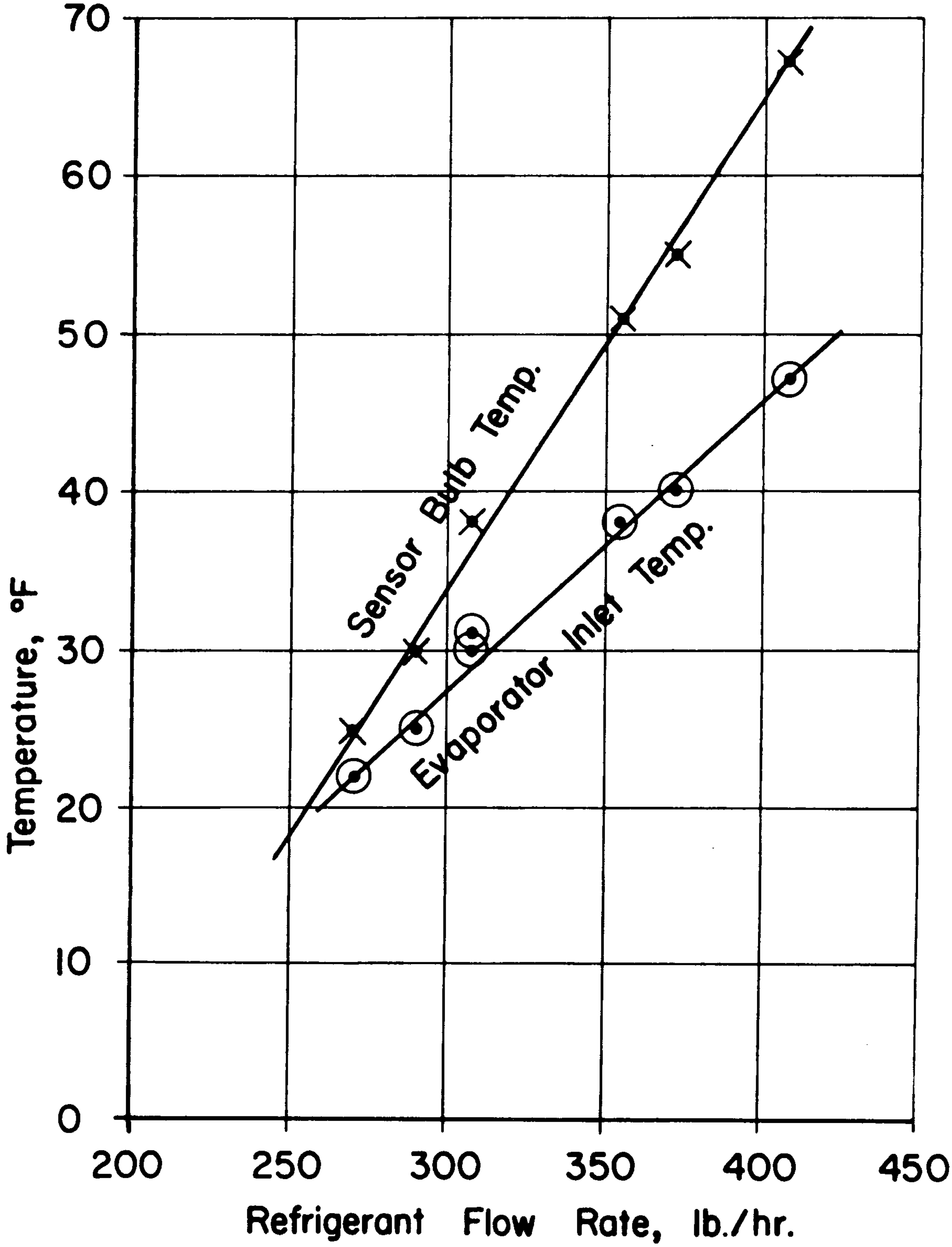


Figure 1-18 Performance of Thermostatic Expansion Valve with R-11/Air Charge

obtained in tests at lower heat inputs as well, the possibility of finding a bulb location better suited to the superheat characteristic was investigated.

Part of the problem related to the difference in operation between compression systems, for which the TEV's were developed, and absorption systems using an absorbent liquid of low but significant vapor pressure. In the latter case, the presence of some absorbent in the refrigerant results in liquid always overflowing from the evaporator. As contrasted to the condition in a compression refrigeration system in which the thermostatic expansion valve bulb senses either superheated vapor or cold liquid refrigerant, the bulb at the end of the evaporator in an R-133a/ETFE absorption system is contacted by a liquid whose boiling point changes gradually as a function of the amount of absorbent left in the refrigerant. The boiling point of the liquid refrigerant at a given pressure rises as the volatile refrigerant evaporates from the comparatively non-volatile absorbent, but the effect is not great until the major portion of the refrigerant has evaporated. Figure 1-6 shows the effect of evaporation on the evaporating temperatures of refrigerants which enter at purities of 98% to 99.9%. As indicated by that diagram, thermostatic expansion valve control at "superheats" of 15°F or more can only happen after some 95% evaporation has occurred. Unless the inlet chilled water temperature is very high, enough evaporation to raise the boiling point of the refrigerant to the 55°F - 60°F range cannot occur in the evaporator. To use such a valve efficiently, the sensor bulb must therefore be placed at a location warmer than the evaporator outlet. The vapor outlet of the precooler could be one such spot. Other locations might also serve. The locations that were tried or considered were the following:

1. Exit of precooler - This location gave a large temperature difference to work with, but resulted in large amplitude cycles because of the long delay between valve action and effect on bulb.

2. Refrigerant line leaving precooler - Resulted in cycles of large amplitude for same reason as #1.

3. Chilled water outlet line - Required low superheat.

4. Chilled water inlet line - Time lag excessive, and chilled water inlet temperature was too low.

5. Dual control between outlet chilled water line and refrigerant line worked relatively well, but extremely sensitive to amount of contact to the individual lines.

All of the alternative bulb locations were therefore found to be unsatisfactory. The evaporator outlet remained the best position. Its use would require low superheats over the full range, and as tried earlier, it had produced somewhat erratic operation. The latter appeared due to the well for the sensing bulb which, located below the bottom evaporator coils, might have been wetted inconsistently by the liquid dripping from the coils. A collecting pan was therefore placed below the coils to catch and funnel the dripping liquid over the control bulb well, as shown in Figure 1-19. This arrangement resulted in much more stable operation and has been incorporated in subsequent evaporator designs.

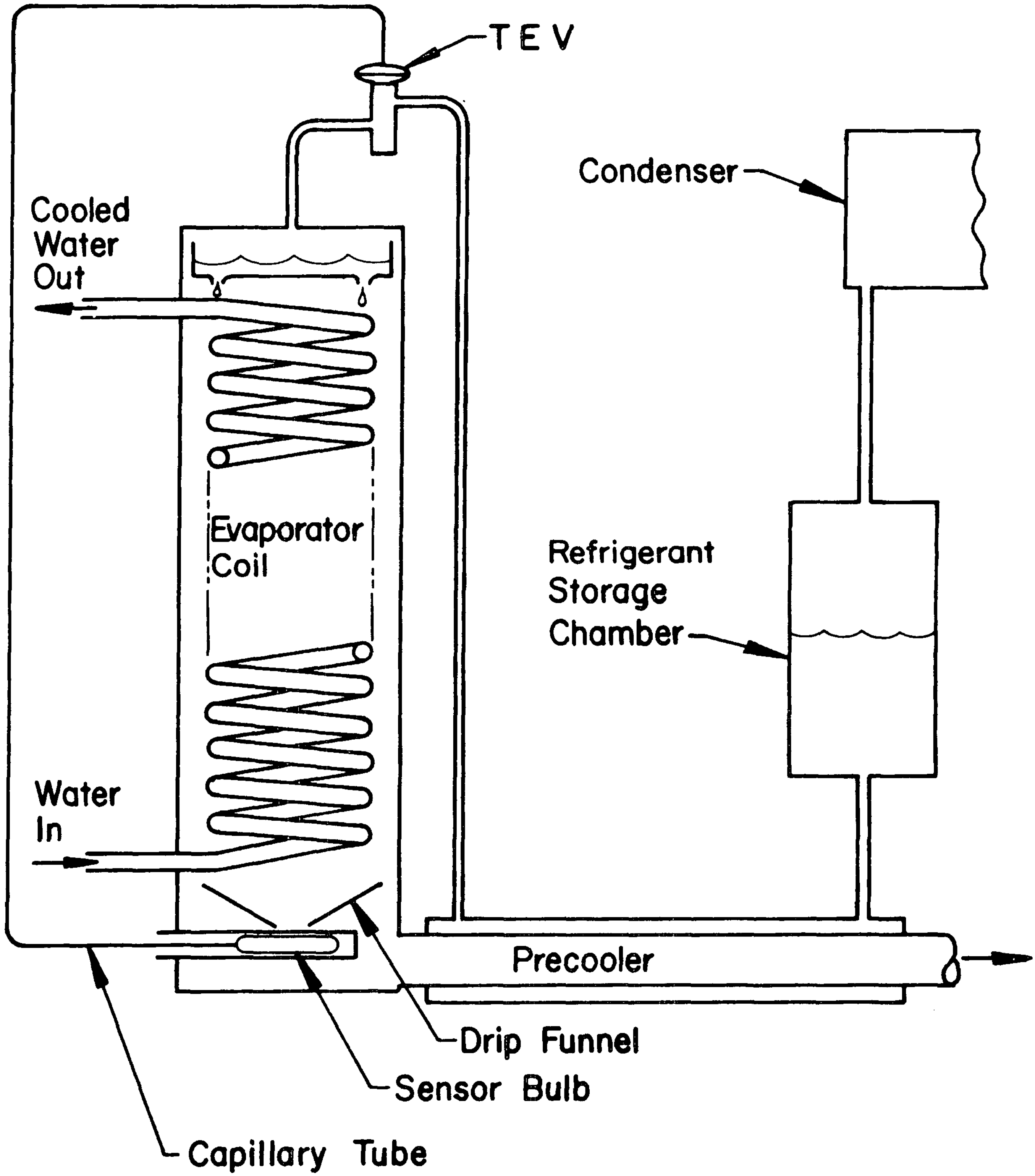


Figure 1-19 Thermostatic Expansion Valve Installation

While the above work was underway, further discussions with Sporlan had resulted in their suggestion that R-21 might provide results closer to the requirements. Samples of power elements charged with R-21 were therefore supplied.

The problem of reducing the superheat and its rate of change to the low level required at the outlet of the evaporator was also attacked by analyzing the spring forces required to produce superheat. The valve springs used in the past had been those used by the manufacturer in the R-12 valves. Normal valves are intended for use at one temperature setting, or over a relatively narrow range of temperatures.

As shown in the schematic diagram of Figure 1-17, the thermostatic expansion valve is designed so that the valve seat position is determined by the balance of the forces acting on the diaphragm. The factors determining the forces are:

P1 = Condenser (high-side) pressure.

P2 = Evaporator (low-side) pressure.

P3 = Bulb pressure.

Fs = Spring force.

AD = Area of diaphragm.

AO = Area of valve orifice.

At equilibrium the upward forces equal the downward forces:

$$Fs + AD \times P2 + AO \times P2 = AD \times P3 + AO \times P1$$

Solving for Fs:

$$Fs = AD\,(P3 - P2) + AO\,(P1 - P2)$$

This equation is not exact in that the spring force of the diaphragm is not included, or known. But if the diaphragm is near its neutral position, its spring force is small, and this equation can be used for selecting a first-order combination of spring force and charge for the bulb. Adjustments may then be necessary.

To determine the necessary valve openings, an equivalent valve, to be manually adjusted through a micrometer screw, was made up. Then by applying the vapor pressure conditions at the high and low evaporator temperatures, the spring force and spring rate necessary to produce the desired superheat change over the evaporator temperature range were calculated. The spring rate was calculated to be 212 lbs/in., much higher than for normal valves.

After construction and installation of the new spring, the test results obtained were close to the performance desired for the Field Trial units. The data are listed in the following table.

TABLE 1-16

THERMOSTATIC EXPANSION VALVE PERFORMANCE

Heat Input, Btuh	72580	72360	72580	71780	72000	71860
Evaporator Inlet Temp, °F	41	36	26	18	12	8
Bulb Temp., °F	53	48	38	30	21	15
Superheat, °F	12	12	12	12	9	7
Precooler Vapor Out, °F	81	79	76	74	68	62

For comparison with Figure 1-18, these data are plotted in Figure 1-20. The superheat is seen to decrease from 13°F at a 40°F evaporator temperature to 7°F at a 8°F evaporator temperature. That rate of reduction was essentially the superheat characteristic desired. The refrigerant flow rate is seen to have dropped from 475 lb/hr at a 43°F evaporator temperature to 200 lb/hr at

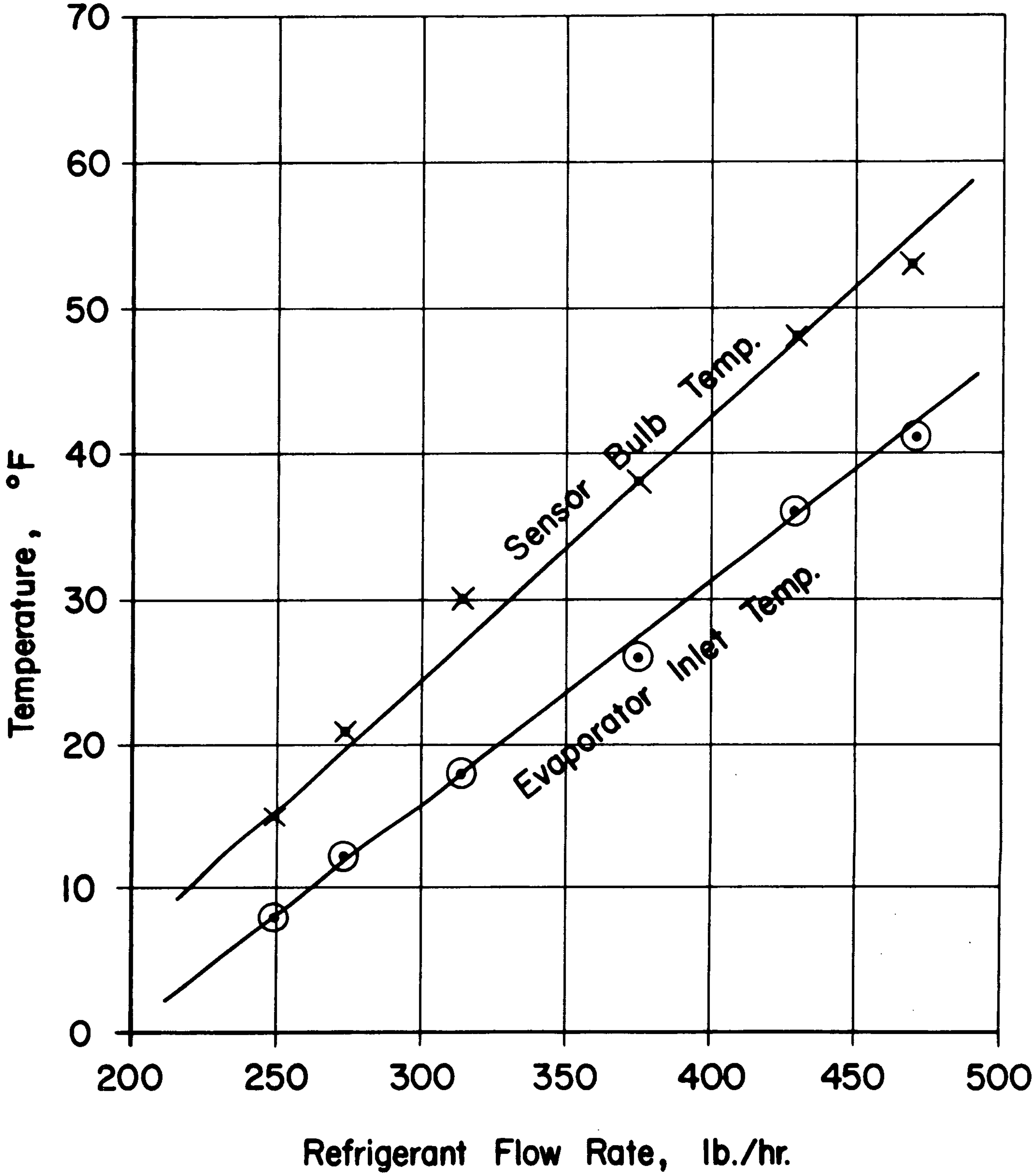


Figure 1-20 Performance of Thermostatic Expansion Valve with R-21 Charge

0°F, which also agreed closely with the reduction in refrigerant rate expected of the Field Trial units.

In the meantime, the stick-slip problems had been greatly reduced by improvements in the Teflon coating of wearing parts. Graphite liners had also been tried. Only occasional small problems were being encountered.

Six thermostatic expansion valves were then built to the design of the above test model. The performance of these valves on the Field Trial units was essentially equivalent, the actual superheats and flow rates being adjusted to the performance of the individual units. While the superheats listed above were close to the original plans for the Field Trial units, superheats of 8°F to 10°F at 40°F evaporators were also used.

Meanwhile, unit development and testing were indicating that even lower superheat levels would be needed for the ultimate prototype models. Lower rates of superheat change would be required under those circumstances. It would probably also be necessary to pick another fluid to charge the power elements. A trial of one of the above expansion valves set at a low superheat at 40°F confirmed that expectation; no superheat remained at 0°F.

Additional thermostatic expansion valve work had been scheduled to develop the valve designs for the M^3 units. That work needed to be done in close cooperation with the manufacturer. It was therefore logical to combine the work on further superheat reduction with the M^3 design studies. The first phase development of thermostatic expansion valves was therefore terminated at this point.

6.0 WEAK LIQUID FLOW CONTROL

INTRODUCTION

The rate of flow of the weak liquid from the generator to the absorber is a major determinant of system performance. When operating with the R-133a/ETFE fluid pair in a heat pump system, the optimum solution flow rate varies with the operating conditions. In heating mode, the solution flow rate must be increased as the outdoor ambient temperatures drop to achieve the best performance and to limit the peak boiler temperature. In addition to affecting the system equilibrium conditions, the weak liquid flow also determines the pumping power required and the heat exchanger sizes. Thus, the flow must be kept at the lowest rates required by the unit components, while being varied to match the operating conditions.

The fluid flow can be controlled either by a variable metering pump or by an automatic throttling device. In the case of the metering pump, the flow of weak liquid returning from the high pressure of the generator to the low pressure of the absorber must be throttled in a manner to match the pumping rate. If instead, an "intelligent" valve is used to control the weak liquid flow, the pump must be capable of pumping whatever flow the valve sends to the absorber. A dynamic pump is preferable for that purpose.

In the previous programs two flow control valve concepts had been investigated: a float valve for use with a positive displacement pump which controlled the flow, and a control valve to meter the weak liquid flow as a function of low-side pressure. Neither valve had reached completely satisfactory performance. The float valve had received the main emphasis immediately preceding this program because the pump being developed was the positive displacement vane pump. A choice was to be made early in this program between the two pumps and the corresponding valves. This report describes the basis for the decision and the development investigations on the weak liquid control valve.

OBJECTIVES

The project objectives for weak liquid flow control valves may be listed as follows:

- To select the valves to be given first priority for utilization in the Field Trial units.
- To develop the flow control valves selected for controlling weak liquid flow to match system requirements for operation under all weather and load conditions.

STATUS AT BEGINNING OF PROGRAM

A float valve had been under development in the preceding program. A float in the solution pump sump at the bottom of the absorber controlled the valve opening to adjust the weak liquid flow into the top of the absorber in a manner to maintain a constant level at the pump inlet. A system of solid lubricant bearings had been developed for the valve, and a spool valve design had eliminated most of the effects of high-side pressure on the valve action. Some inconsistency of operaton when the valve was almost fully closed still remained. When the vane pump was also ready, the valve and pump had been installed on a test unit. In those tests, it had been found that the time delay in flow from the top of the absorber to the bottom was greater than had been allowed for in the bench testing. Due to the area and many crevices in the tube coils, a change in valve flow had been found to take up to two minutes before affecting the solution level at the pump. The resulting overshoot problems had been partly solved as this project was started.

In earlier work using the regenerative turbine pump, a weak liquid flow control valve based on the pressure regulator principle had been under development. Instead of controlling pressure, the valve had been designed so that the low-side pressure controlled flow. As the low-side pressure was a function of the evaporator temperature, the flow rate could be related to the outdoor ambient temperature. The valves being used in the development were commercial refrigerant expansion valves of the automatic, or constant pressure, type manufactured by Sporlan Valve Company. The problem of operation without lubrication in the bearings, and of matching the flow rates to the cycle requirements had been partly solved, but the operating temperature range was still limited. The valves had been in frequent use on test units, with the regenerative-turbine pump, and had performed relatively well over much of the heating temperature range.

INVESTIGATIONS DURING THIS PROJECT

Choice of Valve

The testing of the vane pump and float valve underway at the start of this program was continued while a decision was being made between the vane pump/float valve and the regenerative-turbine pump/automatic valve combinations. During those tests the magnitude of the problems of pulsing flow from the vane pump and of lag in the float valve control sequence were defined more completely. It was concluded that the probabilities of solving those problems in time for the Field Trial units was low. In comparison, the problems remaining in the regenerative-turbine pump/automatic valve combination were more manageable within the time period. The decision was therefore made in favor of developing the low-side pressure-actuated flow control valve.

Pressure-Regulated Control Valve

The decision to use the regenerative turbine pump placed the burden of volumetric flow control on the valve. As mentioned earlier, it is necessary to increase the solution flow rate at low evaporator temperatures not only to obtain peak performance, but also to avoid excessively high generator temperatures. The reason is that the concentraion of the rich solution exiting the absorber is determined by the evaporator, or low-side pressure, and the aborber exit temperature. Since the exit temperature is fixed by the temperature of the warm coolant returning from the house, the concentration of the rich liquid must vary as a function of the low-side pressure, and will decrease as that pressure decreases. At any fixed weak liquid flow rate, the weak liquid concentration would follow the rich liquid and could approach zero refrigerant concentration and/or reach excessive temperatures in the generator. Thus, to avoid such excessive reduction of the concentration of the weak liquid, its flow rate must be increased. The low-side pressure was found to be one basis for control.

The minimum requirements, for the weak liquid control valve are: a) that it be sensitive to low-side pressure and, b) that it increase the flow to the proper extent as the low-side pressure decreases. Though not used in that manner, these are characteristics of pressure regulators and of the automatic expansion valve used for control of refrigerant flow to evaporators intended for operation at a constant temperature. Commercial automatic expansion valves were therefore tested, specifically the Sporlan COFE-23 and CPF-12 valves.

A test fixture which provided for control of the downstream pressure (to simulate actual absorber pressure), as well as the upstream pressure, was set up. The COFE-23 valve was tested first. Its flow range was too wide, and most of the flow variation occurred between zero and five inches Hg of vacuum rather than in the required range of 5 to 25 inches Hg of vacuum. Also, the valve showed marked hysterisis effects.

The CPF-12 valve was checked as received on the test fixture, and two earlier problems were re-encountered:

1. The pressure range over which the flow range occurred was too short.
2. The valving action was erratic.

Increasing the piston spring force (by adding spacers) brought the flow variation within allowable limits, but the erratic diaphragm action was still a problem. Solid lubricant surfaces had been applied to the moving parts, i.e. push-pins and pin carrier, to eliminate stick-slip due to the lack of lubricant. Similar problems with the thermostatic expansion valve had led to direct measurement of diaphragm movement versus small changes in pressure. Sudden, snap action, "oil-canning" type of movement had been encountered. Discussion of the problem with Sporlan engineers led to the conclusion that the oil-canning was probably due to diaphragm stresses from welding into the housing. The effect was imperceptible at the operating pressures of R-12 or R-22 for which the valves had been made, but became a problem at the below-atmospheric pressures of R-133a. Push-pin lengths were changed in an attempt to find a diaphragm working range where "oil-canning" would not occur, but no complete range was found. The irregular motion previously blamed on stick-slip friction was probably partly caused by irregular diaphragm action.

Due to the urgent need for valves for the Field Trial units, it was decided to design and build a valve specifically for the weak liquid flow requirements. A detailed design (drawing # 850-5-1) was made, and a prototype valve was built. This prototype was similar to the design of the valves subsequently installed on the Field Trial units, Figure 1-21 (drawing # 850-7-1).

Inasmuch as the basic problem with the commercial valves had been found to be erratic diaphragm motion, a lot of attention was given to the design of the diaphragm and the selection of material for it. The diaphragm was designed to be clamped in place between flat flanges with an "O" ring for sealing, in order to avoid the warping caused by welding around the edge of the diaphragm. The first diaphragm tried was 0.010" stainless steel. The diaphragm was made from coiled stock. A test apparatus using water was set up so that the inlet pressure was controlled by a pressure regulating valve, and the downstream pressure was controlled by discharging into a partially evacuated vessel. Flow was measured with a rotameter. The tests revealed the following:

1. The valve did respond to downstream pressure.
2. The range of response was too short, indicating that the diaphragm was too thick, i.e., too stiff.
3. This diaphragm also had an "oil-canning" motion.

The oil-canning was attributed to the fact that the diaphragm was made from coiled stock and therefore had a slight bias in one direction.

A Teflon (PTFE) diaphragm, 0.065" thick, was substituted for the stainless steel diaphragm. The assembly was tested on the test apparatus with results as follows:

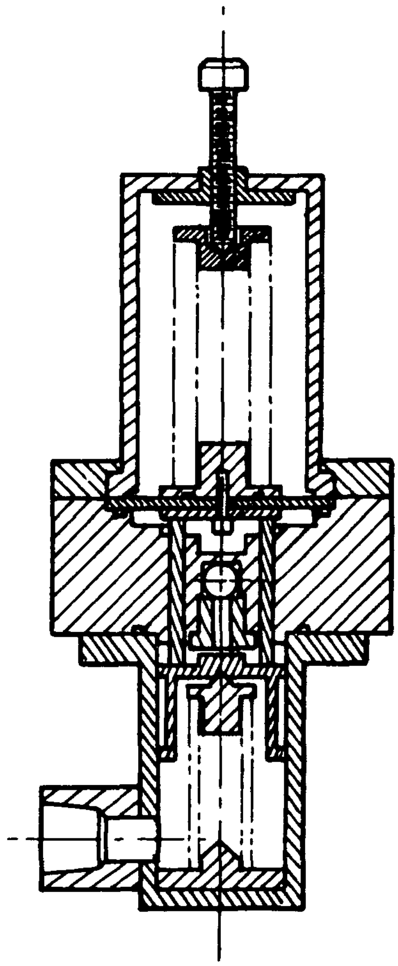


Figure 1-21 Weak Liquid Expansion Valve

1. The valve responded with smooth, uniform action.

2. The valve controlled over most of the desired range; response at the two ends of the range was slightly reduced.

3. There was no "oil-canning" effect.

This valve was put on Test Unit 1 for unit performance testing, and satisfactory operation was obtained. After a couple of weeks the valve was disassembled for examination. The only problems found were that the PTFE diaphragm had assumed a convoluted shape with the circular convolution extending toward the atmospheric pressure side, and the periphery of the diaphragm which had been held between the metal flanges had thinned considerably. The convolution was found to have been caused during unit leak testing, when a pressure of a 30-35 psig of nitrogen is applied inside the unit. This pressure would make the diaphragm bulge outward. The thinning of the periphery was due to the clamping force of the housing flanges making the PTFE cold-flow. The diaphragm was retained around the outside edge, forcing the cold-flow radially inward into the flexing area. That cold-flow toward the center could have contributed to the convolution.

It was also noted that at start-up, the low-side pressure would increase, tending to close the valve, and the flow of weak liquid was diminished. As the evaporator supplied vapor to the absorber, the pressure would rise, tending to close the valve even more. Thus, the system could not recover of itself. This problem was solved by milling a slot across the face of the valve so that the flow could not drop below the minimum required. No further starting problems have been encountered.

Three more valves were constructed for test purposes. Because of the uncertainties regarding the use of the PTFE, diaphragms of other materials and configurations were tested with the following results:

1. A convoluted diaphragm was obtained from a thermostatic control for a chicken "brooder". This diaphragm was made from brass about 0.008" thick with concentric convolutions. It was readily adapted to the valve. Response was good in the middle of the range but poor at the ends, indicating too stiff a diaphragm.

2. A flat diaphragm of brass was too stiff and would not control over the full range.

3. A die set was made to produce a brass diaphragm with two concentric convolutions. Brass ranging from 0.003" to 0.007" was tried. The thinner sheets would take a permanent set; the thicker sheets were too stiff and would restrict motion at the ends of the range.

4. A new die to produce diaphragms with four concentric convolutions was made. Again various thicknesses were tried; diaphragms in the 0.006" to 0.007" thickness gave good response.

It was found in all cases, that these corrugated diaphragms were permanently distorted by the pressures applied for leak testing. A back-up plate was built in so that normal motion would be permitted, but extreme reverse extensions during leak testing would be prevented. The back-up plate did prevent the excessive excursions, but some loss of response occurred after leak testing, indicating that a smaller permanent deformation had occurred anyway.

The Teflon diaphram was thus found to be the most practical at the time, and a valve with a PTFE diaphragm was installed on Field Trial unit #1. The flow at low evaporator temperatures was found to be less than desired, so the valve orifice was drilled out from 7/32" to 9/32". This change also decreased the valve motion required, making the valve more responsive at the two ends of the range.

The other problem with the PTFE, the cold-flow in the flange area, was solved by designing the flanges so that the PTFE diaphragm was snugly contained around the periphery, and the squeeze from the flanges was mechanically limited to 0.005 - 0.007" on the 0.065" thick diaphragm. Sealing was attained by use of an "O" ring directly against the PTFE.

A new design, (drawing #850-7-1) shown in Figure 1-21, was made to reduce the size, simplify assembly, and improve the "O" ring sealing by increasing the "O" ring thickness from 1/16" to 3/32". Six valves of the 850-7-1 design were constructed for the Field Trial units. They have operated satisfactorily, except that some cold-flow may have occurred because resetting has been necessary between winter and summer operation in some cases.

7.0 CONCENTRATION CONTROL

In an absorption heat pump intended to operate to 0°F, means are required for adjusting the concentrations of the solutions in the circuit to match the operating conditions. Change of the solution concentrations can be accomplished by storing out varying amounts of refrigerant. Provisions must also be made for the accompanying change in volume of solution. The location and operation of the storage chambers are dependent on the pumping and flow control means in use. Since the original project plan included the option of using one of two pump/flow-control systems, the development of a concentration control means was included in the plan.

The decision to use the dynamic pump, with a pressure regulator type weak liquid control and a thermostatic expansion valve for the refrigerant made it possible to use a previously developed concentration control concept. That system (2) shown in Figure 1-22, includes a liquid refrigerant storage chamber at the outlet of the condenser and a solution storage chamber in the generator. The amount of refrigerant removed from or released to the circulating solutions is controlled as a function of the operating temperatures, evaporator load, solution circulation rates, etc. The direct agent of that control is the thermostatic expansion valve which controls the flow out of the refrigerant storage chamber in response to evaporator load and temperature.

With the pump/flow controls decision, which determined that this existing concentration control system would be used, the need for development of the concentration control was reduced to sizing the storage chambers to the requirements of the fluids and the unit components. That was done, the Field Trial units operating properly over a range of evaporating temperatures from 0°F to 45°F.

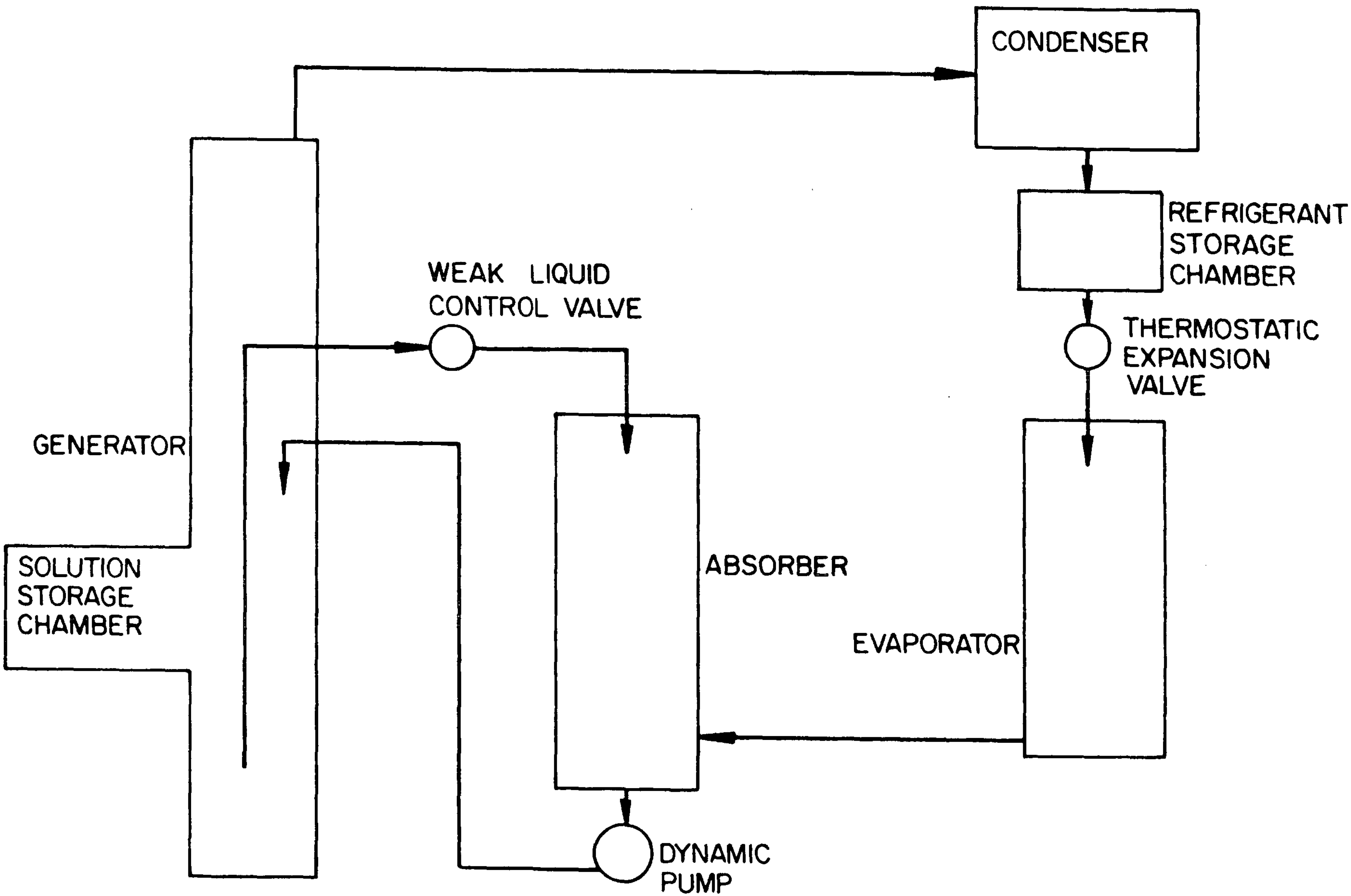


Figure 1-22 Concentration Control System

2.0 AUXILIARY COMPONENTS

EIGHT-WAY VALVE

INTRODUCTION

The year-round heat pump was planned as a system that would utilize hot and cold water (water/glycol brine) to transfer the heating and cooling to the building. The block diagram of Figure 2-1, shows the chilled and hot water circuits interconnecting the absorption system components and the indoor and outdoor heat transfer coils. The diagram shows the need for and location of a valving system to switch the outgoing streams as needed to match the heating and cooling needs. Because defrosting of the outdoor coil in winter could be done best by means of the hot water in the system, the flow switching means was also an integral part of the defrost process. Past investigations had determined that use of multiple commercial valves to switch the eight water connections would be complicated, bulky, service-prone and excessively costly. Considerations of those factors, as well as heat losses between the hot and cold streams had led an earlier program to the development of a prototype eight-way valve molded of plastic.

This report covers the investigation made during this project to develop water-circuiting valves for the six Field Trial units on the basis of that earlier development.

OBJECTIVES

The objectives of this portion of the project were to find means of utilizing the previously-developed eight-way valve, improving its reliability and eliminating cross-leakage between the hot and cold water streams. The design was to be suitable for use on the Field Trial units.

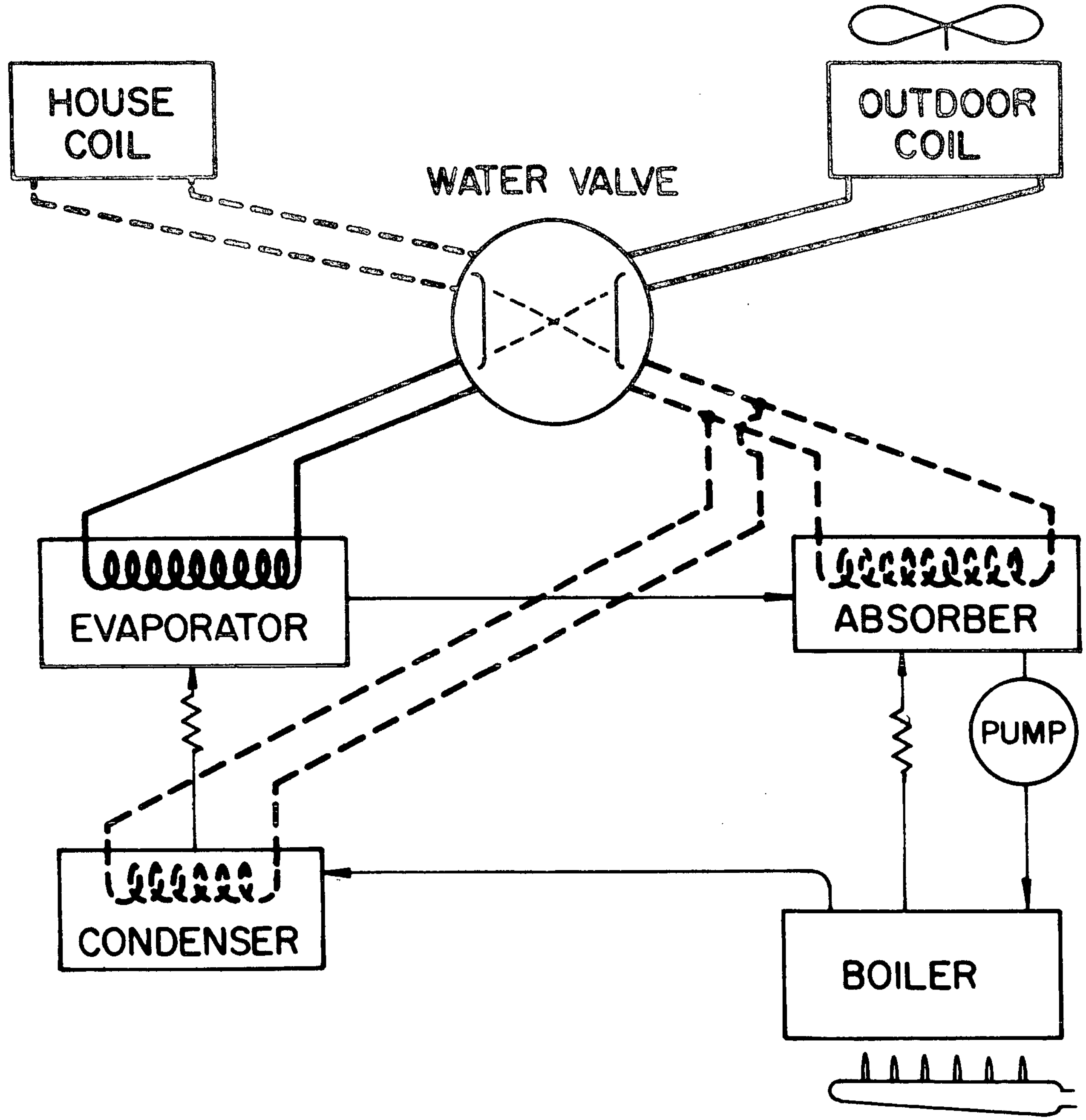


Figure 2-1 System Block Diagram

STATUS AT BEGINNING OF PROGRAM

An eight-way valve, as shown in Figure 2-2, had been developed at Whirlpool Corporation in the late 1960's. Prototype tooling had been made, and the valve had been used in various installations. A number of difficulties had been encountered, cross leakage between the hot and cold sides being one of the most common. Because the valve operated only a few times during the year, in the Spring and Fall, and during a few defrosts in the winter, difficulties in starting, and overloading of the motor drive were also possible problems.

INVESTIGATIONS MADE

A short analysis was made of potential valve concepts with the conclusion that the requirement to have valves in time for the Field Trial units made it necessary that the program be based on the Whirlpool valves if at all possible. Whirlpool was contacted, and it was determined that there were no valves left, but that the tooling for the plastic components was still at the prototype molder's plant. The tooling was found to be in usable form, and an agreement was reached to purchase it for this program. A run of 50 sets of parts was ordered for valve development and for use on the Field Trial units.

As shown in Figure 2-2, the Whirlpool valve was equivalent to two, four-way plug valves placed end-to-end in a single housing. Instead of plugs, wobble plates at a 45° angle were mounted on a common shaft. O-ring seals were used on the wobble plates, partition and end caps to prevent leakage between streams or to the outside. By rotating the shaft and wobble plates one half turn, the flows of the streams leaving and returning were exchanged between the two coils. The valves were operated by a small geared motor, the total assembly mounted on a metal frame.

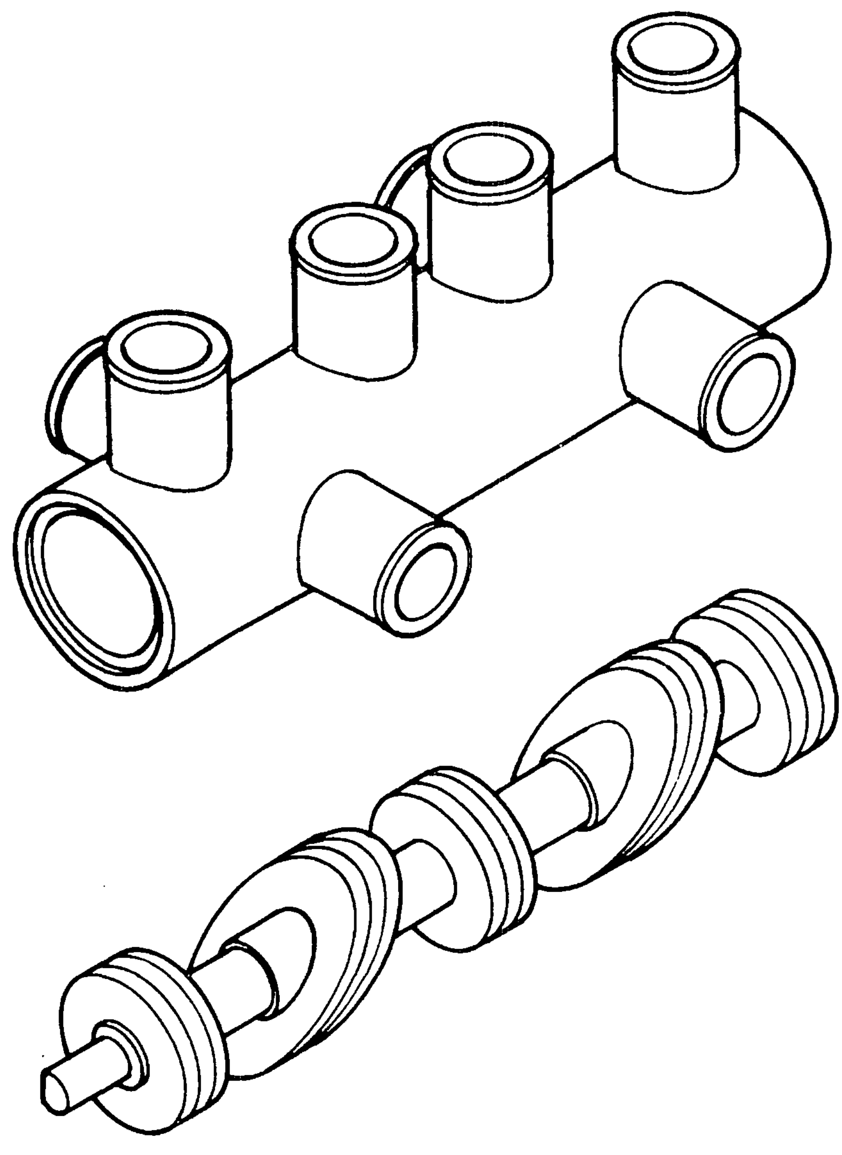


Figure 2-2 Eight-Way Valve

A variety of problems were encountered as the valves were constructed and tested. The molded valve housings were found to be defective. Shrinkage had occurred during molding, leaving a number of depressions on the inside cylinder surface. It became necessary to bore out the cylinders because there was no certainty that perfect parts could be made in time, even with considerable experimentation.

The O-ring seals on the wobble plates posed a series of problems. Apparently due to the leverage on the angled wobble plates, leaks could occur at the outer ends of the elliptical plates. The O-rings were also often pinched as they went past the inlet and outlet ports. The result might be scuffing or tearing the O-ring, or sometimes stopping the rotation altogether. To correct those problems, changes were made in the O-rings and the wobble plates. The wobble plates were fitted more tightly to the shaft and the housing, and with closer tolerances. The O-ring groove in the wobble plate was adjusted to a closer O-ring fit. The O-ring diameter was reduced so that it would be stretched in the groove. Its thickness was also adjusted for a better seal to the cylinder. The sealing was improved with those changes, but scuffing or cutting of the O-rings was not eliminated. A number of O-ring materials were tried with only minor successes. Teflon O-rings performed relatively well when first applied but would take a set after a time and would then leak. Teflon-coated rubber O-rings did not solve the problems either.

The next step was enlarging the O-ring thickness to 3/16 inch. That improved the stability of the O-ring and eliminated or greatly reduced the scuffing and attendant leaks. However, the fits were so tight that there was very little tolerance between being leak-tight and having excessive friction. Continued investigation of materials and hardnesses was indicated, but good results were being obtained with 3/32 inch O-rings by adjusting the groove depth. The 3/32 inch O-rings were giving the best results when the time for installation on the Field Trial models arrived and were used on those systems.

As will be reported in Section 4, a variety of similar problems were encountered on the Field Trial units, including leaks. Further development was therefore necessary as part of the M^3 prototypes, which will be reported in a subsequent report.

The motor drives also required considerable development. The valve design had been intended for operation with a reversing motor. The motor was to rotate the valve 180°, being brought to alignment on hitting a mechanical stop. The motors were geared motors to operate at a 2 rpm shaft speed. They were also specified for continuous operation in a stalled condition. In that manner, only a reversing switch was required. The motor would stop at both ends of its travel, stalling against the lugs that aligned the valve passages in the two operating positions.

It was found important to increase the speed of rotation in order to reduce the mixing of cold and warm liquids during the changeover. This was especially important in the case of defrosts when the inventory of hot water was an important factor in completing the defrost. The cost of maintaining the motor in the stalled position, at 25 watts input, was still minor even at present electric rates. But as motor amperage was increased to increase the speed to 5-6 rpm, problems of motor overheating began to appear, in addition to the increase in operating cost. The increased torque also began to impose a twist, or racking, of the valve and motor assembly.

The investigation therefore turned toward designs in which the motor would be off except during changeover. A Merkle-Korff gear motor with built-in brake, operating at 5-6 rpm, was used along with micro switches to locate the end of travel. Mechanical stops were used as backups. Figure 2-3, shows the valve and motor assembly.

By those means, higher speed operation was obtained at Spring and Fall temperatures, when the heating and cooling changeovers would occur. But at the colder temperatures at which defrost was likely to occur, the rate of rotation was found to be noticeably slowed, apparently affected by increased friction in the valve. During the second winter of operation, the 1981-1982 winter, unusual storms at below 0°F temperatures plugged the finned coils with fine snow and ice crystals, actuating the defrost controls. The defrost system did not operate properly, and the units shut down. The temperatures were far below the 25°F to 35°F range at which most defrosting occurs. An eight-way valve was therefore tested after soaking at 0°F in a freezer. The valve operated, but at an almost imperceptible rate. A test of the geared motor at those temperatures found it to operate at normal rate. The friction was thus all in the valve itself. Trials of Teflon-coated O-rings at 0°F determined that they provided no improvement. The valves were not found to be stuck at the start, only moving at a very slow rate. An investigaton of materials, their friction factors relative to temperature, expansion coefficients, etc. is obviously called for. Another solution to this problem would be the application of a much higher torque motor. Since the motor operates only a few times a year, operating costs would not be a factor. To do that, the plastic valve and mounts would also have to be redesigned so that the assembly could take the high torque under extreme circumstances.

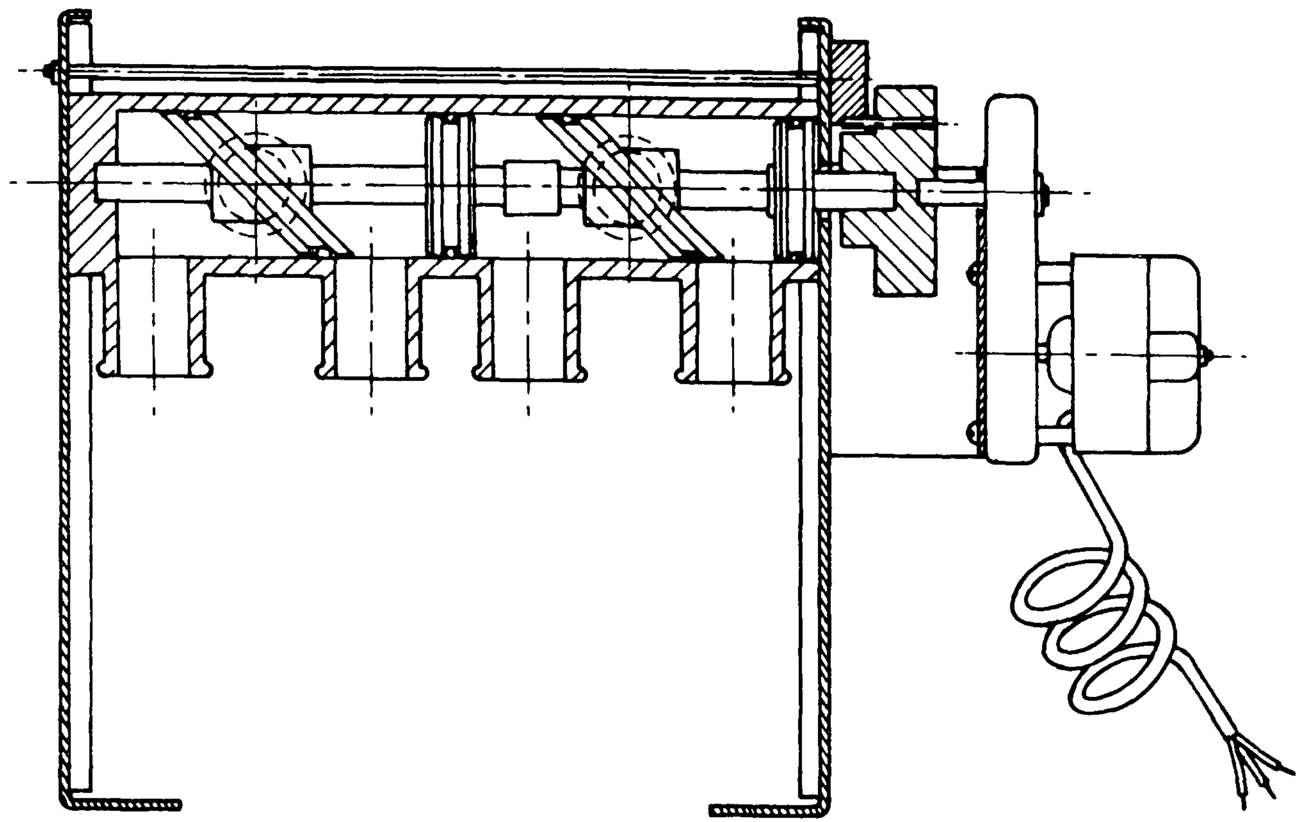


Figure 2-3 Eight-Way Valve and Motor Assembly

Eight-way valves incorporating the best variations listed above were installed on the Field Trial units. They operated relatively well for a period, but ultimately problems were encountered. Leakage and cutting or scuffing of the O-rings were experienced. In one or more cases the valve stalled before reaching its final location, resulting in cross-over leaks. The valves were repaired and replaced when the weather permitted, but further development was obviously necessary. The problems were therefore tackled again as part of the M^3 development. That work will be reported in a later volume.

2.0 FINNED COILS

INTRODUCTION

As shown in the system block diagram of Figure 2-1, the heat output and the cooling effect of the absorption unit are transferred concurrently to two streams of antifreeze solutions. Those two streams in turn transfer heat to or from air by means of two finned coils, one in the house and the other outdoors. The coils both function to transfer heat to air and from air, depending on the season. The heating capacity goal of the planned units was 90,000 Btuh at rating conditions, and the cooling capacity 36,000 Btuh. The house coil might therefore be expected to transfer each of those heat quantities to or from the house air at different times of the year. The outdoor coil would also transfer heat at equivalent rates, extracting heat from the outdoor air in winter at up to 36,000 Btuh and dissipating heat to the outdoor air in summer at up to 90,000 Btuh.

The outdoor coil would normally be, and was designed as, a component of the heat pump unit. It would thus be a standard, built-in part of the heat pump, as seen in the picture of Figure 2-4. The indoor coil is a separate component. It serves as an interface between the heat pump and the house, and must thus match both. Because houses, climates, installation methods and indoor coil locations vary across the country, it can be expected that a number of models of indoor coils will be needed when the absorption heat pump goes to market. In addition to size variations, A-coils, slant coils and horizontal coils are all likely to be required.

For this project the indoor coil development was restricted to the program needs. For use with the Field Trial units, two types of coils were required. In the residential installations, A-coils were needed for up-flow in the

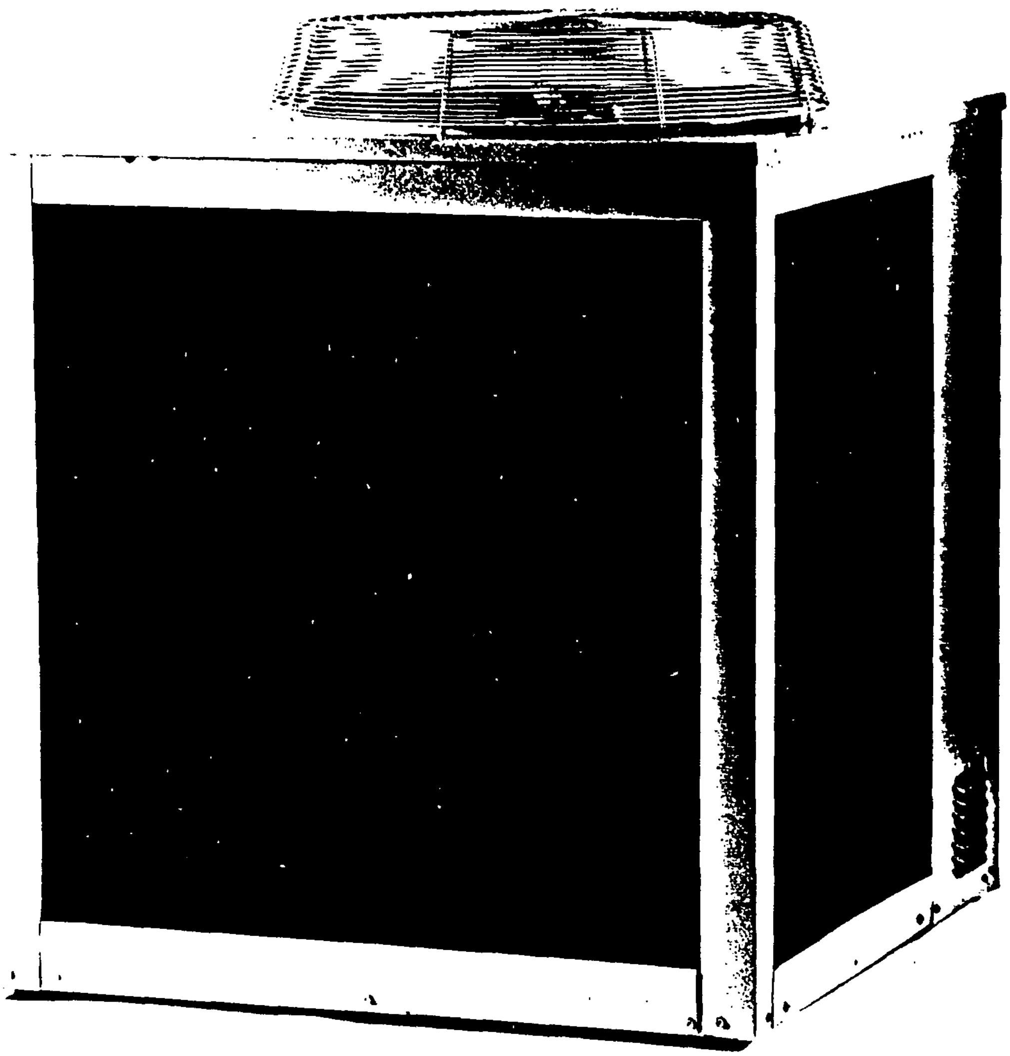


FIGURE 2-4

PROPOSED DESIGN OF PROTOTYPE HEAT PUMP

existing house systems. For the life-test units, blower-coils were needed which would be able to provide all the heat transfer requirements of the units during operation under continuous-run conditions at all building and outdoor temperature and humidity conditions. A blower-coil with horizontal coil configuration was also to be developed. It was developed later in the program for the prototype units and will be reported in that section.

INDOOR COILS

FUNDAMENTAL REQUIREMENTS AND STATUS AT BEGINNING OF PROGRAM

Based on the projected unit capabilities, the indoor coil heat transfer requirements could be 90,000 Btuh in heating mode and 36,000 Btuh in cooling mode. Early calculations showed that a 90,000 Btu heating coil with water entering at 125°F and leaving at 95°F was a very large coil relative to a standard three-ton coil for 36,000 Btuh loading in cooling mode. Further, as seen in the diagram of Figure 2-5, the 90,000 Btuh unit output capacity was to occur at a 47°F outdoor ambient, at which the house load could be of the order of 20,000 Btuh. From the house standpoint, the 90,000 Btuh coil is thus excessive in normal use. There could be an occasional value in having a coil that size, from the unit side, but for that infrequent usage it would add unnecessary cost, take up space and often be awkward, or impossible, to install. Therefore, consideration was given to reductions in coil sizes to arrive at a better compromise between the house and unit requirements.

It was seen that the unit's potential need for high coil capacity at the warmer winter temperatures could be reduced by controlling the unit capacity at those temperatures. Means such as partial modulation of the heat input to

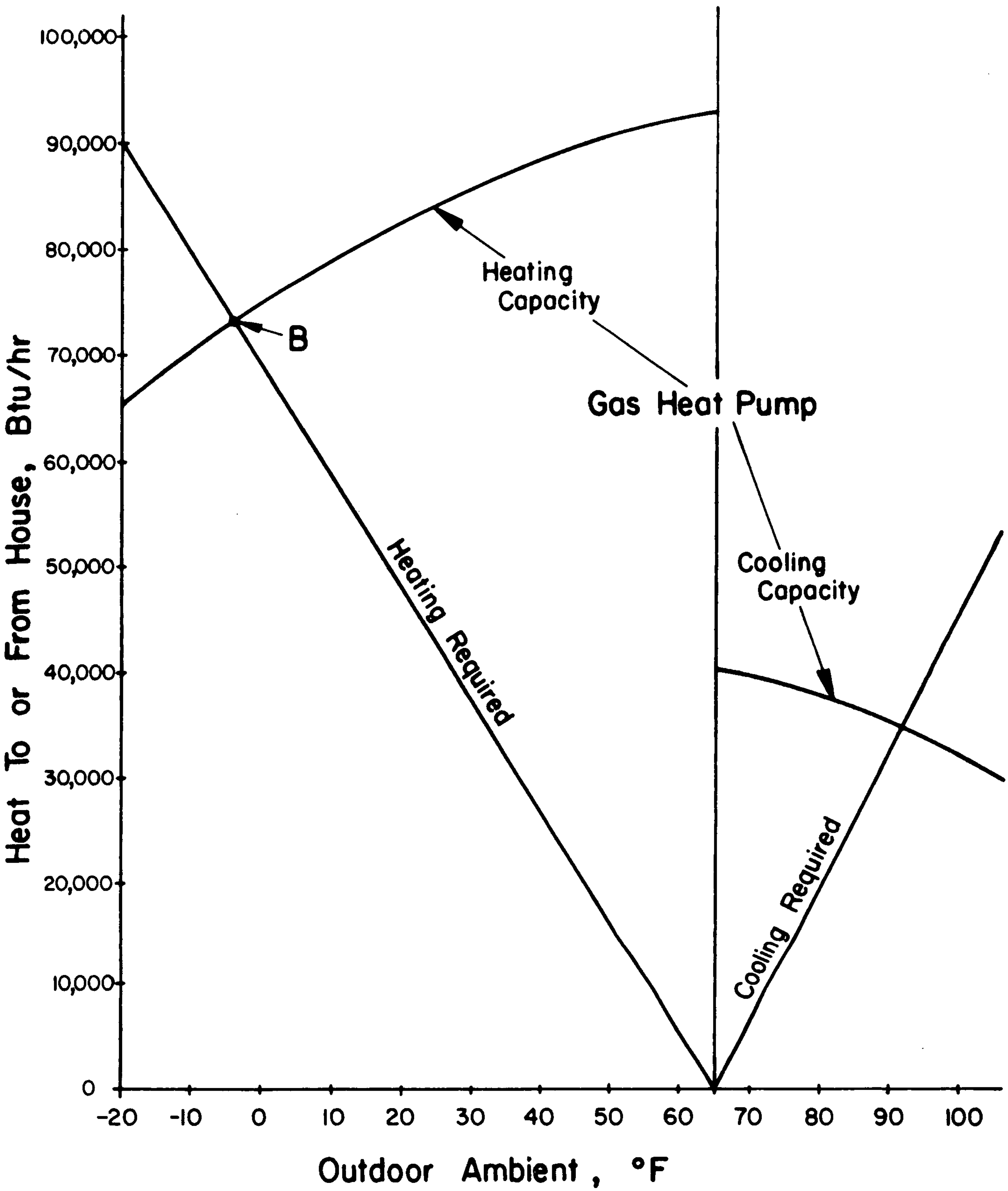


Figure 2-5 Planned Gas Heat Pump Performance Related to House Heating and Cooling Loads

the unit or control of the cycling periods and blower operation could serve to that end. The coil might then be sized to a lower heat output and be a better fit with summer cooling and dehumidification.

The minimum coil size for winter operation could then be related to the condition at which full unit operation under continuous run would occur. That condition is represented by the cross-over point of unit capacity and house load shown as point B in Figure 2-5. That point is at a coil capacity of 72,000-75,000 Btuh.

The coil was to operate with an antifreeze mixture of 35% glycol or higher. The low heat transfer coefficients for glycol solutions showed that the tube surface should be significantly higher than is used in standard air conditioning coils. Manufacturers were surveyed, but the desired primary-to-secondary surface ratio was not found. The maximum tube surface was found on coils manufactured by Sundstrand Heat Transfer Division, which had staggered 3/8 inch tubes placed on 3/4" centers.

Using the Sundstrand Design Manual, A-Coil designs for 36,000 Btuh cooling and 72,000 Btuh heating had been calculated. The heating coil size remained larger than the cooling coils, but they could be similar if the fluid temperatures were 135°F in and 95°F out on heating and 50°F in and 58°-60°F out on cooling. Due to the size and depth of the coil, the latent load was indicated to be 25% or higher under rating conditions even with those high coolant temperatures.

An A-Coil had been designed on that basis. It had the dimensions shown in Figure 2-6, with six rows of tubes and 12 fins per inch. The coil was larger than a standard 36,000 Btuh cooling coil, but not excessive for most retrofit up-flow applications. Coils had been built for test and for a projected field trial.

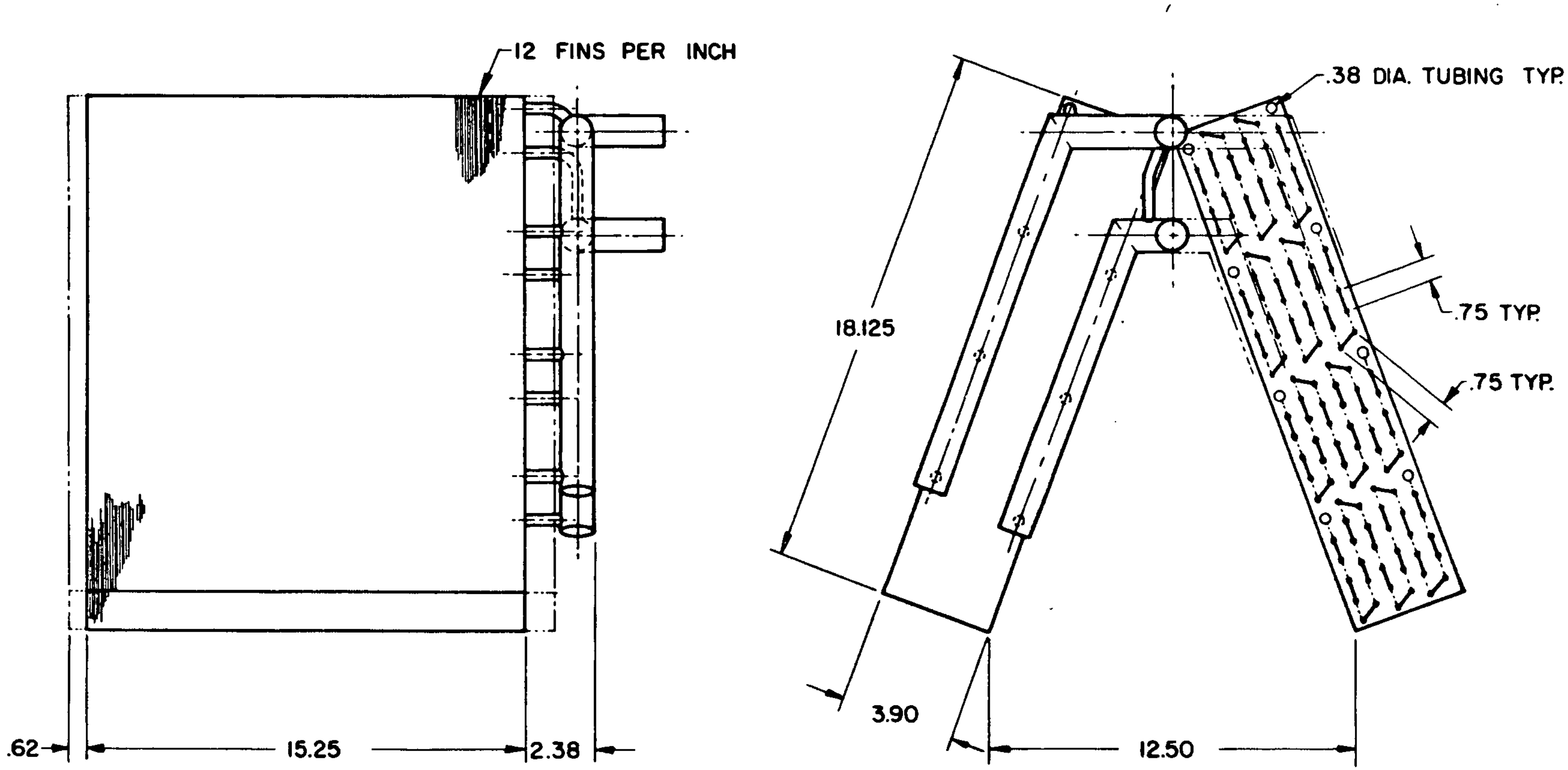


Figure 2-6 A-Coil Dimensions

INVESTIGATIONS DURING PROGRAM

This project's requirements for the Field Trial program were for A-Coils for the residential applications and blower-coil systems for the life test operations. A survey was again made of commercially available coils or coils that might be made on special order. For the A-Coils, it was found that to meet the Field Trial schedule the options remained essentially the same. (Tools were found at Sundstrand Heat Transfer Division which could be modified for use in the development of outdoor coils and indoor coils for the prototype program.)

It was therefore decided that the A-Coils from the earlier stage would be used for the two residential installations. The coils had been tested under both heating and cooling conditions. Some of the results of the cooling tests, made in the psychrometric room at the Buffalo Laboratory, are shown in Table 2-1.

TABLE 2-1

A-COIL COOLING PERFORMANCE

Coolant	----------WATER----------			----------40% GLYCOL----------		
Flow, lb/hr	4049	4066	4060	4039	3976	3952
Temp In, °F	50.2	50.3	49.7	49.8	50.0	50.2
Temp Out, °F	60.7	60.3	58.3	59.2	59.5	60.0
Air In, °F	80.3/68.8	80/67	80/67	80.1/67.1	80.1/67.1	80.3/67.2
Air Out, °F	----	----	55.3	56.6	57.5	58.0
Total Capacity, Btuh	42600	40400	35200	32500	32500	34100
Latent Load, Btuh	8929	10078	9715	7781	8632	8953
% Latent	21.0	24.8	27.6	22.0	26.6	26.3
Approx. Air Flow, Ft3/mm	1600	1600	1300	1300	1300	1400

The tests show that with water as the coolant, the coils had ample capacity. By further adjustment of the air and water flows it would also be able to produce latent percentages in the 25%-30% range with water inlet temperatures of 50°F. With 40% glycol/water, at a specific heat of .835 Btu/lb°F, the coil did not perform up to capacity at the 4000 lb/hr coolant flow rate. The test results indicated that the full capacity should be obtained by increasing the glycol/water flow rate, to close to 5000 lb/hr to make it equivalent to the water loading. The coolant outlet temperature would probably be reduced while maintaining the inlet temperature. The latent load would also be increased.

In heating tests, the coils met the 72,000 Btuh goal marginally with water but not with glycol. The objectives were a 95°F outlet coolant temperature with 68°F inlet air. The tests were run in an uncontrolled temperature space. The temperatures were then corrected to the 68°F inlet air conditious. The following two tables list results obtained with the two coolants.

TABLE 2-2

A-COIL HEATING PERFORMANCE

Coolant	WATER	WATER	WATER	WATER	WATER	WATER
Flow, lb/hr	2000	1800	1600	2000	2200	2400
Temp. In, °F	131.6	138.4	137.7	129.1	130.0	127.8
Temp. Out, °F	95.2	96.5	93.1	93.9	97.9	98.0
Air In, °F	68.0	68.0	68.0	68.0	68.0	68.0
Air Out, °F	109.0	113.5	109.2	106.5	108.7	108.0
Capacity Btuh	72800	75420	71360	70400	70620	71520
Approx. Air Flow, CFM	1540	1510	1645	1665	1580	1620
Δp H20, PSI	----	1.3	1.0	1.8	2.0	2.3
Δp Air, In W.C.	.27	.28	.29	.29	.30	.30

The required capacity was thus obtained with water as the coolant, but the water inlet temperatures had to be higher than the temperatures being developed by the units at best COP's. Also, the air flows were on the high side, and the air outlet temperature therefore somewhat cooler than desired.

TABLE 2-3

A-COIL HEATING PERFORMANCE

Coolant	40% GLYCOL	GLYCOL	GLYCOL	GLYCOL
Flow, lb/hr	2000	2200	2400	2200
Temp. In, °F	131.0	127.9	129.6	127.8
Temp. Out, °F	95.5	95.8	98.8	94.2
Air In, °F	68.0	68.0	68.0	68.0
Air Out, °F	105.0	104.5	106.7	105.0
Capacity, Btuh	61770	61440	64310	64316
Air Flow, CFM	1520	1530	1510	1570
Δp H20, PSI	2.0	2.4	2.6	2.3
Δp Air, In W.C.	.28	.28	.27	.28

With 40% glycol/water the coil performance was well below the objectives. With that coolant, the coil is of 60,000 - 65,000 Btuh capacity. The air flows used were on the high side for the capacity, and the air outlet temperature was lower than desired. Higher air temperatures would have been produced at lower flow rates, with perhaps only a small reduction in capacity.

The tests thus determined that those A-Coils could perform satisfactorily at a cooling load of 36,000 Btuh, but would not perform to the 72,000 Btuh objective with glycol-water. Heating operation was thus controlling. It could also be concluded that a coil of adequate heating capacity can operate to over 25% latent load with an inlet coolant temperature of 50°F.

Because the houses in which the Field Trial units were to be installed did not have three-ton cooling loads in the Michigan climate, the units were to be derated to 2 1/2 tons or less. At the reduced input, the 60,000-65,000 Btuh capacity of the existing A-coils would be adequate. They were therefore used for those two installations and performed well. Detailed coil data were not taken because the houses were not instrumented for that purpose. The coolant temperatures were maintained at the design temperatures, and dehumidification and comfort were good year-around. The potential cold drafts from the somewhat low temperatures of the outlet air did not occur as long as the blower operation was coordinated with water pump operation. The blowers used were those in the existing furnace systems.

No specific provisions were made to prevent the unit output from rising to its peak in the warmer ambients. Anticipator adjustments were made on the house thermostats to prevent long cycles. No high water temperatures were then encountered in normal operation. If the unit was manually kept on for longer than a 15 or 20 minute "on" time, to check out unit operation, water temperatures would rise above normal levels.

To match the full 72,000 Btu load, a coil of 20% greater capacity would be needed. If the cycle "on" time is to be longer than 15 or 20 minutes, the coil might need to be longer yet. Alternatively, the gas input could be reduced at the higher ambients, or a water temperature limit control could be used to shut off the gas for short periods. The water and air pressure drops in these coils were near the limits, so designing for larger coils would require control of the pressure drops as well as the coil dimensions.

LIFE TEST UNITS

In the continuous operation of the life test units, the function of the indoor coils was primarily to provide the loads for the units. The coils also supplied part of the heating and cooling of the building, but a good match to the building was not required. Under the continuous run conditions, the full heating output would be applied to the coils at the warmer ambients. To provide the full 90,000 Btu capacities, full-ton (60,000 Btuh) blower coils were bought which had space for the addition of a heating coil. Extra five-ton coils were bought and installed in the space provided. The coils were connected in series, with the water connected for partial counterflow to the air stream.

The coils served the purpose satisfactorily. There was excessive coil capacity in the summer, so it was necessary to reduce the air flows to bring the chilled water temperatures down to the proper range. In heating mode, there were conditions when the coils did not reduce the cooling water to the design 95°F. The conditions involved the locations of the coils, building usage and outdoor temperatures. The coils were mounted at the 12-foot ceilings, and stratification of the air resulted in inlet air temperatures up to the 80°F range when other devices were not circulating the air. At the warmer ambients, the higher unit capacity and reduced building load also played parts in overloading of the coils.

At the warmest heating mode ambients, the heat pumps alone were sufficient to overheat the building, so the thermostats had been connected to change the operation from heating to cooling when that occurred, rather than shutting off the units. However, if the return water temperatures happened to rise above the design 95°F, no harm was done to the life test or to the units. The only effects would have been a small loss in COP and rise in average boiler temperature.

OUTDOOR COILS

BASIC REQUIREMENTS AND STATUS AT BEGINNING OF PROGRAM

The outdoor coil had load requirements similar to the indoor coil. The 90,000 Btuh load occurred in cooling mode operation at the 95°F rating ambient. The heating mode design load was a 36,000 Btuh cooling requirement at a 47°F rating ambient. During previous work, the coil size and design had been fairly well established relative to a unit the size of a 40 inch cube. The coil was designed to cover most of two faces of the cube, with a total face area of 16.8 square feet. The coil sizes were based on available finned slabs. The fins were 1 1/2 inches wide and 18 inches high with two rows of staggered 3/8 inch tubes, 3/4 inches apart. The coils were L shaped, as shown in Figure 2-7. Each face of the coil was composed of four of the slabs, two deep and two high. The water entered the inner of the four rows of coils and exited from the outer row, flowing through eight parallel circuits.

INVESTIGATIONS DURING PERIOD

To reduce the pressure drop on the water/glycol side and allow higher flow rates, a tubing system was developed with 16 parallel circuits. The water was distributed to all circuits at the inside row of tubes from a header the full height of the coil. It flowed back and forth across both faces of the coils, making three passes in each row of tubes. The coil assembly for the field trial units had one side that was 34 3/4 inches wide and the second 32 1/2 inches wide, both being 36 inches high. Test results with a 35% glycol/water coolant are shown in Table 2-4.

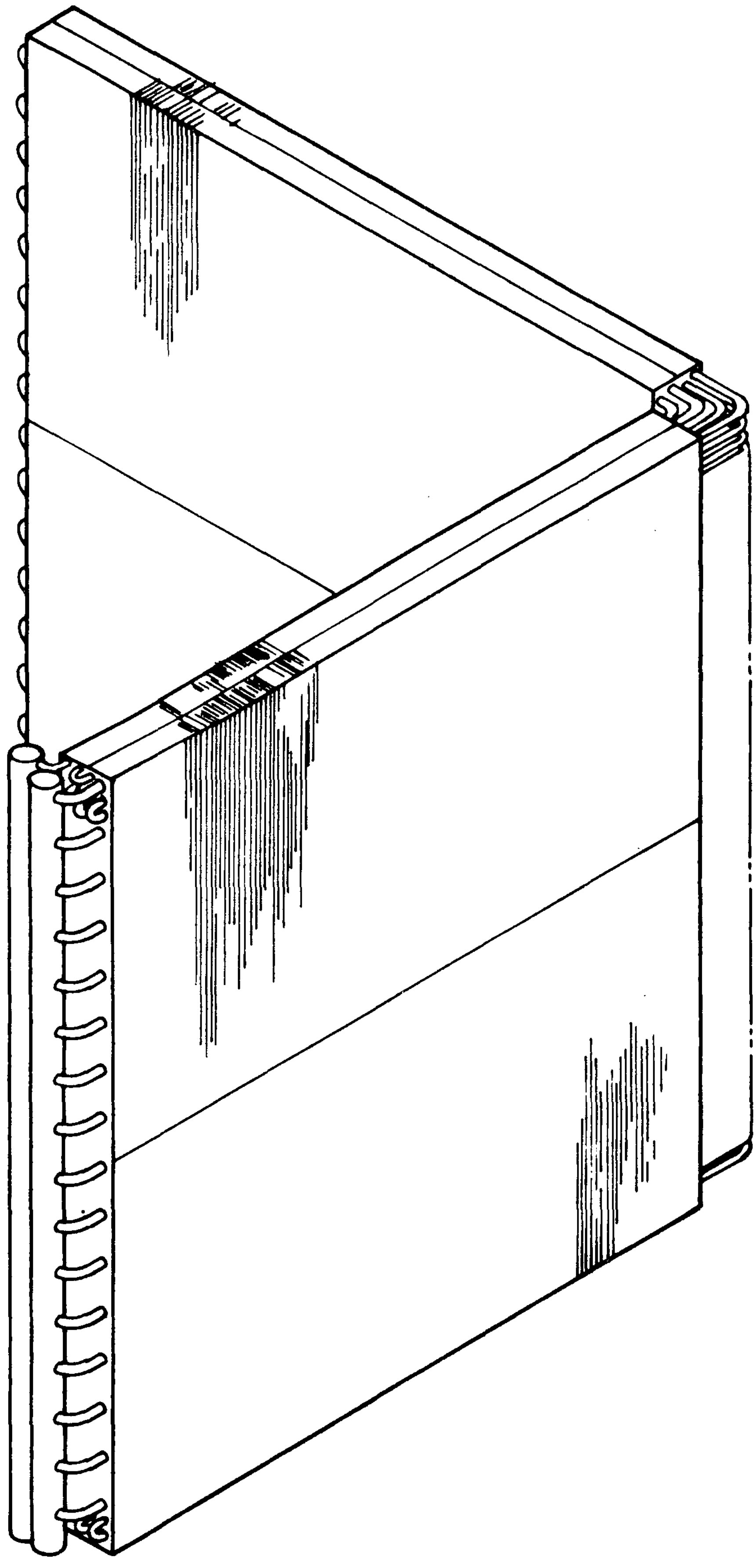


Figure 2-7 Outdoor Coil for Field Trial Units

TABLE 2-4

OUTDOOR COIL PERFORMANCE UNDER HEAT PUMP COOLING MODE CONDITIONS

35% GLYCOL/WATER

Coolant Flow Rate, lb/hr	5073	4504	3925	3987	4012
Temperature In, °F	131.3	132.4	131.6	130.8	129.7
Temperature Out, °F	106.9	106.7	104.8	104.8	105.8
Pressure Drop, Psi	4.5	3.9	3.0	3.3	3.4
Air Temperature In, °F	95.2	96.0	95.5	96.1	96.5
Air Temperature Out, °F	115.5	115.0	112.4	113.0	113.0
Pressure Drop, In W.C.	.13	.13	.15	.16	.13
Calc. Air FLow Rate, CFM	5380	5380	5490	5410	5130
Heat Transferred, Btuh	110160	103020	93620	92260	85340

The outdoor coil performance thus met the design requirements, but the water-side pressure drop was somewhat higher than desired. The data were run at the original design conditions of 130°F coolant inlet temperature and 95°F air inlet temperature. The objective of a 105°F coolant outlet temperature was held. As would be expected, the coil capacity was a function of the coolant flow rate and inlet temperature. It was also dependent on the air flow rate, as shown by the final column. The results of the first column indicate that a 90,000 Btuh output could be obtained at a 125°F liquid inlet temperature by operating at a liquid flow rate of 5000 lb/hr.

Trial tests were also run later under winter conditions, but the source of the chilled glycol/water was not able to provide the full loads required. Data from those preliminary tests are listed in Table 2-5.

TABLE 2-5

OUTDOOR COIL PERFORMANCE UNDER HEAT PUMP HEATING MODE CONDITIONS

35% GLYCOL WATER

Coolant FLow Rate lb/hr	3934	3948	4915	4915
Temperature In, °F	35.5	35.6	37.6	37.4
Temperature Out, °F	43.9	44.2	44.0	43.9
Air Temperature In, °F	46.5	46.7	46.7	46.5
Air Temperature Out, °F	41.5	41.3	41.5	41.5
Calc. Air Flow Rate, CFM	4660	4415	4250	4490
Heat Transferred, Btuh	27360	28010	25950	26357

On first consideration, these test results appear to contradict Table 2-4 because increasing the liquid flow rate from the 3,900 lb/hr rate of the first two tests to the 4,900 lb/hr of the last two tests appears to have resulted in a drop of capacity. However, coil capacity was determined by the capacity of the chilled water source. As operated, the capacity dropped when the increased flow rate reduced the temperature change in the antifreeze solution from about 8.5°F to about 6.5°F. Of greater importance were the temperature differentials at the inlet and outlet. At the outer surface of the coil (the coolant outlet and the air inlet location), the differential remained essentially a constant at the 2.6°F level over all four tests. That Δt is probably a minimum, or close to it, since the average change in the water temperature across each of the four rows of tubes was 1.5°F to 2.0°F. At the inner side of the coil (the coolant inlet and air outlet side), the Δt dropped from close to 6.0°F to about 4.0°F when the coolant flow was increased. The overall Δt was thus reduced to match the lower load.

As opposed to the situation of the indoor coil, the outdoor coil is thus seen to be limited at its low temperature operation, rather than at the high temperature. The primary causes are the extremely low Δt's under which it operates. To reach the 36,000 Btuh level would require a major increase in air flow as well as a rise in the liquid flow rate. For use in the Field Trial units, that full capacity was not needed. As listed earlier, those units were operated derated. The outdoor coils performed satisfactorily there, but the highest air flows and liquid flows were applied.

3.0 ELECTRONIC CONTROLS

INTRODUCTION

The control system for the organic fluid absorption heat pump was to have the functions of operating the system winter and summer to match the building needs, and to make corrections or shut down the unit in case of malfunctions. Considerable redundancy was believed desirable in the malfunction controls for the first Field Trial and prototype units. Flexibility of the controls was also considered important in case adjustments to the control methods were found helpful in improving the cycling efficiency of the systems.

The control system was planned as a solid-state system as a means of ultimately reducing the service requirements of the controls. The work performed in the solid-state controls area can be divided into six categories: logic circuits, power circuits, operational sensors and interfaces, malfunction sensors, and control of electrical noise and voltage spikes.

Commercial components were used wherever possible. The system was developed in-house, because it affected and was affected very directly by the operation of the absorption unit. Also, a few special sensors were needed and were being developed as part of the program.

OBJECTIVES

The primary objective of this part of the controls program was to develop reliable operational and malfunction controls which would operate test and Field Trial units efficiently and safely. A second part of the objective was to lay the groundwork for the control systems of the forthcoming prototype units.

STATUS AT START OF PROJECT

The solid-state control systems design had progressed by simplification of the circuits, reduction of components by use of more powerful chips and assembly of fully-operational circuit boards. Promising results on bench tests had been obtained on the custom sensors being developed: the boiler level sensor, the draft sensor and the magnet decoupling sensor. However, the development was not complete, and problems remained. The thermistor-type draft sensors would be expensive to duplicate consistently, the five-volt DC to 220-volt AC interfaces were questionable, and the defrost system had not yet been proved out.

PROJECT INVESTIGATIONS AND DEVELOPMENTS

The report is divided into the major categories of the program. A flow chart of the relationship between the logic circuits, power circuits, sensors, interfaces, and operating loads is found in Figure 2-8. A general schematic diagram of the control circuit and its components is contained in Figure 2-9. The design intent had been for the control system to consist entirely of five-volt circuits up to the interface with the power circuits. However, some sensors and controls were not available in five-volt rating. It was therefore necessary to include 24-volt circuits in the system.

Logic Circuits

The logic circuits controlled the order in which devices are turned on and off, and processed signals from the thermostat and sensors on the unit. Diagrams are shown in Figures 2-8 and 2-9.

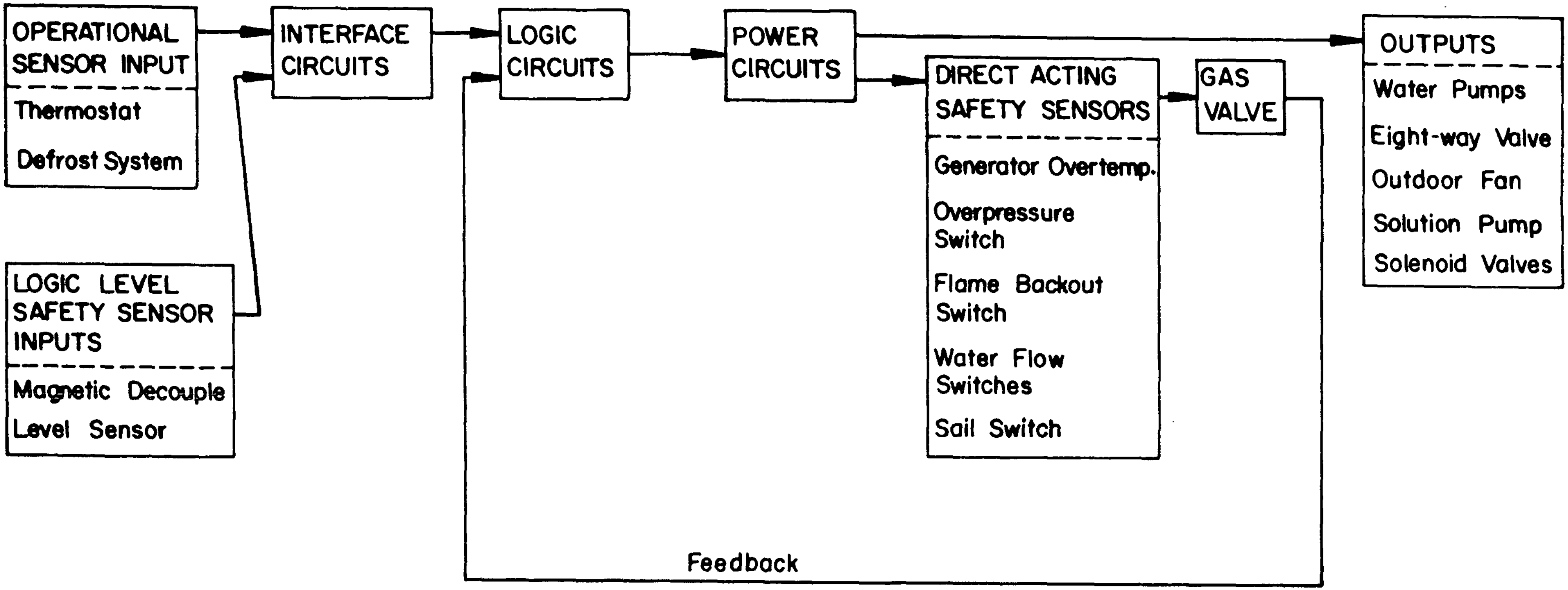


Figure 2-8 Flow Chart for Electronic Controls of Field Trial Units

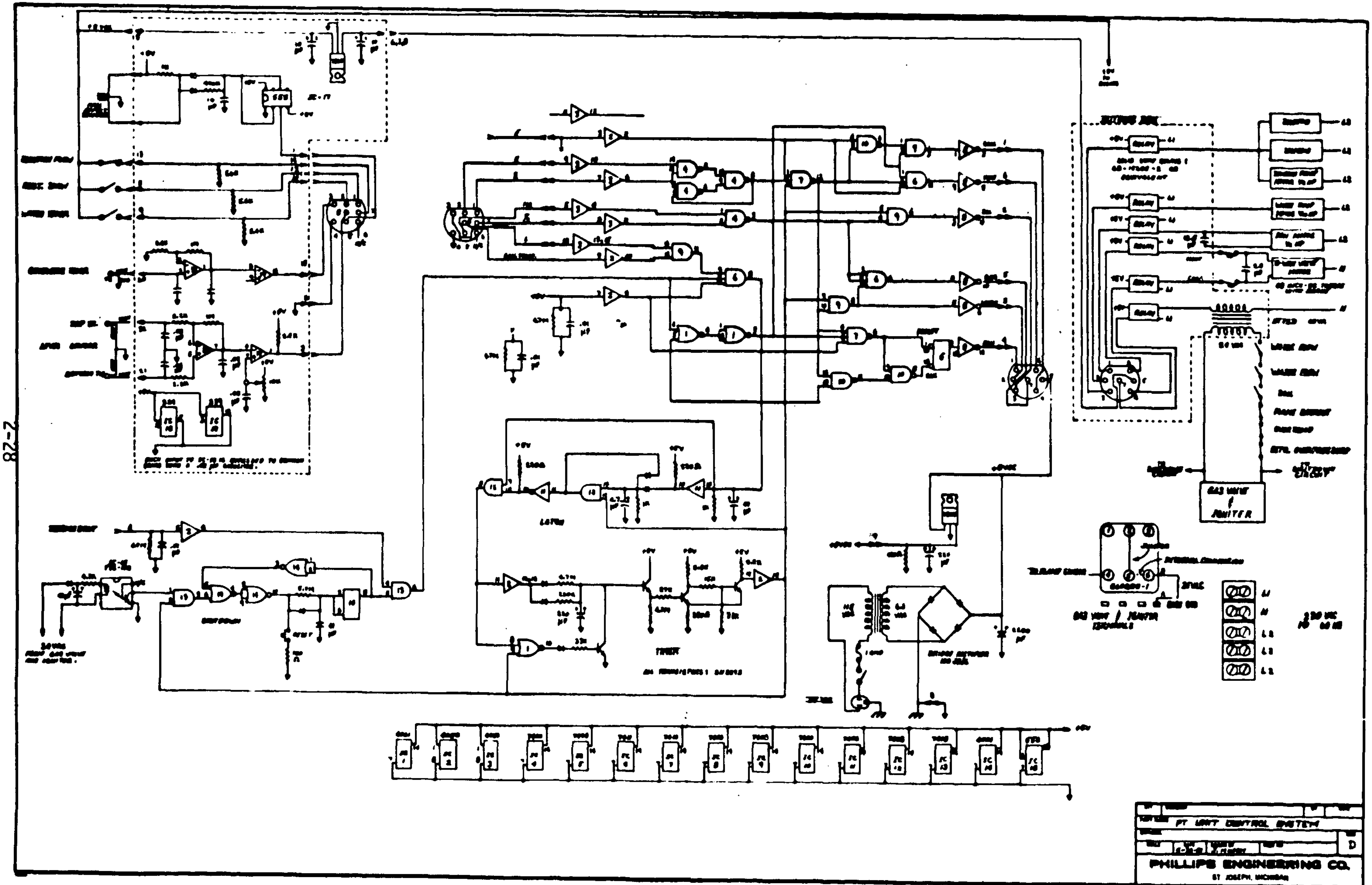


Figure 2-9   Schematic Diagram of Solid-State Control System

The logic control had the function of controlling the timing and sequencing of the unit start-up and shut-down and of defrost operation. The actuation of the house thermostat, in either heating or cooling, started the "on" cycle sequence. Commonly, the fan alone was first turned on. This was to ventilate the system and establish a flue draft before the burner was turned on. Fan operation then closed the sail switch or actuated the draft sensor. After 30-40 seconds, other operational components, the solution pump, water pumps and solenoid valves, were turned on. The setting of the eight-way valve, or a check on its setting, occurred at this time also, but setting it when the fan came on was also tried. If none of the sensors detected problems, the gas valve was then turned on. The unit than operated normally until the house thermostat called for shut-off. On shut-off, the gas valve shut off first. The controls then followed a spin-down sequence that was varied to improve the cycling efficiency and reduce power consumption. The timing of the shut-off of the solenoid valves, the fan, the solution pumps and the water pumps were varied individually to make the best use of the energy input to the unit during the "on" period. As will be reviewed in the Field Trial section, the optimum spin-downs were found to vary with the outdoor temperatures and indoor loads. For optimal annual efficiency, the spin-down sequence will probably have to change with operating conditions. A two-level operation may be found sufficient, but the greater flexibility that a micro processor would provide might improve seasonal performance enough to make that the lowest-cost control.

The logic circuits for the Field Trial units consisted of integrated circuit devices mounted on two printed circuit boards, one indoors and one outdoors. With some modifications, these boards were used on all six units. First-generation designs of the logic system processed all signals through the logic circuitry. Later, in order to improve the reliability of some of the safety system sensors, six sensors were placed as switches directly in the 24-volt AC gas valve circuit. These sensors were the boiler overtemperature switch, overpressure switch, flame backout switch, sail switch, and water flow switches. The change of the switches from low current carrying contacts in five-volt DC circuits to contacts carrying the 24-volt AC current increased reliability. Reliability was also increased in that the switches directly interrupt the gas valve circuit current rather than providing logic level inputs only.

An additional logic circuit was added to reduce the effects of interruption by short-time transients. This circuit required the unit, if interrupted during operation, to go through the shut-down portion of the cycle before restarting.

When the six safety sensors were put directly in the gas valve power line, the logic system lost knowledge of their status. It became possible for one of them to shut the gas off, leaving the other components controlled by the logic circuit running, i.e. fan pumps and motors. A circuit, which sensed the presence of 24-volts AC at the gas valve, was therefore designed and added to the logic circuitry of two units. If the thermostat was calling for operation and there was no power at the gas valve (implying one of the sensors was open-circuited), then this circuit would shut the power off the unit until the problem could be repaired.

The control system was divided between the two circuit boards, one located outdoors in the unit and the other inside the house. The indoor board contained most of the TTL circuits. A number of the feedback diagnostics were thus designed for checking indoors with a voltmeter. For further information concerning the control system and its operation in the Field Trial units, please see the section entitled Field Test Units - Design, Construction and Testing.

Power Circuits

The power circuits were designed to accept low-level inputs from the logic circuits and to switch power to the electrical devices in the units. Those devices were, for the most part, electric motors; such as, the motors for the fan, solution pump, water pumps and eight-way valve. However, other devices, such as the level sensor heater and the gas valve, were also switched by the power circuits.

The power switching devices were first designed to be triacs. The first Field Trial units used five-volt light sources and photo transistors to provide the optical isolation between the logic and the triac power circuits. Problems were experienced on the those units, including triac failures on lightly loaded circuits. For the last four units, a design change was made. The optical isolator circuits and the triacs were replaced by commercial solid-state relays with built-in opto-isolation circuits. For the four units using this design, only two failures were encountered. One relay was found to be defective after a storm accompanied by a power outage. The cause of the relay failure may have been lightning or a voltage transient in the power line. A second relay failure was due to loose connections at a terminal, which resulted in overheating at that point.

The reliability of the power circuits was thus greatly increased through the use of solid-state relays. The change to solid-state relays resulted in a significant reduction in circuit components and in the number of solder connections. From a service standpoint, the new design would also be much easier to trouble-shoot and repair in the field. Failures in the original circuits had usually required bench work by an electronics technician rather than field repair.

Malfunction Sensors and Interfaces

The malfunction sensor portion of the control system was planned for high redundancy. To prevent serious problems during the life test and field test program, it was proposed to provide the sensors and redundancy needed to sense any abnormal operating conditions when they began. The safety system included nine sensors or controls. They consisted of:

Overtemperature Sensor	- To shut off the gas if the boiler overheated.
Pressure Relief Valve	- As required by code, to vent off the vapor if excessive pressure developed.
Overpressure Sensor	- To shut off the unit before such excessive pressures were reached.
Cooling Water Flow Switch	- To assure coolant flow while the gas was on.
Chilled Water Flow Switch	- To assure a load for the evaporator before the unit started operating.
Draft Sensor or Sail Switch	- To assure proper draft for the burner before turning on the gas.
Flame Backout Sensor	- To protect from any plugging of the flue or combustion air passages if a sail switch was used.
Magnet Decouple Control	- To turn off the motor if a decouple occurs, allow the magnets to recouple and restart the motor.
Generator Level Sensor	- To shut off the unit in case of low boiler level, due to leak or circulation failure.

As is evident, a number of these sensors served as backups. The sensors and their operation are described in greater detail in the section on Field Trial Unit Design, Construction and Testing. As listed in the Logic portion of this section, three of these sensors fed into the logic system, while the other six acted directly on the gas valve. Reliability was the major criteria for deciding how best to interface sensors with the controls. Most sensors were commercial products, but three were specially developed for this heat pump. They were the magnet decouple sensor and circuit, the boiler level sensor, and the flue draft sensor.

The decoupling sensor started as a curved iron armature piece which actuated a microswitch when the magnetic field changed on decoupling. It was difficult to set correctly in the field in case of motor changes, or other servicing. That system was replaced by a reed switch which supplied a greater built-in reliability. However, locating it properly remained a problem, and it was not completely consistent. A Hall-effect sensor was then applied. It has been found significantly more successful. As the system is now all solid-state, there is no potential of mechanical or switch contact failure. The sensitivity of the Hall-effect device has eliminated much of the problem in sensor location and adjustment. A change in its mounting also eliminated a possible overheating problem.

The boiler level sensor developed for the Field Trial units was based on the large difference between the heat transfer rate to the boiling liquid and that to the vapor. Two thermocouples were placed in wells in the generator and heat provided to one of the wells by an electric heater element. When both were submerged, the temperatures were close together. At a low-level condition, the temperature difference measured by the thermocouples increased, and a fault signal was produced by the electronic circuitry.

Other than response not being quite as fast as desired, the level sensors performed well on bench tests. The sensors installed on the Field Trial units caused problems, however, primarily due to the 50-watt electric heaters. One heater element shorted to its well, burning a hole which caused a leak in the generator. Another element failed in open circuit, making the sensor inoperative.

Continued experimental work resulted in a similar sensor which used temperature differentials in the generator without the addition of heat from an electric heater. The new sensor located two thermocouples at different levels in one well, with a differential temperature between them of less than 5°F at normal levels, but of 15°F or more when the level was low. The only disadvantage was that the sensor worked only when the gas was on; it did not indicate levels on start-up. However, the possibility of an electrical short to the generator was eliminated, and the electrical power consumption was reduced by 50 watts. The new sensor circuits were installed on four of the Field Trial units. Low levels were detected on three occassions, and there were no known instances of level sensor failure.

Efforts to develop a good, reliable electronics flue draft sensor at an acceptable cost were not successful. Thermistor air flow sensors operated well during operation at fixed temperature, but components which would sense flow equally from ambient to flue gas temperatures were not found. Solid-state pressure sensors may become useful for this purpose in the future, but, at the present, do not provide the sensitivity required. A diaphragm-type pressure switch appears to be suitable from a sensitivity viewpoint, but the price is somewhat high. As a result, a sail switch to sense fan operation was used on the Field Trial units. A back-flame sensor was used with the sail switch to detect plugging of the flue system by soot or foreign materials. Although this combination had had use on gas air conditioners in production, the sail switch was found unsuitable for snow and ice conditions in the winter. Development of a direct all-weather flue draft sensor is thus indicated as a necessary future development.

Defrost Control

Defrost control on the Field Trial units was accomplished using a Robert Shaw series DS-10 demand defrost control switch. The DS-10 switch is pressure-initated, temperature-terminated. Defrost initation was set to require an air pressure drop of 0.35 inches of water across the coil and a bulb temperature at the outlet of the coil below 32°F. Defrost termination occurred when the pressure drop decreased and the bulb temperature at the coil outlet increased to a 55°F setting. In this application, a temperature sensor in the cooling water storage tank was used to make the logic system unresponsive to the DS-10 switch if the storage tank water temperature was below 95°F. If the water temperature was over 95°, the defrost control could put the Field Trial unit into a defrost cycle.

With the Field Trial unit in the heating mode and storage tank water over 95°F, initiation by the DS-10 defrost switch caused the logic control to do the following:

1. Turn off the exhaust fan and circulating chilled water pump.
2. Reverse the eight-way valve (to cooling mode).
3. Turn off the gas burner.

Warm water from the storage tank and unit was then circuited through the outdoor coil. The defrost normally occurred in no more than three minutes, terminating when the water/glycol flowing through the coil cooled to no less than 55°F.

The defrost process had the limitation that only heat stored in the tank and unit was available for defrosting. That was sufficient for normal temperature defrosts. In storms in which the coils were plugged by driving snow at 0°F the cycle was not terminated, evidently because the DS-10 bulb did not get up to 55°F. For such conditions, the future control should include a solid-state temperature sensor and logic to terminate the defrost cycle, return to normal operation, and then repeat the defrost after the water has been reheated. Other potentials for improving the reliability of demand defrost controls were also investigated.

Other Controls

Since the integrated circuit logic levels were five-volt, and most house thermostats required 24-volts for proper operation, an interface was designed for the house thermostat and five-volt logic circuits in the units. This interface allowed the house thermostat to be used unchanged. The heat and cool anticipator circuits in the thermostats then operated normally at 24-volts.

In the residential installations, the control of the indoor blower was made independent of other systems. Blower operation was controlled by the temperature of the water flowing from the indoor coil back to the unit. Commercial aquastats were used for the two installations. The water temperatures at which the blower would turn on and turn off in both heating and cooling modes were set in relation to the house needs and provision of good comfort levels. In cooling mode, that adjustment assured good dehumidification. In winter, use cold drafts could be eliminated. Very satisfactory operation was obtained from the comfort standpoint. Further investigation of the blower control will be important because the running time and on/off schedule of the blower affect the absorption system cycling efficiency as well as the costs of operating the blower.

Electrical Noise and Voltage Spikes

A considerable difficulty was experienced with electrical noise and voltage spikes coming in over the power lines during the development of the controls and the logic and power circuits. The power line problem was worse than normal by the electrical utility company standards, so they replaced incoming power lines and re-routed ground wires. However, since the quality of the electrical power to field installations will not always be controlled, the problems encountered presented an opportunity to develop the electric controls for the most adverse circumstances. Filters were added to inputs of logic signals to reduce the electrical noise. Better isolation and transient suppression were obtained by the use of solid-state relays. Those modifications essentially eliminated the reliability problems of the electric controls even with the remaining noise caused by welders in the shop.

3.0 SOLUTION PUMPS

SOLUTION PUMPS

INTRODUCTION

Historically, the solution pump has been the absorption system component that has been the most difficult to develop to the reliability, performance, operating life, sound level and cost needed in a residential system. Its function is to pump fluids essentially at their boiling point, commonly also pumping vapor, without vapor locking and without lubricants. It must be hermetically sealed and require no maintenance or service over a minimum heat pump life of 15 years. The fluids may be chemically aggressive, and may carry abrasive suspended particles. The pump must also be efficient, quiet, small and of low cost. For year-around heat pump operation using outdoor air as the heat source and sink, the solution pump must also be able to operate over a range of solution flow rates.

The development of a pump for the R-133a/ETFE organic fluids of this project had been expected to be less difficult than for ammonia/water. However, many of the requirements were found to be as difficult to achieve as with ammonia/water. Advantages did reside in the smaller pressure rise of R-133a/ETFE solutions and the higher volumetric flow required for those solutions. The flow rates were five to seven times as high as ammonia/water, and the pressure rise one fifth that of ammonia/water. Thus, the theoretical pumping power was of the same magnitude as ammonia water, but the flow rate and pressure rise fell in ranges closer to common pump applications.

In addition to meeting the operating requirements, it was desired to run the pump at the 3450-3500 rpm of two-pole motors to reduce the size and cost of both pump and motor. Operation at that speed would also reduce the torque, and with it, the size and cost of the magnets used for hermetic drive.

Essentially all pump types had been considered in early investigations. The requirements for maximum efficiency, relatively low flow and moderately high head pointed toward displacement pumps. However, displacement pumps are generally not well suited to zero net positive suction head or to capacity modulation, except by changes in speed. Except for diaphram pumps, they are also more subject to wear than dynamic pumps, and their discharge pulsations can be a problem in absorption systems.

Dynamic pumps are generally less efficient than displacement pumps, especially so at the low specific speeds and flow rates of this application. Centrifugal pump efficiency would be expected to be of the order of 10% at the flow and pressure rise specifications. On the other hand, dynamic pumps have less wear, are generally well suited to varying flow rates, and can be designed for some vapor intake.

Since neither category of pump specifically matched the application, a more detailed analysis of individual pumps had led to investigations of two pumps, one from each of the two categories. Of the displacement pumps, vane pumps were concluded to be preferable to piston pumps or gear pumps. Compared to piston pumps, vane pumps were expected to be less expensive, more reliable, better suited to designs for low NPSH, and to produce less pulsation in the third stream. In relation to gear pumps they had the advantage of being capable of being designed for variable flow, and were perhaps better suited to the materials requirements and wear problems.

Of the dynamic pumps, the regenerative-turbine pump was chosen because its performance is much better at the relatively low flow rates and moderately high heads of the R-133a/ETFE fluids than that of centrifugal or axial pumps. At the specific speed of near 200, its efficiency should be at least double that of centrifugal pumps. A major disadvantage was that commercial regenerative turbine pumps are often subject to wear. In this application, it was believed that the pump could be designed for minimum wear and that the R133a/ETFE fluids could be maintained free of particles.

Past developments had thus centered on vane pumps and regenerative turbine pumps. Neither type had been developed completely; both exhibited operating advantages, and potentials for meeting the requirements. In this project, a choice was to be made between the two pumps. The decision was to be made on the basis of usability on the Field Trial units as well as suitability for development for use in prototype and production units.

OBJECTIVES

The objectives of the pump investigations during this period were to decide between the vane pump and the regenerative turbine pump, and to develop the pump chosen for use on the Field Trial units and also for subsequent prototype or production designs. The target performance had been established as 1800 lb/hr (3.5 gpm) flow at a 50 psi pressure rise, with a power input of no more than 300 watts to the drive motor. The 300 watts were to include motor inefficiencies and magnetic drive losses. Operating life was to be a minimum of 50,000 hours and 200,000 on/off cycles. Other requirements included hermetic operation, the ability to run unharmed when liquid intake dropped to near zero, and to operate without vapor locking and at a low noise level. Since it was desired to operate the pump by means of a magnetic coupling, operation without magnet decoupling was also important.

STATUS AT BEGINNING OF PROGRAM

Earlier versions of the vane pump and the regenerative turbine pump had been designed for horizontal operation at 3,500 rpm with magnetic drive from an external motor. (2) They were constructed of aluminum, with Teflon-impregnated, hard-anodized aluminum bearing surfaces running against graphite. They were of almost identical size, and both appeared likely to be of lower cost than other concepts. The wattage required by the early regenerative turbine pumps had been of the order of 700 watts, and in 1978 the possibility of achieving the 300 watt objective appeared unlikely. During 1979 the development work had therefore been centered almost entirely on the vane pump.

The past development of the regenerative turbine pump had brought it to good operation, other than wattage, and it was serving as the standard experimental pump for operation of the absorption test units. Four life test regenerative turbine pumps had also been built and were being run, two continuously and two on start-and-stop cycles.

During 1979, the development of vane pumps had included efficiency improvement investigations, life testing relative to materials of construction, and testing of the pumps on operating absorption units. A group of efficiency improvement studies had been carried out in parallel paths. Those investigations had been individually successful, but the developments had not been combined into a single operating pump when this program was initiated. In one of the investigations, liquid drag on the internal drive magnet had been found to be of the order of 135 watts. Operation of an improved pump, with the magnet in a vapor space, had subsequently resulted in a power input to the drive motor of 345 watts when operating at the target 1800 lb/hr and 50 psi pressure rise. In a separate study, the replacement of the hard-anodized, Teflon-impregnated cylinder by a Macor glass cylinder (to simulate a ceramic cylinder) had reduced the power requirements by 80 watts. The replacement of the .7 hp test motor with a high efficiency 1/2 hp motor was expected to provide further improvement. Combination of those developments was expected to reduce the motor input power to below the 300 watt objective.

In the life test studies, four vane pumps had been put on life test in 1978. Two failures had occurred early in 1979, but the other two pumps, and two replacements of improved design, were in operation at the beginning of this project. One pump had operated 9,600 hours, a second 6,000 hours, and the third and fourth 5,100 and 4,700 hours, respectively. All pumps had graphite vanes, three of them had hard-anodized Teflon-impregnated aluminum cylinders and the fourth, a Macor glass cylinder. Macor glass is a machineable glass. It had been used to simulate ceramic cylinders in the pump. The experience with it, together with investigation of the operation and life of graphite/ceramic mechanical seals, had led to the conclusion that vane pumps made with graphite and ceramic materials at the wear surfaces should operate for 50,000 hours or more with organic fluids. The hard-anodized aluminum cylinders were being run in attempts to qualify lower cost materials.

The third part of the vane pump investigation consisted of testing the vane pumps on operating absorption units. The development work had been done on bench test fixtures pumping R-133a/ETFE solutions. In the unit operation a problem had soon been encountered. Pulsations in the pump discharge, though small, and not significant in the bench testing, had set up standing waves in the long lengths of tubing that made up the heat exchanger passages to the generator. High noise levels and vibrations occurred. The pulsations were reduced greatly by pump modifications and by installation of a muffler in the discharge line, but at the beginning of this project, the levels of vibration and noise remained excessive.

While the vane pump development was underway, work on the regenerative turbine pump had been only secondary in nature. The four pumps on life tests had remained in operation. No failures or difficulties had been encountered. Two of the four pumps were on continuous-run operation and had reached 21,000 hours of operation. The other two were on cycling operation. They had operated 12,000 hours and logged 140,000 cycles with no difficulties. Regenerative turbine pumps had also been in use on the absorption test units. Operating problems encountered had been solved as part of operating the units. Magnet decoupling and vapor lock, the most common start-up difficulties, had been reduced by strengthening the magnets and by improving the intake conditions at the pump. The high wattage appeared to be the only remaining important disadvantage of the regenerative turbine pump.

CHOICE OF PUMP

The pump program for this contract began by continuation of the vane pump testing and development from the preceding project. Reanalyses of the vane pump and regenerative turbine pump concepts were started immediately so that a decision might be made between the two pump systems.

In the vane pump program, the unit testing uncovered another difficulty. It involved the control of the liquid flow to the pump intake. Because the vane pump is a positive displacement pump, it had been necessary to prevent, or to hold to a minimum, the entry of vapor to the pump. (The collapse of the vapor bubbles in the pump chamber as the pressure rose would produce cavitation type shocks that could be expected to harm the pump). To prevent entry of vapor to the pump, a float valve had been developed as a weak liquid flow control to maintain a constant level of liquid above the pump intake. When the unit vibrations had been reduced sufficiently for normal testing, it was found that the operation of the float valve and the vane pump did not coordinate properly. The float valve tended to hunt, rather than modulate the solution flow smoothly. A significant time lapse was found between a change of the flow to the top of the absorber and the effect of that change on the solution level at the bottom of the absorber. The result was a somewhat uncontrolled rising and falling of the solution level above the pump, with excessive overshoot at times. Increasing the sensitivity of the valve and providing anticipation reduced the problem. Other modifications of the valve so that it did not shut down completely, and damping of the valve swings also helped. However those corrections were not a complete solution. It remained uncertain whether the time lapse problem and the vibration problems could be solved within the time available for installation on the Field Trial units.

Meanwhile, the turbine regenerative pump had been under consideration. From the standpoint of the schedule for construction of the Field Trial units it was clearly the best choice. The decision also involved evaluating the pumps relative to possible production in the future. The potential pump efficiency was a major factor and had been under investigation. The 650 to 700 watt power input to the horizontal shaft pump motors included a magnet drag loss, similar to the 135 watt loss that had been measured in the vane pump development. A similar savings could be expected on the turbine pumps by designing for vertical shaft operation with the magnet in the vapor space. Replacing of the 0.7 hp test motor by a high efficiency motor of under 1/2 hp to match the pump load might be expected to reduce the motor losses by another 50 to 75 watts. Those two energy savings should therefore drop the power to the motor to the 450-500 watt range. Pump modifications to improve efficiency also appeared possible. The 300 watt performance of the vane pump did not appear within reach, but a 400 watt level seemed a possibility. At a maximum operation of 3000 hours per year, that 100 watt increase in power over the vane pump objective would add 300 kw per year, which would increase the annual operating costs by $12 to $25 at the 1979 electric rates of 4 cents to 8 cents per kwh.

The regenerative turbine pump had in its favor the trouble-free life test operation for 21,000 hours versus several failures of vane pumps within a few thousand hours, and a maximum life test experience of 9600 hours at the time. It did not have the unresolved pulsing flow and flow control problems of the vane pump and was much quieter in operation. The overall analysis then led to the decision that the reliability and operational advantages of the regenerative turbine pump would be worth more to the customer than the increase in annual operating costs. The choice of pump was therefore made in favor of the regenerative turbine type, with the requirement that the power input be brought down to 400 watts.

The development program on vane pumps was stopped, but the life tests were continued. Development of the regenerative turbine pumps was immediately reinstituted.

REGENERATIVE TURBINE PUMP

Description and Operation

The regenerative turbine pump is a dynamic pump, but it differs significantly from the usual centrifugal or axial flow pumps. Figures 3-1 and

3-2 are illustrations of the pump and its operation. As compared to centrifugal pumps, the regenerative turbine pump is a low-volume, high-head pump. Those characteristics are of special importance in this application.

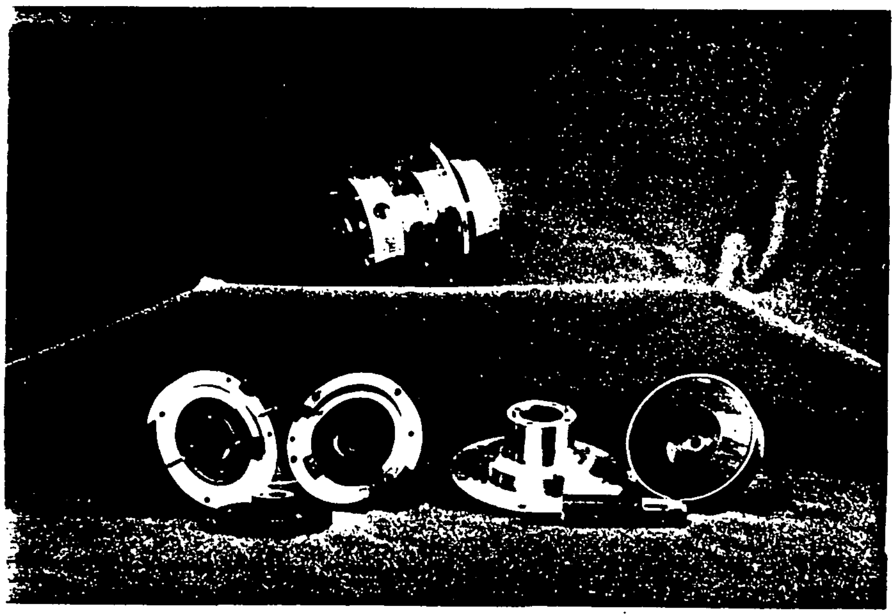


FIGURE 3-1

REGENERATIVE TURBINE PUMP

As indicated in the picture and drawings, the pump consisted of an impeller wheel with a bladed periphery, the blades forming buckets on both sides of the wheel. There is a passageway or raceway formed in each casing alongside the set of blades. The liquid to be pumped enters the buckets and the raceways at the periphery, proceeds through the raceways and the buckets around the circumference and exits close to the intake. Each raceway is interrupted by a stripper which closes off the raceway, separating and defining the intake and discharge. Except for the stripper, the raceway is an open path from intake to discharge. As indicated in Figure 3-2, the liquid is given a helical path

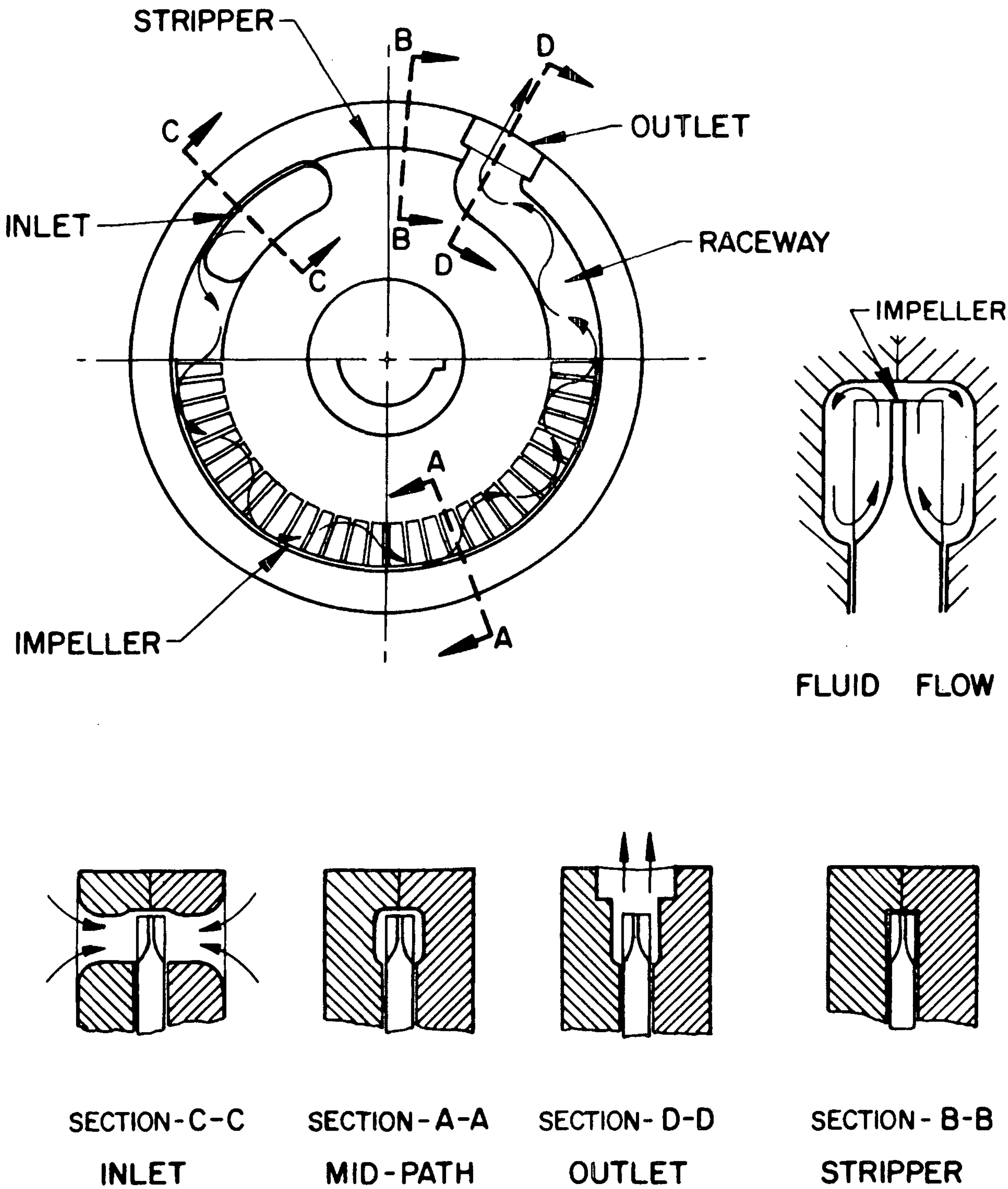


Figure 3-2 Regenerative Turbine Pump Operation

through the buckets and the raceway. Liquid from the raceway enters the impeller buckets at their roots and is accelerated both radially and tangentially, by the blades. The energy thus imparted to the liquid in the buckets is in turn transferred to the stream in the raceway, accelerating it and increasing its pressure as it travels the circular path from intake to discharge. The pump performance is determined by the many factors that affect the transfer of energy from the impeller to the raceway stream.

To meet the schedule of the Field Trial units, the six pumps for those units had to be built to the horizontal design and thus would include the magnet drag losses. The development to the target efficiencies was carried out as a separate, but concurrent program.

EFFICIENCY INVESTIGATIONS

Efficiency studies took three paths. Two of the reductions in input power were expected to result from the elimination of the liquid drag on the internal magnet and from the use of an efficient, smaller motor to drive the external magnet. The third part of the improvement was expected to come in smaller increments from detail design improvements to the pumps. The primary step, to locate the magnet in the vapor space, required the design and construction of a vertical shaft pump with magnet in a vapor space. That program was started immediately.

A search was made for an existing high-efficiency 0.4 hp motor but was not successful. Emerson Motors was then contacted and agreed to design and supply two-pole, permanent-split-capacitor motors of high efficiency to fit the application.

For the investigation of pump efficiency parameters, two regenerative turbine pumps of the horizontal configuration, which had been partly constructed in the previous program, were completed for use as test vehicles. Representative performance of the two horizontal pumps, as built, was 1700 lb/hr flow at 50 psi pressure rise at a power input of 660 watts with a 0.7 hp test motor. The flow rate was below the target 1800 lb/hr, and the wattage was 65% above the goal.

No quantitative theory of the operation or optimization of regenerative turbine pumps was found during a limited literature search. Subsequently, engineers in that field have stated that there is no published design information, and that regenerative turbine pumps remain an art rather than a science. Two manufacturers were contacted, but did not wish to develop these smaller pumps to the required specifications. The in-house investigation of the pump design parameters was based partly on qualitative analysis of flow fundamentals in the various sections of the pump. Parameters investigated were;

Intake port size and shape
Location of intake port
Raceway cross sections
Impeller diameter
Dimensions and number of buckets in the impeller
Pressure rise along the length of the raceway
Thrust bearing location and frictional losses
Taper of raceway
Offsetting of buckets, from one side to the other
Magnet size and strength
Number of poles on magnets
Axial vs. radial magnets

Most of these design variables had some effect on the performance of the pump. The trials of axial vs. radial magnets were done early in the program while overall wattages were still high. Estimates of prices from a magnet manufacturer at that time were that in production, radial magnets were likely to be ten to twenty times the cost of axial magnets, which are low-priced due to very high production. When the wattage reduction by use of radial magnets was found to be small relative to the total improvement required at that time, axial magnets were chosen. As the pump efficiency has improved, the energy losses in the thrust bearings needed for axial magnets has become more significant. The choice of magnet type should therefore probably be reconsidered in future work.

The pump improvement studies were first made on the horizontal pumps, but were shifted to the vertical shaft pump when it was completed; the elimination of magnet drag losses made the vertical pump a more sensitive test bed for investigation of changes to the impeller, raceways, etc. The pump component designs were adjusted, often by trial and error, until the required performance was obtained. The most important efficiency gains resulted from the following developments:

Location of the magnet in the vapor space
Use of a .4 hp motor of 70% efficiency
Low friction thrust bearing
Optimizing impeller diameter and bucket dimensions
Tapered raceway of optimum cross section

Improvements made to the pumps generally resulted in higher solution flows, along with the efficiency gains. Thus, as gains were made it became necessary to develop smaller capacity wheel and raceway systems to reduce the capacity and wattage into the target range. The pump efficiency requirement was first reached at flows well above the objective, using vertical shaft pump #T-1-7 with a special Emerson 1/2 hp permanent split capacitor motor. Table 3-1 lists that performance.

TABLE 3-1

VERTICAL SHAFT PUMP PERFORMANCE

PRESSURE RISE	FLOW RATE LB/HR	POWER TO MOTOR, WATTS	OVERALL EFFICIENCY, %
50 psi	2375	456	23.0
50 psi	2375	442	23.7

The pump performance objective of 1800 lb/hr of 35% R133a/ETFE solution at 50 psi and 400 watts corresponds to an overall efficiency of the pump-motor-drive system of 19%. As can be seen in Table 3-1, the pump performance at the excess flow rate exceeded the target efficiency by 20-25%.

In the tests of Table 3-1, the peak efficiency of the pump system occurred at 50 psi pressure rise. The 50 psi, 1800 lb/hr, 400 watt objective had been set as a rating specification, but heat pumps must also operate at conditions beyond ratings, and to higher pressures. It had been planned that the pump should reach its maximum efficiencies at 60-70 psi, while still meeting the rating specification. After the above results, development was continued to optimize and balance the major parameters against each other to meet the efficiency at the rating flow rate, while also moving the point of peak efficiency to the 60-70 psi range.

Adjustments to the impeller wheel, the raceways and the location of the thrust bearing gradually produced the improvements required. The data of Table 3-2 are from tests on three similar pumps. Each test represents a variation in one or more of the pump design variables.

TABLE 3-2

PERFORMANCE OF SOLUTION PUMPS

PUMP NUMBER	PRESSURE RISE, PSI	FLOW LB/HR	POWER WATTS	EFFICIENCY %	PRESSURE RISE AT PEAK EFFICIENCY, PSI
T-1-7 AC	50	1815	395	20.3	50
	50	1870	397	20.8	50
	50	1840	397	20.5	58
T-1-7 AF	50	1800	397	20.0	63
	50	1800	390	20.4	67
	50	1825	369	21.8	61
T-1-16	50	1825	410	19.7	65
	50	1887	380	21.9	60

The pumps of Table 3-2 all provided flow at, or slightly above, the objective and exceeded the efficiency requirements. Pumps T-1-7 AF and T-1-16 also had the peak efficiencies within the 60-70 psi range.

The pump tests were generally run at room temperature with the solution at about 75°F. Because in operation the pump will usually operate at 105°-110°F, tests were also made at higher temperatures on pump No. T-1-16. The following Table lists the performance at room temperature and at higher temperatures with three different thrust bearings.

TABLE 3-3

EFFECT OF TEMPERATURE ON PUMP PERFORMANCE

PUMP	TEMPERATURE °F	PRESSURE RISE, PSI	FLOW LB/HR	POWER WATTS	EFFICIENCY %	PRESSURE RISE AT PEAK EFF. PSI
T-1-16	75	50	1885	415	20.1	61
	130	50	1820	400	20.9	--
	70	50	1880	380	21.8	65
	120	50	1795	379	21.7	66
	70	50	1880	375	22.1	65
	117	50	1800	361	22.8	60

The effects of operation at the higher temperatures were to reduce flow rates and wattages, but generally increase efficiencies. These effects are presumably due to the reduced viscosity and density of the solution. The pump performances did not drop below the objectives.

As previously indicated, the efficiencies listed incorporate the motor and drive losses. Correcting for motor efficiency and drive losses results in an efficiency for the pump alone in the 33% range at the target conditions. The peak efficiencies would be slightly higher. Figure 3-3, which represents the second test in Table 3-3, shows the pump characteristics of this type of pump.

The pump development program thus achieved the project efficiency objectives. It also provided indications that further efficiency improvements could be made. The optimum design of the impeller and raceways had not been reached. Gains in these components would then allow reduction of the magnet size, reducing thrust bearing losses. (If radial magnets were used, the thrust bearing loss could be essentially eliminated.) Those improvements would, in turn, allow the use of a smaller motor with lower motor losses. The compounding of such potential gains provides reason to believe that the original displacement pump goal of 300 watts may be attainable with turbine regenerative pumps.

PUMP LIFE TESTING

Due to concern about the wear characteristics of regenerative turbine pumps, four pumps had been placed on continuous-run operation in April and May 1977. They had been built for the R-21/ETFE pair of working fluids, which required a flow rate of 2.5 gpm. The purpose of the test was to obtain wear data prior to a field trial of operating units. It had been intended to

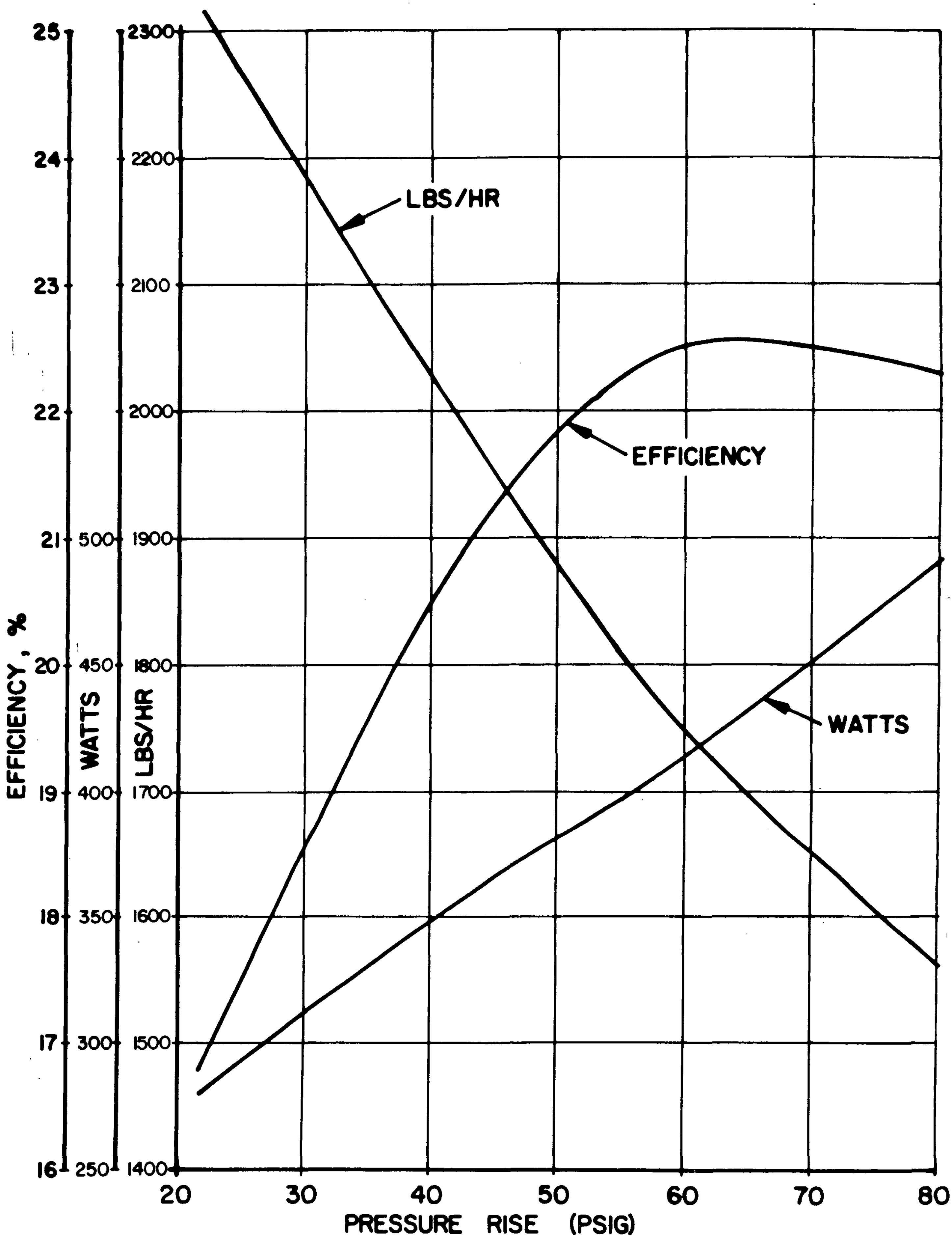


Figure 3-3 Performance Curves of Regenerative Turbine Pump

accumulate up to 2000 hours of pump operation as a basis for forecasting whether unit operation of 4000 to 6000 hours might be expected without excessive service problems. To begin accumulating hours, the pumps had been constructed and immediately put in operation without testing for performance. They were the same as other pumps that had performed above the 1300 lb/hr minimum flow rate required by the R-21/ETFE pump specification. No wear had been found on the four test pumps when inspected after a few hundred hours of operation, only polishing of the bearing surfaces. Plans were then made to continue the test for a longer period. After 2400 hours of satisfactory operation, further plans were again made. Two of the pumps, T-1-1 and T-1-2, were changed to on-and-off cycling operation, 3 3/4 minutes on and 3 3/4 minutes off. The objective was to try for 200,000 on-off cycles on those two pumps. The other two pumps were left on continuous run, with an objective of 50,000 hours of operation. Both objectives represent over 15 years of residential use. Table 3-4 lists the operating times on the pumps as of November 30, 1982.

TABLE 3-4

CUMULATIVE OPERATION TO NOVEMBER 30, 1982

PUMP NO.	CYCLING OPERATION		TOTAL HOURS
	CYCLES	HOURS	
T-1-1	232,810	17,911	29,618
T-1-2	243,551	18,055	33,463
T-1-3	-------	------	48,304
T-1-4	-------	------	47,171

After the 230,000-240,000 cycles shown were completed on pumps T-1-1 and T-1-2 at total times of 17,911 and 18,055 operating hours, those two pumps were put back on continuous-run operation and have continued in that mode. Pumps T-1-3 and T-1-4 have operated with no problems due to the pumps themselves.

The four pumps have all had shut-downs during the five and one-half years of operation, for reasons including test fixture failures, disassembly and inspection, power failures, magnet problems, etc. After the first power failures, the life test pumps were connected for manual restart in case of power failure, because, on unit test, vapor lock or magnet decoupling had been encountered as a result of rapid restarts during momentary power failures. (The decoupling control had not yet reached a satisfactory state.)

The pumps have operated at normal two-pole motor speed with intake pressure at about atmospheric pressure or below and discharge pressure at about 60 psi. They were disassembled and examined periodically. Run-in polishing of the bearing surfaces was noted in the early examinations. There were also a few scratch marks on the sides of the impeller wheels and on the graphite side plates, indicating the entrance of fine aluminum cuttings or other foreign particles. Subsequent examinations have shown little change or wear of the bearing surfaces. The four pumps were last disassembled and inspected during June and July 1980. Table 3-5 lists the operating times and performance of the pumps at that time.

TABLE 3-5

LIFE TEST PUMP STATUS AT 1980 INSPECTION

PUMP NUMBER	OPERATING HOURS	CYCLES	PRESSURE PSI	FLOW LB/HR	WATTAGE
T-1-1	15,128	189,548	50	1325	456
T-1-2	15,270	182,713	50	1470	530
T-1-3	26,500	-------	50	1430	530
T-1-4	26,542	-------	50	1325	494

The pump performances measured were normal for that pump design. As indicated earlier, these four pumps were not originally planned for a full-scale life test. The flow rates and performance shown in Table 3-5 are of the same magnitude and have the same dispersion as other 1976 individually hand-built pumps for R-21/ETFE. Whether there were small changes in the pumping rates during that first period cannot be determined. It is intended to continue operation to 50,000 hours on the continuous run pumps. At that time, they will again be disassembled, inspected and tested. A comparison of that performance against Table 3-5 will determine whether there has been any degradation over the second half of the life test and may allow an estimate of performance change over 50,000 hours.

The wearing surfaces appeared little different than they had at the first inspections. One of the pump bodies was found to have a fine uniform pitting of the aluminum. Corrosion had occurred in a brass valve and copper fittings in the solution circuit. Apparently moisture had entered the solution charge from a cooling water leak. The copper ions in solution had presumably caused the aluminum corrosion. No changes had occurred on the surfaces of the impellers or raceways. All the pumps were carefully reassembled, new filter-driers were installed in the liquid circuits and the pumps were continued in operation.

On July 12, 1982, pump T-1-1 was stopped at 29,618 hours of operation because it was running hot and the external magnet was rubbing on the stainless steel plate. Nothing was done to the pump at the time because the project was then on hold. The pump has now been disassembled and examined. It was found that about half the solution charge had leaked out. Leaks were found at mechanical fittings in the tubing circuit. No harm had occurred to the pump. It was tested and gave the performance shown in Table 3-6.

TABLE 3-6

RECENT PERFORMANCE OF PUMP T-1-1

PRESSURE PSI	FLOW RATE LB/HR	WATTAGE
30	1830	350
40	1675	386
50	1502	425
60	1333	474
70	1150	520

As can be seen, the performance was better than it was in 1980. That might have been due to the cleaning and refilling with fresh fluid before testing. It may be desirable, however, to carry out a short performance investigation when all four pumps are stopped.

It was noted that small sections of the hard-anodize coating had chipped off within some of the buckets of the impeller. The photographs of Figure 3-4 show those small spots. Possible causes for the chipping would include overheating or cavitation due to pumping high quantities of vapor as a result of the low liquid level. The locations of the spots at the bottom of the buckets also implies the possibility that the anodize was less adherent at those interior spots. The spots will be examined again at the final disassembly. Anodizing may not be necessary in the bucket areas since they are not bearing surfaces, but it is possible that the hard-anodize has served to prevent erosion of the blades, even though there are no indications of erosion on the bare surfaces of the raceways and strippers. The pump was put back in operation and is running at the normal 65 psi pressure rise.

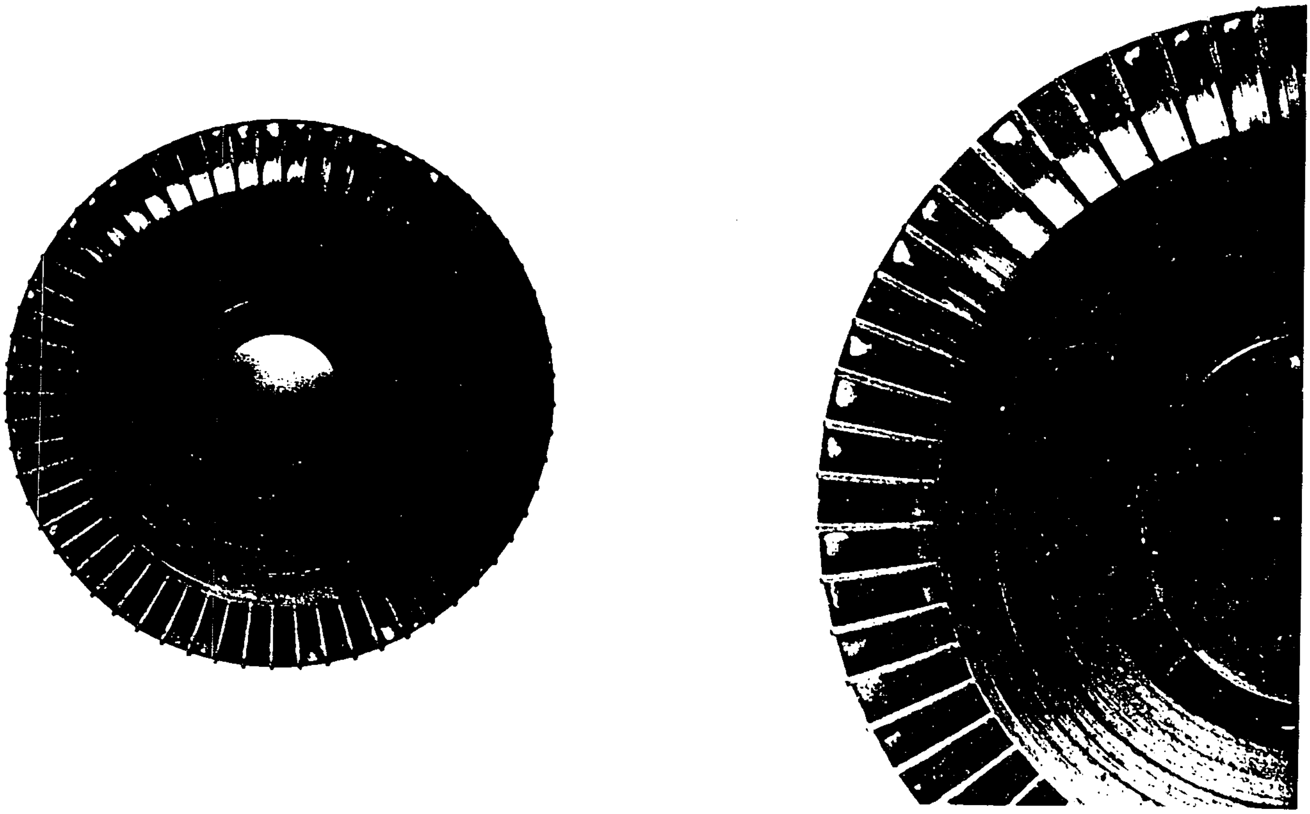


FIGURE 3-4
IMPELLER SHOWING FLAKING OF HARD-ANODIZE COATING

The bearing surfaces of the pump parts, both aluminum and graphite, were machined surfaces. The hard-anodizing process builds up the aluminum surface by about .001 inch. The graphite bearing surfaces were then final-machined to fit new impeller thickness with clearances of .001 inch. The anodized surfaces were not polished or ground. The normal hard-anodizing process produces a surface with microscopic roughness. Much better wear surfaces can be obtained by applying a final diamond hone operation to remove the roughness. Similarly, the finish on the graphite surface can be much better, as on mechanical seals. This life test experience with the relatively rough surfaces could therefore be taken to mean that even longer operation is possible with optimized finishes. If true, the solution pump would not be a limiting factor on the life of the sealed absorption system.

OPERATIONAL PROBLEMS

Vapor Lock

Vapor lock can be expected whenever a saturated liquid is being pumped with near zero NPSH. In these developments, the ability of the regenerative turbine pump to pump a significant amount of vapor reduced that potential problem to a manageable level without having to provide an intake head of more than one or two inches.

Vapor lock problems were encountered on test units when operated with horizontal shaft pumps. The causes of vapor lock that were identified fell into three categories. One was the sudden entrance of a richer solution to the pump, such as a sudden or excessive overflow of refrigerant from the evaporator. That problem was corrected by making provision to thoroughly mix the overflow with the absorber outlet solution before it reached the pump inlet. A second cause related to start-up shortly after a shut-down. That situation might happen during testing, but was expected to result primarily from momentary power failures. Back-flow of hot solution into the pump was apparently the specific cause when that happened. That was eliminated by the installation of a check valve in the discharge line of the pump. Occasionally vapor lock also occurred on cold start-ups. Those could be due to very rich solution held up in the evaporator or precooler before the heat pump started, or to a rapid drop in pressure on entry of weak solution into the absorber.

It was assumed that there were other conditions that might also initiate vapor lock. Once established, the vapor lock condition generally became stable and required manual intervention to overcome it. Rather than attempting to foresee and solve all other incidents that might trigger a vapor lock, means of preventing the vapor lock from becoming stable were investigated. Two such means were developed and applied. An intake standpipe similar to the concept of S. Briggs Patent #3,357,203 was used to make it more difficult for a vapor lock to become established. Secondly, a .062 inch diameter vent hole was drilled into each of the two raceways. Those small vents, close to the intakes, caused only a minor pumping loss, but discharged sizable streams of bubbles. With those two modifications, the remaining vapor lock problems were apparently eliminated in the horizontal shaft pumps.

In the vertical shaft pumps, there have been indications but no proved cases of vapor lock. The questionable cases appear to have been due to the presence of non-condensibles rather than vapor. Small standpipes have been installed as safety measures.

Magnet Decoupling

Magnet decoupling can occur whenever a drive is overloaded; but if the proper magnets are applied, normal overloading should not be a problem. In this application, entry of large quantities of vapor to the pump presented the primary decoupling problem. Vapor entry would normally represent a condition of reduced load, but the vapor apparently caused rapidly changing loads which shocked the magnetic coupling. Presumably the entry of a slug of liquid, or of succeeding slugs, combined with the inertias of the driving magnet and the rotor of the motor, caused displacement or oscillation of the magnets to the point of decoupling. Magnets with four, six and eight poles were tried, with six proving to be the best. The reduction in the size of the motor also produced a noticeable improvement. The use of a weir ahead of the pump intake, to smooth the entry of liquid to the pump, was also helpful. The vapor-triggered decoupling was essentially eliminated by those means; but because decoupling can also be due to other factors, such as momentary power failures, a decoupling sensor was installed, as will be reported elsewhere. The decoupling sensor shut off the motor, and a related control allowed it to stop and recouple before being restarted.

An interim magnet problem, cracked internal magnets, related to the use of cemented magnets. To obtain the required torque with stock magnets no larger than the pump diameter, it had been necessary to bond two magnets together with an epoxy or cyanoacrylate cement. That had worked well in the shorter term laboratory test operations. In pump and unit life tests, and also in longer term laboratory testing, swelling of the cement occurred, causing contact of the magnets with the stainless steel partition. When disassembled, the magnet contacting the stainless plate was commonly found to be cracked. The cracking was apparently due to a combination of friction heating against the stainles steel partition and the swelling forces. The problem was reduced by sealing of the cement from the fluids. It was solved when it was found that Indiana General Corp. had thicker magnets from a special run. No further difficulties were encountered after that change was made.

Noise

The pump is quiet, except for a whistle at 3500 hertz, which is due to the 60 vanes in the wheel circumference, combined with operation at 3500 rpm. Due to its pitch, the sound is very directional and is easily attenuated by baffling and sound insulation. The unit panels alone reduce it greatly. However, attempts were made to reduce the sound. Use of shallower buckets was an improvement, but pump efficiency was reduced. Other pump modifications had little or no effect. It was concluded that baffling and absorption within the housing of the heat pump would be the best solution during the present project.

Cleanliness

Standard turbine pumps are known to wear if the fluids are not clean, and are commonly designed for either replacement of parts or adjustment to compensate for wear. However, the experience with the life test pumps has been that regenerative-turbine pumps with graphite and hard anodized aluminum bearing surfaces and pumping clean R-133a/ETFE in sealed aluminum systems show essentially no wear. In the testing of units that had been cut open for repair or modification, however, pumps were often harmed by aluminum cuttings or particles that had not been cleaned out and entered the pump with the rich solution. The effects were scoring of the graphite and hard anodized surfaces of the impeller side bearings, and the bending or tearing of blades if larger particles entered.

A filter at the pump intake was found impractical because at the low vapor pressures of R-133a/ETFE a flow restriction can easily cause vaporization and vapor lock. Therefore "clean room" assembly procedures are required for the pump and absorber, and possibly the precooler. The liquid entering the absorber is filtered by a filter-drier placed just upstream of the weak liquid control valve, so cleanliness further upstream is of lesser importance. It is necessary that the fluids also be dry and the system free of leaks so that aluminum oxides will not form. A drier was found to be necessary in the solution because ETFE can absorb a significant amount of water. The dry fluids have shown no signs of reactions with aluminum, but the presence of moisture, and perhaps oxygen, has resulted in corrosion. In production, clean, dry and leak-tight assembly can be part of the processing. Under those conditions, the potentials for long operating life should be readily realized.

4.0 FIELD TRIAL UNITS

FIELD TRIAL UNITS

INTRODUCTION

The design, construction and testing of six Field Trial units using organic working fluids is a continuation of the testing, development and evaluation in controlled indoor test rooms. The primary purpose of the program was to obtain real world experience about the operation of heat pumps with R-133a/ETFE working fluids. It was considered necessary to expose units to outdoor conditions so that unforeseen design requirements might be encountered.

In order to obtain a reasonable amount of operating time in the field and life test, it was necessary to begin construction of the Field Trial units almost immediately. Therefore, certain component designs were based on existing knowledge from the previous program. (2) As component developments were made, some were incorporated into later models. However, existing component designs required that the components be built from commercially available aluminum tubing and extrusions. In order to approach the target heating capacity of 90,000 Btuh in a 47°F ambient and cooling capacity of 36,000 Btuh in a 95°F ambient with those materials of construction, relatively tall component designs were used to accommodate larger heat transfer surfaces. Continued development of a production prototype would yield components of a more normal size. The first priority of this phase of the program was to build and put units on test as rapidly as possible in order to accumulate hours and experience concerning the operation of organic fluid absorption heat pumps. Overall size and heat pump performance were secondary considerations from this standpoint. The six Field Trial units (hereinafter referred to as PT units) were built and placed on test on that basis.

STATUS AT THE BEGINNING OF THE PROGRAM

At the conclusion of the previous program, sponsored by the Gas Research Institute, (2) the use of R-133a as the refrigerant and ETFE as absorbent had been demonstrated on breadboard type test units. The overall system performance had reached over 92% of the target heating efficiency and over 84% of the target cooling efficiency. Full heating and cooling capacities had been demonstrated. The positive displacement vane pump was indicated to be at the target efficiency, but problems in unit operation were being encountered and had not been solved. The regenerative-turbine pumps were performing well on test units and had reached 21,000 hours on life test without difficulty, but the efficiencies were low. A decision was made shortly after the program started that turbine-regenerative pumps would have to be used on the six PT units to assure a reasonable certainty of continuing field and life test operation.

Detailed physical and thermodynamic data were developed on the R-133a/ETFE fluid pair. Data classifications included solubility, vapor pressure, viscosity, density, thermal conductivity, and enthalpy of solutions.

OBJECTIVE

The main objective of this phase of the project was to design, build, and test six Field Trial units in order to obtain real world experience concerning the operation and performance of gas fired absorption heat pumps. Two units were to be placed on continuous operation life test, two others in residential operation, and two on cycling life and laboratory test. Emphasis was placed on keeping the units operational for as long as possible, rather than improving performance as the state of the art advanced. In this way, large numbers of operational hours could be accumulated under a variety of actual conditions.

Other objectives of the program were to characterize and improve the cycling performance of the system, to develop the electronic controls to a high level of reliability, to identify potential problems with the secondary components, and to monitor the working fluids and sealed system for decomposition and/or corrosion.

DESIGN

Component design for the Field Trial units began immediately when the project started. Since the objective of the field test was to obtain real world experience with these organic fluid absorption heat pumps, component designs which use round tubing and commercially available extrusions were formulated. Recognizing the limitations that this presented in component design, the cabinet was sized 66" high by 40" wide by 42" deep. This size allowed adding height to the absorber, evaporator and generator, which were the performance limited components. These three components established the height of the cabinet.

Generator

The PT unit generator is shown in Figure 4-1. As developments occured in the generator program which allowed performance improvements in the generator, those developments were incorporated into the later generators. However, the early PT units had generators similar to Figure 4-1. The boiler section is externally finned, with continuous fins in the lower part to limit the rate of heat transfer where the flue gases are hottest. The upper fins are segmented and twisted. Inside the boiler section, there is fluid recirculation in order to maintain a close approach between the fluid and the wall temperature and to promote a high heat transfer coefficient. The

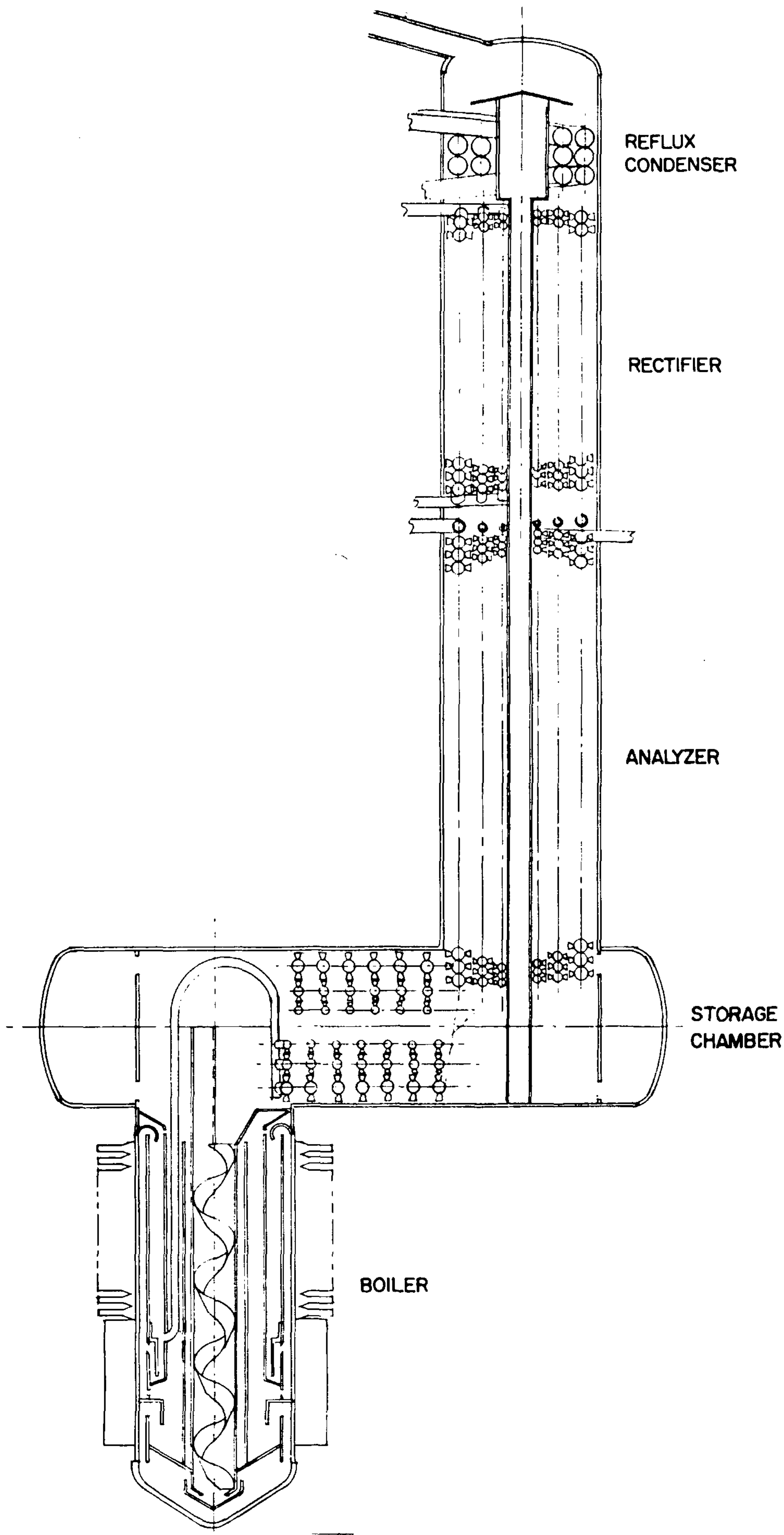


Figure 4-1 Field Trial Unit Generator Design

horizontal and vertical analyzer sections consist of helically-wound, finned tubing. The rectifier is also three, concentric, helically-wound coils. The reflux is smooth 3/4" tubing.

Absorber

The first generation PT unit absorber is shown in Figure 4-2. The upper half is a solution-cooled absorber heat exchanger made from four, concentric, helically-wound coils. The lower half is a water-cooled absorber of the same design. A regenerative turbine type solution pump fits into the pump chamber, which is attached to the absorber section. A diagram of this pump can be found in Figure 4-3.

Evaporator

The evaporator used on all PT units is diagrammed in Figure 4-4. This design uses four, concentric, helically-wound coils enclosed in a shell. The refrigerant evaporates on the shell-side while the water/glycol solution is chilled on the tube-side. The liquid refrigerant flow rate is controlled by a thermostatic expansion valve, the sensing bulb of which fits into the well at the bottom of the evaporator as shown in Figure 4-4. The thermostatic expansion valve itself is shown in Figure 4-5. A refrigerant precooler connects the absorber and the evaporator. The function of the precooler is to chill the liquid refrigerant with the vapor and a small amount of liquid leaving the evaporator to reduce the flashing of the liquid refrigerant as it enters the evaporator. In this way, the high quality refrigeration effect is saved as much as possible for chilling the water. The precooler consists of a helically-wound tube in a horizontal shell. Figure 4-6 is a drawing of the precooler as used in the PT units.

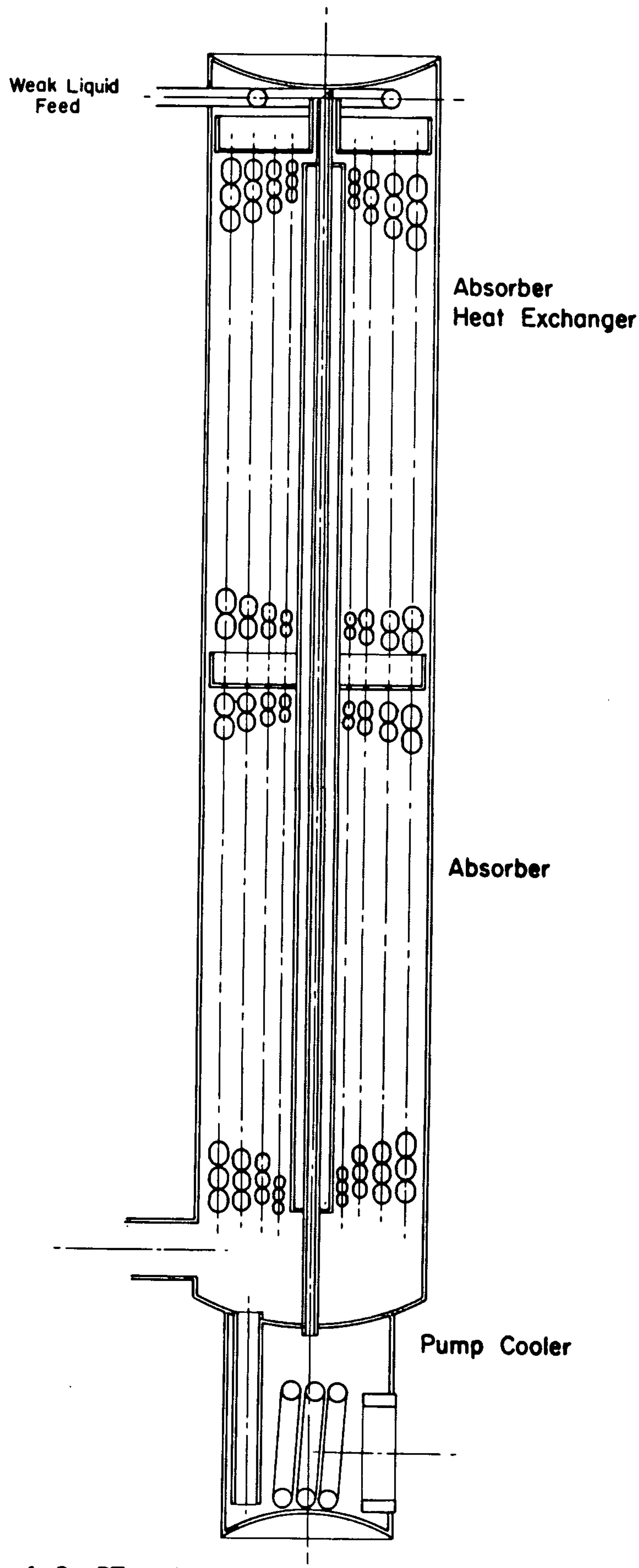


Figure 4-2 PT Unit Absorber/ Absorber Heat Exchanger

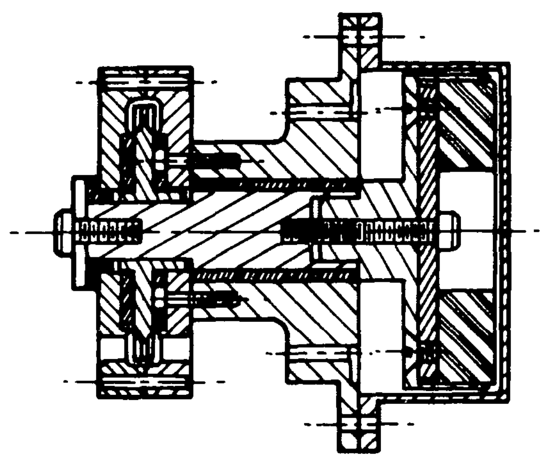


Figure 4-3 Regenerative Turbine Pump

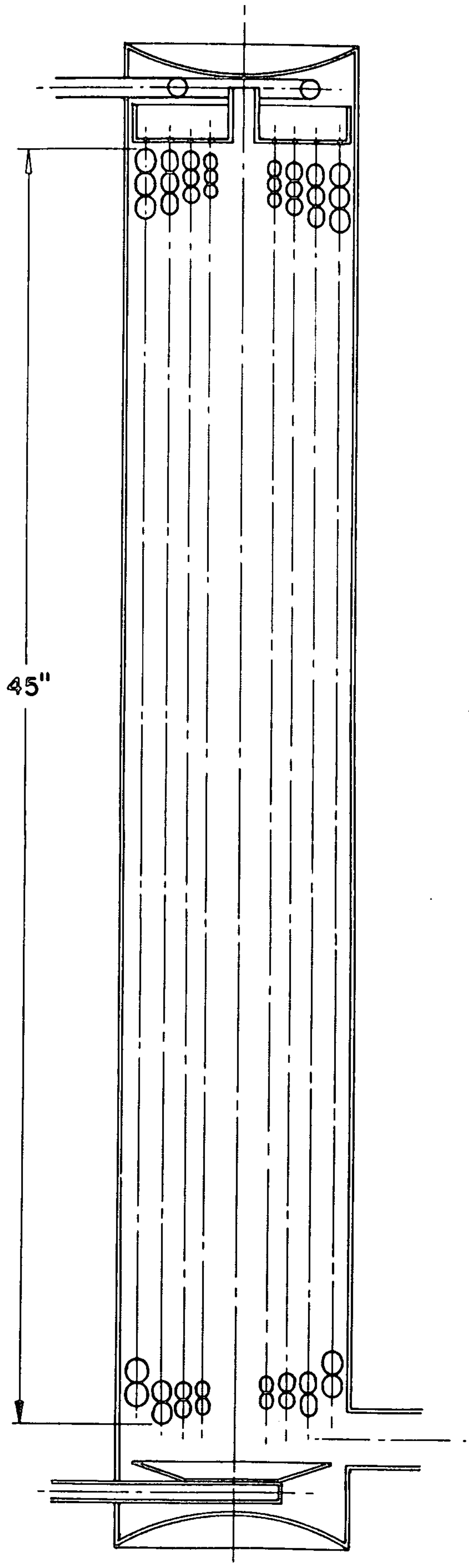


Figure 4-4 Evaporator - Round Tube, Four Coil Design

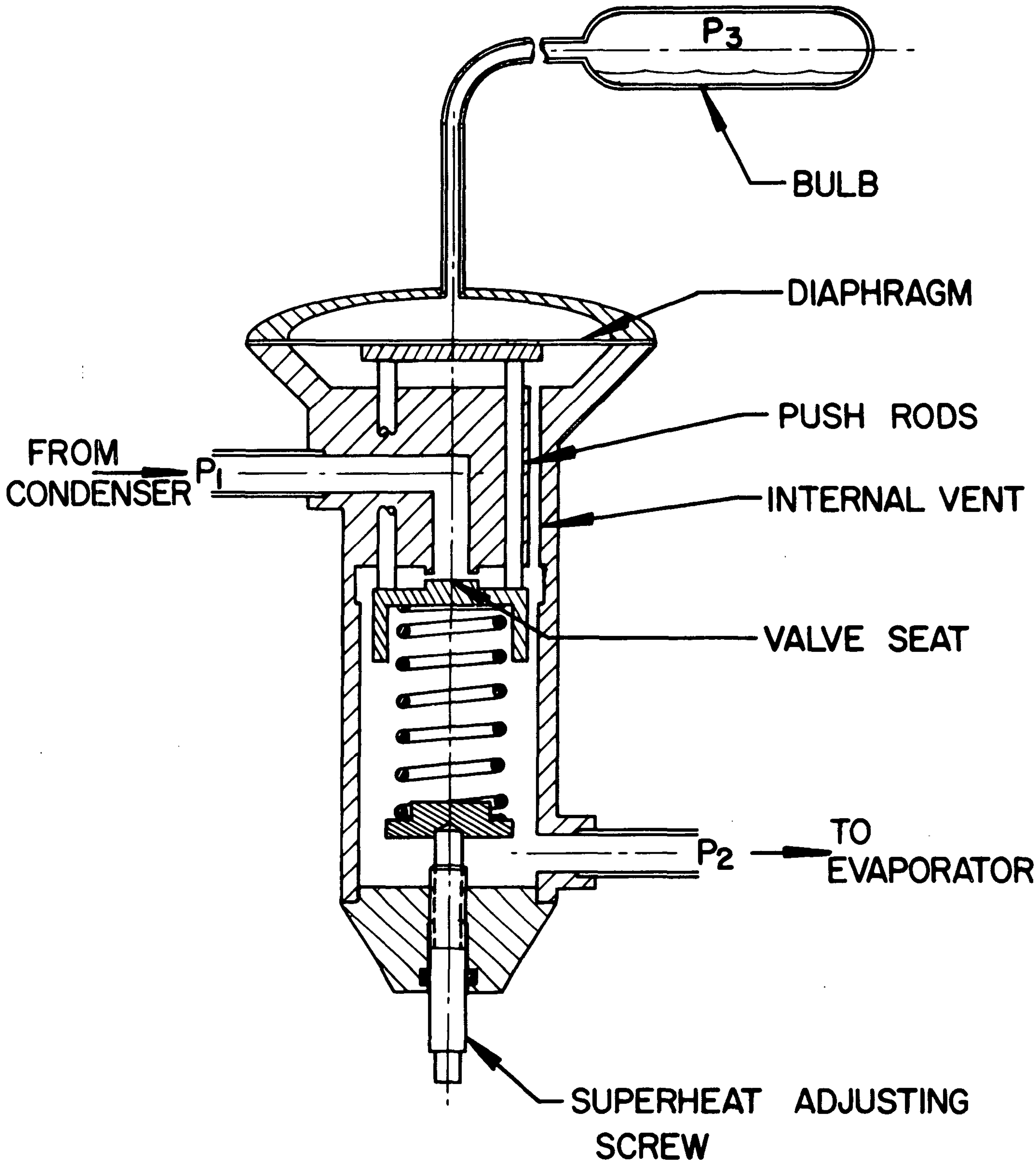


Figure 4-5 Thermostatic Expansion Valve

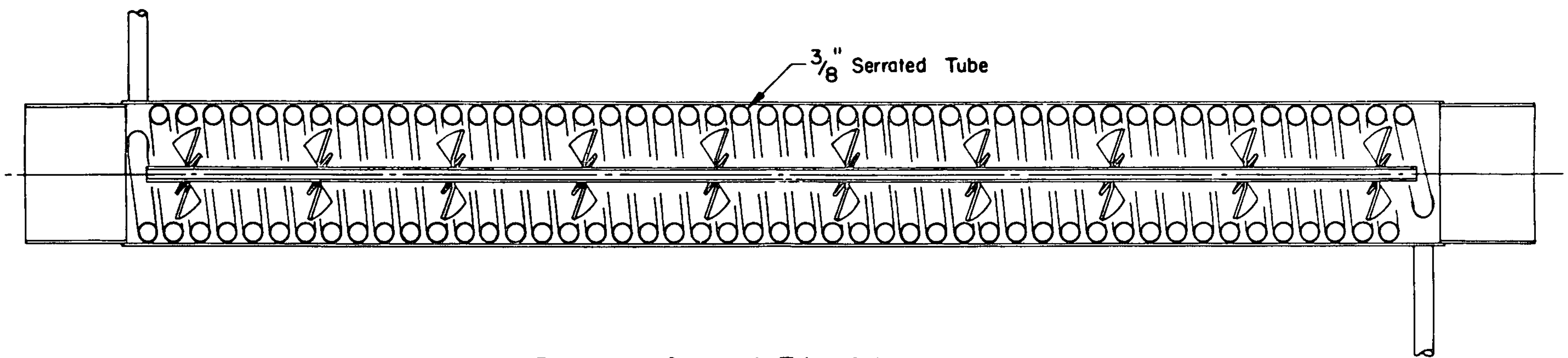


Figure 4-6 Precooler, Serrated Tube Coil

Liquid Heat Exchanger

The liquid heat exchanger was built from a spirally-wrapped extrusion. Heat exchange occurs between adjacent passages in the extrusion, since rich solution and weak solution are contained in alternate passages of the extrusion. A drawing of this extrusion and the final wrapped assembly is shown in Figure 4-7.

Condenser

The refrigerant condenser construction is similar to that of the liquid heat exchanger in that it consists of a spirally-wrapped extrusion. However, the refrigerant occupies all the internal passages of the extrusion, and the cooling water is contained in the passages which are made when the extrusion is wrapped. Please see Figure 4-8 for the extrusion and final assembly concept.

Other Components

The flow of weak liquid from the generator to absorber is controlled by a weak liquid expansion valve. This device senses low side pressure changes and adjusts the flow of weak liquid accordingly. The weak liquid expansion valve is shown in Figure 4-9.

The interrelationship of these sealed system components to the final package is shown in Figure 4-10. The evaporator and absorber occupy opposite corners of the frame, being connected by the precooler. These two components require the full 66" height of the frame. The third large component is the generator, which lies between the absorber and evaporator in another corner of the unit. The condenser and liquid heat exchanger are stacked upon each other vertically. The fluid lines and filters are also shown in Figure 4-10.

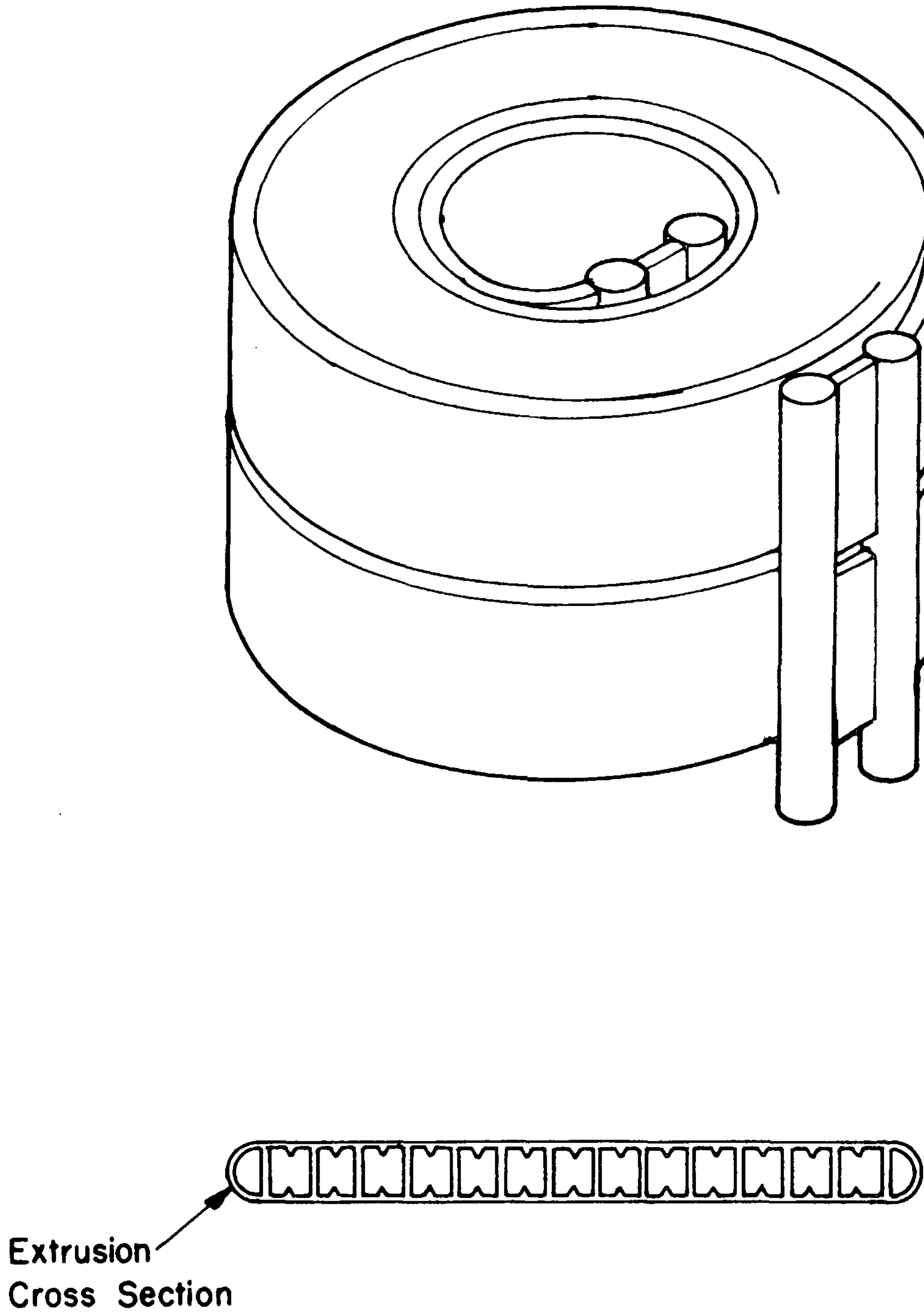


Figure 4-7 PT Unit Liquid Heat Exchanger

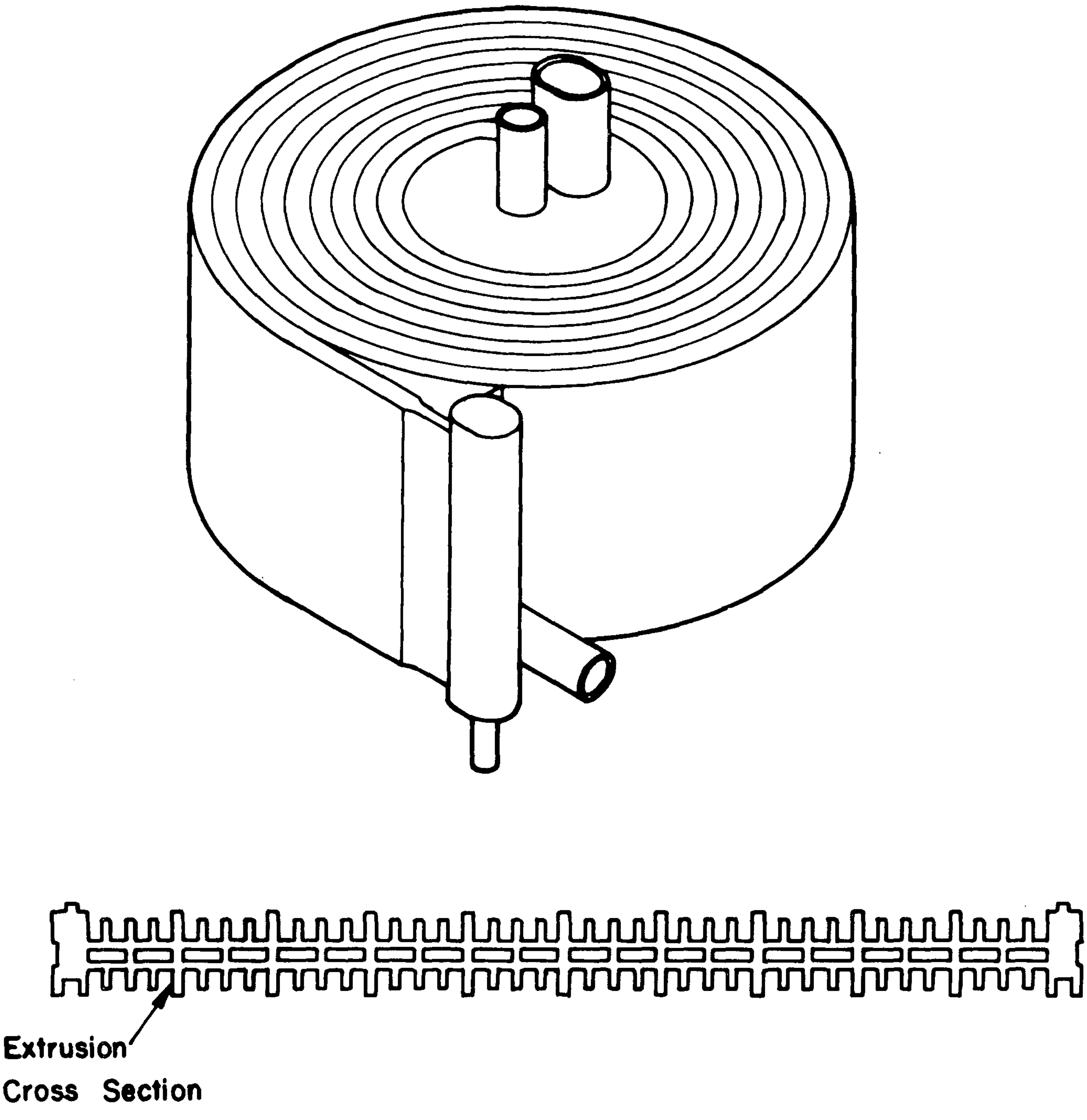


Figure 4-8 PT Unit Condenser

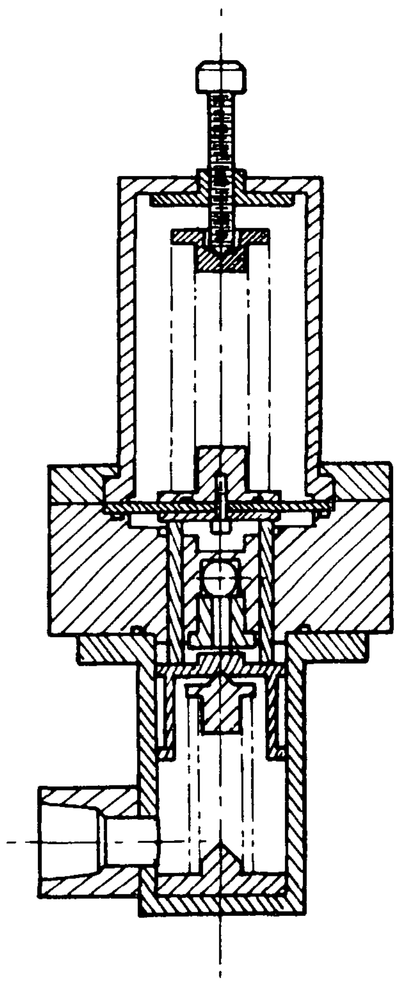


Figure 4-9 Weak Liquid Expansion Valve

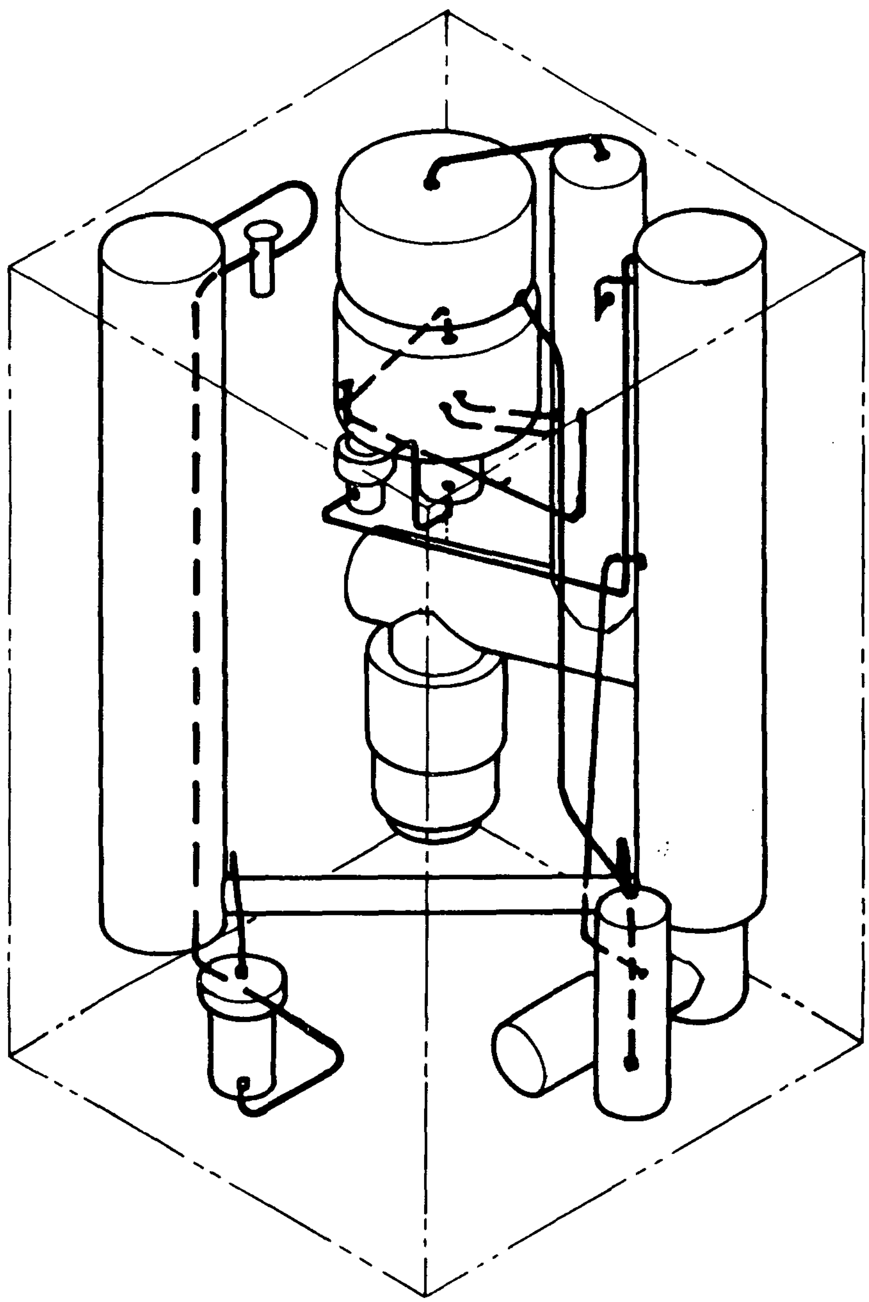


Figure 4-10 PT Unit Sealed System Assembly

Other major components in the PT unit design relate to the water/glycol circuit. The outdoor coil is an L-shaped coil which is oriented across from the L-shaped arrangement of sealed system components. This coil is sized to dissipate 90,000 Btuh as waste heat in the summer air conditioning mode; therefore, it is large enough to allow close approaches to ambient temperature in winter heating. Another advantage of this large coil is that in the winter it seldom requires defrosting. This coil is pictured in Figure 4-11. An indoor coil is shown in Figure 2-6; this coil is for vertical ductwork installations. Since it is sized for 72,000 Btuh heating load, its characteristics in air conditioning are quite good, allowing a warm evaporator temperature while still removing latent heat from the conditioned air.

The cooling water and chilled water are switched between these two coils and the sealed system by means of an eight-way valve. This eight-way valve and motor assembly are shown in Figure 4-12. The assembled cooling water and chilled water systems located with respect to the PT unit frame are shown in Figure 4-13.

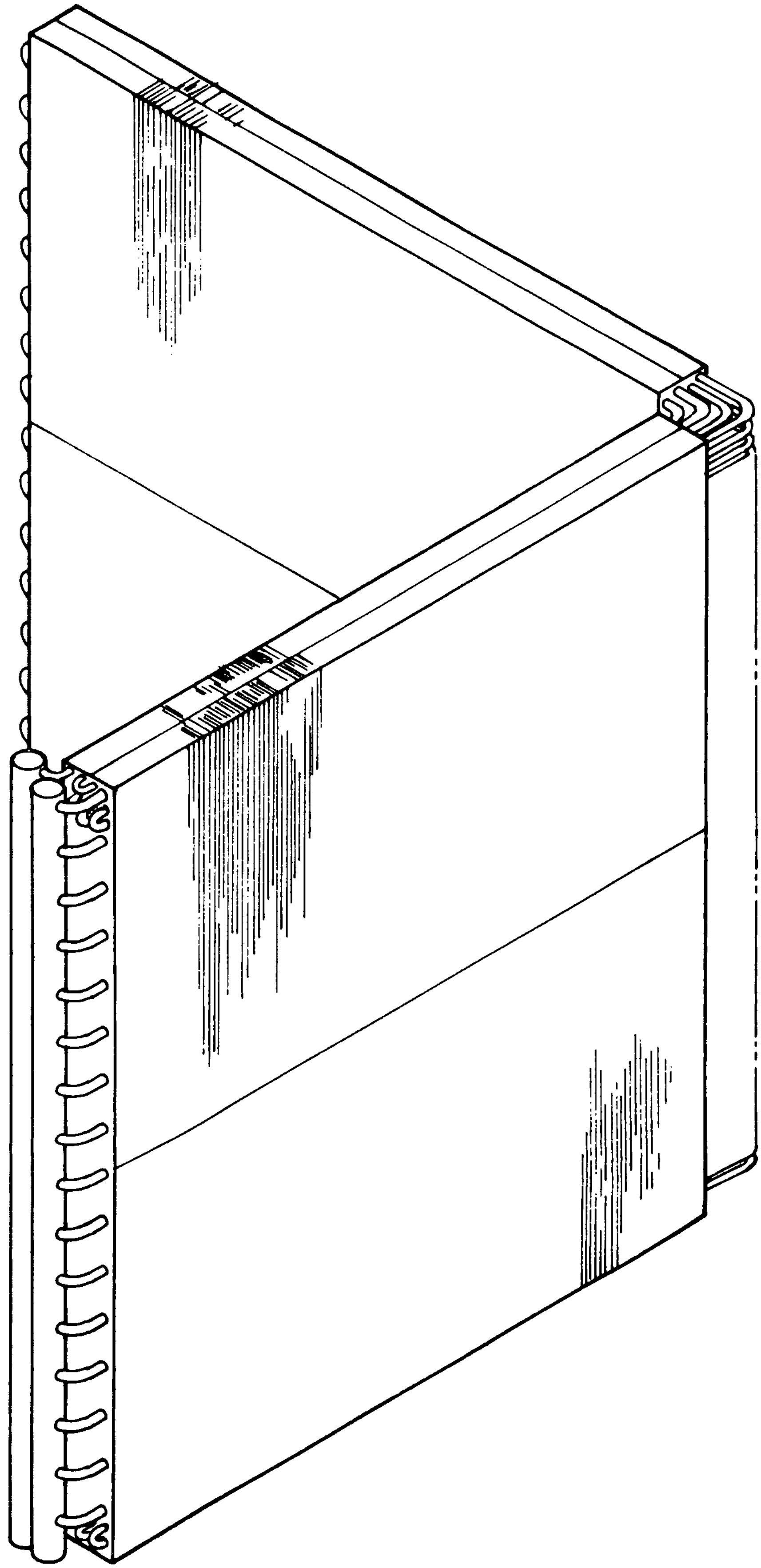


Figure 4 - 11 Outdoor Coil for Field Trial Units

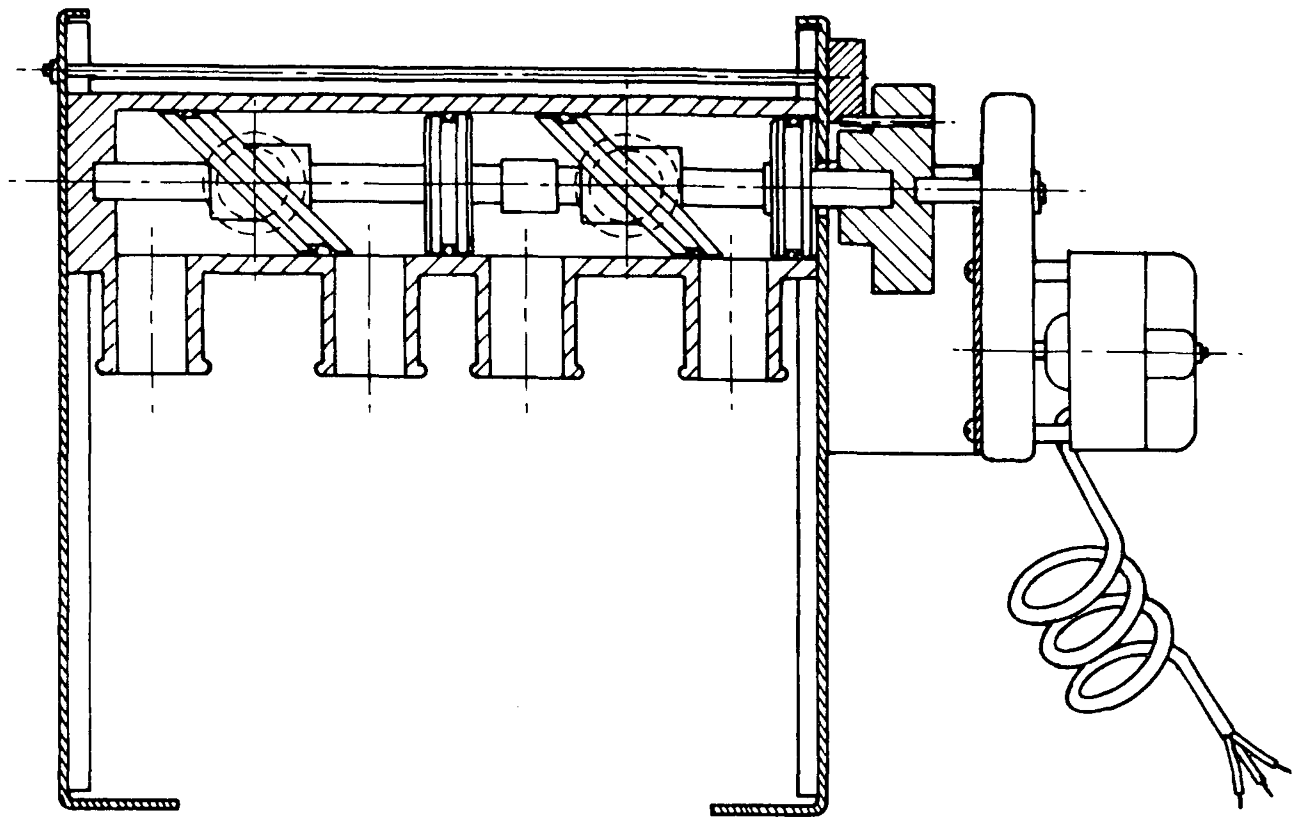


Figure 4-12 Eight-Way Valve and Motor Assembly

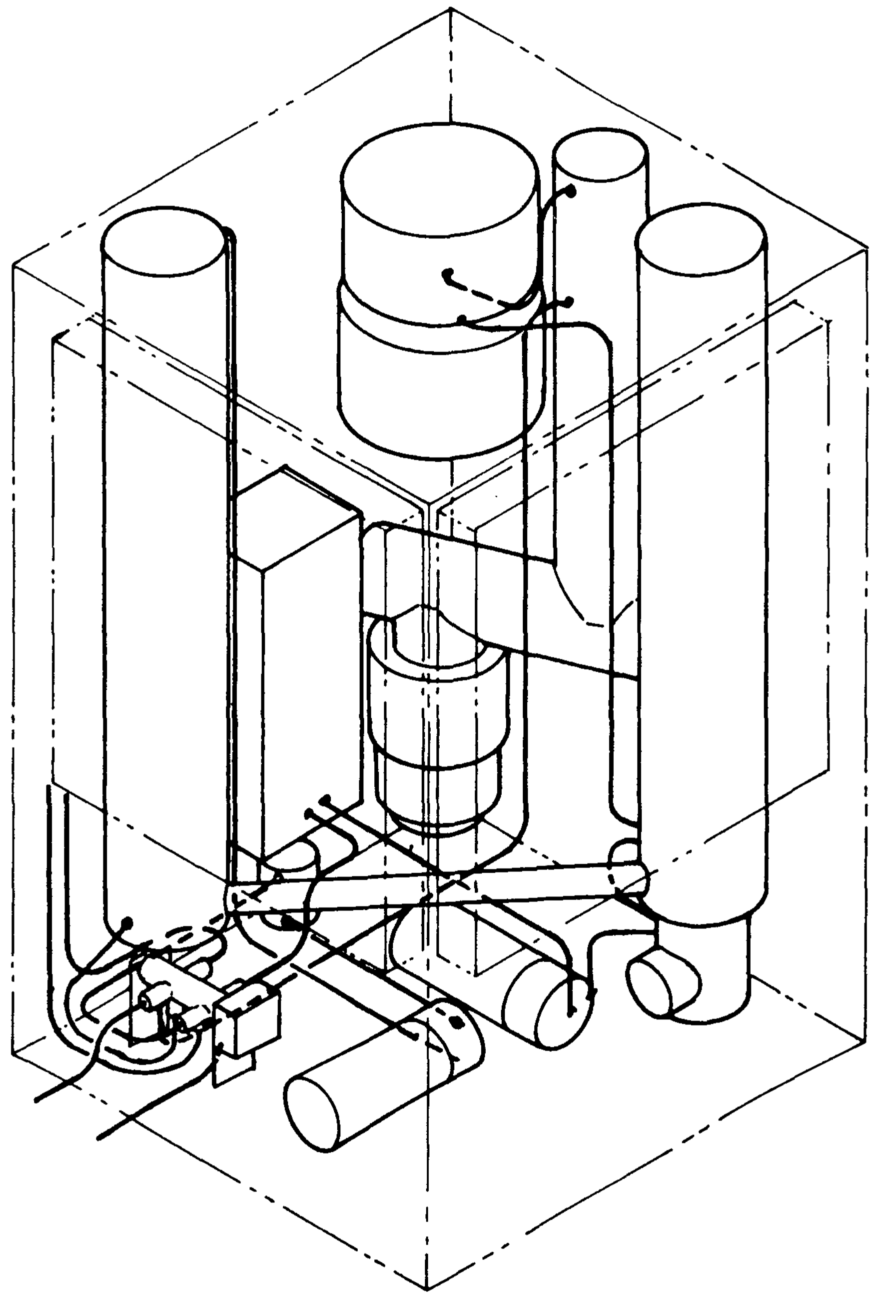


Figure 4-13 PT Unit Water System

CONSTRUCTION OF FIELD TRIAL UNITS

Construction of PT units began as soon as the program started. The availability of raw materials and component parts determined the order in which subsystems were built. Also, subsystems requiring little or no engineering design were built first if the materials were available. Construction of frames, partitions, panels and filter housings began first.

Due to the simultaneous component development program, the design, development, construction and testing of the sealed system became the critical path which determined the completion dates for the PT units. The program began by sequentially testing component designs on the one available Test Unit. Information from those tests aided in the development of other new components, but the procedure was found too time consuming. Delays developed while trying to design and develop an absorber which would perform satisfactorily. In order to speed up construction and testing of the PT units' sealed system, it was decided to do some system development work on the PT units themselves. In this way, the PT units would be modified as developments occurred. In many cases, design modifications were tried on the PT units first. So, rather than build six identical units, as component design and development progressed, subsequent PT units were changed to reflect the progress. The sealed systems of the six PT units were completed and tested in the laboratory at approximately two to three month intervals. Between the time of the first sealed system test and final assembly of the unit with outdoor coil, fan and panels, a considerable amount of lab testing and development was done. Development work was done not only on the sealed system, but on the electronic controls and auxiliary components as well. Shortly after the total package tests of the PT units, they were installed in their field or life test locations. The locations, and disposition of the units from this point on will be discussed in a following section.

Sealed System

The absorption sealed systems were tested by themselves as soon as they were completed. Thus data were obtained more quickly, and any problems in the sealed system were identified before final assembly of the PT units. Those tests also served to document the performance of each unit before the life or field tests.

The sealed systems were assembled in the PT unit frames. Motors and pumps were installed along with the electronic controls. The outdoor coil, fan, and cabinet panels for each unit were not assembled. The PT units were installed in test rooms instrumented to obtain more data than would be possible in a fully assembled unit. Each unit had thermocouples to measure working fluid temperatures, thermometers for water/glycol temperatures, weak liquid and refrigerant rotameters, water/glycol rotameters, and high and low side pressure gauges. Data was collected under a variety of conditions, and modifications were made based on the results of these sealed system tests.

The sealed system of PT-1 was tested first. This unit represented the component designs which had been formulated at the end of the earlier program. The performance of PT-1 was measured at various evaporator temperatures in the heating mode; Table 4-1 is representative of the performance of this unit.

TABLE 4-1

HEATING MODE PERFORMANCE PT-1

EVAP °F	COP C	COP H
46.5	.411	1.16
40.0	.381	1.11
20.5	.215	.978
10.5	.217	.929
5.0	.132	.904
4.0	.181	.898

The above data were with 95°F cooling water.

The effect of cooling water temperature on performance was also investigated in several runs. Table 4-2 contains the results of some of these tests.

TABLE 4-2

EFFECT OF COOLING WATER TEMPERATURE ON PERFORMANCE, PT-1

COOLING WATER °F	COP C	COP H	EVAP °F
87.3	.434	1.15	40.6
95.4	.381	1.06	39.6
105.6	.330	.960	40.6

Both sets of data were taken at inputs of 60,000 Btuh

Performance at the level of these data would normally not be accepted, but it was more important to start the continuous run life tests. If time were taken to improve this unit sufficiently all the units would be delayed excessively.

The sealed system of PT-2 was tested next. This unit was run at 72,000 Btuh. Performance data is presented in Table 4-3 which follows.

TABLE 4-3

HEATING MODE PERFORMANCE, PT-2

EVAP °F	COP C	COP H
39.0	.378	1.12
20.5	.251	.969
2.5	.147	.838

Problems were experienced with solution pumps on PT-2. The original solution pump, which was on the unit when the sealed system tests began, did not have enough capacity at low evaporator temperatures (high pumping rates). It was replaced with a pump of a slightly different design with more capacity. The magnetic coupling was then found to break at low pumping rates because of the sudden variations in pumping load when pumping both liquid and vapor. In order to run sufficient tests, the pumping rates were arbitrarily increased at the expense of performance.

The sealed system of PT-3 differed from that of PT-1 and PT-2 in two ways. It had the first absorber divided into upper and lower sections. The absorber heat exchanger comprised the upper half and the water-cooled absorber, the lower half. The second difference was in the rich liquid flow pattern. The rich solution leaving the solution pump was routed first to the rectifier, then to the absorber heat exchanger, on to the liquid heat exchanger and finally into the generator.

The first tests on the sealed system of PT-3 indicated that this particular rich liquid flow pattern hurt the performance. The relatively cool, rich liquid entering the rectifier coils caused too much refrigerant vapor to condense, and consequently reduced the amount of refrigerant going to the condenser. Another rich liquid flow pattern was tried in which the rich liquid went from the solution pump to the absorber heat exchanger, to the rectifier, to the liquid heat exchanger and finally into the generator. This pattern produced better performance than the first one. Heating mode performance data is contained in the following table 4-4.

TABLE 4-4

HEATING MODE PERFORMANCE, PT-3

EVAP °F	COP C	COP H
45.0	.417	1.200
25.2	.313	1.056
9.9	.188	.968
5.1	.168	.936

All data were with 95°F cooling water and 72,000 Btuh input.

The performance of the PT-4 sealed system was somewhat disappointing when compared to that of PT-3, in spite of the fact that both systems were of similar design. When operated at about 60,000 Btuh input and 95°F cooling water, the performance of the PT-4 system was only slightly better than that of PT-1 or PT-2. Analysis of the data did not indicate any major problems. With two more units to be built, there was not time to interrupt the program for hardware changes and possible extended optimizaton. Therefore, PT-4 was set aside in order to continue work on the remaining two units.

The intended installation for PT-5 was a residence where the full output capability would not be needed. Therefore, that unit was run and tested at an input of 60,000 Btuh Table 4-5 contains representative performance data for PT-5.

TABLE 4-5

HEATING MODE PERFORMANCE, PT-5

EVAP °F	COP C	COP H
41.0	.421	1.206
21.1	.314	1.059
2.2	.208	.926

All data were with 95°F cooling water and 60,000 Btuh input.

The final Field Trial unit was PT-6. This unit represented the latest developments in component design. A new design absorber was tried on this unit for the first time, and the combustion system was somewhat improved. The performance of this unit is found in Table 4-6.

TABLE 4-6

HEATING MODE PERFORMANCE, PT-6

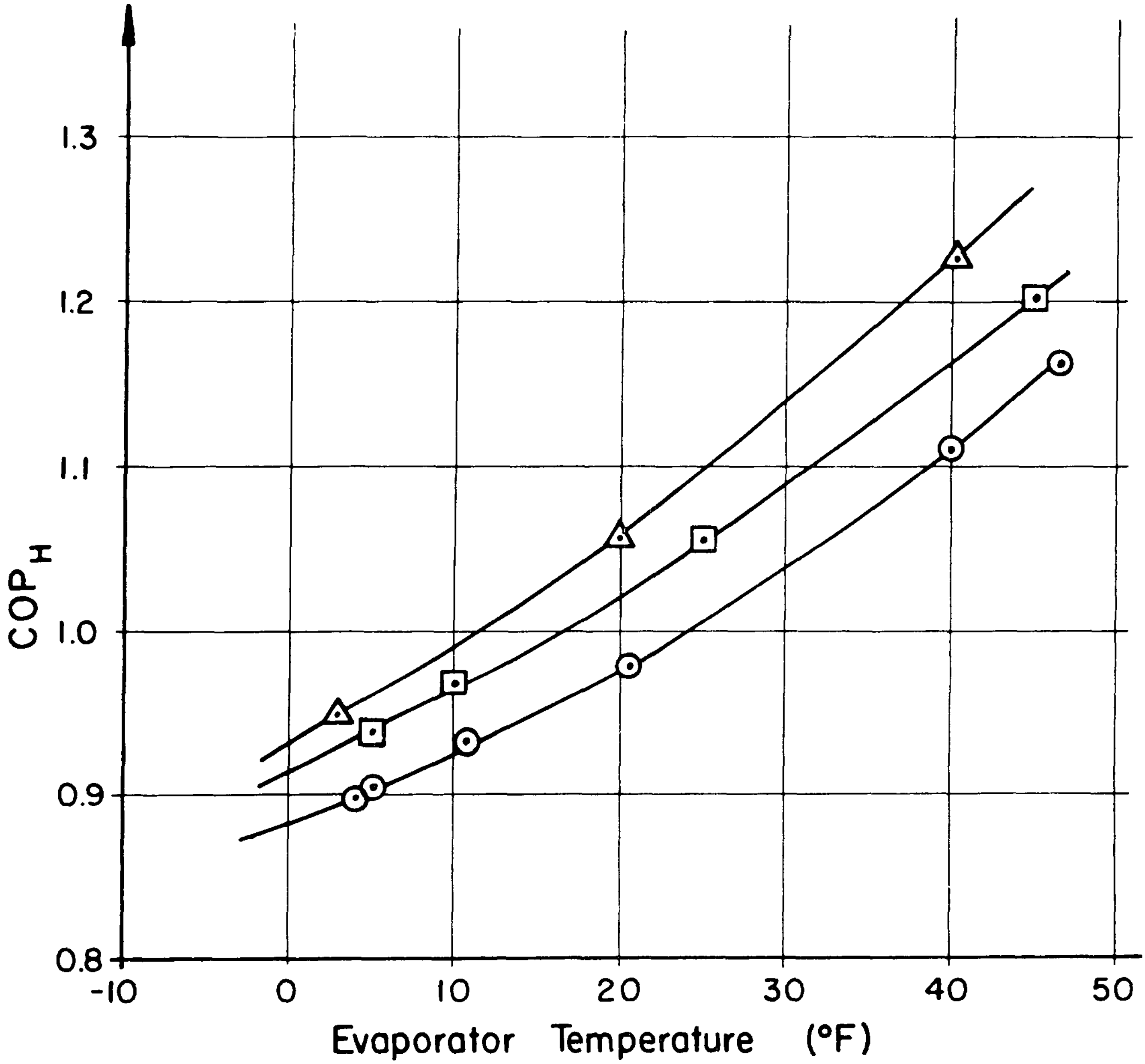
EVAP °F	COP C	COP H
40.0	.426	1.227
20.0	.269	1.059
3.0	.198	.945

All data taken at 60,000 Btuh input and 95°F cooling water temperature.

A comparison of the heating mode performances of PT-1, PT-3, and PT-6 can be found in Figure 4-14. The diagram shows the performance improvements made during the period of construction of the six units.

Aside from performance data, other significant information came from the laboratory tests of the sealed system. Due to the large component sizes, the total charge for the PT units ranged from 90 to 100 pounds of ETFE and R-133a. That quantity is too large for the concentration control chambers, and the concentrations would not match the full range of outdoor temperatures. The charges were adjusted to the lowest temperatures. Overheating of the fluids at low temperature was thus prevented, but operation at the warmer ambients was held to less than the unit potentials.

Other important information obtained from laboratory testing of the sealed system had to do with the weak liquid expansion valve and the refrigerant thermostatic expansion valve. The detailed information on the development of these valves is contained elsewhere in this report. However, it should be



PT-1 60,000 Btu/hr input, heating mode

PT-3 72,000 Btu/hr input, heating mode

PT-6 60,000 Btu/hr input, heating mode

Figure 4-14 PT Unit Performance Comparison mentioned that a considerable amount of test and development work on these valves was carried out during this laboratory test phase of the sealed system.

Minor, but troublesome problems occurred when the weak liquid filter plugged with chips and other debris from the construction of units. This was solved by using a wound glass fiber filter, which had a greater capacity to filter particles without plugging.

Electronic Controls for the Field Trial Units

Electronic controls were designed and installed on the PT units to provide for efficient operation of the unit under all conditions and to insure safe operation at all times. The control system was based on transistor/transistor logic (TTL) for design flexibility and potential high reliability operation. The control logic system on the first two units was optically isolated from switching triacs which were used to switch electric motors, solenoids and other devices within the unit. The laboratory test program allowed for an evolutionary development of controls and sensors to occur while the sealed system was tested, developed and optimized.

The operation of the controls during start-up, shut-down, and steady state running have a major influence on the cycling efficiency of the unit as a whole. For this reason, the operation of the controls during these times has been covered in detail in the Cycling Operation section of this report.

The original method of control, using optical isolation of the logic and switching triacs, was susceptible to electrical noise and not reliable in operation. As an intermediate solution to this problem, 24-volt mechanical relays were added between the switching triacs and components. This improved the reliability of the triacs, but the electrical noise was still a problem in the TTL system. A third generation control system, which used solid-state relays with built-in optical isolation and transient suppression, was developed. The operation and reliability of the control system was much improved using these solid-state relays. This progression of designs in the control system is schematically presented in Figure 4-15.

The safety system was designed to be highly redundant, so that any conceivable failure mode could be detected by at least two safety system sensors. The following table is a brief description of the safety system sensors and their action.

TABLE 4-7

SAFETY SYSTEM OPERATION

SENSOR	ACTION
Overtemperature Control	Shuts gas off when generator shell/fin temperature reaches a preset point. Reset: Automatic
Overpressure Control	Shuts gas off when high side pressure reaches 85 psig. Reset: Manual
Cooling Water Flow Switch	Keeps gas off until sufficient cooling water flow has been established.
Chilled Water Flow Switch	Keeps gas off until sufficient chilled water flow has been established.
Magnetic Decouple Sensor	When magnetic decouple occurs, the gas is shut off and the solution pump motor stops, recouples and restarts. When magnetic couple is re-established, the gas turns back on.
Sail Switch	Shuts gas off when the fan does not operate, turns gas on when draft returns.
Flame Backout Sensor	Shuts gas off when flame backs out of generator draft shield. Reset: Manual

TABLE 1.3-7, (Continued)

SENSOR	ACTION
Generator Level Sensor	Shuts gas off when the solution level in the generator drops below a certain level. Reset: Automatic
Pressure Relief Valve	Vents high-side pressure to fan compartment if high-side pressure reaches 100 psig.
Defrost Sensor System - Pressure Transducer - Temperature Switch in Outdoor Coil - Temperature Switch in Cooling Water Storage Tank	In the heating mode, if the outdoor coil should frost up as detected by the pressure transducer and if the temperature of the water/glycol from the outdoor coil is below 32°F and if the temperature in the cooling water storage tank is above 95°F, then the gas, fan and chilled water pump shut off and the eight-way valve changes to the cooling mode. When the temperature of the water from the outdoor coil rises above 55°F, then normal heating mode is resumed.

As a part of the laboratory testing program, before any of the PT units were installed in their ultimate location, they underwent a safety feature qualification test. In this test, conditions to simulate various failure modes were created, and the safety sensors were checked for proper operation. All safety features except the pressure relief valve were checked in this way.

The defrost sensor system deserves special attention. It is a demand defrost system which senses if several conditions are present before initiating a defrost cycle. The chilled water temperature must be below 32°F, indicating possible frosting conditions; the pressure drop across the outdoor coil must be .35 inches of water column or greater, indicating significant restriction of air flow due to the presence of frost; and the temperature of the cooling water in the storage tank must be greater than 95°F, indicating that there is enough hot water to complete the defrost. This combination of sensors insures that defrosting will occur only when needed and, therefore, the energy consumed for defrost will be kept to a minimum.

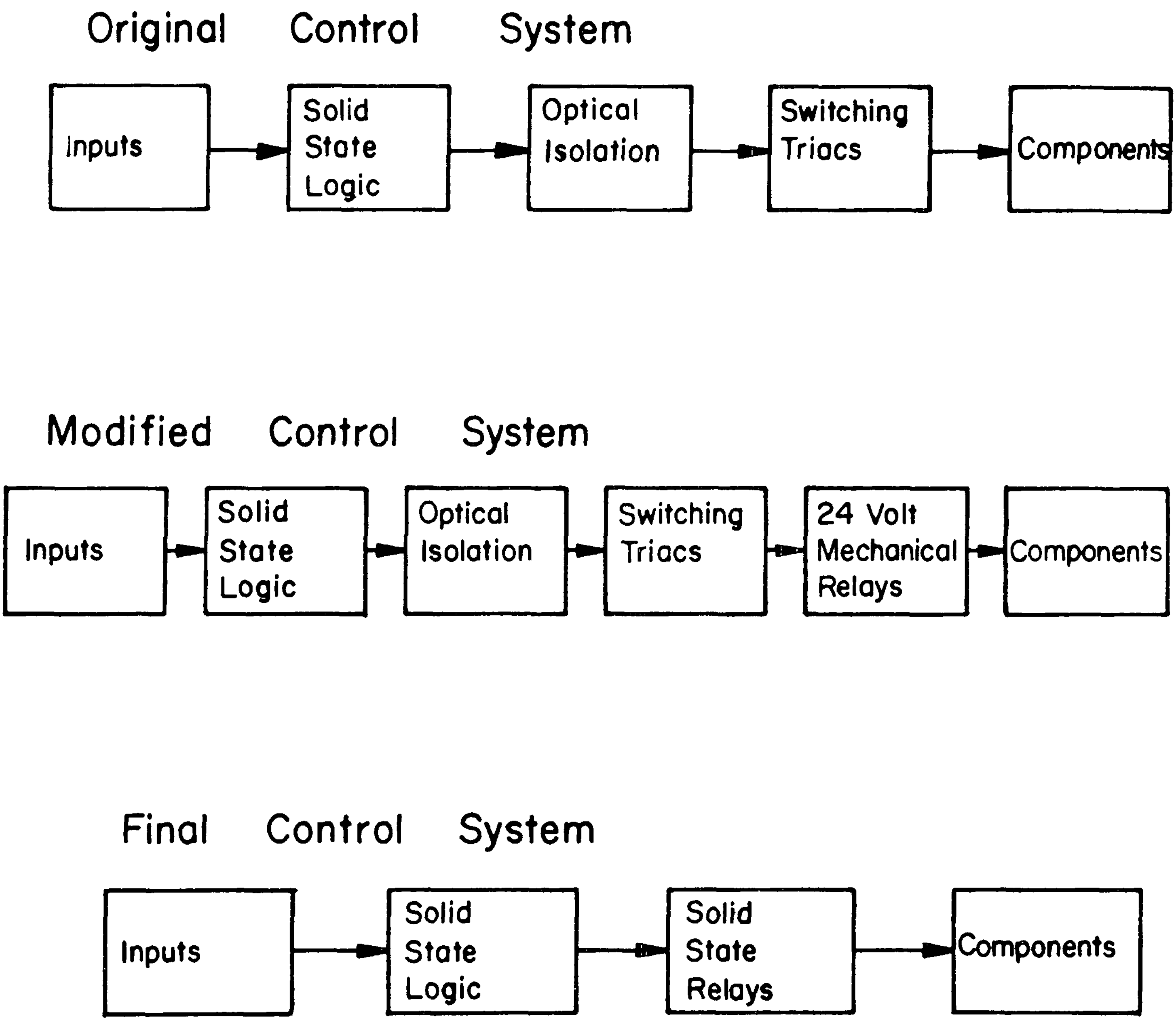


Figure 4-15 PT Unit Control System Evolution

The most difficult sensor to optimize and develop was the generator level sensor. The generator level sensor consisted of a heated and an unheated thermistor pair in a Wheatstone bridge circuit. When the generator level dropped below the level of the heated thermistor and its resistance dropped, the bridge circuit became unbalanced and caused the logic to shut off the gas. When the level in the generator rose to the level of the heated thermistor, the bridge became balanced and the gas was started again. In order to cause a sufficient temperature difference between the thermistor pair with a low generator level, a 45-watt, 220-volt heater was used. To further accentuate this temperature difference, the location of the heated and unheated thermistors was changed until an optimum was found.

The operation of the level sensor when a solution pump failure was simulated was found to be a function of evaporator temperature. For PT-1, running in a steady-state mode with a 34°F evaporator temperature, it took 54 seconds for the gas to go off after the solution pump was turned off. At a 5°F evaporator temperature, with PT-1 running in a steady-state mode, it took approximately 28 seconds for the gas to go off. These times were fairly typical for the level sensors in general. They were longer than desired, but were the earliest signal of low level.

The level sensor and its circuit did not perform satisfactorily in other ways, however. The system was susceptible to electrical noise, and there were heater reliability problems. Trying to find an alternative to the thermistor level sensor became an important part of the laboratory test program. An improved concept used two thermocouples which sensed temperature differences at two levels in the boiler when the liquid level dropped. This worked well for rapid changes in boiler level, but not for slow changes. This type of level sensor was used on two PT units with some success.

The level sensor problem was not completely solved on the PT units. However, the redundant safety features, the generator overtemperature, and solution pump controls served as good backups. Further investigation of the need for the level sensor is required. If concluded to be important, more development work remains to be done to improve its operation and reliability.

Auxiliary Components

Concurrently with the testing and development of the sealed system and electronic controls, several important auxiliary components were tested and refined. The auxiliary components tested during this phase of the program included fans, indoor and outdoor coils, eight-way valves, and electric motors for pumps and fans. The objective of this testing was both to improve the performance of these components and to integrate them with the absorption sealed systems for optimum performance of the PT units as a whole.

Coil Testing

A considerable amount of testing was done on both indoor and outdoor coils of various designs in order to determine their suitability for application to the system. Indoor coils of "A" and "H" configurations were tested and L-shaped outdoor coils as well. Variations in air flow rate, fluid flow rate, fin type, number of rows and tube spacing were investigated for a number of custom and standard coils.

A study of particular interest concerning outdoor coils was carried out in the environmental test room. Ambient temperature was varied over a wide range in order to determine pressure drop versus flow rate characteristics for two different design outdoor coils; these results are summarized in the following table. This study highlights the fact that outdoor coils with low pressure drops at cold ambients are required if pumping power is to be held to a minimum.

TABLE 4-8

OUTDOOR COIL PRESSURE DROP

COIL DESIGN	FLOW RATE (GPM)	AMBIENT TEMPERATURE (°F)	PRESSURE DROP (PSI)
420-6	10	70	5.1
		0	13.3
820-12	10	70	2.5
		0	6.9

All data taken with 40% ethylene glycol solution

Fan Testing

A variety of fans from different suppliers were tested in order to pick the fan with characteristics best suited to the air flow requirements of the outdoor coil. Fan parameters, including power requirements, speed, pitch, pressure drop across the coil and noise, were considered in the selection of the fan. More information on fan and coil testing is contained in the section Field and Life Test Experience with Auxiliary Components.

Parasitic Electric Power

Parasitic electric power consumption has always been a major concern in the absorption system. In order to minimize this power consumption, sources of more efficient electric motors were found. Some of these were incorporated into the PT units as time allowed. Some fan motors and solution pump motors were the first to be changed when more efficient motors were found. Developments which occurred later in the program included the use of an optimized 0.4 hp motor for the solution pump. This motor worked so well it was incorporated into many of the PT units. A further example of the drive to reduce parasitic power is the use of switches on the eight-way valve motor. Previously, the motor had run stalled in the heating or cooling position. The addition of switches at the limits of rotation cut the power to the motor for a savings of 25 watts. The eight-way valve was the subject of further testing, as different O-rings were tried in order to improve the seals and reduce the torque required to turn the shaft.

Conclusions

The laboratory testing program was carried out in considerable depth. The extensive test and optimization period was an important aid in bettering the performance of the later units. Overall, there was a noticeable improvement in performance from PT-1 through PT-6. Figure 4-14 shows that performance improvement for PT-1, PT-3, and PT-6.

In addition to the knowledge gained about the sealed system operating under laboratory conditions, the laboratory testing phase highlighted areas in need of improvement in the electronic controls, safety system, and auxiliary components. The need for improved reliability of the electronic controls became readily apparent in lab testing. This led to the development of more reliable controls. Various safety sensors, such as the level sensor and magnetic decouple sensor, were improved during the laboratory testing phase. The critical requirement for more efficient water pumps, fans and electric motors was also addressed at this time.

The lab test phase was also an important aid to the component development program. It allowed for additional data on the performance of individual components to be gathered while Test Units were occupied with other testing. The main value of the laboratory test phase was to prepare the six Field Trial units for operation in their ultimate locations. While improved operation was achieved, the main value was undoubtedly to reduce the problems encountered in the field trial. Nevertheless many difficulties did occur.

Installation and Instrumentation of PT Units

The objective of the field and life test phase of the program was to obtain as much experience as possible concerning the operation and performance of the six PT units under actual outdoor operating conditions. This was accomplished by installing units on the roof of the laboratory for continuous run and cycling life tests and by putting two units in residences for actual domestic heating and cooling. These units were instrumented to varying degrees, depending upon what kind of information was required from each installation.

Units were installed in two residences in Southwestern Michigan, one at 190 Hunter Drive, Benton Harbor and the other at 2210 Mount Curve Avenue, St. Joseph. These units were instrumented with BTU meters, watt-hour meters for both the entire unit and water pumps only, 24-point recorders which monitored thermocouple temperatures within the unit, separate gas meters, elapsed timers and continuous digital readouts for generator temperature with emergency cut-offs.

The units installed on the roof of Phillips Engineering were equipped with gas meters, BTU meters, multi-point temperature recorders, elapsed timers and digital temperature readouts. Not all of the instrumention was installed when the units were placed in service on the roof. Some was added as the instrumentation became available and as required by the needs of the program.

The chronology of events in the field and life test program, from the first sealed system test to the present, is presented in Figure 4-16. The hours of operation as of November 30, 1982 are also given in this figure. The first field installation occurred when PT-1 was placed on the roof of Phillips

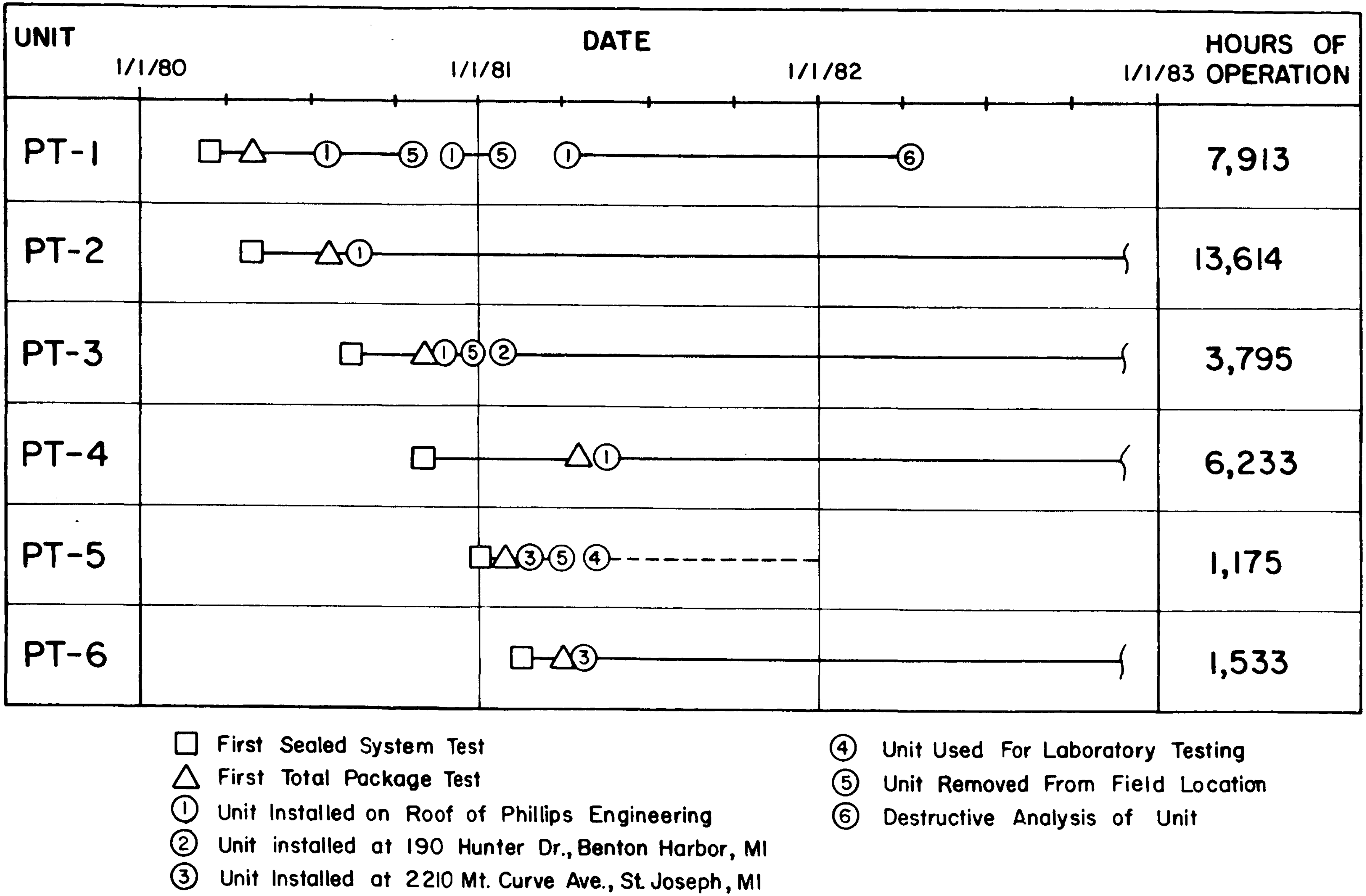


Figure 4-16 Field and Life Test Chronology

Engineering for continuous run life test. This unit was removed from the roof twice for repairs. PT-1 was eventally removed from the roof for destructive analysis of the major components. PT-2 was placed on the roof of Phillips Engineering for life test; it remains there today. PT-3 was tested a short while on the roof (about one month) and then removed and installed at 190 Hunter Drive, Benton Harbor. Unit PT-4 was used as a cycling unit on the roof of Phillips Engineering, and remains on the roof today. PT-5 was located for a while at 2210 Mount Curve Avenue, St. Joseph. Due to problems, it was removed and used as a laboratory test unit inside the plant. PT-6 was then installed at 2210 Mount Curve Avenue, St. Joseph where it is today.

Residential Installations and Operation

This instrumentation of the residential units has been described in the preceding section. The purpose of installing these two units in residences was to expose them to demand cycling and any other special situations which might not be enountered with the units which were installed for continuous run and timed cycling at the laboratory. Also, comfort levels could be qualitatively observed by the occupants throughout the heating and cooling seasons.

The residence at 190 Hunter Drive in Benton Harbor is a single story, stone, ranch style home. Originally it was an all-electric home and consequently was well insulated. An "A" style finned coil was already located in the ductwork of the home, and very little work had to be done to prepare for the installation of the absorption heat pump. Due to the insulation of the home, the input of the heat pump was reduced to 60,000 Btuh, but even this was high for the 1800 square foot home.

The residence at 2210 Mount Curve Avenue in St. Joseph is a two-story, Cape Cod of wood construction. The home is partially insulated. The house heating plant was a gas forced air furnace. An "A" type coil was installed in the furnace plenum prior to the installation of the heat pump. The load presented by this house during the heating season was also insufficient for the full heat pump output; therefore, the input to the heat pump was reduced to about 60,000 Btuh.

In both homes the existing thermostats were used. Difficulties were experienced with the heating and cooling anticipator circuits, either causing the unit to short-cycle or allowing too much swing in the house temperature. Eventually an interface circuit was built which corrected these problems. Control of the indoor blower was based only on temperature of the water leaving the indoor coil to the house. The summer and winter comfort levels experienced were equivalent to conventional methods of space conditioning. Due to the large area of indoor coil, a considerable amount of dehumidification was accomplished during summer air conditioning.

In general, the experience in the two homes paralleled that of the life test units at Phillips Engineering. Many of the same problems occured at both locations. Because changes had been made to increase the house loads, neither of the two homes had historical energy consumption records or load information. The performance of PT-3 at 190 Hunter Drive, Benton Harbor is discussed in the Cycling Operation section of this report. The performance of PT-6 at 2210 Mount Curve, St. Joseph was not as good as PT-3.

Field and Life Test Experience With the Sealed System

Experience with the sealed system during field and life test falls into four major categories; sealed system integrity, flow and concentration controls, solution pumps and magnets, and fluid integrity. Knowledge about each of these categories was a product of running six units in various locations for about two years.

Integrity of Sealed System

Perhaps one of the most important areas concerning the reliability of the heat pump system is the integrity of the sealed system. The sealed system must prevent inert gases from entering the system, and it must prevent any loss or contamination of the working fluids. The PT unit field test program experienced problems in two areas concerning the sealed system integrity. The first area was with mechanical tube fittings, and the second was with welds on generators.

Leaks of inert gas into the sealed system were caused many times by tube fittings which did not seal properly. Mechanical fittings had been used to install rotameters and make changes during testing and to ease assembly. In part, those leaks were caused by the disassemblies during the testing phase. In general, this was not a serious problem, only a troublesome and time-consuming one.

A sealed system integrity problem of a more serious nature was that of cracked welds in the generator. PT units 1, 2, and 3 all experienced solution leaks due to cracked or porous generator welds. Some of these leaked solution into the combustion area of the burner causing the solution to burn.

The generator was the only component which had these problems with welds. This was probably because the generator experiences the highest heat fluxes and thermal stresses due to the larger temperature gradients it is exposed to during cycling operation. Leaks in other components were found during pressure and leak testing of the units. The generator weld problems were not found during pressure and leak testing they occurred in the first months of operation.

Corrections were made by using a different filler rod, heating the general weld area and having a more experienced welder make the repairs. No further problems occurred over the two year period. That welding procedure was then used for the rest of the generators. The component parts were first pre-heated before welding, and the more experienced welders were used for those joints. The changes apparently solved the problem of cracked and porous welds in the generator. PT units 4, 5, and 6 had no cracks or leaks in any of the welds associated with the generator.

Flow and Concentration Controls

The second category in the PT unit field and life test is the experience with the flow and concentration controls. The weak liquid flow control valve and refrigerant thermostatic expansion valve, along with the refrigerant and solution storage chambers, comprised the flow and concentration controls in the PT system. Slip/stick movement of the valves had been observed early in the laboratory development of the flow control valves. The problem was the subject of much attention and resulted in modifications of the commercial valves. Valve pins and pin carriers were Teflon-coated, and pin bearings were enlarged. Graphite sleeves were also used, and the problem appeared solved. However, in the PT program symptoms of slip/stick still occurred. The valve would not follow the temperature of the evaporator until re-adjusted or struck. That became a rather common problem with the PT unit field and life test. Eventually, the determination was made that slip/stick was not all of the problem, that another part of the problem was being caused by non-uniform motion of the power element diaphragm. "Oil canning" best describes what was found; the motion of the power element diaphragm on certain elements jumped from one position to another in a step-like manner. This is similar to slip/stick behavior but is not caused by friction. This kind of stepwise action of the diaphragm was demonstrated in the lab on a fixture which measured deflection of the diaphragm as the bulb was warmed and cooled. The manufacturer of the thermostatic expansion valves was consulted, and it was determined that during the welding of the power element, the diaphragm is always warped to some extent. This warpage cannot be avoided. It is not important in the normal R12 and R22 controls, but in the low pressure R133a system it has been found to be a serious problem.

The resolution of this problem will be to re-design the refrigerant thermostatic expansion valve. A bellows type valve was designed and built later for use in the prototype program, but the conventional refrigerant thermostatic expansion valves remained on the PT units. In spite of some resetting for winter and summer operation, the evaporator temperatures and solution flow rates of the Field Trial units were often found incorrect when checked, and undoubtedly affected performance.

Solution Pump and Magnetic Drives

The third major category under experience with the sealed system during the field and life test concerned solution pumps and magnets. A detailed discussion of solution pumps, including their performance during field and life tests, is contained elsewhere. Therefore, this section will deal with the magnetic drives for the pumps used in the PT field and life tests.

As previously reported, the motors for solution pumps were changed to optimized 0.4 hp motors. Those motors, and pump design, essentially eliminated any magnet decoupling of the Field Trial units. The magnets used were two-piece magnets cemented together and then cement-sealed into an aluminum housing. The anaerobic cement used had performed well in short operations, but since the refrigerant, R133a and the absorbent, ETFE, are organic solvents, the anaerobic cement used to hold the two-piece magnets together became swollen after months of operation and pushed the driven magnet into the stainless steel magnet cover. The spacing between magnet and cover is held to about .020 inches or less, so even a small amount of swelling can cause the magnet to rub against the cover. When the magnet begins to rub against the cover, it generates heat which tends to cause the pump to vapor lock. Eventually, the magnet will crack, and the pump will sieze. This happened to nearly all the two-piece driven magnets on the PT units. This problem was solved when a source for a one-piece driven magnet was finally found. The aluminum magnet holder was then shrunk-fit onto the magnet, using no cement at all. This eliminated the problem with the driven magnet.

The drive magnet also occasionally gave problems with spacing, rubbing against the magnet cover and cracking. The motor mount would not hold to the tolerances required. This problem was solved by putting standoff spacers on the motor to keep the magnet the proper distance from the magnet cover. Those problems with the drive and driven magnets were the only problems observed.

Fluid Integrity

The fourth and final category of interest during the field and life test of the sealed system was fluid integrity. Since the absorption pair being used was organic, there was concern with long-term stability during extended hours of use under actual operating conditions. Fluid samples from running units were periodically analyzed to check for any evidence of decomposition. A good indication of fluid decomposition was found by the Buffalo Laboratory to be the color of the fluid sample. The fluids, as received, are water white in color; as soon as they are run in a heat pump for a few hours, they turn a light straw color. The color seems to be an indication of time at a particular temperature. The higher the temperature and the longer the time, the darker the sample gets. Although no actual analytical connection between color and decomposition has been found, color is used as a quick check of the fluids.

Field Trial unit PT-2 had the most operating hours, and the fluids were sampled periodically and analyzed. Table 4-9 contains the results of this analysis.

TABLE 4-9

ANALYSIS OF R-133A/ETFE SOLUTIONS FROM PT-2

OPERATING TIME (HOURS)	IMPURITIES (%)
3319	1.2
5152	0.8
7688	1.0
10,141	1.1
ETFE as received	1.3
R-133a as received	0.9

Clearly, the level of impurities in the solution after 10,000 hours of operation remained small. Visual observation of the color of the solutions also tended to confirm the analysis. All solutions were a light straw color. Even the pure components, as received, show the same levels of impurities as the fluids which have been run for over 10,000 hours. The stability thus indicated may be due in part to the installation of standard refrigeration filter-driers in the weak solution circuit. They were installed primarily to dry the ETFE which is somewhat hygroscopic, but apparently had additional value.

During the field and life test program, there were two incidents in which units overheated and fluid temperatures exceeded 400°F. The first incident occurred with PT-3 after a brief electrial power outage, which caused either a solution pump magnet decouple or a vapor lock from which the unit did not immediately recover. The generator temperature, as indicated by the chart recorder, was above 400°F for almost an hour. Eventually, the unit recovered and returned to normal operation. Inspection of the fluids showed some darkening. The weak liquid filter/dryer and the refrigerant filter/dryer were changed, and the solution returned to the normal light straw color. The filter/dryers used during the program thus appeared to do more than just filter particulates and remove water; they had the ability to lighten the color of the solutions once the solutions had become darkened, and perhaps also limited decomposition.

The second case of unit overheating was on PT-2 at 13,185 hours of operation. The circumstances were similar in that a power outage had occurred. This time, however, it appeared as if the gas valve had remained open while the rest of the unit had been shut off. The generator temperature had peaked at 520°F before starting to cool down. This time, the over-heating appeared much more severe. The solution was dark brown in color and had brown/black particulate matter dispersed in it. As a result, the fluid was drained from the unit, and the unit was flushed with clean solution. Fresh ETFE and R-133a were used to recharge PT-2, and the unit was put back into operation.

A solution sample from the overheated soluton was filtered and analyzed for impurities. Surprisingly, only 2.24% impurities were found in the sample after it had been filtered. No analysis was made of the particulate matter, but it was small in quantity.

With those exceptions, fluid decomposition was not a problem during the operation of the PT units. The filter/dryers were a considerable help in keeping the fluids only lightly colored throughout the 34,000 hours of total operation. Even under severe overheating, the organic fluids held up well, with impurity levels less than 3% in the liquid phase.

Field and Life Test Experience With Electronic Controls

The operation of the electronic controls during the field and life test program provided indications of weather conditions that caused problems with the operation and control of the unit, as well as indicating what sensors were in need of improvement. During the laboratory test phase of the program, several areas for improvement in the controls were identifed and the improvements were undertaken.

Some of the most troublesome and difficult-to-find problems occurred in the controls and sensors during outdoor operation. In general, the first generation control system using switching triacs and separate optical isolation gave the most problems. The addition of mechanical relays helped, but the system was still susceptible to electrical noise and erratic operation. PT-1 and PT-2 operated with this modified first-generation control system. Later PT units used solid state relays with built-in optical isolation and transient suppression, which greatly increased their reliability.

The component which had the most reliability problems was the commercial gas valve/electronic ignition assembly. Several recurring problems were experienced. The 24-volt transformer used for the gas valve assembly burned out on several units. The proof-of-pilot sensor did not always indicate the presence of a pilot flame when there was one. The high voltage spark cable often lost voltage to the base or surrounding parts, due to moisture or snow, causing a weak or non-existent spark. During severe blowing snow conditions, the gas valve got ice in it and would not open properly. These problems indicate the need for improvement of the gas valve/electronic ignition assembly or design of a special housing in that part of the heat pump.

Reliability problems were experienced with the generator level sensor, generator overtemperature control and the magnetic decouple sensor during the outdoor field and life tests. These sensors were the subject of redesign, and in some cases, entirely new approaches were used. The results of the work on these sensors was subsequently incorporated into the prototype unit program.

The electrically heated level sensor proved to be of uncertain reliability under the use conditions. Heaters and fuses were burned out, and the electronic interface was difficult to adjust. The operation of the level sensor was satisfactory part of the time. On at least two occasions, the level sensor was found to operate properly when pump problems occurred on PT-2.

The original overtemperature control was a commercially-available oven thermostat. This control was difficult to set and was found to corrode due to its location around the top of the fins in the boiler section. A later development used a thermocouple in a well in the generator shell as an overtemperature control.

The first-generation magnetic decouple sensor was a reed switch mounted near the drive magnet. The second-generation magnetic decouple sensor was a solid-state, Hall-effect sensor, which provided more reliability and sensitivity.

The operation of the defrost sensor system under actual conditions, for the most part, gave very satisfactory results. Several defrosts of units on the roof of Phillips Engineering were observed during snowstorms when heavy, wet snow would accumulate on the outdoor coil. The need for defrost in this snow belt area of Michigan during operation in homes and even on continuous-run life test was found to be very low, probably due to the large coil area of the outdoor coil. Only one problem was encountered with the defrost system on PT-3, during an ice storm where large amounts of freezing rain accumulated on the coil. The amount of heat available in the hot water to accomplish defrost was not enough to completely remove the thick layer of ice on the coil; consequently, the unit did not come out of the defrost cycle and remained inoperative. A change was made in the TTL system to allow for this possibility and instruct the unit to return to the heating cycle until enough heat was regenerated to complete a difficult defost.

The sail switch, which indicated whether or not there was draft in the fan compartment, occasionally become clogged with snow or ice and had to be cleaned out manually. Its location was changed on several units, in order to prevent that from happening.

Field and Life Test Experience With Auxiliary Components

A considerable amount of new information was added to our knowledge of the auxiliary components through the field and life test program. These components were exposed to conditions that we were unable to simulate in laboratory testing. This new information was used to redesign and improve the auxiliary components whenever possible.

Installations of units in the two homes emphasized the need for an indoor coil of sufficient capacity. The temperature of the water returning to the unit should be as low as possible, preferably not greater than 95°F. In one location, the indoor blower speed was increased in order to reduce the temperature of the water returning to the unit. Indoor and outdoor coils directly affect pumping power requirements for the water pumps. The need for more efficient water pumps became evident during the field trial. Water pumps were also subject to leaks on occasion, which were mostly due to shaft seal problems.

The eight-way valve was subjected to the full range of outdoor ambients and conditions. Both cross leaks and external leaks were encountered on the eight-way valve. The underlying cause of most of the leaks was O-rings. Various types of O-rings were tried with varying degrees of success. Some made it difficult to rotate the shaft, and others took a set and would not seal well. It was also discovered that the I.D. of the body of the eight-way valves varied somewhat, and that this could cause cross leaks. This was solved by boring out the valve body to a uniform inside diameter. However, the selection of completely suitable O-rings for the eight-way valve remained incomplete.

In a somewhat unusual problem, frost accumulation was observed on the fan grill in cold ambients with little or no frost on the coil. It seems that at some temperatures below freezing, the air would be cooled to the point of forming frost. Mixing with the moisture in the flue gases, it would then freeze on the grill. In some cases the frost would form on the inside, trailing edges of the fins while the exterior remained clear. A large enough area of the grill or coil became covered with ice in a few cases so that the flow of air became restricted, and the sail switch would stop the gas. This happened on two of the Field Trial units on the roof of Phillips Engineering. A trial was made, without success, of coating the fan grill with Teflon to prevent the adherence of frost to the grill. The effect of the flue gas exhaust can be reduced by spreading and thoroughly mixing the flue gases with the air drawn through the outdoor coil. The delayed frosting of air going through the coil may require further investigation. It occurred only at temperatures close to 0°F.

The design of panels for the PT units was shown to be important through several blizzards during the field test. Panels which were not tight-fitting and those that had too much inlet area for combustion air allowed snow to penetrate to the inside of the unit and accumulate around the gas valve and burner venturi, clearly not an acceptable situation.

PT-1 Tear Down

After 7,913 hours of operation, PT-1 was removed from service and taken off the roof of Phillips Engineering for destructive analysis. The unit was disassembled, and the major component parts of the sealed system were cut apart and visually inspected. There were two main areas of concern, the working fluid side and the glycol/water side. Each side was inspected for corrosion of the aluminum and/or deposits which would inhibit heat transfer. The working fluid side was also inspected for any evidence of fluid decomposition.

The results of the visual inspection of the major components were very positive. On the glycol/water side, no evidence could be found of any corrosion whatsoever on any of the components; the surfaces were clean and free from any deposits which would inhibit heat transfer. This result means that the inhibitor systems in the "Prestone", ethylene glycol automotive antifreeze were very suitable for the 1100 and 3003 grades of aluminum used in the heat pumps. Inspection of the working fluid side of the major components revealed no evidence of corrosion or deposits. Evidence of decomposition of the working fluids was found in only two areas of the generator. The first was black tar-like deposits around the well which held the thermistor heater for the level sensor. The second area in which similar deposits were found was in the lower end of the boiler section of the generator where the design included a small vapor space, one wall of which was the finned outer wall of the boiler. Aside from these two small areas, no other evidence of fluid decomposition was seen. The electric heater had been left out of later units; the vapor space can also be eliminated from the design. In general, the destructive analysis showed that the materials of construction, working fluids, ethylene glycol solutions, and the designs of the major components can be integrated without adverse consequences.

Conclusions

The PT unit field and life testing phase of this program placed six, organic fluid, absorption heat pumps in various field locations where they were exposed to a full range of northern climatic conditions. The purpose of the field test was to encounter as many problems as possible with the real world operation of the units. A considerable amount of experience was gained during the 34,263 hours in which the PT units were operated.

The results indicate that organic fluid, absorption heat pumps can be developed for field use to a state of high reliability. No insoluble problems were encountered throughout the field test program. The areas in need of further development were identified, and in many cases, this development was carried out where it was feasible. The problem areas were found to be primarily in the auxiliary components. Other than the flow control valves, no reliability problems were found on the absorption sealed system components.

CYCLING OPERATION

Early in the PT unit program, the cycling characteristics of organic fluid, absorption heat pumps were investigated. It was recognized that, in addition to high steady-state efficiency, cycling characteristics were extremely important to seasonal efficiency. Much heat energy goes into heating or cooling components, and the fluids in them, to their operating temperatures, and into separating the solution into the optimum concentrations and quantities of weak and strong solutions and pure refrigerant. If those operating conditions are then lost, or rapidly degraded, during the "off" portion of a cycle, the efficiency of cycling operation can be much lower than at steady state. Development under cycling test conditions was therefore important to yield the best cycling efficiencies possible. The first tests of unit behavior during cycling were run on Test Unit #1. The results of these tests were used to improve the PT units. Later, cycling tests were run using PT-5 in controlled ambients. Those tests allowed for further refinement of the control system.

The first investigation of cycling behavior on Test Unit 1 was made to confirm the need for valving in addition to the flow control valves. The Test Unit was equipped with manual controls, except for the refrigerant thermostatic expansion valve and the weak liquid expansion valve. Sight glasses in major components and thermocouples were used to determine what was happening. Normally, when the gas was turned off, the rest of the unit was allowed to continue for a short period to prevent the boiler temperature from rising due to accumulated heat. Then the pumps and fan were turned off. It was seen that weak liquid from the generator continued to flow to the absorber through the liquid heat exchanger because the weak liquid flow control valve opening was a function of the low-side pressure. Also, rich solution would flow backwards through the dynamic pump. The build-up of liquid in the bottom of the absorber might flow to the bottom of the evaporator and heat the bulb of the refrigerant expansion valve causing it to waste refrigerant liquid into the evaporator. Thus much of the refrigeration capacity generated during the "on" part of the cycle could be wasted, and cycling efficiencies could easily be much lower than steady state. To correct the problems, solenoid valves were installed in the refrigerant and weak solution lines, and a check valve was placed in the pump discharge line.

Test Unit #1 was then operated at room temperature with those added valves. The solenoid valves were wired to close when the pumps were turned off. In the tests, the unit was brought to steady-state operating conditions before the cycles were started. During the cycling operation, the gas turn-off was followed by a spin-down period of two minutes before complete shut-off. Some of the energy accumulated in the system during the "on" period was thus recovered before shut-off.

The unit was then left off for varying "off" times (5, 10, 20 minutes). After each "off" period, the unit was run for 20 minutes during which the instantaneous heating and cooling capacities were measured every two minutes. Those capacities, including the spin-down heat output, were then compared to steady-state capacities. The following table summarizes the results.

TABLE 4-10

TEST UNIT 1 - CYCLING TESTS

TEST CONDITIONS	TIME FROM START	HEATING CAPACITY AS % OF STEADY STATE CAPACITY	
		RATE AT END OF PERIOD	AVERAGE OF PERIOD
Steady state established,			
2-minute spin-down,	6 min.	91.4	(94.1)
5-minute shut-down,	12 "	96.7	(94.8)
unit then started and	18 "	98.4	(95.4)
run for 20 minutes.	20 "	98.4	95.5
At end of above 20-minute			
test, 2-minute spin-down,	6 min.	90.4	(94.8)
10-minute shut-down, unit	12 "	94.2	(94.2)
then started and run for	18 "	97.0	(94.7)
20 minutes.	20 "	97.4	94.8
At end of above 20-minute			
test, 2-minute spin-down,	6 min.	87.9	(92.5)
20-minute shut-down,	12 "	92.8	(92.0)
unit then started and	18 "	96.3	(92.9)
run for 20 minutes.	20 "	97.1	93.1

The column labeled as End of Period represents instantaneous measurements at the various periods during the "on" cycles. The column labeled Average of Period is the integrated capacity based on the ten instantaneous heat outputs measured during the twenty minute period, plus the heat output during the spin-down period. The data at 20 minutes of operation include the actual spin-down quantity. The figures in parentheses include estimates of the spin-down energy at those times.

The results were encouraging, indicating that much, if not all, of the potential waste during cycling had been prevented. These test results, obtained on a test stand at room temperature and other favorable conditions, might be better than would be obtainable on a unit operating outdoors, however.

Field Trial Unit # PT-5 was chosen for further investigation of potential cycling performance. Most tests were run in a 47°F ambient temperature room. The heat outputs were measured by an integrating Btu meter calibrated for the application. In addition to the solenoid valves, the unit was also equipped with a flue damper to reduce generator losses during the "off" periods.

A number of the factors affecting cycling losses were studied, but the investigation could not be comprehensive due to program limitations. The time required to use up the liquid refrigerant wetting the evaporator coil was first determined by monitoring the temperature at the thermostatic expansion valve bulb at the outlet of the evaporator. It was determined to be 3 min. 45 sec. That valve was then used as the spin-down time in subsequent work. Tests to time the sequences of fan and blower start-ups and shut-downs, flue damper opening and activation of solution pump, solenoid valves, gas valves and water pumps were made at cycles of 20 minutes "on" and 10 minutes "off" operation. Data were then run at a few other conditions. A number of the test results are given in Table 4-11.

TABLE 4-11

PT-5 CYCLING TESTS IN HEATING MODE

ROOM TEMPERATURE °F	CYCLE MIN. ON/MIN. OFF	PERFORMANCE % OF STEADY STATE OUTPUT
47	20/10	96.1
47	20/20	95.2
47	10/10	91.2
47	10/20	88.1
57	20/10	94.5
32	20/10	90.0

These results indicate that there is a good probability that units can be controlled to operate at cycling efficiencies above 90% under cycling conditions similar to those tested. The cycling efficiency is indicated to be dependent on the length of both the "on" and the "off" periods. For the best results, "on" periods of 15 to 20 minutes may be necessary. Lengthening "off" periods reduces the cycling efficiency. Since much of the operating time will be under mild conditions, control sequences for long "off" periods will probably have to be developed. It would appear likely that the spin-down period would have to be lengthened to use up the refrigerant and weak solution rather than risk loss by diffusion.

Much more investigation of absorption unit cycling is obviously necessary. The long "on" times combined with long spin-down times are likely to be unsuitable for many homes. Hence, design improvements will be needed to make possible high cycling efficiencies with shorter "on" times. The improvements needed will relate to the overall heat pump design, as well as the control system.

REFERENCES

1. Hainsworth, W. R. "Refrigerants and Absorbents," Part I, Refrigeration Engineering 48, 97-100 (1944); Part II, ibid. 48, 201-05 (1944).

   "Absorbent-Refrigerant Combinations," Servel Report. Evansville, Ind. Servel, Inc., 1943 (January 23). Unpublished Company Report.

   Zellhoefer, G.F., "Solubility of Halogenated Hydrocarbon Refrigerants in Organic Solvents," Ind. Eng. Chem. 29, 548-51 (1937).

   Zellhoefer, G.F. and Copley, M.J., "Heats of Mixing of Haloforms and Polyethylene Glycol Ethers," J. Am. Chem. Soc. 60, 1343-45 (1938).

   Zellhoefer, G.F., Copley, M.J. and Marvel, C.S., "Hydrogen Bonds Involving the C-H Link," J. Am. Chem. Soc. 60, 1337-43 (1938).

   "New Working Pairs for Absorption Processes", Proceedings of Berlin Workshop, 1982, Wiktor Raldow, Editor, Spangbergs Tryckerier AB, Stockholm 1982.

2. "Development of a Residential Sized Gas Fired Absorption Heat Pump," Gas Research Institute Annual Report for 1979, Chicago, IL.

3. Sherwood, Thomas K., Pigford, Robert L. and Wilke, Charles R., "Mass Transfer, McGraw-Hill Chemical Engineering Series," McGraw-Hill Book Company, New York, 1975, pp. 184-188.

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