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ENERGY CONSERVATION

MASTER

RESEARCH and DEVELOPMENT of a HEAT-PUMP WATER HEATER

VOLUME 2
R & D TASK REPORTS
AUGUST 1978

Work Performed by
ENERGY UTILIZATION SYSTEMS, INC.
for
OAK RIDGE NATIONAL LABORATORY
operated by
UNION CARBIDE CORPORATION
for the

U. S. DEPARTMENT OF ENERGY

Division of Buildings and Community Systems

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R&D TASKS REPORTS

August 1978

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Prepared under Subcontract 7321 for the

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for the

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FOREWORD

This is a report of work performed to date by Energy Utilization Systems, Inc. (EUS), covering Phase I of the research and development of an electric heat pump water heater on which United States letter patents for prior work have been applied for. The work is being sponsored by the Buildings and Community Systems Division of the Department of Energy (DOE) through the Oak Ridge National Laboratory (ORNL).

This Volume 2 contains the final reports of the three major tasks performed in Phase I.

- In Task 2, a market study identifies the future market and selects an initial target market and channel of distribution, all based on an analysis of the parameters affecting feasibility of the device and the factors that will affect its market acceptance.
- In the Task 3 report, the results of a design and test program to arrive at final designs of heat pumps for both new water heaters and for retrofitting existing water heaters are presented.
- In the Task 4 report, a plan for an extensive field demonstration involving use in actual homes is presented.

Volume 1 contains a final summary report of the information in Volume 2.

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ABSTRACT

The heat pump water heater is a device that works much like a window air conditioner except that heat from the home is pumped into a water tank rather than to the outdoors. The objective established for the device is to operate with a Coefficient of Performance (COP) of 3 or, an input of one unit of electric energy would create three units of heat energy in the form of hot water.

With such a COP, the device would use only one-third the energy and at one-third the cost of a standard resistance water heater. Normal water heating costs average $200 per year and run as high as $700 or more. Thus savings using the heat pump would run between $133 and $467 per year. At an estimated cost of the device between $200 and $250 more than for an electric resistance water heater, the payback period for many applications will be between one and two years with some being less than one year.

In warm climates, the economics is favored by the warmer ambient air and by the value of free air conditioning and dehumidification provided while the unit is operating. In cold climates, the efficiency is favored by the colder supply water and the higher operating savings from higher kilowatt-hour use because of the colder water tend to offset the effect of the less favorable climate. This latter is the more important because, given a minimum average COP of 2, the payback period is more dependent on the amount of energy consumed and the price of electricity than it is on the level of COP achieved.

In laboratory tests, prototype units have achieved an operating efficiency, E_R, of 2.5 in average conditions of 70°F ambient air and 55-60°F supply water. With losses taken into account, the Coefficient of Performance is 2.8 or within ten percent of the design objective. Separate heat pump designs are available for new water heaters and for retrofitting existing ones. For both models, the compressor, evaporator, fan and controls are mounted in a round cabinet set on top of the water heater. The condenser

is a dual tube direct immersion type which enters the tank through a special four-inch hole in the top of new tanks. For retrofit units, the condenser is in the form of a helix and is screwed into the tank through the hole normally used by the lower resistance element.

The potential market for the heat pump water heater is estimated as growing to seven million units per year by 1985. Recognizing that potential will be constrained by restraints on electric utilities from state regulatory agencies, the introduction of a new technology (refrigeration) into the normal water heater sales and service organizations, physical installation limitations and possible labor unit jurisdictional disputes.

To help overcome those constraints and to assure the device does not suffer at the start from misapplication and poor installation and servicing, the device will be marketed initially through merchandising electric utilities. Twenty of these utilities will participate in Phase II of the development -- the field demonstration of one hundred units. Each utility will purchase, install and service five units, and install, service and monitor instrumentation packages supplied by DOE. Instrumentation will be designed to determine the annual COP of the unit including allowance for its impact on the heating and cooling load of the house.

CONTENTS

RESEARCH AND DEVELOPMENT

OF A

HEAT-PUMP WATER HEATER

TASK 2

MARKET STUDY

JUNE 1978

Prepared under Subcontract No. 7321 by

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365 Plum Industrial Court
Pittsburgh, Pennsylvania 15239

for

Oak Ridge National Laboratory

Operated by

Union Carbide Corporation

Oak Ridge, Tennessee 37830

for

The United States Department of Energy

Contract No. W-7405-eng-26

ABSTRACT

The rising cost of electricity coupled with the shortage of energy is focusing new attention on the high cost and energy use of electric water heating. The electric heat pump water heater could use as little as one-third the energy of the resistance type with a resulting payback period measured in months and a significant reduction in total energy use in the United States.

The degree to which these are true will depend on the performance of the device, the potential market and the technical and institutional constraints that will impede its marketability.

This report examines the various parameters that affect the performance, including climate, type of house, location of the unit in the house, supply water temperature and delivery water temperature. The report estimates probable payback periods which are measured in months in homes with above average electricity costs and hot water consumption. The major constraints that must be overcome include restraints on electric utilities from state regulatory agencies, the introduction of a new technology into the normal water heater sales and service organizations, physical installation limitations and possible labor union jurisdictional disputes.

The report estimates a potential market for the heat pump water heater growing to seven million units per year by 1985 and derives an initial target market of rural and suburban single family homes served by merchandising utilities.

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CONTENTS

LIST OF TABLES

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LIST OF FIGURES

1.0 Introduction

Domestic hot water heating is usually the second largest user of energy in the home. In a typical suburban or rural house with an electric water heater, water heating uses between five and ten thousand kilowatt hours. With electricity costing between three and seven cents a kWh, the annual cost is between 150 and 700 dollars.

The heat pump type of water heater, with an average seasonal Coefficient of Performance (COP) of three, would reduce the kWh use, and thus the cost to one-third of these levels. At a projected installed cost of $200 more than for a conventional electric water heater, the payback period would be one year or less in most cases.

The heat pump water heater works much like a window air-conditioner except that it pumps heat into the water tank rather than outdoors and is expected to pump it from otherwise waste heat in an unconditioned space such as the basement, garage, utility room or attic.

The idea of a heat pump water heater is not new. Two firms built them in the late fifties but due partly to the low and still declining cost of electricity, they did not sell well and were taken off the market. Now with the cost of electricity escalating rapidly, the heat pump water heater has become economically feasible in many, if not all, areas of the country. It is the objective of this study to identify those areas where the device is technically and economically feasible and under what conditions, derive the potential market, examine the technical and institutional constraints that will impede its marketability and select a most feasible market for initial targeted marketing. The results of the study are summarized in the following section.

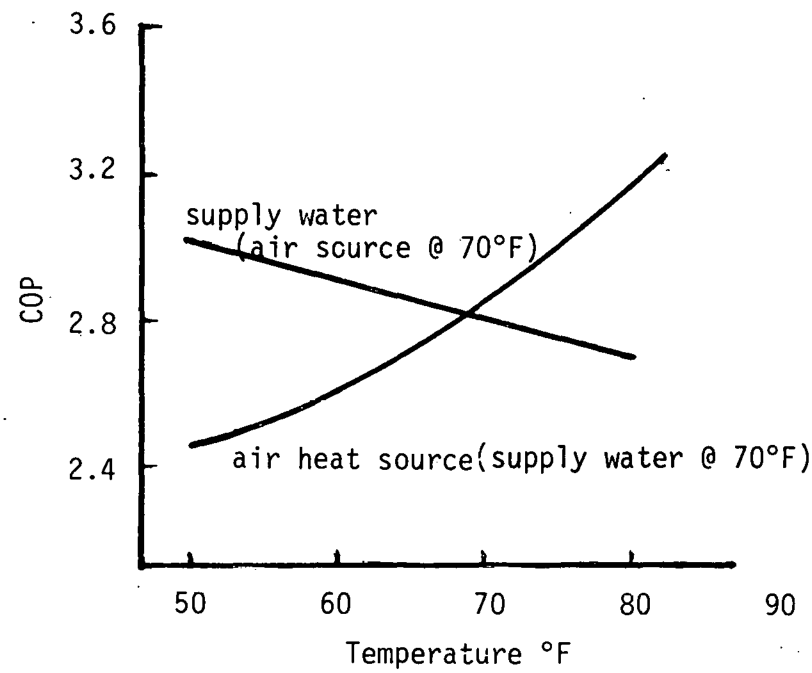
2.0 Summary

The heat pump water heater achieves its greatest feasibility in areas where the cost of electricity is high and in homes where hot water consumption is also high. These are the factors to which the payback period is most sensitive, but their effect is amplified in those areas where the coefficient of performance is high. The Southern portion of the country plus other areas of the country such as the Pacific Northwest and Hawaii are good locations because of year-round moderate climates.

Also in the South, the device could provide the fringe benefit of air-conditioning and dehumidification for most of the year. For installations providing this benefit this represents a savings of $75 a year or one-third the installed cost of the device in addition to saving two-thirds of the cost of water heating. The same principle would be true in the summer in the North. But in the winter, the heat pump would be taking heat from the house that would need to be replaced by the home heating system. While that would be true to some degree regardless of where the unit is installed in the house, it would be particularly true where the unit is installed in a conditioned space such as a heated basement or utility room. But if the unit is installed in an unconditioned space the unit could get much of its heat from the ground or outside air flowing through the basement walls and floor, losses from the furnace and/or ductwork or other interior heat gains. In a worst case analysis where no waste heat is available, the water heater could be switched to resistance heating in the winter with its COP of one. Assuming equal seasons and a summer COP of 3, the annual average COP would then be 1 1/2. A further consideration is that in the North, supply water is colder than it is in the South. This improves the efficiency of the heat pump and offsets some of the disadvantage of the colder climate. This is demonstrated in Figure 2.1 which shows curves of COP versus ambient air temperature and supply water temperature obtained in laboratory tests.

The colder water supply also requires more energy for water heating. This increases the potential for operating cost savings and thus for lowering the payback period. As shown in Figure 3.8 daily energy consumption is increased by 27% going from an average of 70°F in the South to 50°F in the North.

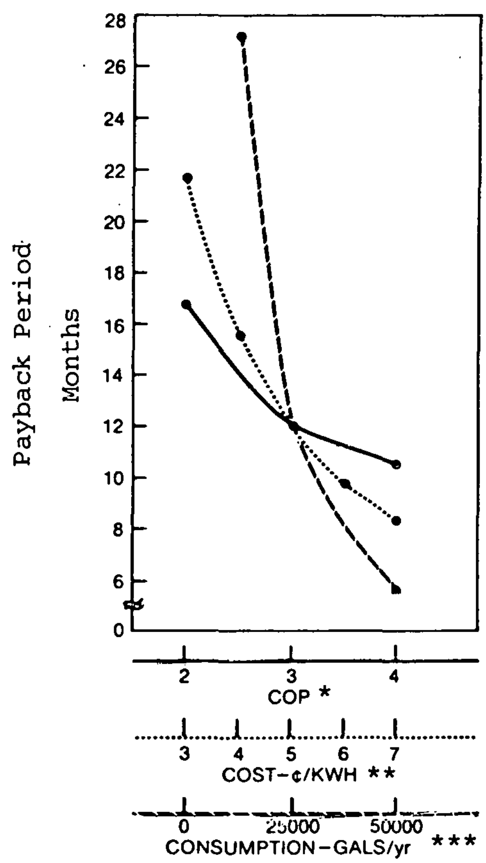
Figure 2.1
COP vs Supply Water and
Heat Source Temperatures



Although the relative value of the heat pump in different climates is not yet determined, the primary criteria, the payback period, is relatively insensitive to the COP. At least it is not as sensitive to COP as it is to the cost of electricity and to the amount of hot water consumed. (see Figure 2.2) In the case of the COP, the payback period (which is measured as the ratio of the additional installed cost to the monthly operating savings, including maintenance costs, compared to resistance type water heating) increases 57% between a COP of 4 and one of 2. These are practicable limits for the device. In the case of the cost of electricity, the payback period increases 161% between the limits of seven and three cents per kilowatthour. For hot water consumption, payback period increases by 110% between 50,000 gallons per year which is representative of a typical rural or suburban family and the national average of 25,000 gallons per year. Thus many homes in the North where hot water consumption and electric rates are both above average will prove to have high potential even if the colder climate does effect the COP.

Figure 2.2

Payback Period vs COP, Cost of Electricity & Hot Water Consumption for New Installations



* Cost = 5¢/kWh; Consumption = 25000 g/yr
** COP = 3; Consumption = 25000 g/yr
*** COP = 3; Cost = 5¢/kWh

With thirty million electric water heaters now in use there would appear to be a large market for a heat pump for retrofitting these existing units. But the majority of these are either too old or too small to justify retrofitting or are eliminated by some accessibility or space limitation. Thus the estimated potential for retrofitting existing water heaters is 2,400,000 units. This is derived from sixty percent being less than 52 gallon or too small to retrofit, fifty percent being more than five years old and sixty percent being located in inaccessible places.

Over the next five years, more than three million electric resistance type water heaters will be installed each year. Most of those will then represent retrofit potential, a potential that will be enhanced as the price of electricity continues to rise, shortening the payback period. Existing water heaters, gas, oil and electric, are being replaced at the rate of 3,500,000 units per year. These represent a much greater potential for replacing with a heat pump type electric unit than they do for retrofitting.

New units for new dwelling construction represent the most immediate potential as large home builders adopt the heat pump water heater and use it as a selling point because of its low operating cost.

Aggregating the above market segments shows a potential market growing to seven million units by 1982, with initial market penetration in those areas served by merchandising utilities.

Realizing the potential will take much longer because of many technical and institutional constraints that will impede the market acceptance. The most significant is the need to train plumbing installers who normally install water heaters to apply, install and service a refrigeration system. Without that training, customer complaints would be multiplied and the device would get a bad image at the start.

To reduce that possibility, electric utility people, each of whom will have people trained by the manufacturer, will participate in a demonstration program involving one hundred units purchased in groups of five by twenty utilities. The units will be installed in homes and monitored for one year to determine actual performance in various climates, house types, etc.

Eighty units will be complete with new 82 gallon tanks. Twenty units will be for retrofitting existing 82 gallon electric water heaters. The initial target market will be rural and suburban single family homes in areas served by merchandising utilities or utilities who will select a reliable dealer, train dealer personnel and supervise the application, installation and servicing of heat pump water heaters for some initial marketing period.

After enough field experience is gained to create a comprehensive application, installation and service training program, personnel throughout the normal channels of distribution of water heaters will be selected for training in this new technology.

3.0 Parametric Analysis

The economic benefit of the heat pump water heater will be a function of:

- Its efficiency versus that of an electric resistance water heater
- The magnitude and daily profile of hot water consumption
- The price of electricity
- The added investment above that for an electric resistance water heater
- Maintenance costs
- Beneficial space conditioning

Each of these is, in turn, a function of some number of variables which makes it difficult at this point to predict an ultimate cost/benefit ratio for the device.

In this section, the major variables are analyzed using empirical data where they exist and postulated data where empirical data do not exist. The aim of the analysis is to quantify -- at least on some relative basis -- the sensitivity of each on the ultimate costs or benefits as measured by the number of months to payback the original investment.

The most significant variables include:

- Efficiency of the heating system
    - Coefficient of performance of the heat pump
    - Ambient air temperature
    - Input water temperature
    - Delivery water temperature
    - Availability of waste heat
- The quantity of hot water consumption
    - Family size
    - Appliance saturation
    - Supply water temperature

- Fuel prices
  - Projected increases
  - Rate structure

3.1 Heating System Efficiency

The primary reason for developing the heat pump water heater is the belief that its operating efficiency will be substantially higher than that of an electric resistance type water heater. These efficiencies, which are expressed as E_R, are based on an NBS developed test for conventional water heaters which cycles test units over a 48-hour period simulating average family use. The E_R equals the ratio of energy content in the hot water drawn during the test and the electrical energy input.

The COP of the heat pump water heater, as used in this study, is derived from the ratio between the E_R of the heat pump water heater and the E_R of the resistance water heater. It is this ratio, or COP, that will determine the comparison in energy use and operating cost between the two systems. Fortunately, recent test work has provided evidence for clarifying the efficiency of electric water heating. So there is now a more accurate base from which to derive the COP of the heat pump water heater as explained in this section.

3.1.1 Efficiency of Resistance Water Heaters

Electric resistance type water heaters are considered 100 percent efficient in converting electricity to heat. Estimates of jacket losses have long been in the range of 5 and 10 percent of total annual energy use, or the device has been said to be 90 to 95 percent efficient. But if the losses are constant and only depend on the tank wall construction, the average temperature of water in the tank and the ambient temperature surrounding the tank, then the losses when expressed as a percent of total annual use will vary inversely as the total annual use. Expressed as an equation:

$$\text{Water heater efficiency} = \frac{\text{Annual energy consumption - losses}}{\text{Annual energy consumption}} \times 100$$

Because the three variables affecting losses are cumulative, the percent efficiency should vary more than between the narrow limits of 90 to 95%. This has been demonstrated in the EUS laboratory where tests on two conservation models showed an average loss of 447 Btu/hr at a tank temperature of 135°F and an ambient temperature of 75°F. Extrapolated to 140°F in the tank, the loss is 484 BTU/hr or 1242 kWh per year. total annual electricity use in a resistance type water heater is 5,000 kWh the efficiency as shown in Table 3.1 for the conservation model would be:

$$\frac{5000 - 1242}{5000} \times 100 = 75\%$$

Tests have also been run in the EUS laboratory on standard models. The efficiency for an annual use of 5000 kWh is 70% as shown in Table 3.1. If total annual electricity use is 10,000 kWh, the efficiency for the conservation model would be improved from the 75% to:

$$\frac{10,000 - 1242}{10,000} \times 100 = 88\%$$

Recent research done by D.E. Spann, Jr.(1) at the Oak Ridge National Laboratory found the efficiency to be 78% for a standard-type 66-gal. unit supplying 70 gal. of 150°F water per day. Here too, the efficiency is seen to increase as usage increases as shown in Figure 3.1. Interpolation of other Spann data indicates a heat loss of 1472 kWh/yr when water is heated to 140°F in a 70°F ambient.

Similar results have been obtained by Jameson(2) who tested three 52-gal. water heaters. A standard unit showed a test loss equal to 1518 kWh/yr with 150°F water and 76°F ambient air. This would equate to 1313 kWh/yr if water is heated to only 140°F. On the same basis, a conservation model would lose 1073 kWh/yr.

In Table 3.1, results of the test programs are normalized to 140°F delivery temperature and 75°F ambient temperatures.

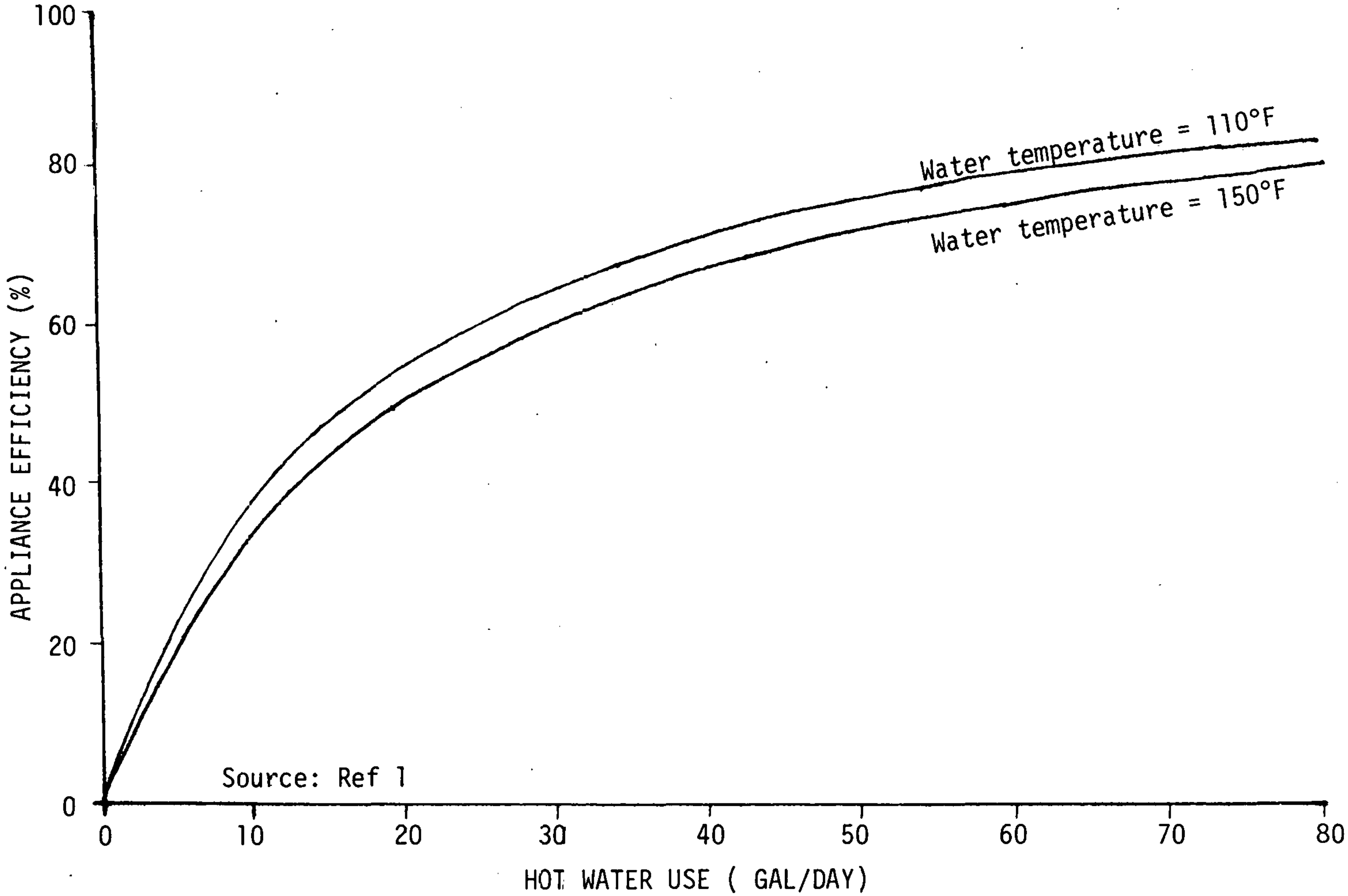


Figure 3.1 Appliance Efficiency of Residential Electric Water Heater (66 gal, 2" insulation)

TABLE 3.1

Yearly Loss of Electric Water Heaters
at 75°F Ambient and 140°F Water Temperature

Test	Size (Gals)	Standard Model kWh/yr	E_R*	Conservation Model kWh/yr	E_R*
EUS	82	1515	70%	1242	75%
Spann	66	1367	73%		
Jameson	52	1333	73%	1090	78%

*Based on total consumption of 5000 kWh/year

It should be pointed out that the three test programs are not completely comparable because (1) the tanks used are different sizes and (2) Jameson and EUS measured standby loss which would be slightly greater than losses from a tank where a temperature thermocline exists much of the time as hot water is drawn from the top and cold water comes in the bottom. But apparently the two differences offset each other or are insignificant.

3.1.2 Efficiency of Heat Pump Water Heater

Because the net efficiency of a heat pump water heater is a function of several variables as listed previously, estimating the performance to be expected of any one application would be highly conjectural. As the tests now running in the EUS Laboratory enable us to select an optimum design and then test some of the variables, we will be able to establish correlations from which estimates can be made with considerable accuracy.

However, based on test results to date coupled with some assumptions, some preliminary conclusions can be drawn.

In Table 3.2 are representative results of cycling tests involving one draw each of 5, 15 and 25 gallons. This is done after the water tank has been brought up to temperature. The test is designed to simulate the normal in-service operation of the heat pump water heater. A compressor cool-down period between the end of one draw and the start of another is used to include the effect of the lower efficiency during the starting warm-up period.

TABLE 3.2

Heat Pump Water Heater Performance from Eight-Hour Cycling Tests

	25 Gal.	15 Gal.	5 Gal.	Total
Output - BTU	14,438	8,663	2,888	25,989
Input - BTU	5,160	3,280	1,501	9,941
E_R	2.8	2.64	1.92	2.61
Losses - BTU				4,340
Output Plus Losses - BTU				30,329
COP				3.05

The E_R is calculated as the ratio of output BTU to input BTU. Output is calculated as the number of BTU required to heat each draw from the source temperature of 70°F to the delivery temperature of 140°F. The three tests are then aggregated and an average E_R of 2.61 is the result. But that does not account for losses during the eight-hour period. They are calculated for each test based on the top two-thirds of the tank remaining constant at 140°F and the average of the bottom third being the reading at 1/6 from the bottom, which in this case averaged 123°F. Normal losses are estimated at 590 BTU per hour as shown for a standard model (1515 kWh/yr) in Table 3.1.

$$\text{Then actual heat loss} = 8 \text{ hours } [(2/3 \times 590) + (1/3 \times \frac{53}{70} \times 590)]$$
$$= 4340 \text{ BTU}$$

When this estimated standby loss is added to the output in Table 3.2, the

COP of the heat pump is seen to be 3.05.

The COP of 3.05 for an average application of 70°F ambient and 70°F input water would meet the objective that has been assigned for the development of the heat pump water heater. However, it has yet to be demonstrated that these results can be achieved consistently. On the other hand, improvement in performance can be expected as further refinements in system design are accomplished. Finally, as will be shown in the discussion of payback period, the economics of the heat pump water heater are not very sensitive to improvements in COP above the level of 2.

3.1.3 Effect of Ambient Air Temperature

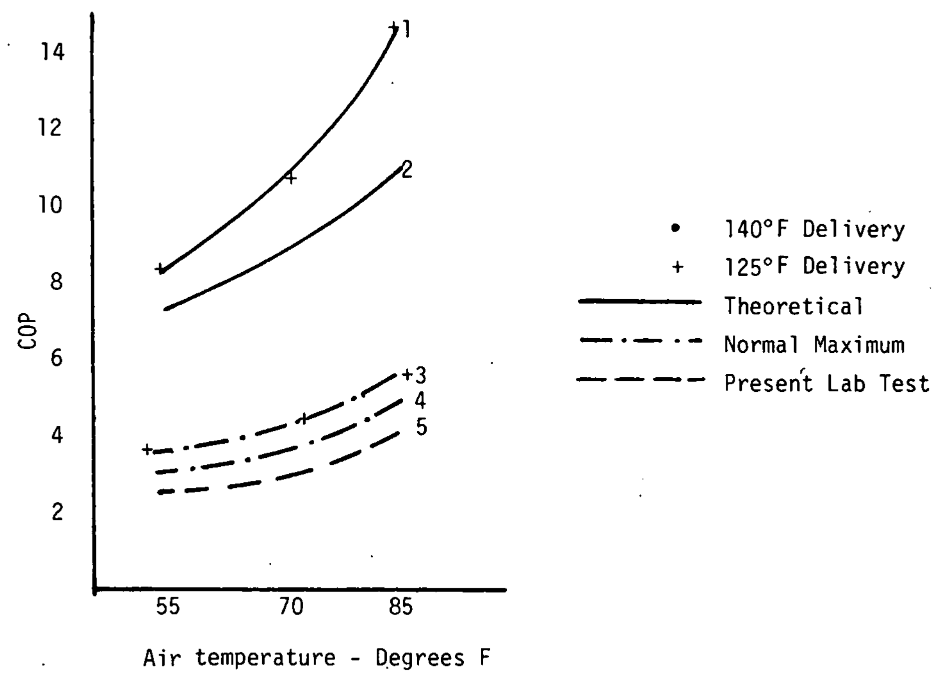
The theoretical COP of a Rankine cycle machine is expressed as:

$$COP = \frac{T_2}{T_2 - T_1}$$

where T_1 and T_2 are delivery and source temperature in deg. R

In Figure 3.2, the two top curves (1 and 2) are theoretical COP's based on delivery temperatures of 140 and 125°F with source temperature varying between 55 and 85°F. In practice, the Rankine cycle COP generally runs 40 percent of the theoretical COP. The second two curves (3 and 4) represent 40 percent of the theoretical curves which represent then about the maximum that can be expected from the heat pump water heater. But the units presently on test at EUS are achieving something less than that as shown by the bottom curve #5.

Figure 3.2
HEAT PUMP WATER HEATER COP VS. AIR TEMPERATURE.



3.1.4 Fffect of Supply Water Temperature

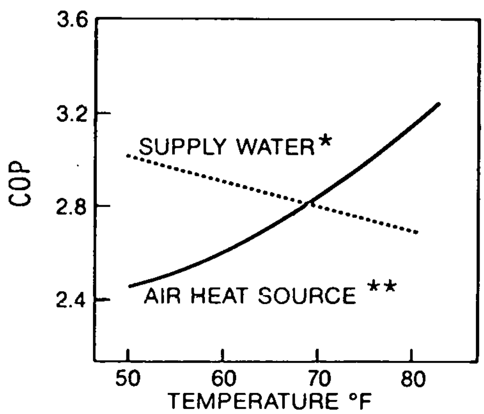
The efficiency of the heat pump is higher when heating water starting at lower temperatures because of the better heat transfer between condenser and water. This is demonstrated in Table 3.3.

TABLE 3.3
Heat Pump Water Heater
Efficiency Versus Supply Water Temperature
Cold Start Tests

	50-140°F	60-140°F	70-140°F	80-140°F
Output - BTU	66,620	59,610	52,595	45,580
Input - BTU	21,310	19,670	17,950	16,180
E_R	3.13	3.03	2.93	2.82

The net effect is seen to be a seven percent improvement in E_R when supply temperature decreases from 70°F to 50°F. That seven percent will offset some of the disadvantage of the colder climate in the North. That is portrayed in Figure 3.3 which shows the relative impact of air and supply water temperatures.

Figure 3.3
COP VS. SUPPLY WATER AND HEAT SOURCE TEMPERATURES



* Air source @ 70° F
** Supply water @ 70° F

The value of this improvement in efficiency in the North from colder water is amplified because colder water also requires more kWh of energy from a resistance type water heater. Thus, the improved E_R is operating on a larger base which has a positive impact on the calculation of payback period.

3.1.5 Effect of Delivery Water Temperature

Because of the basic relationship of the Rankine cycle that the COP is a function of the delivery and source temperatures, the question arises as to what improvement in COP can result from reducing the delivery temperature. Table 3.4 which shows results from a series of tests, indicate a 10% improvement in the COP as the delivery temperature is reduced from 140° to 125°F, regardless of air temperature. Table 3.4 data is plotted in Figure 3.4. The COP's shown here are from cold start tests which are consistently higher than those from cycling tests because of the higher efficiency while the water is still cold.

TABLE 3.4

Heat Pump Water Heater COP Versus Delivery Temperature
(ambient temperature - 70°F)

	50-140°F	60-140°F	70-140°F	80-140°F
Output - BTU	66,621	59,608	52,595	45,582
Input - BTU	21,313	19,673	17,948	16,180
COP	3.13	3.03	2.93	2.82
	50-125°F	60-125°F	70-125°F	80-125°F
Output - BTU	52,600	45,580	38,570	31,560
Input - BTU	15,322	13,680	11,960	10,190
COP	3.43	3.33	3.22	3.1

The effect on operating costs will depend on the total annual kWh consumption and the price of fuel. Table 3.5 compares the energy costs for a 60°F supply temperature and fuel prices of 4 and 7 cents/kWh.

Figure 3.4

Heat Pump Water Heater
COP vs Delivery Temperature
(ambient temperature - 70°F)

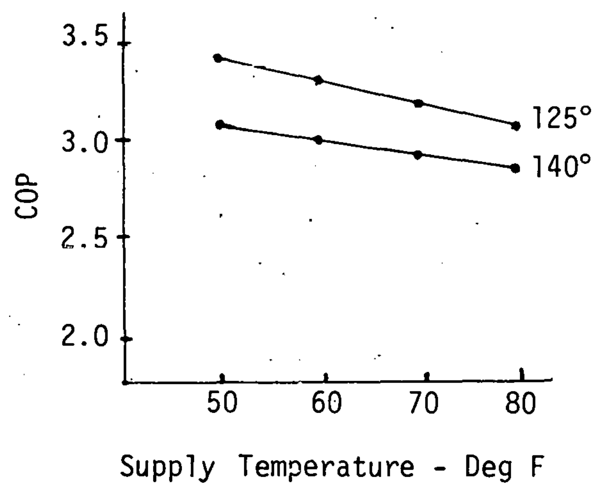


TABLE 3.5

Energy Cost Comparisons for Heating 60°F Water

Fuel Price - Cents/kWh			4	4	7	7
Consumption - Gals/Yr			25,000	50,000	25,000	50,000
Consumption by Resistance Unit kWh/Yr			5,000	10,000	5,000	10,000
System	COP	Delivery Temp. (Deg F)				
Resistance	1.0	140	\$200	\$400	\$350	\$700
Heat Pump	3.03	140	66	132	116	231
Heat Pump	3.33	125	60	120	105	210

From Table 3.5, it is apparent that even with high consumption and high prices (50,000 gals. and 7¢/kWh) the savings are only \$21 per year and only \$6 per year in the more representative case (25,000 gals. and 4¢/kWh). To achieve the cost reduction by decreasing delivery temperature from 140 to 125°F without a loss in heat storage capacity (thus increased reliance on the upper resistance element) requires increasing the tank capacity by 23%, the ratio of the number of degrees the water is heated above 60°F. Thus, a 50 gal. tank at 140°F would compare with a 66 gal. tank at 125°F. Based on an incremental cost of \$40 for a 66 gal. tank versus a 50 gal. tank, the payback period would run from 2 to 7 years for the cases evaluated here. However, the impact of the lower temperature on sanitizing of clothes and dishes would have to be considered before any recommendation is made to go to the lower temperature.

3.1.6. Availability of Waste Heat

The theory of the heat pump water heater is based on recovering what would otherwise be considered waste heat. This raises the question of just where and how much waste heat is available. For as heat is pumped from any given space, it must be replaced if the heat pump is to continue operating. Otherwise, the ambient temperature will be reduced to a point where the unit can no longer function unless the system provides for deicing of the evaporator.

A unit operating outdoors, in a ventilated attic or in a carport, would have an unlimited heat source as long as the ambient temperature remained above the evaporator's icing temperature. Recent tests have determined that temperature to be between 45 and 50 degrees F.

A unit operating indoors would be limited to heat flowing through walls, floor, ceiling and windows in an unheated space. In utility rooms, furnace rooms and basements with exposed ducts, heat is available from the dryer, washer, furnace jacket losses, duct losses, the water tank jacket losses and the effluent water from bath, laundry and kitchen.

The availability of waste heat (defined as energy utilization) was examined by General Energy Associates [3] (GEA) using a computer simulation. Table 3.6 shows that heat would be available above 50°F only 50 percent of the time in the basement of a house in Pittsburgh and Minneapolis. The garage with uninsulated walls and ceiling is better at 58 percent because heat flows into the garage faster whenever the outdoor temperature is above 50°F. Conversely, the utility room in Pittsburgh has heat available only 46 percent of the time, assuming no internal heat gains.

The effect of internal heat gains is shown in Figure 3.5. For example, for a single-story house in Pittsburgh, the utility room shows an availability of 50 percent with no internal heat gains. This increases to 73 percent with internal heat gains from the dryer and furnace.

These numbers appear to be valid, but the number of cases studied represent a small fraction of the number of possible variables that affect the result. Except for the case of the utility room, heat sources other than ambient air and ground temperature have not been considered. Often, furnace jacket losses and duct losses would provide more than enough heat in winter for operating the water heater.

The use of insulated outdoor walls has not been evaluated. Wall insulation would raise the required ΔT across the wall by several degrees to achieve the same heat gain to the basement, utility room, etc.

It has now been concluded that a closer fix on the wide variety of applications, climates, house construction, geographic locations, etc., where the heat pump would be economic (measured by length of payback period) can best come from the field test of units in these different environments.

TABLE 3.6

Utilization Efficiency For a Heat Pump Water Heater
In Two Story House with Basement

City	Net Degree* Days	Utilization Efficiency (%)**			
		Utility Room	Garage	Basement	Attic
Jacksonville	-1625	74	96	--	98
Pittsburgh	4951	46	58	50	--
Minneapolis	7220	41.5	--	50	--

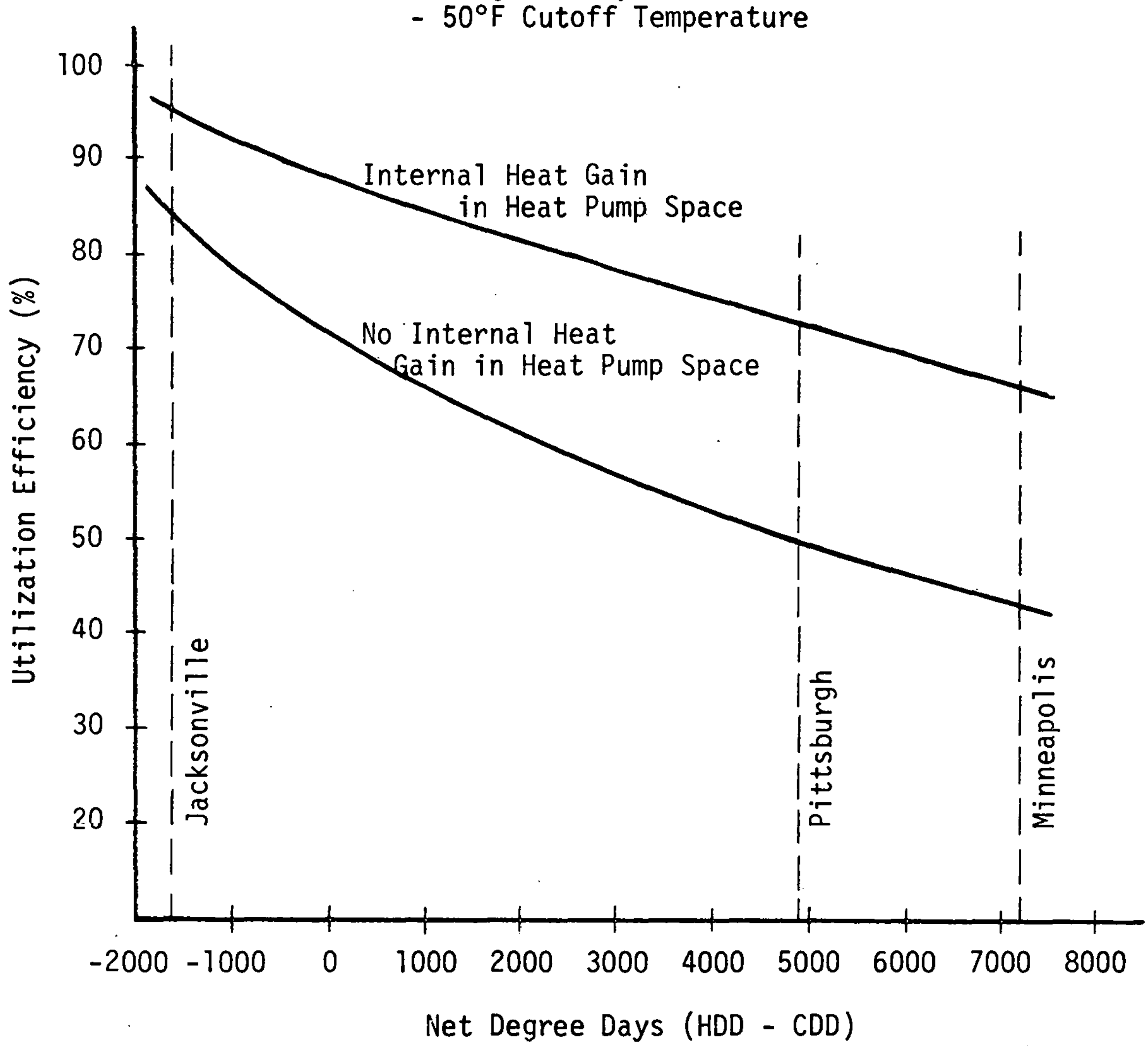
** Fraction of time when temperature is above 50°F with no internal heat gains.

* Net degree days = heating degree days (HDD) less cooling degree days (CDD).

Figure 3.5

Utilization Efficiency of Heat Pump Water Heater
Located in Utility Room

- Single Story House on Slab
- 50°F Cutoff Temperature



3.2 Hot Water Consumption

Very little has been done, or at least reported, in recent years toward determining patterns of hot water use. The latest data available are from the early and mid-sixties when the AEIC Load Research Committee (Ref 4, 1967-68) sponsored tests by several utilities to study the effect of varying supply water temperatures on kWh consumption. The results showed average usages between four and five thousand kWh per year and appear consistant with estimates (5 and 6) as shown in Table 3.7 for an average family of three persons based on heating water to 140°F.

Table 3.7

Daily Hot Water Consumption Versus Family Size
(Gals/day @ 140°F)

Family Size	3[11]	4	6
Food preparation	3	4	6
Hand dishwashing	4	5	8
Automatic dishwasher	15	15	22
Clothes washer	21	28	42
Shower or bath	15	20	30
Face and hands	2	3	4
TOTAL	45/56	60/70	90/104
kWh/Year	3,970/4,940	5,300/6,200	7,950/9,200

Based on our extrapolation, energy consumption for heating water would increase to 7,950 kWh/year for a family of six without an automatic dishwasher and to 9,200 for that family with a dishwasher. With the increase of saturation of dishwashers since the AEIC tests in the sixties (See Figure 3.6) from some 10 percent to today's 40 percent, the average annual consumption should be above 5000 kWh. This market study assumes the average energy use to be 5000 kWh. Based on this average and a typical customer frequency distribution as shown in Figure 3.7, the annual kWh use of the thirty million electric water heaters in service in 1976 would be distributed as shown in Table 3.8.

Figure 3.6

Saturation of Hot Water Using Appliances

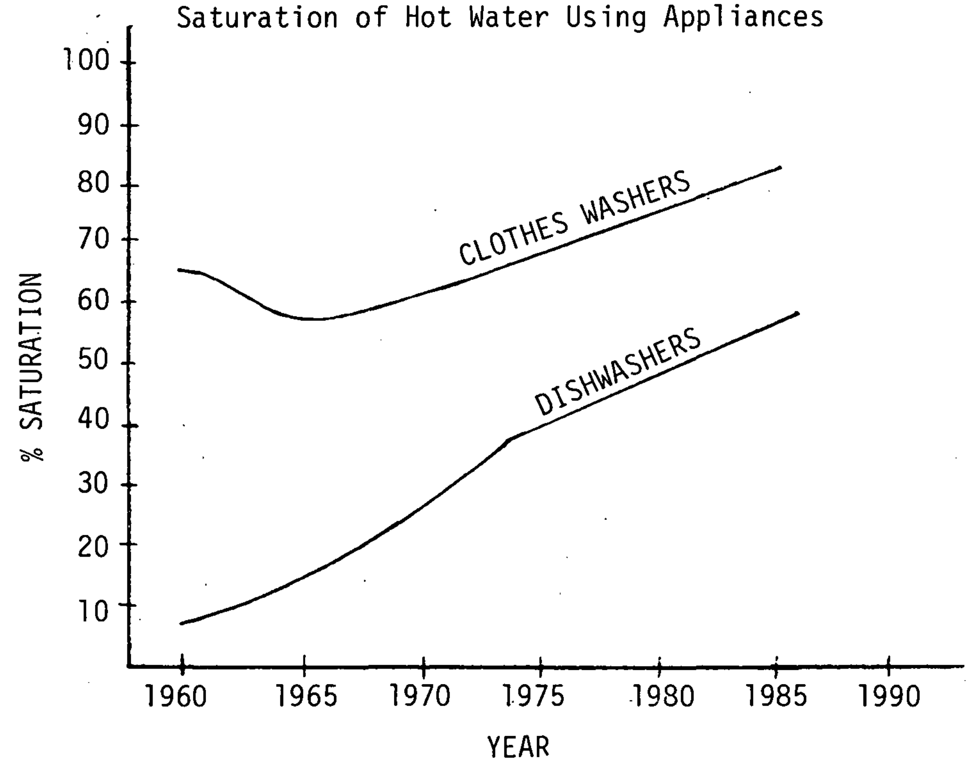
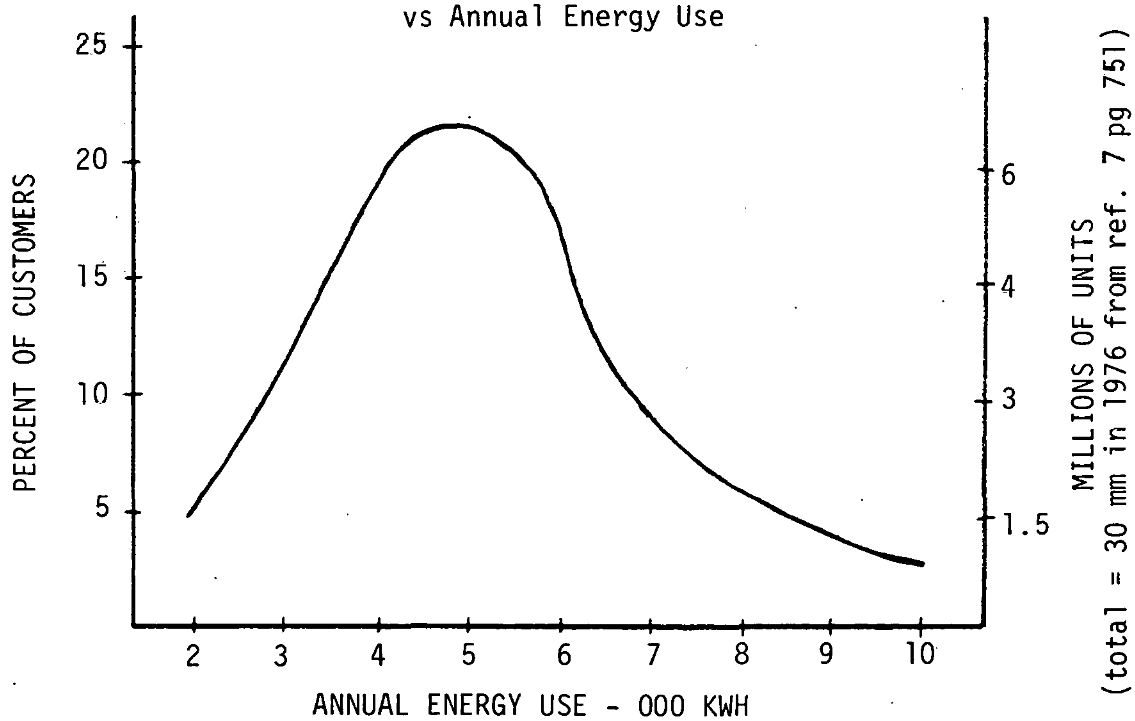


Figure 3.7

Customer Frequency Distribution
vs Annual Energy Use



Source: Reference 4 , Page L-130

TABLE 3.8

Distribution of Electric Water Heating Users Versus
Annual kWh Use

Annual kWh Use	Millions of Customers
2000	1.5
3000	3.5
4000	5.7
5000	6.8
6000	5.0
7000	2.8
8000	1.7
9000	1.2
10000	.8
over 10000	1.0

Projecting the future average consumption per household requires estimating some offsetting influences. Figure 3.6 showed a projected increase in the saturation of dishwashers and clothes washers which would significantly increase the average consumption. Conversely, the Department of Energy program on appliance efficiency will call for lower jacket losses from new units sold and existing units will be retrofitted with insulation, both of which will lower average kWh consumption.

Throughout this discussion of hot water consumption, life style has not been considered. Yet, since the early sixties, the advent of long hair for men, with daily shampoos and long showers for teenagers has had some impact on hot water use. To demonstrate that, an average shower head will have a flow rate of 3 to 5 gpm. Assuming mixing with 60° cold water, a normal 4 or 5 minute shower uses 7.2 to 15 gallons of 140°F water. A fifteen minute shower would use 27 to 45 gallons. If there has been a trend to longer showers, it should be documented by the EPRI* study which will record lifestyle, patterns of use, etc.

* Electric Water Heating: Group Load Research and Analysis. RFP 3472.

3.2.1 Effect of Supply Water Temperature on Energy Use

In 1963-64, the Association of Edison Illuminating Companies (AEIC) ran tests in four cities which compared the kilowatt hours used for domestic water heating with the temperature of the water supplied from the municipal water system. The results are tabulated in Table 3.9 and presented in a more visible fashion in Figure 3.8. On the average, each degree F of increase in supply water temperature lowers the daily electricity use by 0.18 kWh. With a difference of 20°F between the annual average supply temperatures of the South and the North, annual consumption will vary about thirteen hundred kilowatt hours (4,928 kWh and 6,241 kWh). This compares very closely with calculated values of 3,890 kWh and 5,000 kWh based on energy required to heat 23,000 gallons of water to 140°F from averages of 70°F in the South and 50°F in the North. However, these calculated values do not include losses as do the field test values. Conversely, they are based on 63 gallons per day which would be higher than the average during the field tests in 1963-64.

TABLE 3.9

Effect of Supply Water Temperature on Water Heating kWh

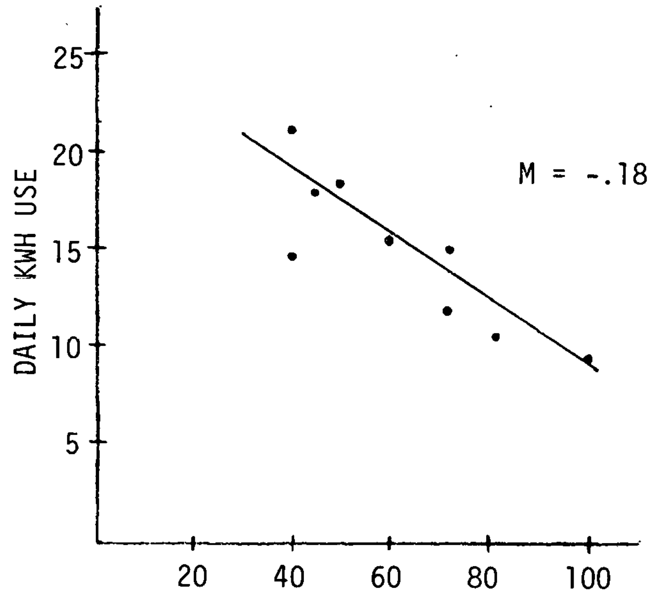
City	No. of Units	Winter: Min. Supply Temp. (°F)	Winter: Daily Use (kWh)	Summer: Max. Supply Temp. (°F)	Summer: Daily Use (kWh)	Average: Supply Temp. (°F)	Average: Annual Use (kWh)
Cleveland	(60)	40	14.5	72	11.7	56	5295
Dallas	(22)	46	18	81	10.5	65.9	4562
Baltimore	(23)	40	21.5	71	15	56.25	5304
Detroit	(50)	50*	18.5	60*	16.5	55	5245

*Average Temperature

Source: Ref 4, 1963-64, Page L137

Figure 3.8

Effect of Supply Water Temperature on Water Heating kWh



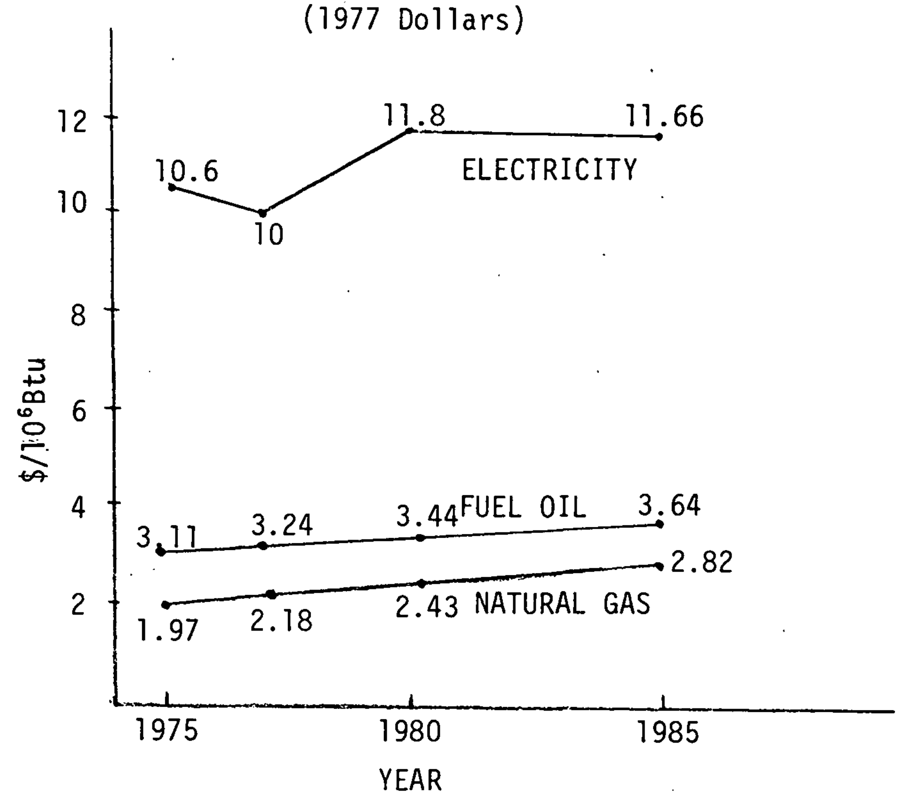
3.3 Price of Fuels

The marketability of the heat pump water heater will depend on its capability for generating operating cost savings compared to other types of water heaters. Those savings will vary directly as the price of electricity and to the relationship between the price and availability of electricity and the price and availability of fossil fuels, oil and gas. In Figure 3.9 are shown projected prices of electricity, gas and oil.

These projections were made in the summer of 1977 before the effect of foreign oil imports on the value of the U.S. dollar was widely recognized. Since then the declining value of the dollar has caused the President to suggest a tax on foreign oil imports and Congress to insist on the gradual deregulation of natural gas. If these occur, both would increase the gas and oil prices forecasted in Figure 3.9 with some lesser effect on the price of electricity.

Figure 3.9

Future Fuel Prices
(1977 Dollars)



Source: Averaged from Federal Register (8)

But even without that, prices of gas and oil are projected to increase faster than electricity so the ratio between the price of electricity and the prices for fossil fuels will be reduced. Table 3.10 shows the reduction between 1970 and 1985.

TABLE 3.10
Relative Price of Electricity Versus Gas and Oil

Year	Electricity $\$/10^6$Btu	Gas $\$/10^6$Btu	Ratio (2÷3)*	Oil $\$/10^6$Btu	Ratio (2÷5)*
1970	6.2	1.05	5.9	1.3	4.8
1975	10.6	1.97	5.4	3.11	3.4
1980	11.8	2.43	4.9	3.44	3.4
1985	11.66	2.82	4.1	3.64	3.2

*Refers to columns.

When the ratio drops below four, the heat pump water heater would cost less to operate than the fossil-fired water heater. This is derived by assuming an E_R of 2.8 for the heat pump water heater and .7 for the fossil units. With the ratios down to 4.1 and 3.2 by 1985, the heat pump water heater will be competitive on an operating cost basis. But that is somewhat academic because, unless the supply of natural gas and oil improves, there will be more use of electricity for residential use, including water heating. So the greatest impact on the market acceptance of the heat pump water heater may actually come from the future price of electricity which will, to a great degree, determine the payback period of the device compared to a resistance type electric water heater.

Projecting the future cost of electricity as it will pertain to water heating is complicated by the possibility of special time of day (TOD) rates (rates that are lower during off-peak hours) being adopted which would substantially reduce the cost. At one time this was common practice, for prior to the decade of the fifties some utilities offered special rates (one cent per kilowatt-hour and less) based on water heaters being controlled by a time switch to restrict their use to off-peak hours -- except for the quick recovery top element. At that time, both elements were 1500 watts -- a significant load in the 1930's. By the fifties, ranges, dryers, television and air-conditioning had seemingly made the control of water heaters unnecessary. Checks of time switches in use showed them to be mostly off-time due to power outages and failure of meter readers to reset them. Thus the resulting diversity was the same as it would be without time control. In any event, the economics of water heater control had suffered as economies of scale reduced the cost of generation and transmission facilities.

During the fifties and sixties, almost all utilities abandoned time control. Generally rate structures were modified to include a block of three to four hundred kilowatt-hours at the same reduced price used with the time switch.

In the decade of the seventies, many states have encouraged, if not forced, utilities to abandon all promotional rates including both special water heating rates and total electric home rates. The resulting additional cost to the homeowner is substantial and the cost increase has been aggravated by the coincident rapid escalation of the price of electricity over the last four years.

Because the price of electricity is projected to continue to increase, (on an actual dollars basis) attention is again being focused on some type of control for electric water heaters. This is one part of a general consideration of load control coupled with TOD rates. Because the water heater tank provides capacity to carry over the three or four hours of a normal peak, water heaters are a primary candidate for off-peak (TOD) rates.

Impact of electric water heaters on the utilities' system peak is variously claimed to be between 1 and 2 KW per unit. With new base load plants now being installed at $700 per KW, and even peaking plants at $300/KW, the control of water heaters again appears attractive. If it happens, the lower rates will raise the payback period of the heat pump water heater and decrease its economic justification.

Conversely, the heat pump water heater provides an alternative for reducing the demand of electric water heating on the utility facilities without requiring the costly metering and control equipment associated with TOD rates, thus decreasing the economic justification of TOD rates. For example, some utilities are now selling water heaters with just one 2500 watt element. Their experience shows this gives very satisfactory service when used in the larger 66 and 82 gallon tanks. A heat pump water heater would use only 700 watts and provide the equivalent of 2100 watts of heating capacity. This would certainly lower the diversified demand to 1/3 that of units with 2500 watts and less than 1/3 of units with 4500 watts.

In summary, TOD rates could reduce the economic justification of the heat pump water heater. Conversely, the heat pump water heater could reduce the economic justification of TOD rates applied to water heating.

Coupled with the anticipated reluctance of utilities to return to TOD rate and controlled water heating, the reduction in demand of the heat pump water heater will enhance its chance for acceptance and promotion by electric utilities at the regular residential rate. If regular rates were to increase, the feasibility would be yet further enhanced.

3.4 Payback Period

It is generally agreed that in the deep South the heat pump water heater could provide hot water at one third the normal cost and practically year round. At the same time, it would be providing free air conditioning and dehumidification. But the practicability of the device in the North (at least in the winter) is not as evident because it would be taking heat from the house that would have to be replaced by the house heating system. For this case the efficiency of the water heater would be the same as the house heating system. This would seem to indicate that the most feasible application will be in the South.

However, there are other variables that tend to offset the effect of ambient air temperature. For example, the availability of waste heat in a furnace or utility room and the effect of colder supply temperatures on the heat pump's efficiency increase annual operating savings.

Even in the worst case in the North, assuming no waste heat is available, the heat pump can be turned off in the winter and the water heated by electric resistance with a COP of one. Assuming equal seasons and a COP in summer of three, the annual average is 1 1/2 which is still attractive.

A COP of two reduces consumption by 50% compared to a resistance unit. Going from there to a COP of three only reduces it another seventeen percent. A COP of four cuts only another eight percent. So the consumers payback period is reasonably insensitive to COP, at least not as sensitive as payback period is to the level of hot water consumption and the price of electricity.

Figure 3.10 compares the payback period for three variables (COP, price of electricity and annual consumption) for what are considered reasonable limits for each variable. The common point is for a COP of 3, a price of five cents per kilowatt hour and an annual consumption of 25,000 gallons.* This analysis is based on increased installed costs of $200 for retrofit units and $150 for new units. Using the latest estimates of $200 to $250 added cost, the payback periods are still very good.

Data for the curves is shown in Table 3.11. Examination of the data shows that the length of the payback period is relatively insensitive to the COP, improving about 37% between a COP of 2 and one of 4. Increasing electricity prices lowers the payback period in excess of 59% going from 3¢ to 7¢/kWh. Increased usage lowers the payback period by more than 50% as water heating requirements goes from 25,000 to 50,000 gal/yr.

This analysis does not take into account the negative effects of taking heat from the home or having insufficient heat available where the unit is installed in a small space such as a utility room. Nor does it give credit for performing the functions of air conditioning and dehumidification.

Still the conclusion can be drawn that the most attractive applications will be those involving large families and in areas with high electricity rates somewhat regardless of the climate and other variables affecting the COP.

3.5 Air Conditioning and Dehumidification

When running, the heat pump water heater will provide over 1/4 ton of air conditioning and dehumidification.

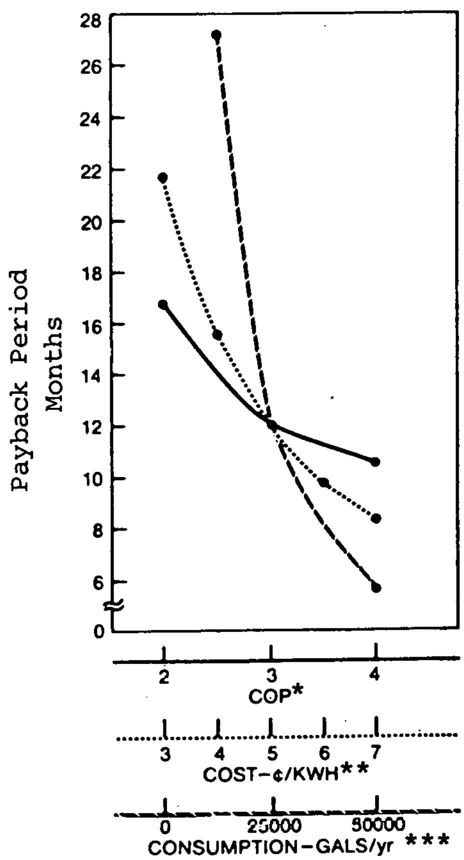
For some situations this could have some value but we have not quantified it because of matching supply to need as well as uncertainty regarding space conditioning need in the heat pump location.

---

* 25,000 gallons of 140°F water would require about 5,000 kWh in a resistance type heater or about the national average. 50,000 gallons or 10,000 kWh in a resistance heater is typical of a suburban or rural house.

Figure 3.10

Payback Period vs COP, Cost of Electricity & Hot Water Consumption for New Installations



* Cost = 5¢/kWh; Consumption = 25000 gals/yr
** COP = 3; Consumption = 25000 gals/yr
*** COP = 3: Cost = 5¢/kWh

There is some value for homes with basements which have high humidity in summer and can require dehumidifiers ranging in electricity rating from 300 to 600 watts. Running continuously for five months, a dehumidifier would use between 1080 and 2160 kilowatt-hours per season. At electricity prices between three and seven cents per kWh, the annual energy cost is between $32 and $151. This can be significantly reduced by the addition of a humidistat control because dehumidification is not always needed, particularly in the perimeter months of May and September. The heat pump water heater could provide the great majority, if not all, of the dehumidification required. Thus, it can save up to that maximum figure of $151 per year now being spent for dehumidification.

TABLE 3.11

PAYBACK PERIOD CALCULATION FOR HEAT PUMP WATER HEATERS.

Electricity Cents/ KWH	Type of Heater	Installed Cost New	Installed Cost Retro	Ann. Own Cost @15%	Heat Pump Maintenance @ 5%	COP=2 Ann Oper. Cost	COP=2 Total Ann. Cost	COP=2 Payback Period (Months) New	COP=2 Payback Period (Months) Retro fit	COP=3 Oper. Cost	COP=3 Total Ann. Cost	COP=3 Payback Period (Months) New	COP=3 Payback Period (Months) Retro fit	COP=4 Oper. Cost	COP=4 Total Ann. Cost	COP=4 Payback Period (Months) New	COP=4 Payback Period (Months) Retro fit
						25,000 GAL/YR; (5,000 KWH)*											
3	Resist	$200	--	$30		$150	$180	31	41	$150	$180	21.7	29	$150	$180	19	25
	Ht Pump	350	$200	53	17	75	145			50	120			38	108		
4	Resist	200	--	30		200	230	22	29	200	230	15.5	21	200	230	14	18
	Ht Pump	350	200	53	17	100	170			67	137			50	120		
5	Resist	200	--	30		250	280	16.7	22	250	280	12	16	250	280	10.6	14
	Ht Pump	350	200	53	17	125	205			83	153			63	133		
6	Resist	200	--	30		300	330	13.5	18	300	330	9.8	13	300	330	9	12
	Ht Pump	350	200	53	17	150	220			100	170			75	145		
7	Resist	200	--	30		350	380	11.4	15.2	350	380	8.3	11	350	380	7	10
	Ht Pump	350	200	53	17	175	245			117	187			88	158		
						50,000 GAL/YR; (10,000 KWH)*											
3	Resist	200	--	30		300	330	13.5	18	300	330	10	13	300	330	9	12
	Ht Pump	350	200	53	17	150	220			100	170			75	145		
4	Resist	200	--	30		400	430	9.8	13.1	400	430	7	10	400	430	6	8
	Ht Pump	350	200	53	17	200	270			133	203			100	170		
5	Resist	200	--	30		500	530	7.7	10.3	500	530	5.7	7.6	500	530	5	7
	Ht Pump	350	200	53	17	250	320			167	237			125	195		
6	Resist	200	--	30		600	630	6.4	8.5	600	630	4.7	6.3	600	630	4	5.5
	Ht Pump	350	200	53	17	300	370			200	240			150	220		
7	Resist	200	--	30		700	730	5.4	7.2	700	730	4	5.3	700	730	3.5	4.7
	Ht Pump	350	200	53	17	350	320			233	303			175	245		

*kWh consumed by electric resistance water heater

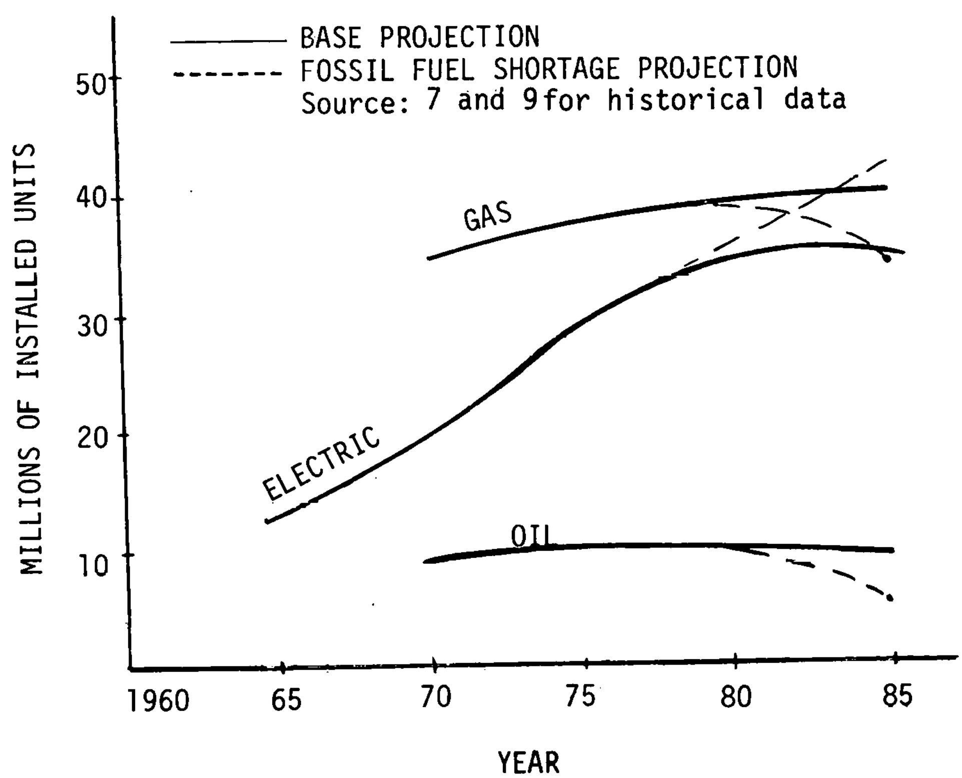
4.0 The Market

In Figure 4.1 are shown curves (solid lines) of current projections of the number of dwelling units with water heaters -- gas, oil, or electric. These projections are based on extrapolation of the past trends of water heater purchases. Thus, they assume no significant change caused by shortages of domestic supplies of gas or oil.

At the other extreme are the dotted line curves in Figure 4.1 which represent what could happen if increasing prices and/or shortages of fossil fuels do cause a rapid switch to electric water heaters. While the actual projection will probably fall somewhere in between these two extremes, the dotted line projections will be used in defining a potential market for electric hot water heaters.

Figure 4.1

Water Heater Installation by Type



Because of the general emphasis on energy conservation, at some point financial institutions may add energy conserving devices to the mortgage without affecting the down payment and financing for a product such as a heat pump water heater with its potential for both energy and cost saving.

Thus, the market for the heat pump water heater is projected to build up in steps as follows:

1.) Large home builders in areas where the payback period is one year or less.

2.) Individual home builders where the payback period is less than one year.

3.) Builders in areas where lending institutions will include cost in the mortgage without increasing down payment requirements and where payback period is no more than two years.

4.) Builders and home owners in areas where local electric utilities will provide financing with repayment from savings in utility bills and where repayment period is not more than two years.

Thus, the primary criterion for the market acceptance of the heat pump water heater is seen to be the expected payback period. As shown in Section 3.4 the payback period is:

- under three years for all new applications
- under 1 1/2 years for all users of 10,000 kWh
- under 14 months for all new units using 10,000 kWh

Based on the criteria listed above and the payback periods in 3.4, the heat pump water heater would appear to be the logical choice in every case. But a wide variety of constraints exist that will dictate otherwise. In general, these are covered in Section 6.0. But the major constraint may prove to be a lack of available heat in many applications as developed in a study by General Energy Associates [3] discussed in Section 3.1.6.

At this point, it is assumed that some economically attractive heat source location will be found in practically every case in new houses; that homes where the present unit is electric, where the tank size is 52 gallons or above, where the tank is less than five years old and where space exists for applying the heat pump unit represent potential for retrofit of a heat pump; and that all existing units, gas, oil or electric, will be replaced over the next ten years.

Thus, the market potential would be projected as shown in Table 4.1.

TABLE 4.1

Potential Market for Heat Pump Water Heaters (in 10^3)

Year	New Units (new houses)	Existing Retrofit	Retrofit New Units	Replace Existing Units	Total
1978	1,500	2,400		2,500	6,400
1979	1,800			2,700	4,500
1980	2,000		1,000	3,000	6,000
1981	2,000		1,200	3,200	6,400
1982	2,000		1,500	3,500	7,000
1983	2,000		1,700	3,700	7,400

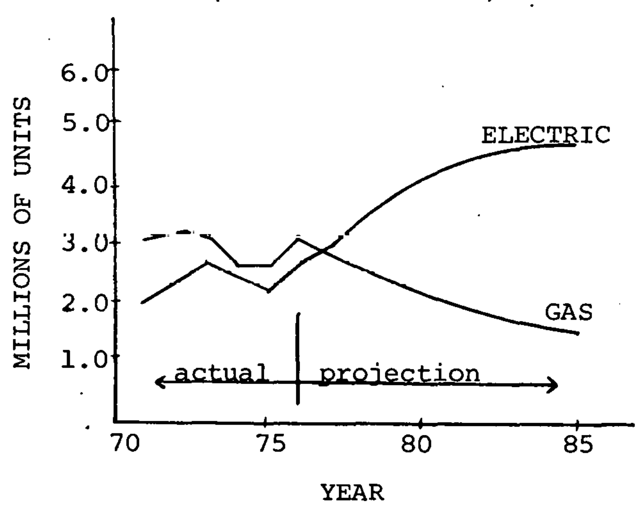
The logic for these projections, in addition to that already expressed is as follows:

- Electric water heaters will increase their share of new housing units.
- Most of those installed over the next five years will represent retrofit unit potential as shown in Column 4.

- Of the thirty million existing electric water heaters, 2.4 million represent retrofit potential. This is derived from sixty percent being less than 52 gallon, fifty percent being more than five years old and sixty percent being in inaccessible places.
- Replacement of existing gas, oil and electric units will represent the greatest potential.

Figure 4.2 shows projected annual sales of gas and electric water heaters. The projection is based on a continued decline in the availability of natural gas for new residences in the 1980's and a coincident gradual trend to replacement of gas water heaters with electric water heaters.

Figure 4.2
Sale of Water Heaters
(Gas and Electric)



In Table 4.1, the potential market for heat pump water heaters is broken down by segment -- whether for new housing starts, retrofitting existing electric water heaters, or for replacing existing electric, gas or oil units. This differentiation is important because of the difference in the marketing channels and key decision makers involved in each segment.

The buying decision in the case of a new home will in the vast majority of cases be made by the builder who is prone to make his decision on first cost as compared to life cycle cost unless the life cycle costs are so favorable as to represent a saleable feature. This would probably require a payback period not to exceed two years or where, with no increase in down payments, the extra monthly mortgage charge would be more than offset by lower utility bills.

To the average homeowner faced with paying for a higher cost product with a lower operating cost, the payback period has to be close to one year except where the electric utility would finance the appliance and take monthly payments as part of the monthly utility bill.

5.0 Building Disaggregation

Before projecting the future market potential of the heat pump water heater, it is necessary to disaggregate the buildings population and examine the characteristics and hot water requirements of each type. The requirements would include a daily maximum and an hourly maximum which provide an indication of the ability of the heat pump to provide sufficient hot water. They also indicate where the demand is great enough to provide enough savings to more than offset the added investment.

This section studies seven major types of buildings to different degrees depending on the amount of data available, the variables within the building type and what is needed to determine the potential feasibility of the heat pump water heater.

5.1 Single Family Residential

Table 5.1 is a breakdown by region of those single family house characteristics which will have some impact on the potential market for the heat pump water heater. The number of other than single family homes is shown for reference and because for some characteristics (i.e., the number of bathrooms) multi-family units are included.

The figures shown have been collected from several Bureau of Census sources. Thus they do not always relate to each other as might be expected. Even so, they provide considerable data for evaluating the potential of the heat pump water heater in different areas or regions.

The more significant numbers include:

- 53,000,000 single family homes
- 50% are air conditioned
- Only 19% have dishwashers, but 55% of new homes in 1975 sold below $40,000 came with them and 93% in houses above $40,000

TABLE 5.1

Disaggregation of Single Family Residential Building Characteristics

Category	Northeast* Total	Northeast* Added 1975	South* Total	South* Added 1975	North Central* Total	North Central* Added 1975	West* Total	West* Added 1975	United States* Total	United States* Added 1975
Single family homes-1975 (10)	9.6		18.5		14.6		9.6		52.6	
Other	7.9		6.5		6.0		4.8		25	
Air Conditioning (%)	41		66		50		29		50	
# bathrooms - one (10,11)	12	.041	14.7	.072	13.5	.062	14.5		48.5	
More than one	4.8	.072	8.7	.286	6.4	.153	5.9		25.8	
# dishwashers-1970 (%) (12)	18.5		17.3		15.8		26.9		18.9	
# clothes washers -1970 (%) (12)	69		70		76		70		71	70%-1975 (7)+
Foundation (10,11)										
Basement (11,12)	14.9	.085	4.6	.071	14.5	.171	3.1	.059	37.1	.386
Slab-1970 (11)	1.0	.018	6.7	.192	1.9	.016	4.7	.081	14.4	.307
Crawl space		.010		.095		.028		.041		.174
		.113		.358		.215		.181		.866
Parking Facility (11)										
Garage		.077		.196		.161		.146		.580
Carport		--		.056		.004		.021		.081
None		.034		.106		.051		.051		.205
		.113		.358		.215		.181		.866
Persons/household (10)+										
1-3	10		15.8		12.9		9.2		47.9	
4	2.5		3.1		3.0		2.0		10.6	
5	1.6		1.9		1.8		1.1		6.4	
6	.7		.9		.8		.5		2.9	
7 or more	.6		.8		.6		.4		2.4	

* Total figures are in millions except where other units are shown

\+ Includes all dwelling units

TABLE 5.1
(cont.)

Category	Northeast* Total	Northeast* Added 1975	South* Total	South* Added 1975	North Central* Total	North Central* Added 1975	West* Total	West* Added 1975	United States* Total	United States* Added 1975
House Heating Fuel										
Gas	6.4	.027	11.4	.104	13.6	.105	9.5	.107	40.9	.343
Oil	8.9	.046	5.8	.014	4.3	.020	1.2	--	20.2	.081
Electric	.8	.037	10.0	.236	1.2	.080	2.3	.070	14.3	.425
Other or none		.003		004		.008		.003		.018
		.113		.358		.215		.181		.866
Type of Heating System										
Warm air furnace	5.9	.036	11.9	.292	14.2	.156	6.9	.142	38.9	.626
Hot water	9.6	.040	1.2	--	3	.011	.6	--	14.4	.054
Baseboard, etc.	.8	.034	1.8	.040	.8	.039	1.6	.029	5.0	.142
Other or none	1.0	.003	10.1	.023	2.3	.009	5.4	.009	18.8	.045
		.113		.358		.215		.181		.866
Water Heating Fuel-1970 (12)										
Gas	6.8		8.8		11.6		7.7		34.9	
Oil	5.8		1.8		1.3		.5		9.3	
Electric	2.3		6.9		4.0		2.8		16.1	
Other or none	.6		1.8		.6		.2		3.5	
Dishwashers included in: (11)										
Houses sold up to $40,000										55%
Over $40,000										93%
Typical Fuel Prices										
Gas-1976 ¢/therm (7)	16-31		15-25		15-18		15-24			
Electric-1975 ¢/kWh (7)	4-8		4-5		3-6		3-6			
Oil ¢/gallon (13)	41		37		38		40			

* Total figures are in millions except where other units are shown

- 71% of houses have clothes washers
- From the census data, it would appear that 71% of homes have basements and 27% are built on slabs. However, it is well known that a large number are built on crawl spaces. For instance, in 1975, 20% were built on a crawl space.
- 67% of houses built in 1975 had a garage.
- 22 million households have four or more persons. This is 29% of the total (includes those in multi-family housing).
- 14.3 million or 18% of dwelling units are heated by electricity; 49% of those added in 1975 were heated electrically.
- Electric prices ranged from 3¢/kWh to 7¢/kWh. These limits may not have changed much since 1975, but the average has risen significantly. The national residential average is now 4.02¢ per kilowatt-hour.(8)

All of these characteristics in addition to other criteria will affect the applicability of the heat pump water heater in any given house. But how the many variables are combined in houses is impractical to quantify.

For example, just how many large houses with large families there are with clothes washers, dishwashers, 2-1/2 baths, an unheated basement with a furnace, etc., etc., and in an area with high electric rates, cold municipal water and high humidity is difficult to predict. Yet, these positive characteristics relative to applicability of the heat pump water heater will probably occur in combination. Conversely, negative characteristics such as a small house built on a slab with low hot water consumption will often occur in combination.

Suffice it to say here that while many of the figures in Table 5.1 would tend to lower the national potential market, none by themselves would do so to a level that would even threaten to eliminate the potential. Nor would any group of negative data do so, for the positive characteristics will always be present in sufficient degree to create a worthwhile potential.

5.2 Office Buildings

Table 5.2 assumes an average office building as being five stories with ten thousand square feet per story. Allowing 200 square feet per person, the average building would hold 250 people. If each person is assumed to use a maximum of two gallons of hot water per day, maximum requirements are 500 gallons/day and 100 gallons per hour.

TABLE 5.2

Office Building Characteristics

Inventory - 1970 (14)	3,380 x 10^6 sq ft
- 1980	5,681 x 10^6 sq ft
Hot water req. - max. day	2 gallons/person (15)
Hot water req. - avg. day	1 gallon/person**
Assume average building*	5 stories, 10,000 sq ft/story
Assume one person/200 sq ft	
Average people/building	250
Hot water req. - max. day	500 gallons
Hot water req. - max. hour	100 gallons (.4/person)

* Glenn (14) assumes three stories of 13,500 sq ft/story.

** EUS estimate based on 2/3 gal/person for hand and face washing (as shown in Table 3.7) plus janitor use.

This is above the capacity of the present 3/4 to 1 H.P. heat pump water heater. But units in the range of two or three horsepower would be adequate given a tank in the 500 gallon size.

But while the average building may be as has been described (5 stories, 10,000 square feet, etc.) the vast majority (number wise) of office buildings are much smaller. Unfortunately, construction and building population statistics are kept on a total square foot basis so the number of office buildings by size is not known.

Approaching the problem another way, if the maximum daily use per employee is two gallons, a 3/4 to 1 horsepower heat pump water heater would be adequate for up to 90 persons or a building of some 36,000 square feet. At the lower end, 25 persons would use an average of 25 gallons/day or about the lower limit of the economic feasibility of the heat pump water heater.

So office buildings with between 25 and 90 persons represent a market potential for the 3/4 - 1 horsepower unit. Further, it should be pointed out that because of high internal heat gains from lights, people, equipment and solar, many office buildings use air conditioning as much as ten months of the year in average climates. So it can be assumed that adequate amounts of waste heat are available.

5.3 Retail Stores

The census of retail stores includes eating and drinking establishments. These use large quantities of hot water at 160 to 180°F. Thus they are not considered potential applications for the heat pump water heater in its present form.

They do represent great potential for larger units with high temperature delivery such as the Westinghouse Templifier. This would provide a high COP by using the restaurant effluent water as the heat source.

TABLE 5.3

Retail Stores Characteristics

Inventory - 1972 (16)	1,912,000 units
Eating and Drinking (16)	- 359,000 units
Total Inventory Less Eating and Drinking	1,553,000 units
Hot Water Requirement	
Max/day/store (14,17)	6 gallons
Max/hour/store (14,17)	1.5 gallons

All other stores use much lower amounts of hot water. Thus, some of the 1,553,000 existing stores and the new ones added each year represent potential for the heat pump water heater because here too high lighting and people loads create large quantities of waste heat. The maximum gallons required are estimates of Glenn[14] where no data exist. It is believed the dispersion about those average values would be quite wide going from individual stores with no public washrooms to large department stores with public washrooms. It is the latter that would represent potential users of the heat pump water heater. Unfortunately, the Bureau of Census data does not differentiate by size of store.

5.4 Educational Institutions

Based on the hot water use estimates in Table 5.4, it can be concluded that schools of any type do not offer potential for the heat pump water heater. The only exceptions are small elementary or secondary schools with less than 75 students. While many of these do exist, new construction is preponderantly in large area or regional schools based on use of school buses over large areas.

TABLE 5.4

Educational Institutions Characteristics

Type	Elementary	Secondary	Higher Level
# of schools (1000)[7]	77	27	2.7
Enrollment (1000) [7]	34,000	15,000	8,500
Students/School	440	550	3,100
Hot Water Req/Student			
Max. day-gallons [14]	1.5	3.6	24 (dormitory)
Max. hour-gallons[14]	.6	1.0	5 (dormitory)
Total Hot Water Req.			
Max. day/bldg-gallons	660	1,980	6,000*
Max. hour/bldg-gallons	264	550	1,250*

* Assumes 250 Student Dormitories

5.5 Hospitals and Nursing Homes

While some nursing homes and small hospitals might seem to offer market potential for the heat pump water heater, their use (like restaurants) is at 160 to 180°F for sanitizing purposes. Thus, they too would require units similar to the Westinghouse Templifier.

5.6 Religious Buildings

Table 5.5 shows 332,000 churches in the United States with an average maximum hot water use of 200 gals/day or 60 gals/hr. Any below the average would represent market potential for the heat pump water heater. But that potential might not be real because (1) the hot water demand in a church is extremely cyclic and thus the total annual consumption is low compared to the maximum demand and (2) most electric utilities give special rates to religious institutions. Both these factors would work against the economic feasibility of a heat pump water heater.

TABLE 5.5

Religious Buildings Characteristics

Number of Churches (7)	332,000
Membership (7)	132,000,000
Members/Church	400
Attendance (average Sunday) @ 50%	200
Hot Water Requirement	
Max/day	1 gallon/person
Max/hour	.3 gallon/person
Total Water Requirement	
Max/day	200 gallons
Max/hour	60 gallons

5.7 Multi-Family Residential

Table 5.6 presents maximum hot water requirements for any size multi-family building. These are above the capacity of the 3/4 to 1 horsepower heat pump water heater when the building requirements are considered in total. But if each apartment provided its own hot water, most would be potential users of the heat pump water heater. In the future this may be the case because for reasons of achieving energy conservation, future buildings are expected to use individual water heaters (and furnaces and air conditioners). This will develop because states will require the use of individual meters on each apartment. But that would not accomplish the objective of stimulating energy conservation unless each apartment is providing its own heating, water heating and air conditioning. Energy conservation has to come from the 70 percent of energy use represented by these three loads.

This leads to the conclusion that the water heater in many high-rise apartments will be located in a furnace room adjacent to the outside wall or on the balcony where it is hot in summer and warm in winter. Where the apartment houses three or more people and includes its own clothes washer (practically all already include a dishwasher), the result is an ideal application for the heat pump water heater.

At present, this definition comes closer to most garden type apartments than it does to high-rise. These generally include a utility room with a furnace, air conditioner and water heater. New units always have arrangements for an automatic laundry.

Thus, it can be concluded that a large number of multi-family buildings, both low and high-rise, will represent potential users of the heat pump water heater.

TABLE 5.6

Multi-Family Residential Characteristics

Units With	Number (10)	Average Units/ Structure	Hot Water Required - Gallons: Per Apt Max/day	Max/Hour	Total/Building Max/Day	Total/Building Max/Hour
2 - 4 units	9,802	3	80	12	240	36
5 units or more	11,792					
5 - 20			80	12	400-1600	60-240
50			73	10	3,700	500
100			60	7	6,000	700
200 or more			50	5	10,000	1,000

CONSTRUCTION 1971 - 75

Number of Units	Average	Number of Bldgs (11)	Avg/Yr	Hot Water Req/Bldg - Gallons: Per Unit Max/Day	Per Unit Max/Hr	Total Bldg (avg) Max/Day	Total Bldg (avg) Max/Hr
2 to 4	3	178,000	36,000	80	12	240	36
5 to 9	7	92,000	18,000	80	12	560	84
10 to 19	15	64,000	13,000	80	12	1,200	180
20 to 29	25	19,000	4,000	73	10	1,800	250
30 to 49	40	9,000	2,000	73	10	2,900	400
50 or more	100	7,000	1,000	60	7	6,000	700

6.0 Constraints to Achieving Market Saturation

To attain full commercialization and public acceptance of the heat pump water heater, a comprehensive national marketing program must be developed and carried out. A number of the essential steps in this program will encounter to a greater or lesser degree obstacles that must be overcome if this effort is to be successful. As these constraints are examined, it appears that in large measure they are institutional in nature, but some may be classified as behavorial, physical or economic*. Timely and competent action will be necessary in each case to prevent undue delay and excessive added expense. For orderlv examination, constraints are arranged in the following seven catagories:

- Compliance with the regulations of federal, state and local safety, health and code authorities.
- Price competition and consumer resistance.
- Reluctance of major manufacturers of electric and gas water heaters to add this higher cost item to their present product line.
- Electric utility constraints.
- Labor jurisdiction
- Inertia of architects and home builders
- Physical limitations

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* Based in part on telephone and personal contacts with thirty electric utility marketing executives, three electric utility trade associations, two water heater manufacturers, one large plumbing supply house and a personal interview with a former water heater promotion specialist of a major appliance manufacturer.

6.1 Regulations of Federal, State and Local Authorities

Before the product can be sold and installed in residences and other structures, it will be subject to the approval of certain authorities. The most important and universally required approval is that of Underwriters Laboratories, Inc. (UL). Preliminary approval can probably be obtained in a few months, but final acceptance will extend to a full year.

While present electric water heater manufacturers depend almost exclusively on UL approval, due to the inclusion of a refrigeration cycle in this device it will be desirable to submit the equipment to American Refrigeration Institute (ARI) for their evaluation. The cities of Chicago, Detroit, Los Angeles and New York each have their own independent testing laboratories. Each will test the heat pump water heater before approving it for use.

While not yet formalized it is likely that the Department of Energy and individual state energy commission appliance efficiency standards will apply. This appliance with its COP of 2 to 4 should easily satisfy such requirements. For inclusion of this device in Federal Housing Administration (FHA) and the Departments of Housing and Urban Development (HUD) programs, the standards and regulations of the respective agencies must be satisfied.

To comply with the requirements cited above and to obtain all necessary approvals with minimum time lag and expense, application will be made to the various authorities, submitting thoroughly documented technical material and, as appropriate, well constructed demonstration production models for testing and examination. The first available units from the pilot run will be used for this purpose. Should product modifications be required prior to approval, prompt action should be taken to correct the design or otherwise resolve the difficulties that arise.

6.2 Price Competition and Consumer Resistance

While it is estimated that this product will retail at $150 to $200 more than equivalent conventional gas and electric heaters, its higher efficiency

(COP) and operating cost savings will return the added cost to the buyer in many cases in less than one year. Despite the attractive payback ratios, current experience with other energy conservation products, e.g., insulation and heat pumps, suggests that this economic incentive will be slow to change the normal behavorial pattern of the public who usually buy on price.

Similarly, consumer resistance to retrofitting present electric water heaters at a cost of $150 to $200, plus installation labor can be expected. Despite the economic attractiveness it is likely that many consumers will prefer to wait for failure of their existing unit before installing a heat pump water heater as a replacement.

To offset this buy-the-cheaper syndrome, point of purchase promotional material should be available for pursuading potential purchasers to avail themselves of the substantial life-cycle cost saving of this product. In view of the sizable energy savings that results in significant reduced operating cost, support should be solicited from federal and state energy agencies and consumer groups.

6.3 Reluctance of Present Water Heater Manufacturers

The expense and time delay that would be encountered by a new company in creating complete water heater production facilities and establishing channels of distribution virtually precludes invoking this option. Thus, for this new product to succeed in the marketplace it is essential that it be produced and distributed by present water heater manufacturers, preferably those who have substantial shares of the market nationally.

Nine manufacturers produce most of the 3,000,000 annual volume of electric water heaters, but they are marketed by a large number of organizations including major appliance wholesalers, plumbing supply houses, retail appliance dealers and national retail chains.

The task of retraining all the distributors, dealers and plumbers in the technology of refrigeration is a formidable one. If it is not done properly, the heat pump water heater will gain the same bad image as did the early space heating heat pump. Initial marketing through electric utilities will minimize this possibility because they will have trained people to insure proper application, installation and servicing. Some fifteen percent of private utilities merchandise water heaters as do a much higher share of rural electric cooperative utilities. Many others have indicated they would return to merchandising for this device. Some have said they will locate capable dealers in their service areas and be responsible for training their personnel and supervising the application, installation and servicing of heat pump water heaters in the early stages.

6.4 Electric Utility Regulatory Restraints

Throughout the history of the electric utility industry, virtually every major domestic use of electric service has been pioneered and promoted by the industry itself. Utility marketing departments have been responsible for the market development of such products as electric ranges, clothes dryers, dishwashers, air conditioners and in earlier days, many small appliances. Over the past twenty years the promotion of electric water heating has been one of its major efforts with greater resources devoted to it than to any other domestic application except space heating.

A possible impediment to electric utility support in this program is the regulatory constraints under which they operate. In a number of states the utilities, by law, regulation or political pressure, are barred or discouraged from promoting any electricity consuming device. On the other hand, they are encouraged to engage in energy conservation programs. Thus it would appear that for the heat pump water heater, not only will the utilities be allowed to promote the device, they will be encouraged to do so. They may even be required to include financing programs.

So it is clear, beyond enlisting the support of electric utilities, the success of this program can be enhanced by obtaining parallel support and acceptance from state regulatory and energy commissions, as well as Federal agencies and possibly the Congressional Committees that have cognizance over energy matters.

6.5 Labor Jurisdiction

Ideally the installation and servicing of electric heat pump water heaters should be done by one trained journeyman. Historically the installation and repair of a water heater has been the province of the plumber who installs both water and gas piping and connections. But in many unionized urban areas, the plumber installs the electric water heater and a union electrician completes the electrical connections. The refrigeration system introduces a third trade and unless this jurisdictional problem is resolved, greater delays and increased costs would inhibit the acceptance of this new product. With the help of local utilities, distributors, dealers and other allies, efforts should be made to prevent this potential labor problem.

6.6 Inertia of Architects and Home Builders

A significant difficulty that is sure to be encountered is that of having electric heat pump water heaters specified for new tract, speculative and custom designed homes. The home buider merchandises his units on a price and a feature basis with little if any emphasis on operating costs. The builder is unlikely to offer it even as an option until buyer pressure mounts.

Similarily, architects are unlikely to specify this product since they are under pressure to design to a price ceiling set by their client and this seldom seen appliance will have little appeal until the economic benefits become widely known and accepted. To meet this serious challenge, support should be solicited from American Institute of Architects (AIA), National Association of Home Builders (NAHB), National Electrical Manufacturers

Association (NEMA), Gas Appliance Manufacturers Association (GAMA), and American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE).

6.7 Physical Limitations

The heat pump water heater must be installed in an upright position where headroom exceeds five feet for lowboy models (52 gallon) and six-and-a-half feet for 82 gallon models.

In present practice, water heaters are located in heated and unheated basements and garages, crawl spaces, unventilated closets, utility and laundry rooms, kitchens, bathrooms, attics, and outdoors; and electric units in both horizontal and vertical positions. Some of the locations have headroom or access limitations that preclude or minimize the possibility of the heat pump water heater. Because of the dehumidification feature of this device, a condensate drain is required and for some locations a piping connection to the sewer could be a significant expense.

To avoid serious misapplication in the installation of the device,

1) a detailed application guide should be prepared and
2) initial marketing done through trained electric utility personnel.

7.0 Selection of Target Market

Anticipating the start of production of the heat pump water heater, it is necessary to make plans for an optimal approach to its marketing. Because of the many variables that will affect the marketability of the device, one element of the marketing plan must be the selection of those market segments that promise the greatest chance of success and to provide for an effective demonstration of the units.

7.1 Methodology

To identify those market segments, a decision-making method developed by Kepner-Tregoe, Incorporated,[18] was used. It starts with the listing of those criteria which will affect the decision. These are shown in column one of Table 7.1.

Then for each criteria, an importance rating is assigned based on the most important being ten and all other between 1 and 9. These are shown in column two. In the main these are subjective judgments, but in this case, the relationships between price of electricity, kWh consumption and the coefficient of performance had been evaluated by parametric analysis in Section 3.

The availability of an appropriate "Channel of Distribution" is rated quite high because starting any new product without that is very difficult. In this case, the complication of application, installation and servicing problems make the availability of thoroughly trained people in the field practically a necessity.

The geographic pattern of present "electric water heater saturation" was rated low because in the near future the increasing cost of other fuels coupled with their threat of being in short supply should cause a trend to electric water heaters in all areas.

TABLE 7-1

Evaluation of Market Segments

Criteria	Importance Rating	Urban	Suburban	Rural	Single Family Home	Mobile Home	Multi-family	Retail Stores	North East	South	North West	West	New	Retro fit-
		Markets												
Price of Electricity	10	10 100	9 90	8 80	7 70	7 70	7 70	10 100	10 100	8 80	8 80	10 100	10 100	8 80
Consumption														
Water Temp(supply)		8	9	10	9	10	8	8	10	5	10	8	10	10
Family Size		7	10	10	10	2	5	5	10	10	10	10	10	10
Appliance Inventory		5	10	10	10	2	8	5	10	10	10	10	10	7
Total	10	7 70	10 100	10 100	10 100	5 50	7 70	6 60	10 100	8 80	10 100	9 90	10 100	9 90
Channel of Distrib.	7	5 35	5 35	10 70	10 70	7 49	7 49	7 49	8 56	10 70	10 70	9 63	10	5
Coefficient of Perform.														
Building Design		5	10	10	10	2	5	5	10	8	10	8	10	5
Climate/ground temp.		10	9	8	10	10	10	10	3	10	4	7	10	10
Water Temperature		8	9	10	10	10	10	10	10	8	10	8	10	10
Humidity		10	8	5	8	8	10	10	7	10	7	2	10	10
Total	5	8 40	9 45	8 40	9 45	8 40	9 45	9 45	7 35	9 45	8 40	6 30	10 100	8 80
Electric Water Heater Saturation														
Avail. of other fuels		5	9	10	8	9	7	10	10	6	8	6	10	8
Cost of other fuels		7	7	10	8	10	7	10	10	7	8	10	10	9
Total	3	6 18	8 24	10 30	8 24	10 30	7 21	10 30	10 30	7 21	8 24	8 24	10 100	9 90
TOTAL		263	284	320	309	239	255	284	321	296	314	307	400	340

Attention is called to three of the criteria in column one which are broken down into the factors that determine the level of the criteria. This is done to facilitate their evaluation later on.

The next step is to determine the market types to be evaluated and the segments of each. Four types are used:

- Population density
- Type of building
- Geographic region
- New or retrofit

Then for each criteria, one segment in each market is selected where that criteria would have the greatest effect. It is assigned a rating of 10 and the other segments in its market assigned values from 1 to 10 as they are judged to relate to the maximum segment. For example, for "price of electricity" under "population density," urban is judged to have the highest cost electricity which would make the heat pump water heater more desirable there, so it is assigned the value of ten with suburban and rural at 9 and 8, respectively.

Next, the rating assigned is multiplied by the importance factor for the criteria to give a net quantification. For "price of electricity" under "population density," urban is 10 times 10 or 100 with suburban and rural at 90 and 80.

For those criteria that are broken down into two or more factors, a rating for each factor is assigned for each market segment. All factors are then averaged before the "importance rating" is multiplied to derive the net quantification.

As noted previously, the assigned ratings to a great degree represent subjective judgments. Generally, the logic behind the judgments comes from the data in other sections of this report. The basic logic behind each criteria included:

Price of Electricity	- the higher the price the more desirable would be the heat pump water heater

Consumption	- the greater the kWh consumption, the greater the opportunity for annual dollar savings
Family size/ Appliance saturation	- is proportional to hot water use and thus kWh of consumption and shorter payback period
Channel of Distribution	- the present availability or the ease of establishing appropriate distribution will affect the length of time required to make the device available--in particular to initial target market(s)
Coefficient of Performance	- the higher the COP, the lower the kWh used and the greater the annual operating savings but recognizing that beyond a COP of 2.0 the incremental value becomes low
Building Design	- homes with basements with furnace rooms in the north and large utility rooms in the south would be optimal
Climate/ Ground Temperature	- assuming a fully insulated house, heat for the heat pump must come from the outside or from the ground. Thus, warmer climates will have more heat available.
Water Temperature (supply)	- the colder the supply water, the more efficient will be the heat pump and the more kWh used per gallon of hot water. Both factors lower the payback period.
Humidity	- humidity in the air is a source of latent heat -- then too, its removal is a fringe benefit to the consumer and an assignable value to the heat pump
Electric Water Heater Saturation	- the higher the present saturation, the lower may be the cost of electricity relative to other fuels or the lower is the opportunity for significant savings

Availability of Other Fuels	- in areas where low cost natural gas is not available, the market potential for electric water heaters of whatever type is highest -- typically true of the rural market -- also of many parts of the northeast region
Cost of Other Fuels	- high cost of those energy sources normal to a given region will both create a market for electric water heaters and raise the price of electricity as these energy sources are used for its generation -- thus the heat pump water heater becomes the economic choice.

7.2 Conclusions

Examination of the bottom line of Table 7.1 will show total values which do not seem very far apart. But based on prior experience with the Kepner-Tregoe method, this is considered normal. Any difference in the vicinity of twenty percent is considered distinctive. On the other hand, a difference of ten percent or less is not considered enough to make a distinction between two elements because of the margin of error of such subjective evaluations. So there is a quite fine line between acceptable difference and that which can be ignored.

Using that distinction, the following are the market segments offering the greatest potential:

- the rural market with suburban just marginal
- the single-family residence
- the market for new units

In the case of geographic regions, all regions appear to be almost the same. It should be pointed out, however, that in almost every case, vast differences exist within the region. Thus, it is necessary to look within regions to identify the best possible applications, such as:

- Hawaii with its year round ambient of 65° to 90° has an inexhaustible source of heat.

- Florida with high ambient and high humidity for ten or eleven months of the year.

- The Atlantic Coast as far north as Massachusetts with heavy reliance on expensive imported oil, with warm humid summers and with winter climate moderated by the ocean.

- In many areas electric utilities merchandise appliances, and in particular electric water heaters, to their consumers. This provides an ideal channel of distribution for a new device with unique application, installation and servicing problems. This is particularly true throughout the South and Midwest with high rural populations.

Thus, we conclude that our initial target market(s) will involve rural (with some suburban) single-family residences and more specifically those rural areas with year round moderate climates served by merchandising utilities.

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RESEARCH AND DEVELOPMENT

OF A

HEAT-PUMP WATER HEATER

TASK 3

DEVELOPMENT AND LABORATORY TESTS

JULY 1978

Prepared under Subcontract No. 7321 by

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for

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Operated by

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Oak Ridge, Tennessee 37830

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ABSTRACT

Energy Utilization Systems, Inc., (EUS) as a sub-contractor to the Union Carbide Corporation - Nuclear Division for the Department of Energy has designed, built and laboratory-tested a dedicated heat pump for domestic water heating.

This report identifies the final design of both a new water heater model and one for retrofit to existing electric water heaters, identifies the physical and performance characteristics of the major components and presents results of the laboratory tests involving all the variables to which performance of the device is sensitive. It also presents (in Appendix C) evaluations of the applicability of the heat pump water heater in several climates and locations in the residence. This supports information in the Market Study (Task 2).

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LIST OF TABLES

1. Summary

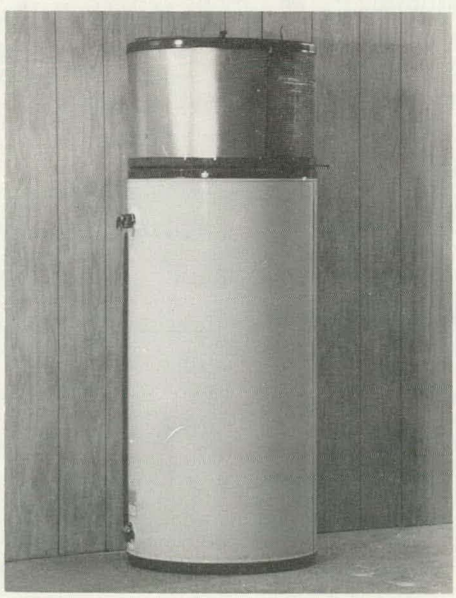
Energy Utilization Systems, Inc. (EUS) has completed the design of an electric heat pump water heater. Seven prototype units have been built and put through a series of performance and sensitivity tests in the EUS laboratory with the test results of two units verified by an independent test laboratory.

The design objective has been to achieve a heat pump Coefficient of Performance (COP) of 3 in an average climate and at a volume production cost to give a retail installed premium selling price of 200 to 250 dollars.

The most recent laboratory tests achieved an efficiency of E_R of 2.5 on a twelve hour cyclic test. The COP for the heat pump which gives credit for losses, would be about 2.8 or within ten percent of the design objective. EUS now believes that continuing refinements in the system design and in particular, in the direct immersion condenser, will achieve the goal of a COP of 3.

The final design for either a new unit or for retrofitting to an existing unit is in a 24" diameter round housing sized to match the diameter of an 82 gallon energy saver electric water heater tank manufactured by Mor-Flo Industries.

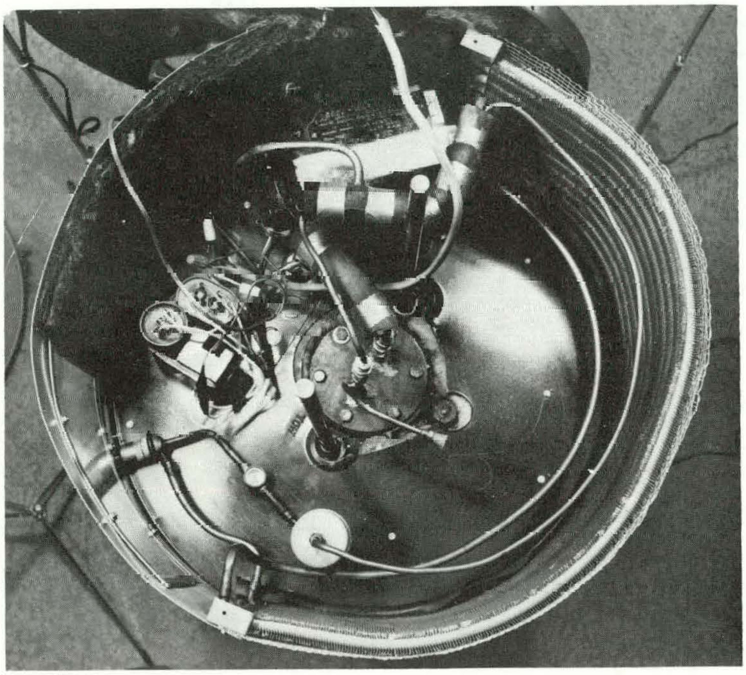
Figure 1-1



82- Gallon Heat Pump Water Heater

The housing contains the compressor, evaporator, expansion valve, sight glass, drier, electrical controls and fan.

Figure 1-2



Top View of Heat Pump Water Heater
(cover removed)

The top and bottom of the housing are the same as the top used for the water heater tank itself. Thus, an aesthetic match between heat pump and tank is achieved and at a cost savings because the covers are already in mass production.

The compressor for a new unit is a standard one horsepower, 230 volt, R-22 model produced by Copeland, Inc. In this application the refrigerant is changed to R-12 because of the high discharge pressures associated with the systems discharge temperature of 200°F. This derates the compressor by approximately one-third. A 230 volt unit was selected to match the voltage of the 2,500 watt resistance element provided for quick recovery or in event of heat pump failure.

For the retrofit design, a 3/4 horsepower compressor is used with an output of 6000 Btu per hour because this provides a closer match to the reduced heat transfer capability of the retrofit condenser as compared to the new unit condenser.

The evaporator is 11 inches high and 36 inches wide formed in a radius to fit the tank top and covering the rear half of the housing. It is constructed of a single row of 3/8" copper tubing with 10 aluminum fins per inch.

The expansion device is a thermostatic expansion valve, factory adjustable, rated at 1/2 ton R-12.

A low ambient cut off control will be used to shut the compressor off at temperatures below 45°F, in which case the upper resistance heater would automatically provide the water heating.

Both a run capacitor and a start capacitor with relay will be used to permit expected quick restarts, as when hot water is drawn from the tank just after the compressor has cycled off.

The fan is rated at 1/30 horsepower, 230 volts, 750 C.F.M. and provides 300 FPM face velocity across the evaporator.

The tank for new units is a standard Mor-Flo 82 gallon energy saver tank except that a special top cover with a 4 1/2 inch hole is used to allow entry of the condenser. The 82 gallon size was selected to allow the heat pump with a nominal heating capacity of 2100 watts to provide as much of the water heating as possible without help from the quick recovery element.

The condenser for new units is a "chimney" type coil having 19 4.25 inch diameter horizontal coils spaced 1/2 inch apart. (See figure 3-7 on page 3-23). The copper is tin plated to protect the tank anode. The coil is wound from 20 feet of 3/4" double copper tubing with fluting between the two tubes and on the outer surface (see figure 3-8, page 3-24) to improve the heat transfer between refrigerant and water. Space is left between tubes (the annulus) for providing a barrier between toxic refrigerant compounds and the potable water. The annulus is filled with colored water and sealed with a pressure relief cap to provide a visual indication of failure if increased pressure from a hole in either tube forces the dyed water from the annulus.

For retrofitting existing electric water heaters the condenser will consist of 20 feet of fluted copper refrigeration tubing inside a 1/2 inch smooth copper sheath with a special double wall return bend having an O.D. of less than 1 1/8 inch (see figure 3-9, page 3-25). It will be in the shape of a 12 inch diameter helix, 22 inches long, in order to "screw" it into the hole left by removal of the lower resistance element. External insulated "risers" will be used to make field connections of the precharged lines to the heat pump.

Operation of the heat pump in conjunction with backup resistance elements can be in various combinations using none, one or two elements, wattages from 1500 to 4500 watts and controlled to be interlocked with or to operate in parallel with the heat pump.

Units to be built in the pilot run will have a 2400 watt upper element. Normally, only the heat pump will be used to heat the water. However, on start up, or on withdrawal of all or most of the hot water, the resistance element will automatically operate in parallel with the heat pump for fast recovery. Total wattage will be approximately 3200 watts while providing effective heating of 4500 watts.

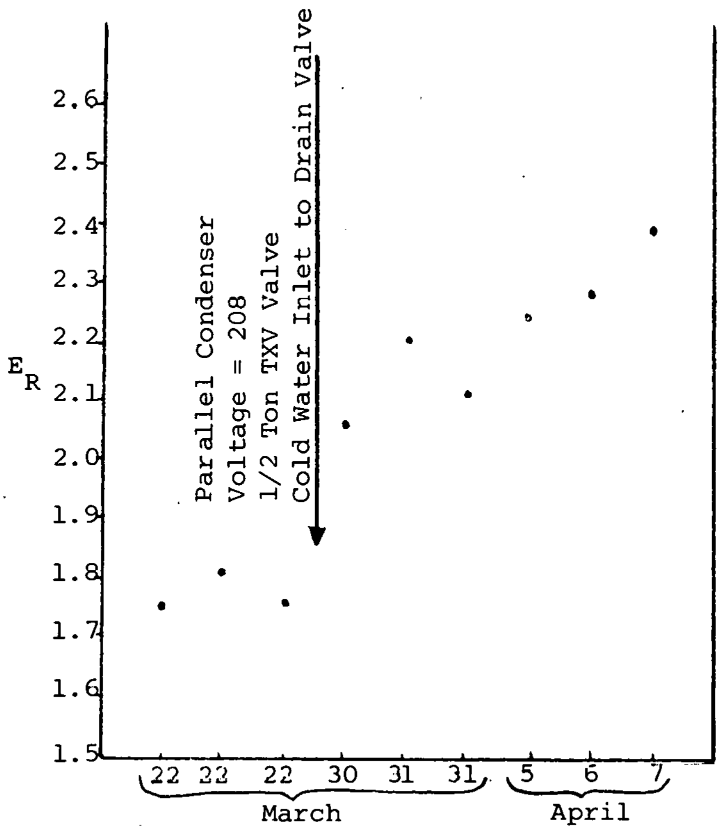
In the event of failure of the heat pump or cut off in below 45° ambients, the upper element will supply hot water. Normally the upper element will only heat twenty-five percent of the tank but EUS and Mor-Flo have designed an element to bend down into the tank to achieve heating of as much as one-half of the tank.

Throughout the laboratory testing, the upper element has been disconnected so that all heating is by the heat pump. Five units have been on life test for periods of time varying from three to ten months. The units operate on a continuous cycle of withdrawal, heating to 140° F, compressor cool down and repeat. Thus far, the only component failures have been two thermostats. Recently, all five thermostats have been raised to a setting of 160°F to accelerate the life test.

Performance and sensitivity tests have been run starting in August of 1977. The results have not been comparable as the design was being constantly modified to incorporate improvements as instrumentation was being changed. As shown in Figure 1-3, the test results showed substantial improvement in the E_R in early April of 1978. Subsequent to those tests a better chimney condenser was built which has achieved an E_R of 2.5.

Figure 1-3

Tests With Double Condenser
(cycling tests)



During that same period, sensitivity tests were run on the then latest design. The optimum refrigerant charge was determined as 26 ounces, the voltage at 220 volts and the expansion valve at one turn open from the factory setting.

Cold-start sensitivity tests of ambient air temperature, supply water temperature, and delivery water temperature were performed by EUS and Associated Test Laboratory (ATL). Summary results from the EUS and ATL data are shown in Appendix A, Tables A-2 and A-3. Figures 1-4 and 1-6 use ATL data. Figure 1-5 uses EUS results for the 40°F supply water temperature test due to an inconsistancy of the ATL data,

Figure 1-4

Heat Pump Water Heater E_R
vs Ambient Air Temperature

(Supply Water 60°F)
(Delivery Water 140°F)

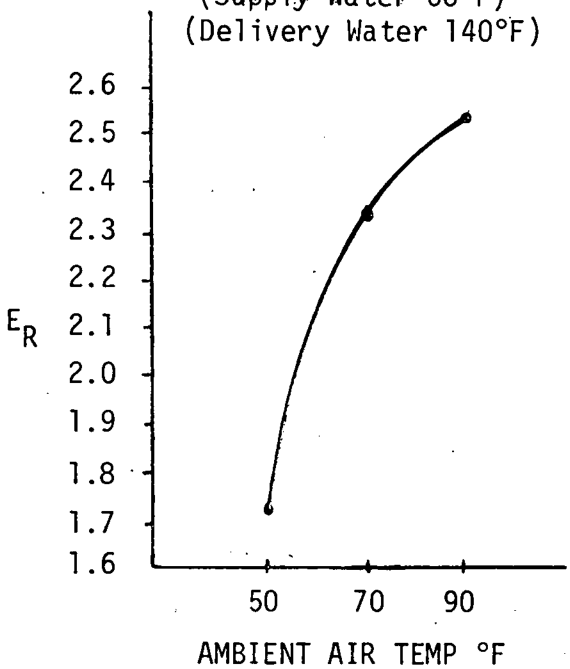


Figure 1-5

Heat Pump Water Heater E_R
vs Supply Water Temperature

(Ambient Air 70°F)
(Delivery Water 140°F)

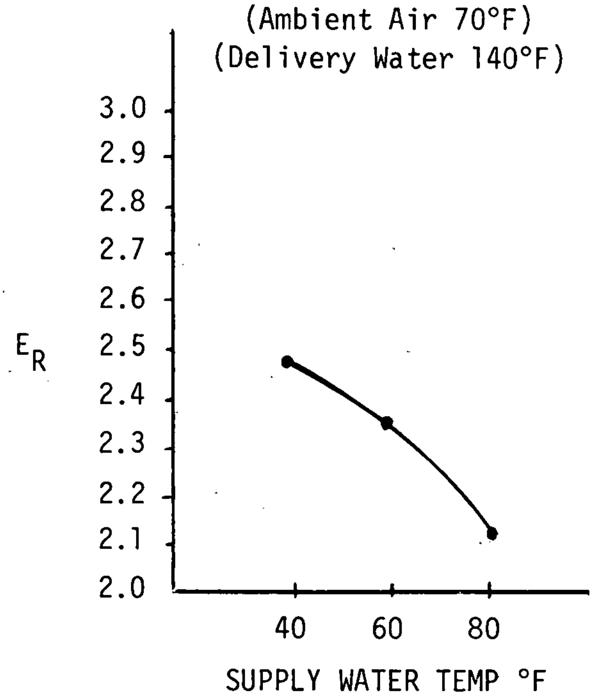
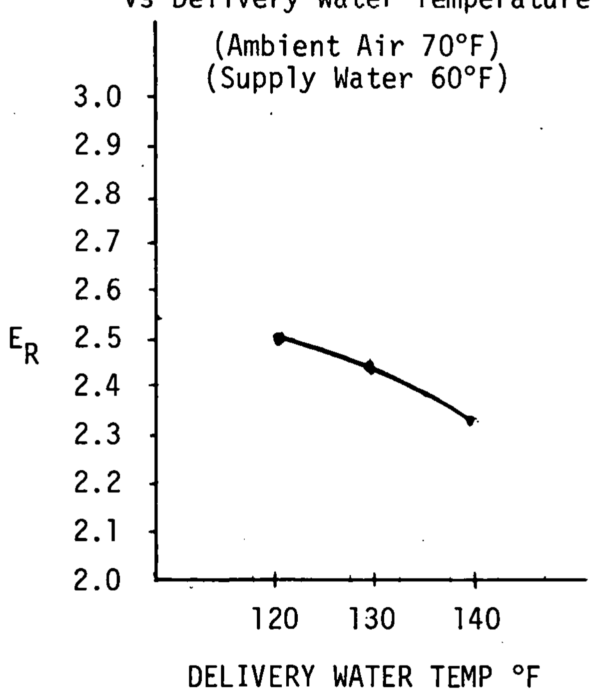


Figure 1-6

Heat Pump Water Heater E_R
vs Delivery Water Temperature

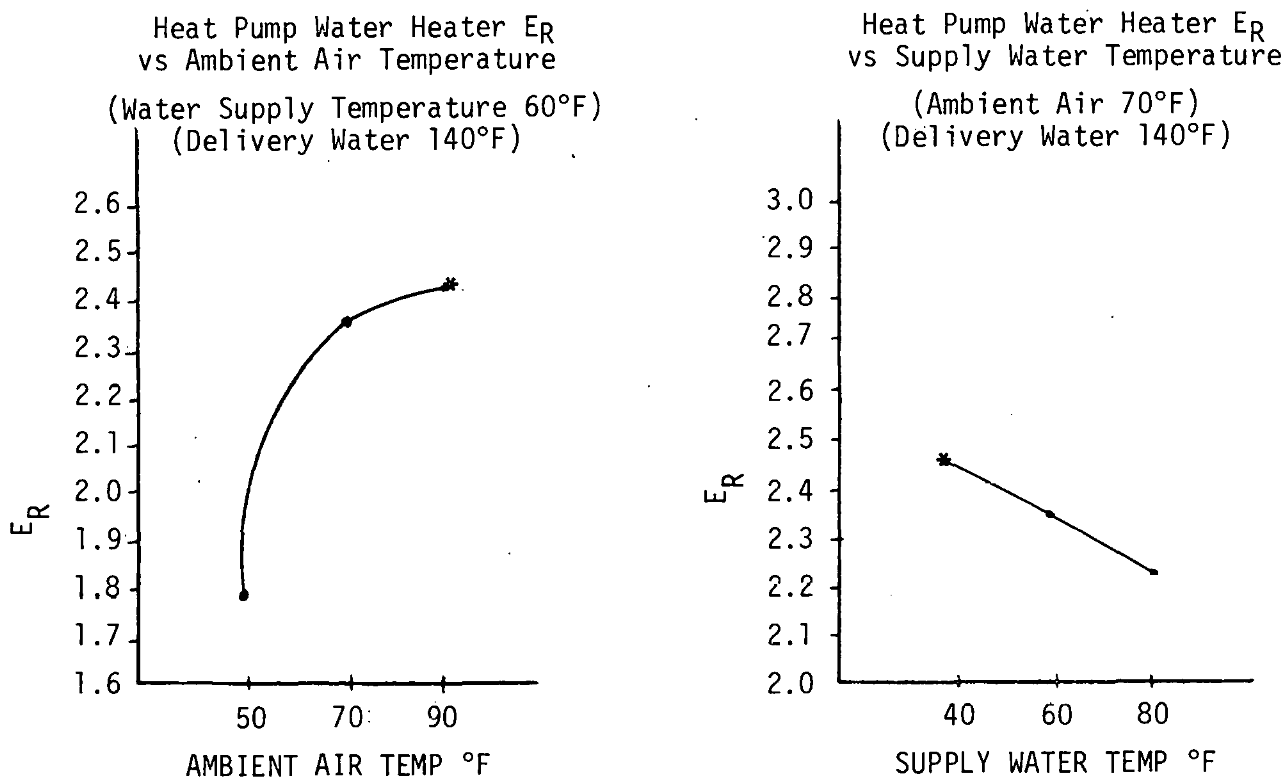
(Ambient Air 70°F)
(Supply Water 60°F)



Although cold start tests, i.e. heating the entire tank from city supply water temperature to final temperature, is the suggested method of rating a water heater, it is not indicative of home use efficiencies, particularly with a heat pump water heater.

Figures 1-7 and 1-8 indicate the efficiencies that might be expected in normal home use at varying ambient air and source water temperatures. The test results are from the latest chimney type fluted condenser and demonstrate considerable improvement over the #1 unit tested by ATL and EUS.

Figure 1-7. Efficiency Measurements of Cycling-Test Water Heating.



* Estimate -- data not available.

Based on test results of the most recent design and the almost faultless continuous performance over ten months life test of the earliest models, it has been concluded that the design objectives are within reach. Because of the significant progress made in the last three months, it is assumed that yet some further improvement in performance will be achieved as other modifications are conceived and incorporated in the design.

2. Introduction

The economic justification for the use of a dedicated heat pump for domestic water heating has recently been enhanced by increasing prices for electricity coincident with improvements in the state of the art of heat pumps. For if a heat pump water heater could achieve a coefficient of performance of 3 at a cost premium of $200 while pumping heat from an unheated space in the home, the reduction in operating cost compared to a standard electric resistance water heater would pay back the added cost of the heat pump in from six months to 30 months depending on the hot water used and the price of electricity.

To investigate the possiblity of achieving such a coefficient of performance, EUS has designed heat pump water heater systems for new and retrofit electric water heaters. Prototype units have been built and tested in a series of sensitivity tests covering variables of ambient air temperature, supply and delivery water temperatures, voltage, compressor size, evaporator size, condenser size and configuration and expansion valve type, size and adjustment. Similar tests have been performed by an independent test laboratory to confirm the EUS test results.

To perform the tests, a test room was created with the capability of controlling temperature, humidity, voltage and supply water temperature; automatically controlling withdrawal cycles; and recovering the heat from hot water as it is withdrawn.

The design objectives were established to achieve a coefficient of performance of 3 from average conditions of 70°F ambient air, 140° delivery water temperature and 60° supply water temperature and at a total system installed cost premium of $200 to $250.

3. Heat Pump Water Heater Design and Analysis

3.1 Background

The concept of a heat pump water heater is not new. Two firms manufactured and marketed them in the 1950's. They met with little success because the cost of electricity was low and declining and/or the performance achieved from the then state of the art of heat pumps was minimal. Units manufactured by the Hotpoint Division of General Electric were tested by Dr. J.B. Chaddock(1) under contract to the Association of Edison Illuminating Companies. In field tests, the COP ranged from 1.26 to 1.39. This minimal performance is believed to stem, at least in part, from the condenser design which consisted of a single tube wound around and brazed at one point to the tank. Thus much of the condenser surface was not in contact with the tank.

A more efficient, but more costly unit was manufactured by the Harvey-Whipple Corporation, Springfield, Massachusetts. It employed a large diameter double wall tube as the condenser with exposure to water on its outer periphery. The condensed liquid refrigerant line returned up the air column in the center of the condenser tube. Further description of this type of unit may be found in ref. 1.

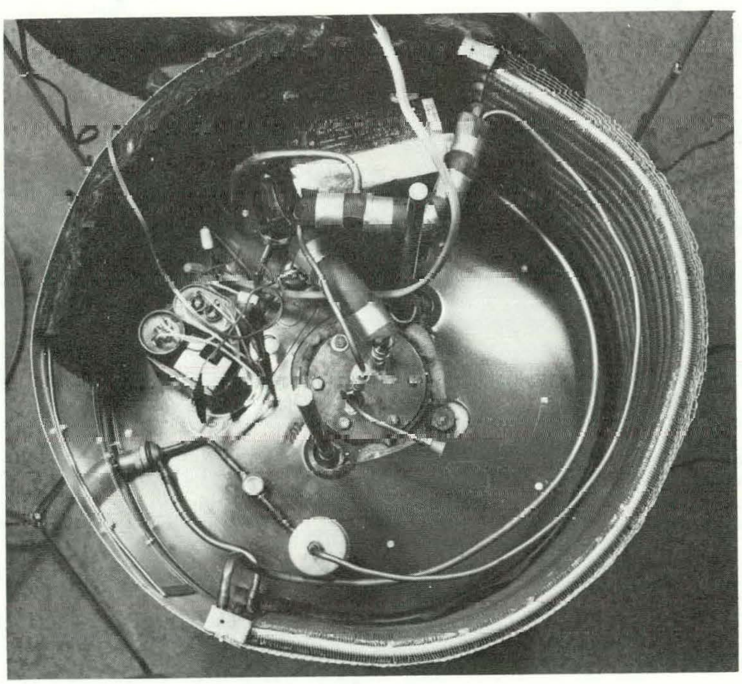
Operational efficiencies (COP) of these units ranged from 1.7 to 2.5 according to tests by American Gas and Electric Service Corporation.(2) The Chaddock feasibility study states that other tests of ten modified Harvey-Whipple units revealed COP's ranging from 1.1 to 3.1 (test conditions not stated).

The Harvey-Whipple design did not provide for a double-barrier between potable water and the refrigeration system but present day plumbing codes in some major cities would require some form of double barrier based on preventing intrusion of refrigerant substances into the potable water. Codes adopted by HUD for solar water heaters have included this requirement between heat exchange fluid and potable water.

It is assumed that codes will require that direct immersion condensers as anticipated for the EUS design will be required by state and local building codes to be of double wall construction approved by HUD, UL and local testing organizations. Due to the heat transfer problems of double wall tubing with relatively large annuluses, it would be desirable to eliminate the double tube requirement. This is a step now being evaluated by HUD aimed at lowering the cost of solar water heaters. Should further investigations lead to acceptance of single barrier systems, the unit could be easily modified with the attendant benefits of improved efficiency and reduced costs.

3.2 Heat Pump Water Heater System and Proposed Design - New and Retrofit Models

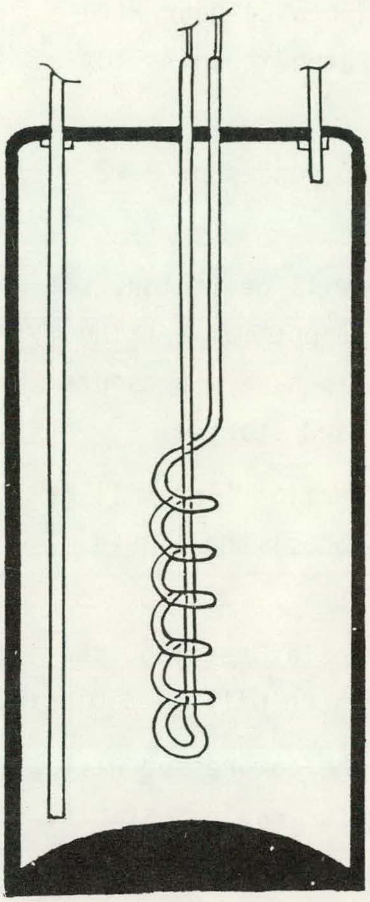
Figure 3-1



Top View of Heat Pump Water Heater (Cover removed).

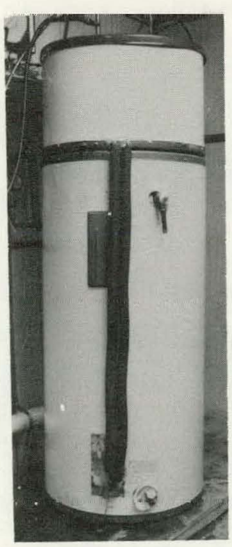
The EUS heat pump water heater will be in a round housing mounted on top of an 82 gallon water heater tank manufactured by Mor-Flo Industries. (see Figure 3-1) For both the new and retrofit models the housing will contain the compressor, evaporator, expansion valve, electrical controls (except thermostat), sight glass and anti-evaporator freeze-up protection.

Figure 3-2 Sketch of Tank with Chimney Condenser



In the new unit, the condenser will consist of a direct immersed chimney type coil inserted through a 4.5" flanged hole in the top of the tank.

Figure 3-3 Unit Showing Refrigerant Lines on Side.



The retrofit unit will use a helix wound, direct immersion condenser, inserted into the lower element hole and connected to the compressor by insulated refrigerant lines running up the outside of the tank.

3.2.1 Detailed Physical and Performance Specifications

- New type units will be tested using a 24-hour withdrawal cycle for an E_R of 2.5. Retrofit units 10% less.
- The system will be charged with 26 ounces of R-12 refrigerant.
- The thermostatic expansion valve will be factory set at 6°F suction line superheat, the optimum setting for this application. It will also have a pressure equalization bleed port to ease hard starts.
- Approximately six inches of the suction line will be soldered to the liquid line to subcool the liquid line and superheat the suction line.
- At 70°F ambient, 140°F tank thermostat setting, the maximum compressor discharge pressure will be 300 psig.
- In addition to a run capacitor, the compressor will have a start capacitor and relay for rapid restarts, plus internal thermal overload protection.
- The evaporator air flow face velocity will range from 300 to 400 feet per minute.
- The evaporator temperature differential (saturated suction temperature to ambient air) will range from 25° to 40°F.
- The condenser temperature differential (condensing temperature to water temperature) will approximate 30°F.
- The heat transfer agent (water) in the condenser annulus will contain a potable red vegetable dye to provide residual evidence of a leak when the seal cap is blown off.

- The maximum tank thermostat setting will be restricted to 140°F.
- The safety overtemperature thermostat will be set at 175°F (standard for electric water heaters).
- An ambient air thermostat will open (cut off the heat pump) below 47°F and close at 52°F.
- All refrigerant lines will be copper, type L with all joints brazed.
- The system will be nitrogen tested to 400 psig.
- The hot liquid refrigerant line will be extended to form an anti-freezeup serpentine coil mechanically secured to the evaporator.
- Evaporator condensate will be collected in a corrosion resistant trough and transferred to a drip tube suitable for the small drain hose provided.
- Hot and cold water copper pipe extensions will be secured into the tank inlet and outlet so that connections can be made outside of and/or above the heat pump cabinet.
- The carton design will be a 12" extension of present Mor-Flo carton with added internal corner protection.
- Additional specifications are given in the following sections.

3.3 Water Heater Tank Analysis and Design

3.3.1 Cold Water Inlet Design

The National Bureau of Standards, while visiting EUS stated that tanks with vertical cold water inlet dip tubes did not prevent mixing of cold inlet water with heated water as well as horizontal inlets located at the drain tap. Since the heat pump provides the highest COP with low water temperatures the N.B.S. suggestion was tested. The E_R and COP of daily withdrawal schedules improved by 10%. Thus, the EUS design will incorporate a redesign of the cold water inlet tube to create the same or similar effect.

3.3.2 Tank Condenser Assembly

To adapt to a mounted heat pump, a large diameter hole would be required in the tank top to facilitate lowering the condenser into the tank. The larger the hole diameter, the larger may be the condenser coil diameter. Therefore, with fewer coils, it will have most of its heat transfer area at the very bottom of the tank where the water is lowest in temperature.

Limitations to the size of the hole are the standardized industry hot and cold water connections 8" apart on centerline. The entry hole must be centered so that any added flange will not interfere with existing major Mor-Flo tooling. It has been resolved that a 6" outside diameter flange is the largest that can be welded to the tank top without interfering with tooling and water connections.

The weld flange will therefore be 6.0" diameter, 0.5" thick mild steel with a 4.5" hole in it. It will have 6 drilled and tapped 3/8" NFT holes spaced 60° apart. The weld flange will be heliarc or submerged arc welded on its I.D. and O.D. to provide a water tight seal. After welding it must be flat $\pm$ .005. Mor-Flo is to test each tank assembly in accordance with standard electric water heater pressure test procedures.

The gasket will be 40-60 durometer, 0.62 thick urethane rubber (or the equivalent) with 0.5" bolt clearance holes, 4.5" I.D. and 6.0" O.D. The gasket material must be suitable for use at temperatures of 32° to 200°F.

The bolt plate shall be 0.25" thick mild steel, with a 6.0" outside diameter. It will have six clearance holes, 0.5" diameter, spaced 60° apart. Two holes, 13/16" in diameter, will be on the centerline of two of the bolt holes, and they will be spaced 2.0" apart on centerline (one inch from center of plate on centerline).

The bolt plate must be held flat to within ± .005 on one face. Six half hard 3/8" NFT hex head bolts, 5/8" in length, with 3/8" lock washers will be used to secure the plate.

The lower resistance heater entry hole will be sealed by a bolt-on seal plate. Both thermostats will have a maximum setting of 140°F, the lower thermostat to be adjusted for a heat pump condenser, not a resistance heater. Tests have shown that the thermostat "sees" the high watt density of the resistance heater and will actually provide lower hot water supply temperatures at the same setting than with the lower watt density condenser.

3.3.3 Water Heater Tank Specification

The new type unit will use the 82 gallon energy saver model tank from Mor-Flo Industries. The use of an 82 gallon tank is predicated on providing a maximum of the hot water requirements from the heat pump and as little as possible from the quick recovery top element. The top element will be a 2500 watt resistance element which will operate in parallel with the heat pump when quick recovery is required. This will provide effective heating of 4600 watts while using a total of 3200 watts.

The selection of Mor-Flo is predicated on:

- The desire for a quality manufacturer approved by and selling to utilities which merchandise water heaters, as they will be the initial marketing channel.
- Adequate in-house engineering capability to design and tool required changes and willingness to make such changes.
- Willingness to share in the costs of the demonstration project.

3.4 Evaporator Analysis, Selection and Design - New and Retrofit Models

Based on the National Bureau of Standards hot water use survey showing an average 64.3 gallons of household hot water use per day, Energy Utilization Systems chose a 12-hour consumption period to calculate recovery requirements. This will require a 5.4 gallon per hour heating rate minimum. If we use a 6 gallon per hour recovery rate with 60°F inlet water, the heat pumping rate required to provide 140°F water will be:

$$q_H = 6(8.3)\ (140 - 60) = 4000 \text{ BTU/hr (condenser)}$$

where:

q_H = heat pumping rate (BTU/hr)

8.3 = heat capacity of 1 gallon water @ 100°F (BTU/gal - °F)

140 = final water temperature (°F)

60 = inlet water temperature (°F)

The minimum condenser capacity for the average family would therefore be 4000 BTU/hr. At a COP of 3.0 this would require 2/3 of the BTU to be supplied by the evaporator, or 2667 BTU/hr. To compensate for winter water supplies of 40°F water or less, larger families and reduce energy input, EUS selected an evaporator design capacity of 5000 BTU/hr (dry capacity at 70° ambient). The advantage of oversizing the evaporator is two-fold. A smaller fan and fan motor can be used reducing power consumption, improving E_R (COP), and reducing fan blade air noise. Larger evaporators also lower the air-to-refrigerant delta T, raising the evaporator temperature, thereby increasing the COP.

The physical size of the evaporator was also dictated by aesthetics. The bench test design was a 2 row 12" x 16" flat evaporator used with a rectangular case sitting on the floor. When the water heater tank top was determined to be the best heat pump location, the evaporator configuration was designed to conform to the shape of the tank, i.e., an arc with a 12" radius for 82 gallon tanks. A shorter radius will be used for smaller tanks. For ease of manufacturing and forming the evaporator, a single row was chosen instead of 2 or 3 rows which would be hard to form. The minimum height is determined by the overall height of the compressor which is 10". To allow for clearance and insulation, an 11" high evaporator was chosen.

Sizing a dry evaporator depends on the T.D. (temperature differential of the evaporator = inlet air temperature minus saturated suction temperature). The design formula gives the capacity (q_H) as the T.D. times the face area heat transfer coefficient at a given average air face velocity.

The ambient air should be drawn into the insulated case to capture the compressor heat and other system losses. With ambient air at 70°F, the air in the case will be 72°F. The saturated suction temperature of R-12 in the EUS heat pump water heater varies between 30° and 50°F at 70° ambient. Therefore, the average T.D. becomes 72° - 40° = 32° at 70° ambient.

Air flow velocity experiments by EUS determined that 500 feet per minute (FPM) was a maximum for comfort when in close proximity to the unit, as may be the case in some installations in living areas. Therefore, the evaporator face area-fan relationship selected was one which provided a 250 to 500 FPM velocity range over the face of the evaporator. Ten fins per inch was selected as the maximum which would not clog easily with lint. Referring to the evaporator sizing charts for wet and dry evaporators (Figures 3-4 and 3-5) from a supplier, Bohn Heat Transfer, we can see that the dry evaporator becomes the limiting case. Evaporator rating will be at 300 FPM to allow for possible future reduction of air flow due to lint.

The point on the wet evaporator chart, single row, 10 fins per inch at 300 FPM face velocity, shows a BTU capacity of 4250 BTU per hour per square foot of face area; at 400 FPM the capacity becomes 4500 BTU per hour. The circled point on the dry evaporator chart, single row, 10 fins per inch at 300 FPM shows a BTU capacity of 68 BTU per hour per square foot of face area. This, multiplied by the temperature differential, T.D., of the evaporator (32°F at 70°F ambient) gives 2176 BTU per square foot per hour. From this data it was determined that an 11" high evaporator with a finned coil area 10" high by 29" wide would be adequate at a 70°F ambient. However, at a 50°F ambient the T.D. falls to 27°F, reducing the dry coil capacity (q_H dry) to 27 x 68 = 1836 BTU per hour per square foot. The coil length was therefore increased to 36" wide x 10" high, for a total of 2.5 square feet. Higher ambients produce higher capacities and potentially better performance. The evaporator capacities therefore become:

$$q_H \text{ wet} = 4250 \times 2.5 = 10{,}625$$
$$q_H \text{ dry } 70°F = 2176 \times 2.5 = 5{,}440$$
$$q_H \text{ dry } 50°F = 1836 \times 2.5 = 4{,}590$$

Although the 50°F ambient air evaporator capacity is lower than the design point of 5000 BTU/hr, operational data show heat pumping capacity also falls with lowering ambient temperature. The 4590 BTU/hr is more than adequate at 50°F for a 3/4 H.P. output compressor. At 50°F and at a COP of 2.0, the condenser will only provide 4500 BTU/hr, 1/2 of this from air and 1/2 from the power supply.

It is worthy of note that after constructing a system with the optimized evaporator and air flow, tests were run with double the air flow. The results confirm the Bohn charts, showing a slight increase in performance. However, the improvement is more than offset by the increased fan wattage which would be part of the system power consumption.

Figure 3-4

EVAPORATOR CAPACITY—WET SURFACE

3/8" O.D. TUBE COILS — TYPE 3L1 FIN—1 × 1 INLINE

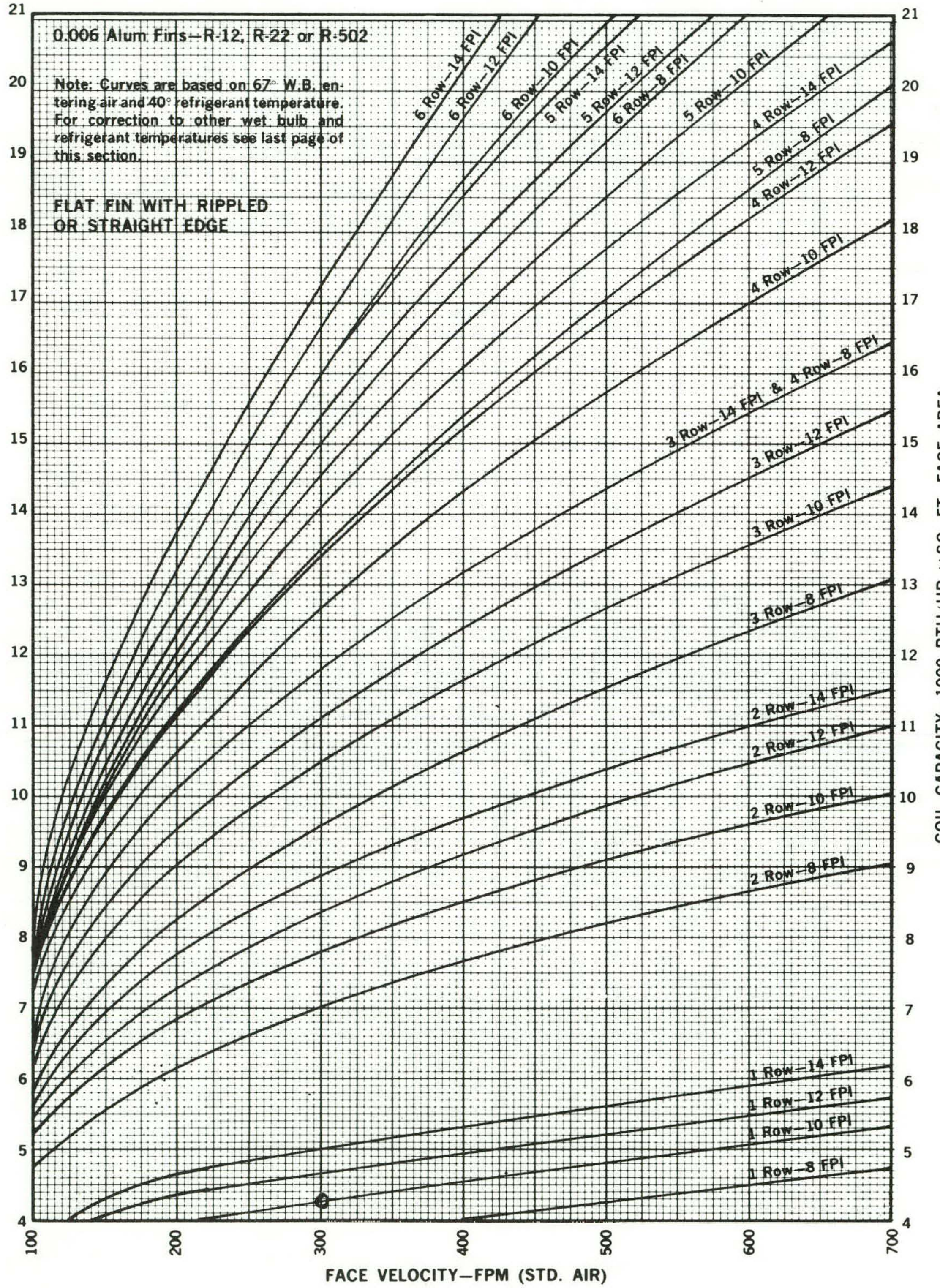
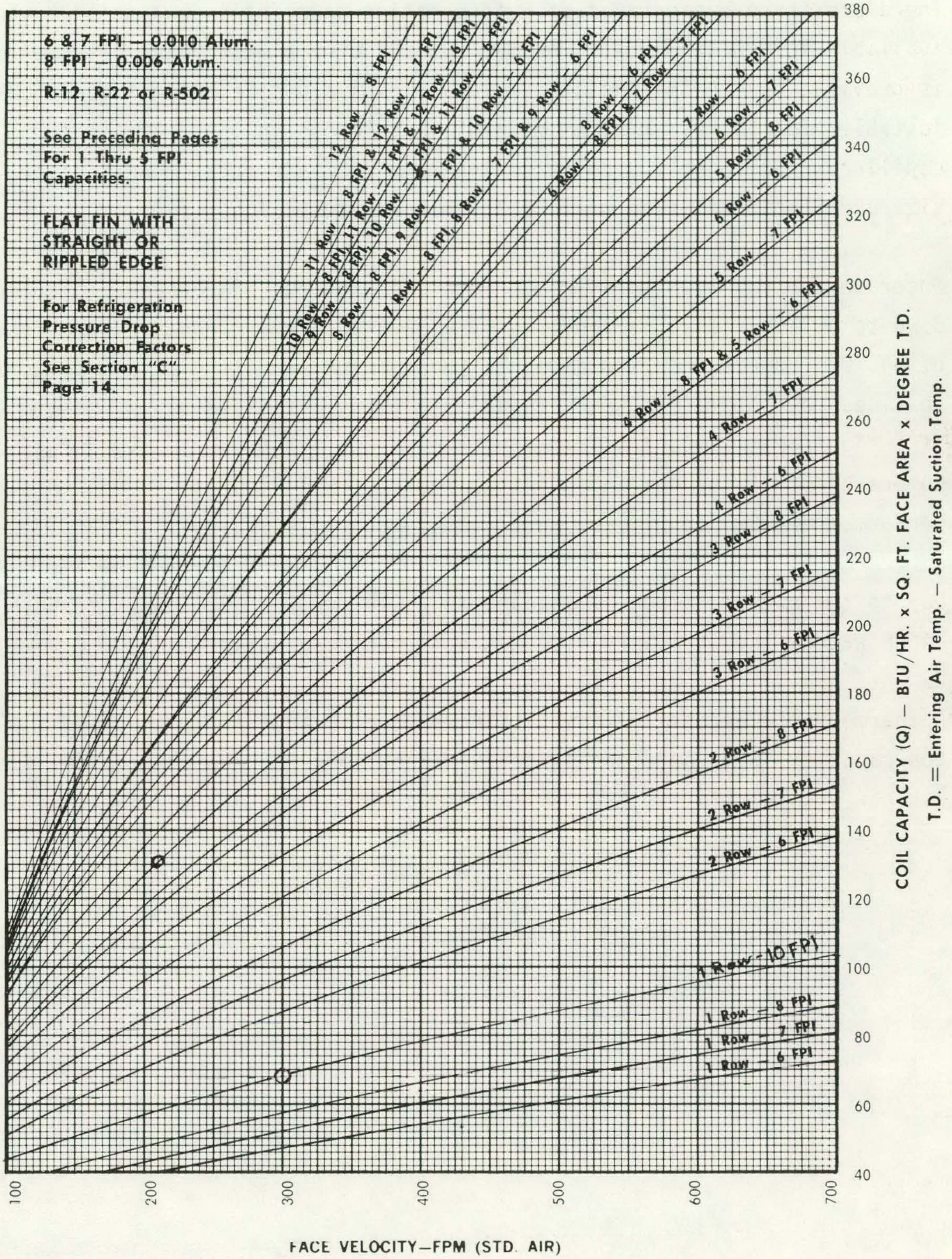


Figure 3-5

EVAPORATOR CAPACITY—DRY SURFACE 6, 7 & 8 FPI

3/8" O.D. TUBE COILS TYPE 3L1 FIN—1 × 1 INLINE



3.5 Refrigerant Expansion Device Analysis and Selection - New and Retrofit Models

There are three basic types of refrigeration restriction-expansion devices available for these units. Maximum control over a wide range of conditions is provided by the thermostatic expansion valve available, not field adjustable, for $6.50. Minimum control, but lowest in OEM cost, is the capillary tube, widely used in central and room air conditioners. The size and length required would cost approximately $1.

Moderate control with anti-frost protection for the evaporator at ambients down to 40°F or less, is provided by a constant pressure valve at a cost of $4.70. However, at low ambients this device permits liquid refrigerant to flood the compressor when its adjustment is optimized for 70°F ambient. If a significant amount of flooding does occur it can seriously damage the compressor. If the device is set to restrict the flow at low ambients, performance (COP) at higher, more normal household ambients is sacrificed.

Due to the extreme range of conditions, 50°F to 90°F ambients, and 40°F to 140°F water, the capillary tube was not considered practical.

A thermostatic expansion valve (Singer model F 233A 1/2 with bleed, 1/2 ton capacity) was therefore selected for optimum COP and compressor life for new and retrofit units.

3.6 Evaporator Anti-Freeze Up Protection - New and Retrofit Models

Even with a low ambient cut off thermostat, the evaporator can still freeze up at slightly higher temperatures when the humidity is very high if a thermostatic expansion device is used.

To prevent this, the hot liquid refrigerant line between the condenser outlet and the expansion device is serpentined across the inside face of the evaporator. There is very little system heat loss since it is captured by the evaporator.

To date, using this method, there has been no evaporator freeze up or even initial formation down to 42°F, at up to 75% relative humidity. There is also no discernable difference in COP with the added liquid line length.

3.7 Cabinet Analysis and Design - New and Retrofit Models

The water heater heat pump cabinet must satisfy several requirements:

1. Attractive appearance
2. Mountable to top of water heater
   a. to minimize floor space required
   b. to capture maximum room air temperature which is near ceiling
   c. to capture heat losses from tank top
   d. to use tank top heat losses to keep compressor warmer than evaporator in order to avoid liquid refrigeration migrating to, and slugging the compressor.
   e. provide easy assembly and packaging by water heater manufacturer.
3. Provide air chamber large enough to induce uniform air flow over the evaporator.
4. Provide an insulated internal cabinet ambient which will permit air drawn in by the fan to capture the heat losses of the fan motor, motor-compressor and refrigerant tubing and transfer this heat to the evaporator.
5. Minimize tooling and unit cost for pilot run and early production.
6. Minimize home installation differences over a standard electric water heater.
7. Provide a means to collect condensate dripping from the evaporator and eject this to an exterior drain off point.
8. Provide easy service access.

Although appearance dictated the round cabinet design, it does satisfy all the other requirements. Since the diameter of the cabinet will be equal to the diameter of each type of water heater tank it sits on, the top and bottom of the cabinet can use the jacket cap or base from the water heater manufacturer. Some modifications have to be made, i.e., the fan venturi and clearance for the tank top hole cover plate. However, considerable tooling savings do result. Unit parts cost is also low due to the water heater manufacturer's high volume production of these parts. About one half of the cabinet periphery is made up of the 11" x 36" evaporator, the balance is a 24 gauge painted, galvanized steel wrapper. The wrapper tooling is also low cost, requiring only screw hole punches and a roll form.

Forming the evaporator into a true arc to match the tank periphery is somewhat more difficult, but nevertheless, a common industry forming operation.

As can be seen in Figure 1-1, the resulting cabinet is an attractive, highly functional addition to the water heater. Attached as it is to the jacket top, it becomes an integral part of the tank assembly and is easily packed by the water heater manufacturer using a 12 inch higher carton.

The retrofit version requires two cartons, one for the heat pump in its round cabinet, and one for the helical retrofit condenser.

3.8 Compressor Analysis and Selection - New and Retrofit Models

There were a number of factors which influenced the selection of the type, size and brand of compressor, for new and retrofit units.

1. Most importantly, the operating temperatures and pressures ruled out an R-22 system. With the condenser operating with a delta T of 30°F above water temperature, to achieve 140° water temperature, condensing temperature must be 170°F. At 170°F condensing, R-22 pressure is over 450 psig. However, R-12 pressure at 170°F is 300 psig which is much more compatible to long compressor life and lower power consumption.

2. Criteria for the compressor, in decreasing order of importance were:
   a. To satisfy larger families using as much as 130 gallons of 140°F water over 12 hours, or to provide adequate recovery without the upper fast recovery electric heater coming on when the ambient air is down to 50°F, requires the capability of raising 10 gallons per hour from 50°F to 140° at 70°F ambient. The system must therefore be able to provide 7500 BTU per hour to the water at 70°F ambient.
   b. The type of compressor used should be one with the highest possible E.E.R. manufacturer's rating.
   c. The model selected should be one produced in sufficient volume to make it low or lowest in cost.
   d. It should operate satisfactorily on a 208 or 230 volt power supply.
   e. It should be compatible with a 115 volt model for those installations where a 230 volt supply is not available.

The compressor selected for new type units was the Copeland model JRL4-0100 PAV with an E.E.R. of 8.52, 208/230 volt, single phase, 60 cycle. This model is a nominal one horsepower (11,000 BTU/hr) R-22 compressor. Derating it to R-12 provides about 7500 BTU/hr. The R-22 compressor was selected because it is produced in much higher volume with resultant lower cost and has a better E.E.R. than a 7500 BTU/hr R-12 compressor.

At one point it was felt that even with R-12, the discharge pressures (and temperatures) might be too high. An alternative was to consider R-114 even though it is not commonly available. DuPont calculated for EUS, Inc. that R-114 would derate a 3/4 H.P. compressor to 0.09 horsepower (1083 BTU/hr) and a one H.P. compressor to 0.14 H.P. (1650 BTU/hr). This was obviously unacceptable and received no further consideration.

For 115 volt installations model JRL4-0100-PAG is a direct substitution for the 230 volt model with identical E.E.R. and BTU output. A representative Copeland calorimeter test is shown in Figure 3-6.

Due to the limited length and heat transfer area of retrofit condensers, compressor size had to be reduced. The heat transfer limitation of this condenser is about 5000 BTU/hr. An attempt to pump more than this through it results in overheating the compressor.

For the above reasons the Copeland JRE4-0075-PAV 3/4 ton R-22 compressor was selected. It has a E.E.R. of 8.88 on a 208/230 volt single phase 60 cycle power supply. When used with R-12 it will provide 5000 BTU/hr.

A 115 volt motor-compressor such as the Copeland JRE4-0075-PAA, with an E.E.R. of 8.88, could also be used.

For use in the heat pump water heater, Copeland has agreed that the compressors will have a normal one year warranty:

1. as long as the compressor does not exceed 350 psig discharge pressure under normal worst condition operation, 90-95°F ambient and 145° delivery water temperature.
2. compression ratios do not exceed 7 to 1.
3. refrigerant system charge does not exceed the capacity of the compressor crankcase (28 oz.).
4. return of liquid refrigerant to the compressor is prevented by adequate suction line super-heat or other means.

Figure 3-6

Copeland

CALORIMETER TESTS

Refrigerant______ Log Sheet No.______

Cond. Unit Model ______, Serial No. 76E 09266, Date 5-12-76

Compressor Model JRL4-0100-PAV-XXX, ☐ A ☐ W ☐ R Cooled, Serial No.______, Protector MRA-8923-309

Motor Part No. 046-5020-07, Mfgr. E.C., Type 071-0370-18

Volts 208/230, Cycle 60, Phase 1, H. P. 1, Run. Cap. 20 MFD

Cal. No. 1, Cal. Factor 61440, Heat Leakage BTU/1° F. T. D.______, Baro.______, Pjt. No.______

~~TEST NUMBER~~ REFRIGERANT	22		12		
TIME START ( )					
ROOM TEMP. (COMPRESSOR)	95°		95°	SAMPLE #45144	
ROOM TEMP. (CALORIMETER)					
R. P. M.	3450		3525	P.O. # 103	
CFH/~~1416~~ 3500 R. P. M.	141.5		144.6		
SECONDARY PRESSURE					
SECONDARY TEMPERATURE					
KWH METER END	6830.5		7025.2	ENERGY UTILIZATION	
KWH METER START	6765.0		6982.6	SYSTEM INC.	
KWH NET	65.5		42.6		
TIME IN SECONDS	369		363	PITTSBURGH, PA.	
SECONDARY TO AMB. T. D.					
HEAT LEAKAGE BTU/HR.					
CAPACITY BTU/HR. (HEATERS)	10906				
CAPACITY BTU/HR. (GROSS)					
CORRECTED TO 115° LIQUID	11233		7210		
BTU/WATT HR.	9.36		8.58		
BTU/CU. FT.	79.38		49.86		
VOLUMETRIC EFFICIENCY					
EVAPORATING TEMPERATURE	45°		45°		
CONDENSING TEMPERATURE	130°		130°		
SUCTION PRESSURE	76#		41.7#		
SUCTION TEMPERATURE	95°		95°		
DISCHARGE PRESSURE	296.8#		181.0#		
LIQUID TEMPERATURE	121°		115°		
VOLTS	230		230		
WATTS INPUT COMP. MOTOR	1200		840		
WATTS INPUT FANS					
WATTS INPUT TOTAL					
AMPERES COMP. MOTOR	5.32		3.80		
AMPERES FAN					
AMPERES TOTAL					
MOTOR TEMP. SEELY					
VALVE PLATE GASKET					
OIL TEMPERATURE					
SUCTION TEMPERATURE @ V. P.					
SUCTION TEMPERATURE (LINE)					
DISCHARGE TEMPERATURE @ V. P.					
DISCHARGE TEMPERATURE (LINE)					
MOTOR HOT SPOT MAIN					
MOTOR HOT SPOT START					
CONDENSER WATER OR AIR IN					
CONDENSER WATER OR AIR OUT					
TEMPERATURE DIFFERENCE					
LBS. OF WATER PER. MIN.					
BTU/AMP.					
POWER FACTOR					

$$*BTU/HR. = \frac{\text{CAL. FACTOR X KWH}}{\text{TIME IN SECONDS}} \qquad \text{POWER FACTOR} = \frac{\text{WATTS}}{\text{AMPS X VOLTS}}$$

LAB NO. A-10,686 CC-737

TESTED BY Q.C. LAB - S. LEE W. ORO

3.9 Condenser Analysis and Design - New and Retrofit Models

The function of the condenser is to transfer heat from the compressed refrigerant to the potable water supply. The design selected shall be optimized to satisfy the following requirements:

1. Provide high performance during daily withdrawal cycles as opposed to rare cold tank start up conditions.
2. Provide inherent corrosion resistance to refrigerant, refrigerant chemicals and water solutions for at least 10 years.
3. Protect potable water supply from contamination by any chemicals in refrigerant system.
4. Have no adverse effect or reaction with standard water heater tank anodes, tank coatings or other exposed components.
5. Provide external failure indication if refrigerant line fails or water barrier sheath fails.
6. Seal heat transfer material in annulus between refrigerant tube and water barrier sheath to contain material at least 10 years against normal operating pressures, but rupturable at higher refrigerant or city water pressures (20 psig or higher).
7. Be designed for easy assembly into the tank.
8. Provide 10 year performance without significant degradation due to scale build up on exterior heat transfer surface.
9. Have sufficient heat transfer area to provide a delta T from refrigerant condensing temperature to the potable water supply which will provide the most effective heat transfer.
10. Be sized to provide good refrigerant oil return to the compressor.
11. Be easily removed for service or replacement.

New Model

Although much hope was held for an extruded aluminum tube-in-tube integral fin condenser, tests by ALCOA of various alloys and platings proved unsuccessful in several types of potable water supplies. Copper became the next most cost effective choice, and although there are no guarantees on its life in varied water supplies, it is commonly used in all parts of the country, and except in rare instances, provides at least a 10 to 20 year pipe life.

The final condenser design is comprised of 20 feet of 3/8" O.D. convoluted copper refrigeration tubing, .032 wall in a 3/4" O.D. convoluted copper sheath .028 wall (Figure 3-7). The annulus between the tubes is continuous from end to end, filled with a free-flowing, thermally conductive, non-toxic substance, i.e., water. The spiral tubing convolutions increase the external heat transfer area per linear foot by 2 to 4 times that of a straight tube of the same diameter. See Figure 3-8.

The condenser tubing is annealed and formed into coils having a maximum outside diameter of 4.25". The 1/4" liquid refrigerant return line and its 3/4" sheath, both smooth copper refrigeration tubing .032 wall, are brazed to the lower end of the coil and rise vertically through the center of the coil to the bolt plate. The hot compressed gas inlet line is 3/8" in a 3/4" sheath and brazed to the top of the coil. The inlet and return refrigerant lines shall extend 6.0" above the bolt plate when the condenser is just above the tank bottom (1" - 0 + 1"). The sheaths shall terminate at and be brazed to the bolt plate and to the refrigerant tubing except for a connection to a 1/4" vent tube attached at that point.

The vent tube tip is lubricated with silicone grease and capped with an external, 1/4" tube cap 5/8" long, Niagara Plastics #275 or equal, polyethylene or polypropylene, heat resistance 220° minimum.

The annulus shall be filled with water to a point not exceeding one foot above the top coil of the condenser. This is for three reasons:

1. Allow for expansion of the water without blowing the cap off under normal operating conditions
2. Prevent reheating of the condensed returning liquid refrigerant as it passes through the hot water at the top of the tank.
3. Avoid dissipating the condenser super heat in the water at the top of the tank which is normally already at the desired delivery temperature.

The final condenser assembly is tin plated to prevent electrolytic reaction of the tank anode to the copper. Small imperfections will not cause problems since it is the magnesium anode which is etched, not the copper.

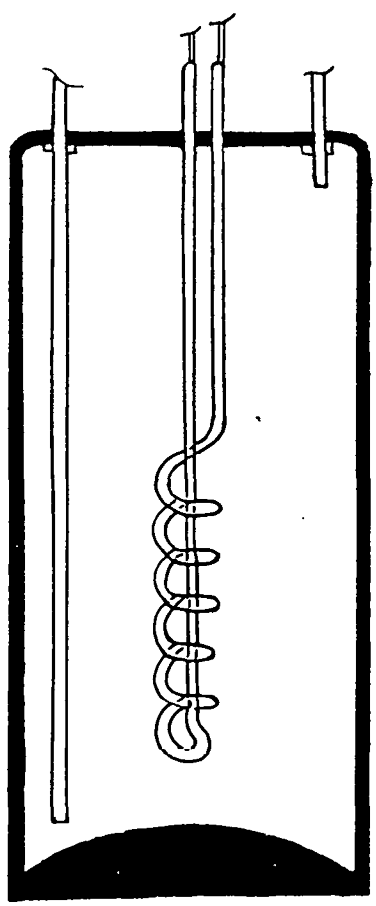


Figure.3-7

For new water heaters, condenser is formed into "chimney" type coil which increases convective flow and achieves a maximum of heat transfer.

Schematic of Consenser Inside Tank

Fluted Copper Tubing

Figure 3-8
Condenser is formed from double tubing with annulus filled with water to improve heat transfer.

Retrofit Models

The final condenser design will be tin plated, copper refrigeration tubing as in the new type condenser. However, since the retrofit condenser must be entered into the tank through the lower resistance heater hole its configuration is radically different.

The length and diameter of the retrofit condenser is limited by the diameter of the retrofit tank and the size of the heater hole (usually 1 1/8" O.D.). Forming of the condenser is very critical in order to provide easy entry of maximum length. Basically the detached condenser is "screwed" into the hole and therefore is helical in shape. Experimentation demonstrated that a helical coil comprised of 2 - 1/2" O.D. smooth tubes provided one solution for maximizing the condenser length and area inside the tank. (See Figure 3-9).

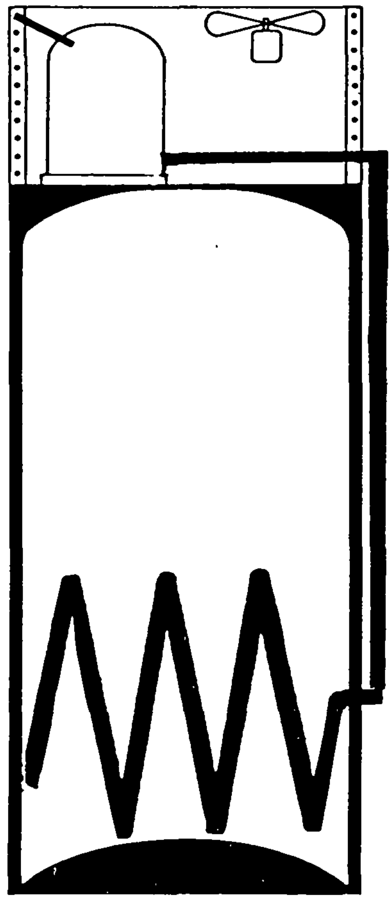


Figure 3-9

Sketch of Helix Condenser

Unlike the new type condenser, the sheaths on the retrofit must be smaller (0.50 O.D.) and smooth on the outside in order to slide through the small heater hole. The refrigerant tube is 3/8" O.D. with a .032 wall and convoluted to provide improved contact to the sheath. The annulus between them is filled with water and sealed with a vent tube and cap as in the new type condenser. The maxiumum inscribed length is 19.5 to 20.0 feet.

A major problem was encountered designing the return bend of the tube and sheath. No known devices would fit through a 1 1/8" hole. Special return bends were designed which satisfy all the criteria of double tube protection and fit through the small entry hole. All connections are brazed. The return bend is shown schematically in Figure 3-10.

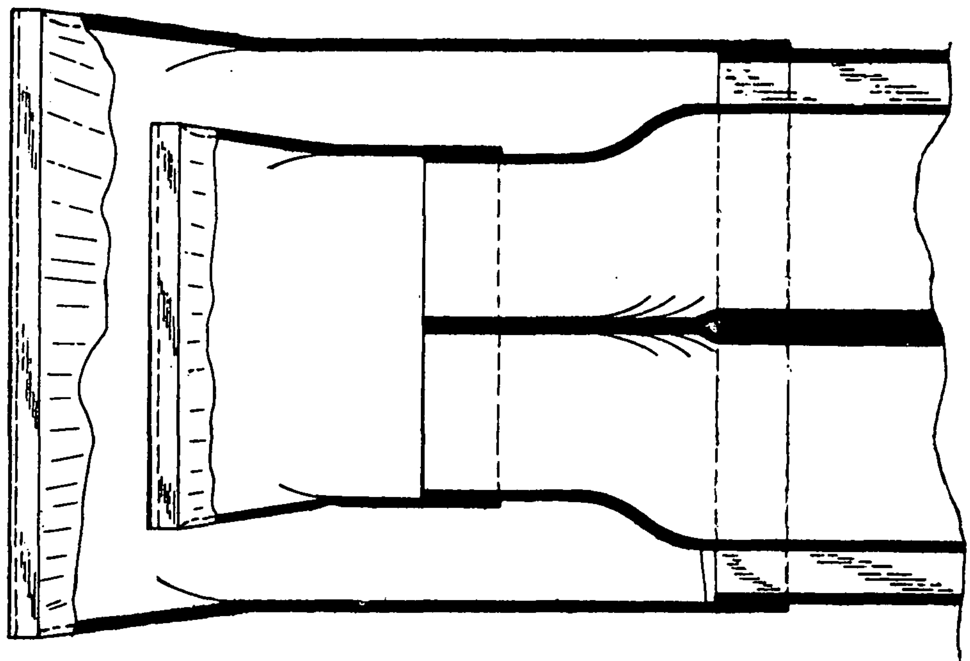


Figure 3-10

Sketch of Return Bend

The seal to the tank required two solutions, one for a bolt plate seal, the other for a 1 1/8" (1" pipe size) threaded hole. In both instances the adapter plate is brazed to the sheath. Retrofit condensers must be ordered with a differentiation determined by type of seal.

Connections to the heat pump system are made by way of a 3/8" compressed gas line and 1/4" liquid line, both insulated, external to the tank and jacket, extending from the condenser up to the heat pump cabinet.

Precharged condenser and system lines will be sealed with Aeroquip 3/8" connectors, similar to split system central air conditioning system connections. This permits installation of the separate condenser and the heat pump on top of the tank with the added task only of connecting the two sealed precharged systems.

The installation requirements of retrofitting the condenser must not be minimized. The tank must be drained, which requires one to two hours. There must be at least three feet of space available extending outwardly from the heater hole. The tank should be less than five years old to justify the cost of change-over commensurate with the balance of the tank life. However, a retrofit can be converted to a new type tank at only the cost of a new condenser.

Only 24" diameter or larger tanks (82 gallon or 50 gallon low boys) are suitable for retrofitting because smaller diameter tanks do not permit the introduction of sufficient condenser to be effective.

As noted in the compressor design section, retrofit heat pump water heaters, because of their less effective condensers, are suitable only for 5000 BTU/hour output compressors. Thus, their recovery rates are not as good as the 7500 BTU/hour new type units.

3.10 Electrical Components - New and Retrofit Models

1. Compressor Run Capacitor - 20 MFD, 270 Volts. Supplied by compressor manufacturer.

2. Compressor start relay - supplied by compressor manufacturer. GE, 180 volt minimum pickup (cold), 200 volts maximum (hot). Drop out volts 55 minimum, 115 volts maximum. Coil volts 336, 60 hertz.

3. Compressor start capacitor - supplied by compressor manufacturer. 41 - 53 MFD, 220 volts.

4. Compressor motor overload protector - automatic reset, supplied by manufacturer, from Texas Instruments.

5. Low ambient cut off thermostat - Cut off at 47°F, cut in at 57°F. Manufactured by Therm-O-Disc or Texas Instruments.

6. Water heater tank controls and neaters - supplied by tank manufacturer, Mor-Flo.
    a. thermostats - standard except with 140°F maximum temperature setting
    b. upper heater - 2500 watt electric resistance heater will operate in parallel with heat pump for fast recovery.
    lower heater - none.

4. References

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C-1. TRNSYS, A Transient Simulation Program, Solar Energy Laboratory, University of Wisconsin, Madison, Wisconsin, (Engineering Experiment Station Report 38).

C-2. "Solar Heating and Cooling of Residential Buildings - Design of Systems", Solar Energy Applications Laboratory, Colorado State University, U.S. Department of Commerce, October 1977.

C-3. Climatological Data, National Summary, Vol 26, National Oceanic and Atmospheric Administration, National Climatic Center, Asheville, North Carolina, 1975.

C-4. Tamami Kusuda, NBSLD, The Computer Program for Heating and Cooling Loads in Buildings, U.S. Department of Commerce, Washington, D.C. July 1976.

APPENDIX A

Summary of Task 3 Work

Heat Flow Analysis

This task was performed by sub-contract to General Energy Associates at Drexel University and DuPont Corporation, in conjunction with EUS. The data and calculations are included in Appendix C. The conclusions are incorporated in each of the pertinent design analyses.

Develop Test Procedure

After consultations with the Oak Ridge National Laboratory and the National Bureau of Standards, a heat pump water heater test procedure was submitted to the Oak Ridge National Laboratory Technical Manager (ORNL-TM) and approved in October, 1977. It is essentially the same as the proposed procedure published in the Federal Register dated April 27, 1977, Part II, pertaining to water heaters. The revisions were only those necessary to assign the proper energy equations to the heat pump water heater and inclusion of a water draw cycle. A copy of this test procedure is included in Appendix B.

All tests performed by EUS (and the sub-contractor, Associated Test Laboratories), have followed the guidelines established in the above test procedure since its acceptance by ORNL.

Since heat pump water heaters have a better E_R in cold water, daily cycling E_R or COP can never get as high as cold start E_R on a heat pump water heater. Unless a tank is extremely well baffled and the condenser is only in the new cold water section, the mixing effect raises the water temperature the condenser sees and results in a lower E_R.

E_R rating on cold start or cycling tests includes tank losses. To compare to a resistance heater to get COP, either divide by the E_R of a resistance heater in the same tank or assume the COP of a resistance heater is one (1) and add the total hourly tank losses to the measured heat pump output. Thus COP, as used in this project, refers to the efficiency of the heat pump system to transfer heat to the water (i.e., it can be compared to a resistance heater COP of one).

Figure A-1 shows the decline of heat pump COP (E_R) with increasing hot water temperatures. Voltage effects are also shown.

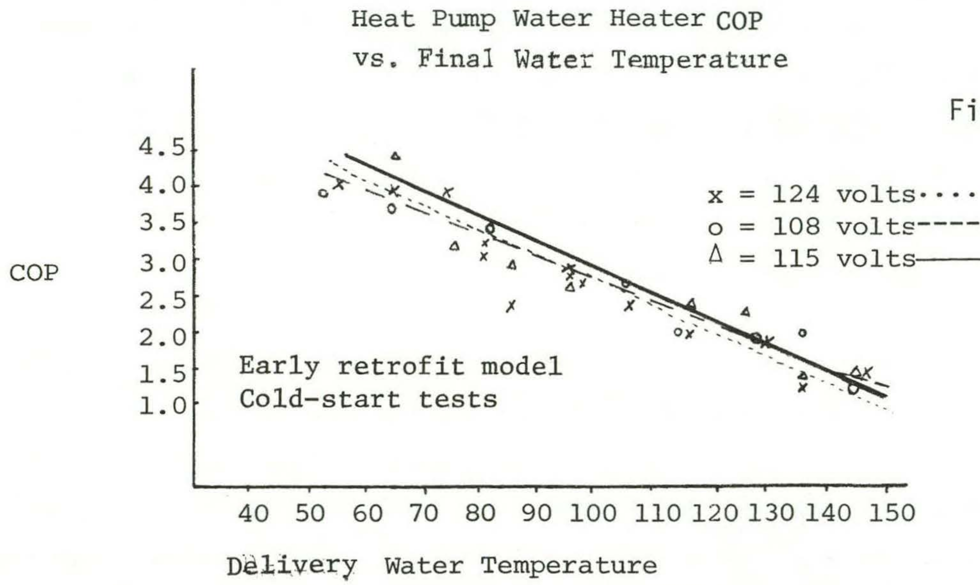


Figure A-1

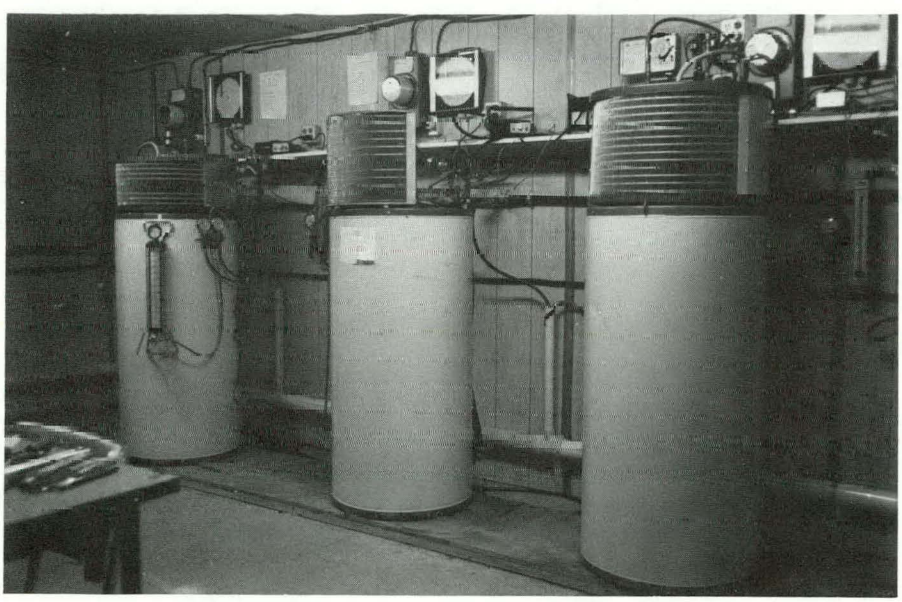


Figure A-2

EUS Test Room

Construct Test Room

The test room at the original EUS location was replaced at the new EUS site with a much larger room, 12 feet by 24 feet. (see Figure A-2) The new room is automated to perform daily withdrawal cycles of 64 gallons and cold starts day and/or night. Voltage, ambient air temperatures, water supply temperatures, and withdrawal water temperatures are also automatically controlled to less than the plus or minus 5°F allowed by the procedure.

Tank water inlet and outlet temperatures are measured and recorded on TRERICE disc chart recorders. The temperature probes are located in the water flow paths.

Room ambient temperatures and humidity are recorded on a BACHARACH servex disc recorder located four feet off the floor, five feet away from the nearest tank per NBS.

Tank water temperatures are measured with OMEGA 10 point digital readouts with iron-constantan thermocouples. The points measured are on the centerline 1/6, 2/6, 3/6, 4/6, and 5/6 down from the top of the tank along with three inches from the bottom and the top.* A thermocouple is also located at the lower tank thermostat. (This usually reads 3°F lower than the average tank water temperature at shut off).

Heat pump system temperature measurements, also using OMEGA 10 point digital readouts, are taken at (1) discharge line six inches from compressor for super heat, (2) return liquid line at condenser exit from tank, (3) liquid line just before expansion devices, (4) evaporator refrigerant gas inlet, (5) evaporator air outlet, (6) evaporator refrigerant gas outlet.

Discharge pressure and condensing temperature, suction pressure and temperature, and system pressure drops, are measured with SUPERIOR refrigeration pressure - temperature gauges.

Power supply is controlled with 230 volt variable transformers.

Laboratory heating is accomplished by collecting the 120° to 140°F tank withdrawal water in a 175 gallon insulated tank. This water is then pumped through a fan coil in the laboratory as the room thermostat calls for heat.

*These locations are different from those of the procedure (Appendix B).

Cooling is supplied (down to 45°F) by the heat pump water heater evaporators as they run on their various tests. Additional cooling is provided by a through-the-wall air conditioner.

All tanks are mounted on 3/4 inch plywood platforms resting on 2" x 4"'s laid flat, per NBS specifications.

Tank water inlets and outlets enter the tank vertically at the top (except for the latest experiments with bottom tank tap horizontal water inlets) and become horizontal within one foot of the top of the heat pump (24" of the top of the tank).

Build and Test Retrofit Unit

A bench model 3/4 ton heat pump water heater was used to test the various condenser configurations. Data from a ROBINAIR volt-wattmeter was used to calculate performance. Based on later conversion to DUNCAN watt hour recorders at ± 0.1% accuracy, the early ROBINAIR meters read 10 to 15% low, giving an E_R (COP) 10 to 15% high -- but comparison data on the same meters is still valid -- two tests each were run.

The condenser configuration and E_R cold start results are as noted in Table A-1, all condensers 20 feet long (2 - 10 foot lengths):

TABLE A-1

Condenser Configurations and E_R Cold Start Results

(Retrofit Unit)

Type Condenser (20 feet long)	Annulus	E_R	% Dev. (+)
Base Cu. 3/8" refrigerant R-12 condenser tubing	None	2.49	
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath	Air	1.86	0.5
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath	Water	2.22	1.0
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath plus external 0.15 x 1" longitudinal fins	Water	2.13	1.0

Type Condenser (20 feet long)	Annulus	E_R	% Dev. ±
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath	Si. Grease	1.95	0.5
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath	Al. Powder	*	
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath	Al. Powder + Mineral Oil	2.55	2.0
3/8" R-12 tubing in 1/2" 0.32 wall cu. sheath plus external 0.15 x 1" longitudinal fins	Al. Powder + Mineral Oil	2.14	2.0
New Spiral R-12 tube in 1/2" .032 wall cu. sheath	Air	2.32	
New Spiral R-12 tube in 1/2" .032 wall cu. sheath	Water	3.12	.000
Special external pumped condenser 3/8" spiral cu. R-12 in 3/4" cu. spiral sheath in 1" plastic water pipe	Water	2.60	25.0

* Test had to be terminated due to system over-pressure, over-temperature conditions. The condenser heat transfer was inadequate.

Blow out tests were performed on all of the condensers to simulate a city water or refrigerant leak into the annulus. (20 psig water - 70 to 350 psi, R-12)

Annulus Filling	Gauge Pressure PSI	Failure Indication R-12	Water	Failure Mode
Air	0	none	water	safe
Water	2	water	water	safe
Silicone grease	100	grease	none	not acceptable
Aluminum Powder	100	Al. Powder	none	not acceptable
Aluminum Powder-Oil	1000	none	none	not acceptable

Based on the blow out tests the only acceptable annulus filling of those tested is air or water. Since water is superior in heat transfer ability to air, the annulus will be filled with potable water in all cases.

The external condenser using a recirculation pump was not considered acceptable for three reasons:

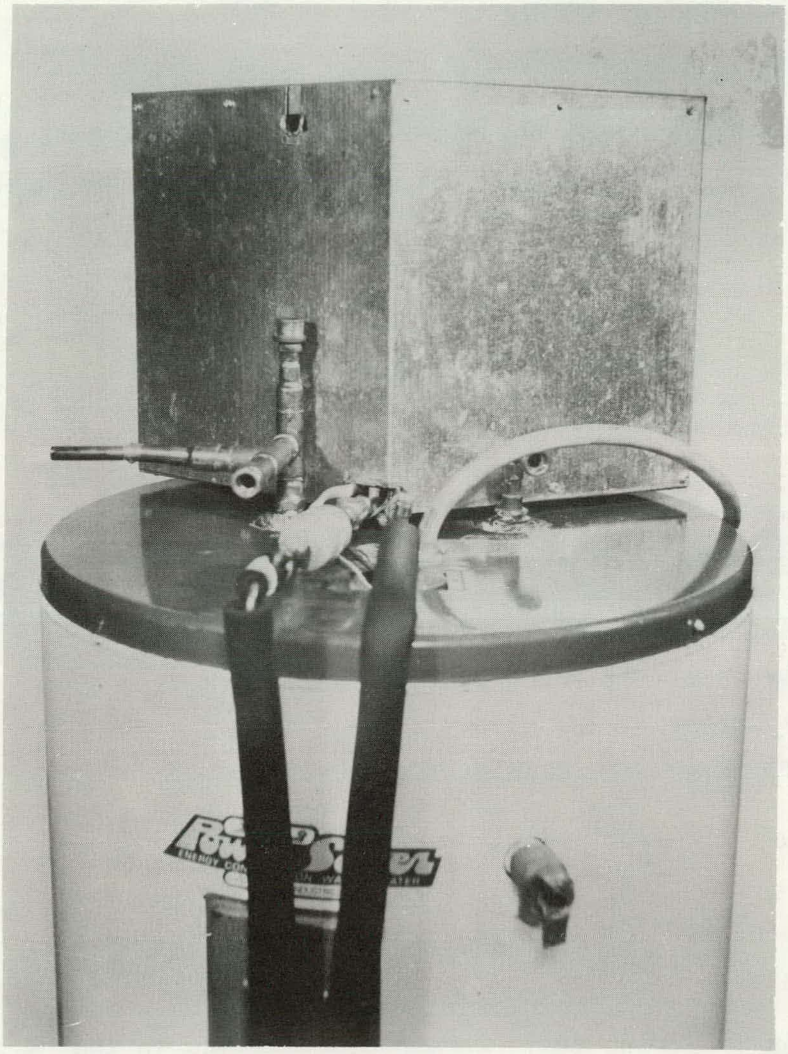
1. The estimated added retail cost of such a system was estimated to be more than $75 (3 times base labor and material)
2. It did not perform as well as the spiral tube condenser, and
3. To achieve the E_R noted, the compressor ran continuously at pressures and temperatures close to specified limits, even in a 70°F ambient.

Fins are not included in the final design because of the shape of the helical coil in the tank. Approximately 1/2 the length of tubing is vertical or nearly vertical and 1/2 is horizontal or nearly so. Longitudinal fins block the convection flow of water in the horizontal tubing. Radial fins cannot be entered through the retrofit hole. General Energy Associates calculated that a helical retrofit condenser could have a maximum benefit of 10 to 15% with longitudinal fins. It would be possible to have a net decrease if the heat transfer to the fin was not adequate (a line contact). The EUS tests confirmed the latter condition.

The new copper internal fluted tube in sheath condenser (by Spiral Tubing Corporation, See Figure 3-9) provides at least 18% better E_R than the smooth tube-in-tube design and will therefore be the component used for retrofit condensers.

One prototype retrofit unit (Figure A-3) was designed and built using a 3/4 H.P., R-22 compressor, a single row three square foot evaporator and a 700 C.F.M. fan per General Energy Associates calculations. The system was mounted on top of the tank with insulated copper discharge and liquid line extensions down to the sheathed spiral tube retrofit condenser.

Figure A-3
Prototype Retrofit Unit



The prototype retrofit unit was modified to operate with variations in evaporator-fan size relationships, condenser lengths (area) and compressor size. (See design analysis for details.)

Using a 3/4 H.P., R-22 compressor charged with R-12 refrigerant, the condenser length was also varied. The cycling (25 gallon draw) E_R varied from 1.64 at 15.5 feet, 1.67 @ 17.5 feet, to 1.72 at 19.5 feet.

Double row 12" x 18" evaporators, 18" x 24" single row and 12" x 36" single row were all tested with air flows from 700 to 1500 ft^3/min. (250 to 1000 F.P.M. face velocity). When the system was fine-tuned for each condition, performance efficiency varied less from one change to another than from test to test of the same component (± 2.0%).

The optimized retrofit units were tested for a series of water supply, ambient air and delivery temperatures. They were also tested for extreme conditions.

The sensitivity tests have also been run by an independent sub-contractor, Associated Test Laboratories, Inc. (ATL). Their tests corroborate the EUS tests very closely on those early design units.

The EUS results of the sensitivity tests are as follows: E_R - cold-start tests for retrofit unit

a.) 60° source water - heated to 140°F
50° ambient - 1.68
70° ambient - 2.27
90° ambient - 2.91

b.) 70° ambient air - water heated to 140°F
40° supply water - 2.45 cold start
60° supply water - 2.27 cold start
80° supply water - 2.18 cold start

c.) 70° ambient air - 60° inlet water
water heated to 120°F - 2.44 cold start
water heated to 130°F - 2.32 cold start
water heated to 140°F - 2.27 cold start

d.) Evaporator freeze up tests were run, resulting in the compressor cycling on its built-in motor protector until the room temperature was raised.

e.) Overtemperature tests were also run to 173°F water temperature. Compressor was shut off by safety thermostat on the tank.

The first new type heat pump water heater was designed and built similar in appearance to the top mounted retrofit design. (Figure A-3, see design analysis for details).

Condenser variations included 20 and 40 foot lengths, 1/2" smooth sheath, and 5/8" and 3/4" fluted sheaths, to obtain changes in condenser heat transfer area.

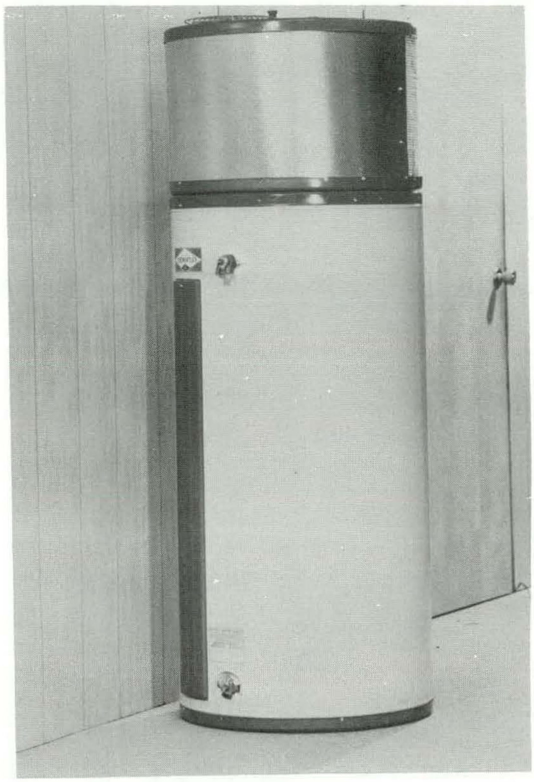
Because of the large condenser entry hole, the total heat transfer area of the condenser can be varied by either length, diameter or fins (fluting). The externally fluted spiral tubing has 2 to 4 times the area per foot as smooth tubing the same size, depending on the number of flutes per inch (see Figure 3-8).

Based on the tests performed, only the one H.P. and the 3/4 H.P. R-22 compressors using R-12 refrigerant were tested for optimization with the 11" x 36" evaporator. The condenser test and selection also included varying the condenser coil shape. Coils 20" in diameter located at the very bottom of the tank, both in 5/8" and 3/4" external fluting gave a cold start E_R of 2.45.

These coils were extremely difficult to install into the tank. The new chimney type coils up to 4 1/4" diameter are more easily installed. They show an E_R of 2.90 and higher (cold start).

The final new type unit (Figure A-4) is therefore top mounted as is the retrofit unit. The evaporator is 11" x 36" single row, a 1 H.P. compressor, a one-half ton thermal expansion valve, a 20 foot long condenser with fluted 3/8" refrigerant tube in 5/8" fluted sheath coiled in a chimney type vertical helix coil 4.25" O.D.

Figure A-4



Final New Type Unit

The newest 20' - 5/8" fluted sheath is equal in area to 80 feet of 3/8" tubing.

A total of eight units have been built. Only three models are not on life test -- the bench test model and two at Associated Test Laboratories. The two units at ATL were shipped to them in December, 1977. As such they represent the development advancement as of that time.

The performance of the former design new model condenser at ATL is not as good as the latest 20' - 5/8" chimney unit. Also, the new cold water inlet suggested by NBS adds 10% to the daily cycling E_R, new and retrofit models.

The tests performed by ATL, even though not on the latest units, serve their function of validating EUS testing methods and accuracy (see Tables A-2 and A-3).

Sensitivity tests performed by EUS and ATL are as follows: E_R-cold start tests for new type unit.

a.) 60°F supply water heated to 140°F

	Preliminary Design	Final Design
50° ambient	1.72	1.98
70° ambient	2.33	2.92
90° ambient	5.53	3.03

b.) 70°F ambient air - water heated to 140°F

40° supply water - 2.48 EUS
60° supply water - 2.33 ATL
80° supply water - 2.12 ATL

c.) 70°F ambient air - 50° supply water

water heated to 120°F - 2.73
water heated to 130°F - 2.67
water heated to 140°F - 2.59

20 foot 3/4" fluted chimney condenser (semi-final design)

All units continue on life test. There has not been any failure of any kind other than one compressor that failed after 24 hours with a bad bearing (analyzed by Copeland on its return and exchanged).

Hot Water Supply Rating

The heat pump 82 gallon Mor-Flo energy saver tank set at 137°F with no upper or lower resistance heaters, delivers 60 gallons of water down to 134°F and 73 gallons down to 97°F per the test procedure, using the improved cold water inlet suggested by NBS.

Standby Losses

Standby losses for the heat pump 82 gallon energy saver tank are 450 BTU/hr with ambient air at 70°F and tank water temperatures averaging 137°F.

At those conditions, to convert E_R to COP, the elapsed hours of the test are multiplied by 450 and added to the calculated heat pump output (gallons x 8.3 x °F rise).

In the sensitivity test series, heating water to 140°F from 50°F had an E_R of 2.59. This becomes a COP of 2.72 as an example.

$$COP = \frac{450 \times 7.5 + 82 \text{ gallons} \times 8.3 \text{ lb/gal.} \times \text{ °F rise}}{3.412 \times 6960 \text{ system watt-hrs}} = \frac{64630 \text{ BTU}}{23748}$$

COP = 2.72

a summary of ATL and EUS test results is given is Tables A-2 and A-3.

Table A-2
ATL and EUS Cold-Start Test Summary
(Temperatures all ± 5°F)

ATL Average E_R		EUS Average E_R	
70°F ambient, 60°F water to ≃ 140°F			
New type unit #1	2.33	New type unit #1	2.35 (55°F H_2O)
Retro type unit #2	2.22	Retro type unit #1	2.27
70°F ambient, 40°F water to ≃ 140°F			
New type unit #1	2.26	New type unit #1	2.48
Retro type unit #2	2.46	Retro type unit #1	2.45
70°F ambient, 80°F water to ≃ 140°F			
New type unit #1	2.12	New type unit #2	2.17
Retro type unit #2	2.10	Retro type unit #3	2.18
50°F ambient, 60°F water to ≃ 140°F			
New type unit #1	1.72	New type unit #1	1.94 (52°F H_2O)
Retro type unit #2	1.66	Retro type unit #1	1.68
90°F ambient, 60°F water to ≃ 140°F			
New type unit #1	2.53	New type unit #4	3.03
Retro type unit #2	2.57	Retro type unit #3	2.91
70°F ambient, 60°F water to ≃ 130°F			
New type unit #1	2.45	New type unit #1	2.48 (53°F H_2O)
Retro type unit #2	2.39	Retro type unit #1	2.32
70°F ambient, 60°F water to ≃ 120°F			
New type unit #1	2.50	New type unit #1	2.55 (53°F H_2O)
Retro type unit #2	2.53	Retro type unit #1	2.44

Table A-3

ATL and EUS Cycling-Test Summary*
(Temperatures all $\pm$ 5°F)

ATL Average E_R		EUS Average E_R	
70°F ambient, 40°F water to ≃ 140°F - 65 gallons			
New type unit #1	≃ 1.50	New type unit #4	1.72
Retro type unit #2	≃ 1.50	Retro type unit #1	1.60**
90°F ambient, 60°F water to ≃ 140°F - 65 gallons			
New type unit #1	≃ 1.56	New type unit #1	2.02
Retro type unit #2	≃ 1.70	Retro type unit #1	2.03**
50°F ambient, 60°F water to ≃ 140°F - 65 gallons			
New type unit #1	≃ 1.10	New type unit #2	1.28**
Retro type unit #2	≃ 1.17	Retro type unit #1	1.23**
70°F ambient, 60°F water to ≃ 140°F - 65 gallons			
New type unit #1	≃ 1.39	New type unit #2	1.78
Retro type unit #2	≃ 1.35	Retro type unit #1	1.60**
70°F ambient, 80°F water to ≃ 140°F - 65 gallons			
New type unit #1	≃ 1.41	New type unit #2	1.67
Retro type unit #2	≃ 1.37	Retro type unit #3	1.96

* Lack of accumulative watt-hour meter data by ATL makes these tests difficult to calculate accurately. Some EUS cycling tests had reduced off time in order to run 2 tests per day. Except on 50°F ambient tests which took more than 12 hours, shortening the cycle time could inflate the E_R (cycling only) by 3 to 5%. #1 units, new and #2 retro, are ATL units. Higher numbered units with improved designs and higher E_R's were used only when good test data was not available for these. Allowable temperature variations of ambient or water can vary E_R's by more than $\pm$ 5%.

** Composites of 2 or more tests.

APPENDIX B

TEST SPECIFICATIONS AND PROCEDURES

These specifications and procedures are designed to establish:

a) The efficiency (Coefficient of Performance - COP) of the heat pump water heater compared to a standard electric resistance water heater using the NBS average of 64.3 gallons per day per family usage at designated hot water output temperatures, an average source water temperature of 55°F in an ambient of 70°.

b) The performance and COP of the heat pump water heater using source water temperatures of 40°, 60° and 80°F, ambients of 50°, 70° and 90°F, and water output temperatures of 120°, 130° and 140°F, compared to an equal size electric resistance water heater. Since the normal maximum temperature setting for this type heat pump will be 140°F water, no performance or COP rating tests will be run at higher temperatures. Accelerated life tests will be run at 150 to 160°F.

All tests will start with a cold tank using the designated water supply temperatures. Both the top and bottom heaters (or heat pump) will operate as required. When the tank reaches the temperature set at the bottom thermostat (equal to the desired temperature at the top of the tank), hot water withdrawals shall be made over a 12- hour span totalling 65 gallons in accordance with the following schedule to determine performance efficiency and energy factor. Some screening tests were run with 5, 15, and 25 gallon withdrawals over an 8 hour time span.

Hour	Representative Draw Schedule*
0	15 gallons (shower or bath)
1	0
2	0
3	5 gallons (miscellaneous)
4	0
5	25 gallons (clothes washer)
6	0
7	0
8	0
9	5 gallons (miscellaneous)
10	0
11	15 gallons (dishwasher)
12	0

c) The hot water rating of the tank, which is the amount of hot water at a selected temperature which can be supplied in a given period of time, also compared to an equal size standard electric resistance heater tank.

d) The energy factor, which is the amount of energy consumed by the heat pump water heater in supplying the NBS average of 64.3 gallons per day, compared to an equal size standard electric water heater.

e) The monthly and annual energy cost for the heat pump water heater compared to a standard electric resistance water heater using the NBS average of 64.3 gallons per day, 30 days per month, 365 days per year at \$.03, \$.05 and \$.07 kWh electric rates.

f) The accelerated life test of the system and its components on a minimum six-month laboratory test of six units.

* Minimum time between draws is determined by thermostatic shut off of power at end of recovery. Low temperature ambients (50°F) will require more time. Total time must compensate for standby losses (i.e. 200-600 BTU/hr) to equate to 12 hours. However, the compressor does not cool down on minimum time schedules and will draw 2 to 3% more power.

I. Definitions

1.1 Heat pump water heater is one in which a nonreversing vapor compression cycle immersion condenser replaces the lower electric resistance element. The upper electric resistance heater (if any) will operate in the normal manner as a fast recovery heater but so connected that the heat pump also operates in this mode. (Total watts not to exceed 4500) A heat pump water heater may also use an external pumped condenser, connected to the water heater tank to provide withdrawal of cold water from the bottom and returning it at the desired temperature to the top.

1.2 Heat trap means a device which can be integrally assembled or independently attached to the hot and/or cold water connections of a water heater such that a portion of this device will develop a cold water seal to minimize the natural convection and resultant heat loss of the hot water stored in the water heater.

1.3 Recovery efficiency expressed as a dimensionless ratio, is the ratio of heat absorbed by the water to the heat input to the heating unit during the period that water temperature is raised from inlet temperature to final temperature.

2. Testing Conditions

2.1 Installation - Install the water heater according to the manufacturer's directions, on a 3/4-inch-thick plywood platform supported by three 2 x 4-inch runners. Install bends in the cold-water inlet and hot-water outlet piping so that these pipes become horizontal as close as possible to the water heater. Provide a quick-acting valve in the hot water outlet piping at a location beyond the point of temperature measurement, but as close to the water heater as possible, and arranged such that the hot water can discharge into a container and be weighed. Insulate the water inlet and outlet connections and piping which is within 4 feet of the water heater, using a material having an inverse of the overall heat transfer coefficient (R) value of not less than:

$$\frac{4°F}{BTU/ft^2\text{-}hr}$$

2.2 Power Supply - For an electric water heater maintain the electrical supply to the water heater within one percent of the center of its nameplate voltage range for the entire operating portion of the test cycle. Assume the heating value, Q_e, in BTU per kilowatt-hour, for electrical energy to be 3,412 BTU per kilowatt-hour.

2.3 Thermocouple Installation - (To be used to calibrate tank thermostats and verify tank temperatures). Daily cycling tests will use outlet and inlet water temperature thermocouples.

Install along the center line of the water heater tank, six thermocouples (herein "center-line thermocouples"), one at the level of the center plane of each of six sections of equal volume from top to bottom of the tank. Install three inches below the highest water level in the tank, an additional thermocouple (herein "tank-top thermocouple"). Install thermocouples in both the cold-water inlet (herein "cold-water thermocouple") and the hot-water outlet (herein "hot-water thermocouple"), not more than six inches from the connections to the water heater, or, where those connections are inaccessible, at the closest accessible point to those connections. Install in the test room a thermocouple (herein "ambient air thermocouple") at a height of 4 feet from the test room floor and at a horizontal distance of at least five feet from the water heater under test.

2.4 Setting of Tank Thermostat - Starting with a tank of unheated water, initiate normal operation of the water heater. At the time that the main heating element is turned off by the thermostat (herein "cutout"), determine whether the average of the temperatures measured by the center-line thermocouples (herein "mean centerline temperature"), is between 135°F and 145°F. If not, turn off the water heater, adjust the thermostat, again fill the tank with unheated water, initiate normal operation of the water heater, and determine the mean centerline temperature at cutout. Repeat this procedure until the mean centerline temperature at cutout is between the 135°F and 145°F, at which time the thermostat is properly set. If the water heater has two thermostats, set the upper thermostat to cut out at a water temperature of 140° ± 5°F in the top six inches of the tank and temperature of 140°F ± 5°F. Repeat settings for heat pump. -

2.5 Energy Flow Instrumentation - Install one or more energy flow instruments which measure, with an error no greater than 1 percent the quantity of electric energy, supplied to the water heater.

2.6 Water Flow Instrumentation - Install a flow control valve in the inlet water line downstream of the expansion tank and calibrate this flow control valve to provide a flow of 3.0 ± .25 gallons per minute. Provide instrumentation for measuring the volume of water withdrawn to within ± .1 percent.

2.7 Room Ambient Temperature - Maintain the test room at an ambient air temperature of between 65°F and 85°F, as measured by the ambient air thermocouples. Maintain room temperature during testing within ± 5°F of the initial room temperature.

3. Measurements

3.1 Tank Storage Capacity - Determine the storage capacity, F, of the water heater under test, in gallons, according to the method specified in 2.26 of the American National Standard for Gas Water Heaters, Volume 1, designated ANS Z21.10.1 - 1975.

3.2 Power Input Determination - Initiate operation of the water heater, and, using the energy flow instrumentation of 2.5 and the actual heating value of 2.2, periodically measure the power input, P, of the heat pump under test when it is in operation, in kilowatts or BTU per hour, as appropriate. To the extent possible, adjust the water heater to use the power indicated by its nameplate rating before obtaining the value of P. Maintain any adjustment so made during the entire test procedure. In addition, measure the Power, P_a, if any, used by any auxiliary electrical system or heaters when the heat pump is in operation, in kilowatts per hour. Measure and record the total energy flow, F_r, at the end of the test.

3.3 Recovery Efficiency For A Heat-Pump Water Heater - With the water heater turned off, fill the tank with water and eliminate any residual air remaining in the tank. Continue to flow water through the tank until the mean center-line temperature remains at the desired initial temperature. Record the mean center-line temperature (herein "initial mean center-line temperature"). Begin measuring the power consumption and initiate normal operation of the water heater. When cutout occurs, record the maximum mean center-line temperature (herein "maximum mean center-line temperature") and record the total input power, F, from initiation to cutout. Record the temperature difference, D, obtained by subtracting the initial from the maximum mean center-line temperature. This test may be performed with only the heat pump operating and/or with the fast-recovery upper resistance element, if any, in the system.

3.4 Standby Loss Test - With the water heater operating normally, such that the water temperature is established in the range specified in 2.4, eliminate any residual air from the tank. At a point in time immediately after a cutout such that all heating power is off, begin the energy flow measurements. Record the time as the beginning of the test period, and record the initial mean centerline temperature and the initial ambient air temperature. At the end of each time interval not exceeding 15 minutes in duration throughout the test period, record the mean centerline temperature and the ambient air temperature. Continue these measurements until the first cutout in which all heating power is off following a period of at least 48 hours from the beginning of the test period. When that cutout occurs, record the time as the end of the test period, record the total energy flow, F_s, from the beginning to the end of the test period, and record the final mean centerline temperature and the final ambient air temperature. Record as the duration, t, of the standby loss test the time, in hours, which elapsed from the beginning to the end of the test period. Calculate the average of the recorded values of the mean centerline temperature and the ambient air temperature taken at the end of each time interval, including in these two averages the initial and final recorded values for these temperatures. Determine the difference, D_a, between these two averages by subtracting the latter from the former, and the differences, D_s, between the final and initial mean centerline temperatures by subtracting the latter from the former.

3.5 Hot Water Supply Rating - Establish normal operation of the water heater such that the water temperature is established in the range specified in 2.4. Immediately after a cutout such that all heating power is off, measure the outlet water temperature (herein "initial outlet water temperature") and record the time and begin withdrawing water from the tank into a weighing container at a rate of between 2.5 and 3.5 gallons per minute. Continue the withdrawal until the outlet water temperature drops to a value (herein "shutoff temperature") 40°F below the initial outlet water temperature, at which time terminate the withdrawal. If a cutout of any upper thermostat occurs less than one hour after the inital cutout, again withdraw water until the outlet water temperature drops to the shut-off temperature. Repeat this cycle until one hour after the initial cutout, at which time turn off all energy flow to the water heater. If the temperature measured by the tank top thermocouple at that time is higher than the shutoff temperature, withdraw water until the water outlet temperature drops to the shutoff temperature. Record the weight, W, of water withdrawn during this test, in pounds.

4. Calculation of Derived Results from Test Measurements

4.1 The Recovery Efficiency E_r is a dimensionless quantity and is defined as:

$$E_r = \frac{k \times V \times D_r}{F_r \times Q}$$

Where:

k = nominal specific heat of water in BTU's per gallon degree F = 8.3

V = tank capacity in gallons per ANS - 21.10.1 - 1975

D_r = difference between the initial and final centerline temperatures, in degrees F

F_r = total water heater input power in kWh

Q = BTU per kWh = 3412

4.2 The Standby Loss S is expressed in hours^{-1} and is defined as:

$$S = \frac{F_s \times Q}{t \times k \times D_a \times V} - \frac{D_s}{t \times D_a \times E_r}$$

Where:

F_s = total energy flow in the standby loss test 3.3 in appropriate units

t = duration of standby loss test in hours per 3.3

D_s = difference between initial and final centerline temperatures as determined in the standby loss test of 3.3 in degrees F

D_a = difference between the average value of the mean centerline temperature and the average value of the ambient air temperature during the standby loss test in 3.3 degrees F

Q, V, and E_r are defined in 4.1

4.3 The Hot Water Supply Rating G is expressed in gallons and defined as G = W/d , where:

W = total weight of water withdrawn during the supply rating test of 3.4 in pounds

d = nominal density of water in pounds per gallon = 8.3

4.4 <u>The Recovery Rate R_{90}</u> - Calculate the 90°F-rise recovery rate expressed in gallons per hour and defined as:

$$R_{90} = \frac{F_r \times Q \times E_r}{k \times \Delta r}$$

where Fr, Q, E_r and k are defined in 4.1

Δr = nominal difference between inlet and outlet temperatures in °F = 90

4.5 <u>The Daily Water Heating Energy Z</u> is the energy required to heat the nominal amount of hot water used daily, expressed in BTU and calculated as:

$$Z = \frac{k \times \Delta r \times U}{E_r}$$

where:

k, E_r are defined in 4.1

Δr is defined in 4.4

U is daily hot water usage = 64.3 gallons

4.6 <u>The Average Daily Energy Consumption C_E</u> expressed in BTU's is calculated as:

$$C_E = Z + m \times (24 - Z/P) - F_H - F_C$$

where:

m = average energy required per hour to maintain stored water temperature in BTU per hour = S x k x a x V

E_r, k, V, P and S are defined in 4.1

Z is defined in 4.5

a = nominal difference between mean centerline temperature and the ambient air temperature in degrees F = 90

F_H = daily energy credit for a heat trap if provided on hot water outlet = 1,311 BTU

F_C = daily energy credit for a heat trap if provided on cold water inlet = 983 BTU

4.7 Average Daily Energy Consumption C_{EK} expressed in kilowatt hours per day is calculated as

$$C_{EK} = \frac{C_E}{3{,}412 \text{ BTU/kWh}}$$

Where C_E is defined in 4.6.

APPENDIX C

HEAT PUMP WATER HEATER

APPLICATION STUDY

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Summary

The following comments and observations are made concerning this study of the application of heat pump water heater in typical houses in three regions of the United States:

- The three cities included in this study are Jacksonville, Florida; Pittsburgh, Pennsylvania; and Minneapolis, Minnesota.
- The heating degree days range from 1188 to 8070 and the cooling degree days range from 2813 to 721.
- A single story house on a slab (1500 ft^2) and a two story house with a basement (1700 ft^2) were the residential units considered.
- The hot water heat pump was analyzed in various locations in these houses:

  i) utility room
  ii) basement
  iii) garage
  iv) attic

- The heat pump does not heat the water is the space (location) temperature decreases below a designated cut off temperature (40 or 50°F). If the temperature drops below that temperature a resistance heater is used.
- It was assumed that water was heated from a low of 50°F to a high of 140°F. Based on test data and an average water temperature of 95°, an average COP of 3.0 was used.
- An utilization efficiency (% of hot water energy requirements supplied by the heat pump) was developed for each application.
- For the single story house with the heat pump in the utility room, the efficiency ranged from 44% to 85% for Minneapolis to Jacksonville, respectively.
- For the two story house under similar condition the efficiency ranged from 41.5% to 74%.

- The garage and attic heat pump location show an increased utilization efficiency in the warmer climates.

- The basement location shows an utilization efficiency very dependent upon ground temperature, cutoff temperature and internal loads generated in the space. In colder climates, under the conditions in the analysis, the efficiency was $\sim$ 50%.

- If internal heat gains (due to location of furnace, clothes washer, dryer) in the space of the heat pump are included, an estimated increase of 10 to 20% points in the efficiency level may be realized. (Based on analysis of single story house, utility room).

- A decrease in cutoff temperature from 50°F to 40°F, may increase the efficiency level by 15 to 20% points (Based on single story house, utility room location in Pittsburgh).

- In analyzing the cost savings of the heat pump, three factors must be considered:

  i) utilization efficiency
  ii) increase in heating requirements
  iii) decrease in cooling requirements

  These three factors were integrated on a cost basis for a sample problem in Pittsburgh ($3.00/$10^6$ BTU fuel and $10/$10^6$ BTU electricity). In this case, there is a net saving of $66.00 per year.

  Parametric cost curves for each of these factors have been developed for a range of fuel and electric costs.

Application of Hot Water Heat Pump in Typical Houses in Three Regions of the Country

A. Introduction

The approach taken in this study of the application of the heat pump is presented in Figure C-1. The task has been directed toward the evaluation of the use of the heat pump water heater under the following conditions:

i) Two residential units were considered for evaluation as defined in Section B.

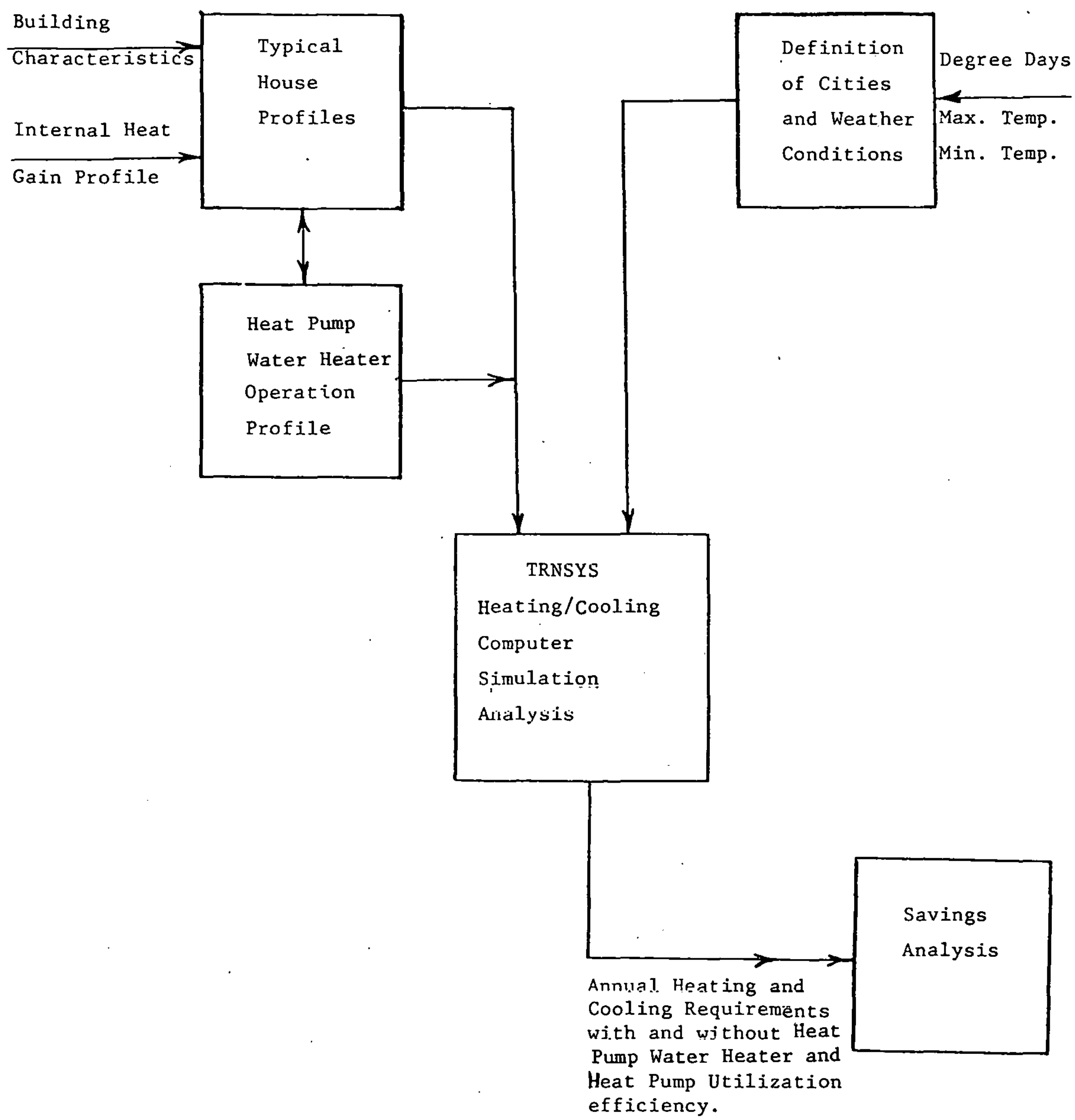
- One story house, 1500 ft^2 on concrete slab with attached garage.
- Two story house, 1700 ft^2 with basement and attached garage.

For each house a profile of internal heat gains has been defined.

ii) Three cities were chosen for analysis and are typical of weather conditions in the nation:

	1975 Heating Degree Days	1975 Cooling Degree Days
Jacksonville, Florida	1188	2813
Pittsburgh, Pennsylvania	5672	721
Minneapolis, Minnesota	8070	850

Figure C-1: Approach to Application Study



iii) The heat pump water heater operation is simulated in each residential unit for each city with the heat pump location varied depending upon the particular city. The various locations within the houses include the following:

- utility room
- basement
- garage
- attic

In all cases the heat pump water heater was not operated if the space temperature decreases below a designated cutoff temperature. Two cutoff temperatures - 40°F and 50°F - were considered for certain cases. All cases were based on 50°F unless otherwise stated. Once this temperature is reached, a conventional electric resistance heater is then used.

A hot water use profile has been generated, and from this, a cooling profile for the heat pump evaporator has been developed.

iv) This information was input to the TRNSYS (Ref C-1) heating/cooling simulation model. Annual heating and cooling requirements were generated for conditions with and without the operation of the heat pump water heater.

v) From the information developed above, the percentage increase in heat load, percentage decrease in cooling load, heat pump utilization efficiency and parametric energy cost savings have been assessed.

B. Typical One Story and Two Story House Profiles

Typical one-story and two-story houses are detailed in Figures C-2 and C-3. The one story house has 1500 ft^2 on a concrete slab with an attached garage and the two story house has 1700 ft^2 with a basement and attached garage. The construction materials and related engineering parameters are presented in Table C-1. A comparison of the two houses is presented in Table C-2

The profiles of internal heat gains (from occupancy and appliances) are defined in Figs. C-4 and C-5. The house has been maintained at 72°F. The temperature of space (utility room, basement, attic or garage) in which the heat pump is located has been allowed to float; that is, these spaces are not heated or cooled to maintain 72°F. The heat pump was only operational if the "location" space was greater than the specified cutoff (50°F for most cases, 40°F for two cases in Pittsburgh). When the temperature decreased below the cutoff, a conventional electric resistance heater was used and the resultant cooling load due to the heat pump evaporator was reduced to zero.

C. Hot Water Use Profile

The daily requirement for hot water use is presented in Figure C-6. This is based upon a peak use of 1.6 gal/hr/person and 4 people in the residential unit. Total hot water use, 78 gal/day, in the residence is represented as 58300 BTU/day (i.e. 78 gal/day with a ΔT of 90°F).

Figure C-2. One-Story House

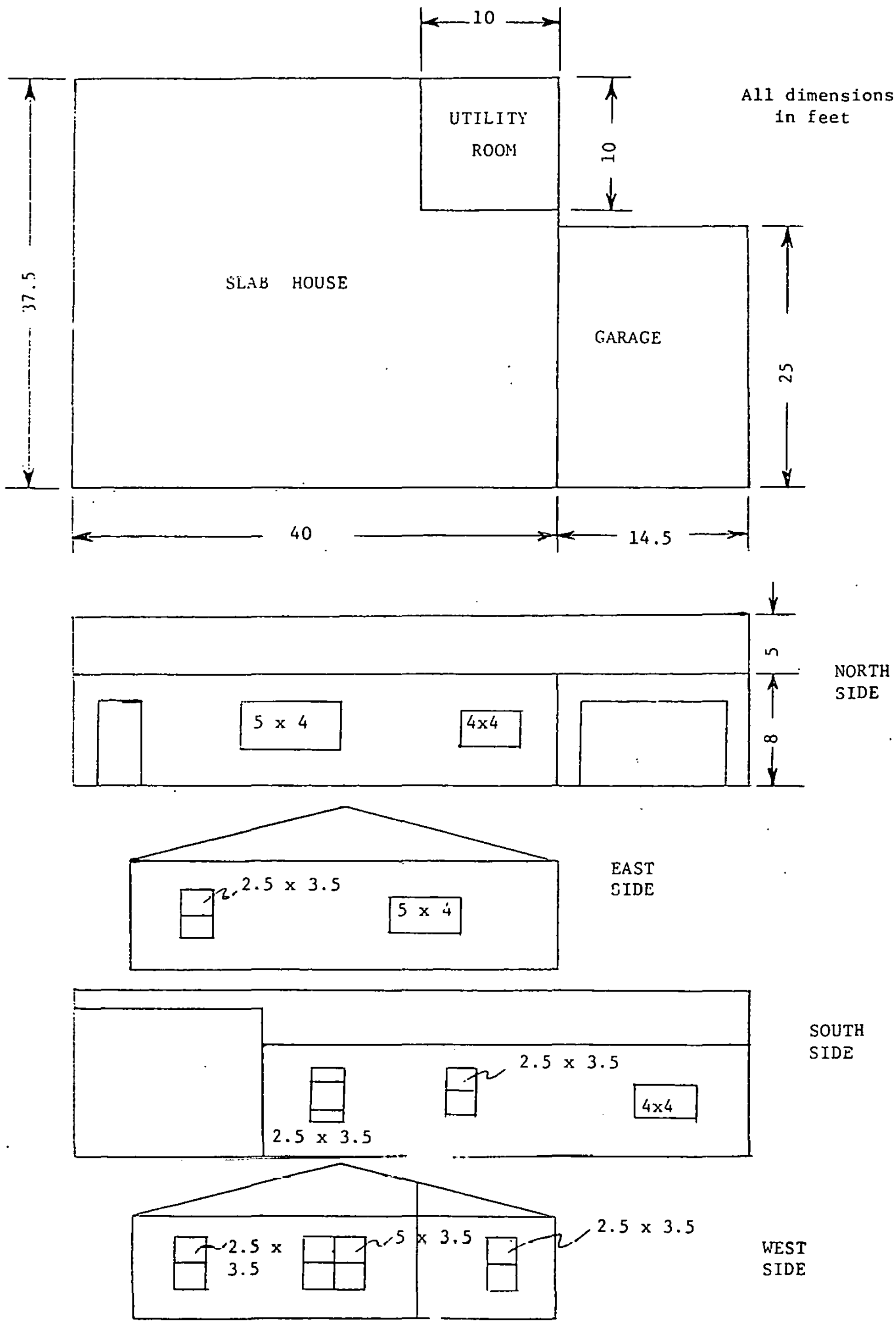


Figure C-3. Two-Story House

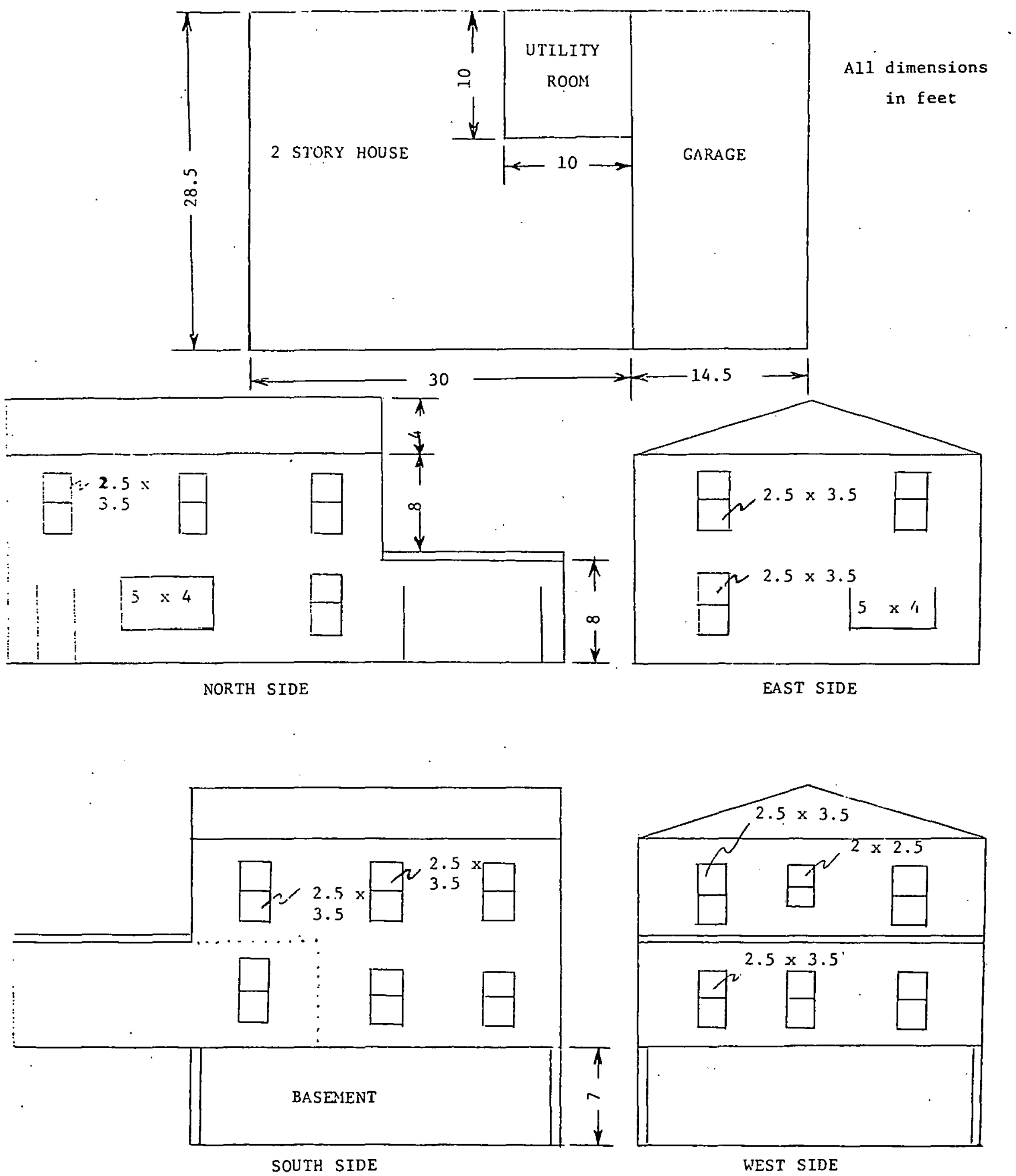


Table C-1

Construction Materials and Engineering Parameters

EXTERIOR WALLS $U = .075\ BTU/hr \cdot Ft^2 \cdot °F$

1/2" Green Siding $\alpha = \varepsilon = .7$
1/2" Sheathing
Stud and Insulation 3 1/2"
1/2" Wallboard

WALL COMMON WITH HOUSE AND GARAGE $U = .076\ BTU/hr \cdot Ft^2 \cdot °F$

1/2" Wallboard
3 1/2 Stud and Insulation
1/2" Wallboard

WALL COMMON WITH HOUSE AND UTILITY $U = .09\ BTU/hr \cdot Ft^2 \cdot °F$

1/2" Wallboard
2" Air Space
1/2" Wallboard

HOUSE CEILING $U = .059\ BTU/hr \cdot Ft^2 \cdot °F$

1/2" Wood
6" Stud and Insulation
1/2" Wallboard

GARAGE WALLS $U = .27\ BTU/hr \cdot Ft^2 \cdot °F$

1/2" Green Siding
2" Stud Backup

HOUSE ROOF $U \sim .27\ BTU/hr \cdot Ft^2 \cdot °F$

1/2 " Wood with Black Shingles $\alpha = .97$ $\varepsilon = .91$

GARAGE ROOF $U \sim .27\ BTU/hr\ FT^2\ °F$

1/2" Wood with Black Roofing Paper $\alpha = .97$ $\varepsilon = .91$

WINDOWS $U = .6\ BTU/hr\ FT^2\ °F$

$\tau = 1.0$

FLOOR INSULATION

6 inches of insulation between house and basement

BASEMENT

4 inch concrete floor

8 inch concrete block walls

Table C-2

House Comparisons

Utility Room	Single Story	Two Story
Wall, Floor, Ceiling Common with Living Space	160 ft^2	260 ft^2
Exposed Wall Facing South (11% Window)	80 ft^2	80 ft^2
Exposed Facing West (11% Window)	80 ft^2	-
Common with Garage	-	80 ft^2
Floor Area	100 ft^2	100 ft^2
Ceiling	Attic	Second Floor

Garage	Single Story	Two Story
North Wall (no Windows)	116 ft^2	116 ft^2
South Wall (no Windows)	116 ft^2	116 ft^2
West Wall (13% Windows)	200 ft^2	228 ft^2
Floor Area	363 ft^2	413 ft^2
Common with House other than Utility	200 ft^2	80 ft^2
Common with Utility	-	148 ft^2

Figure C-4

PROFILE OF INTERNAL HEAT GAINS FOR SINGLE-STORY HOUSE

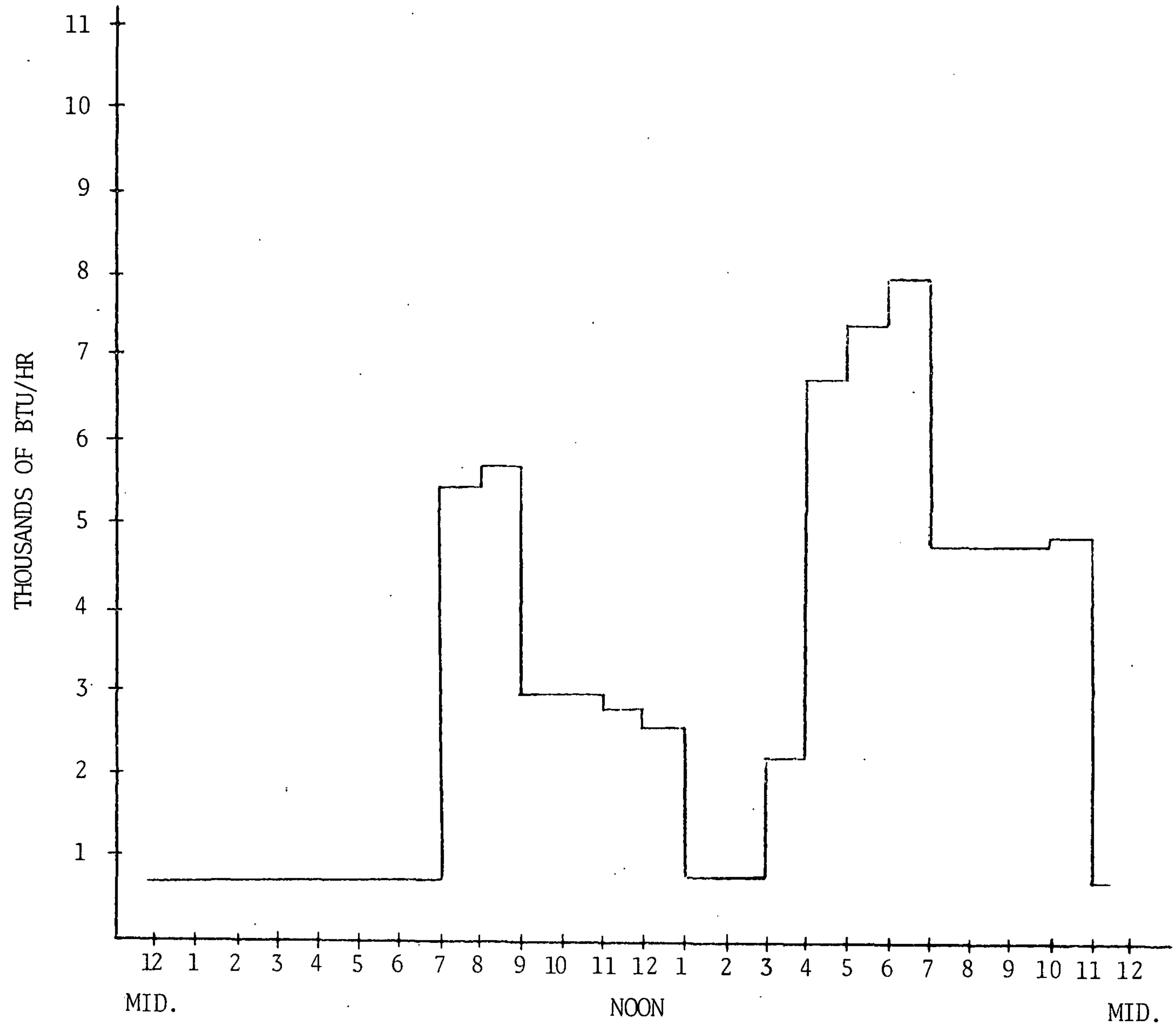


Figure C-5

PROFILE OF INTERNAL HEAT GAINS FOR TWO-STORY HOUSE

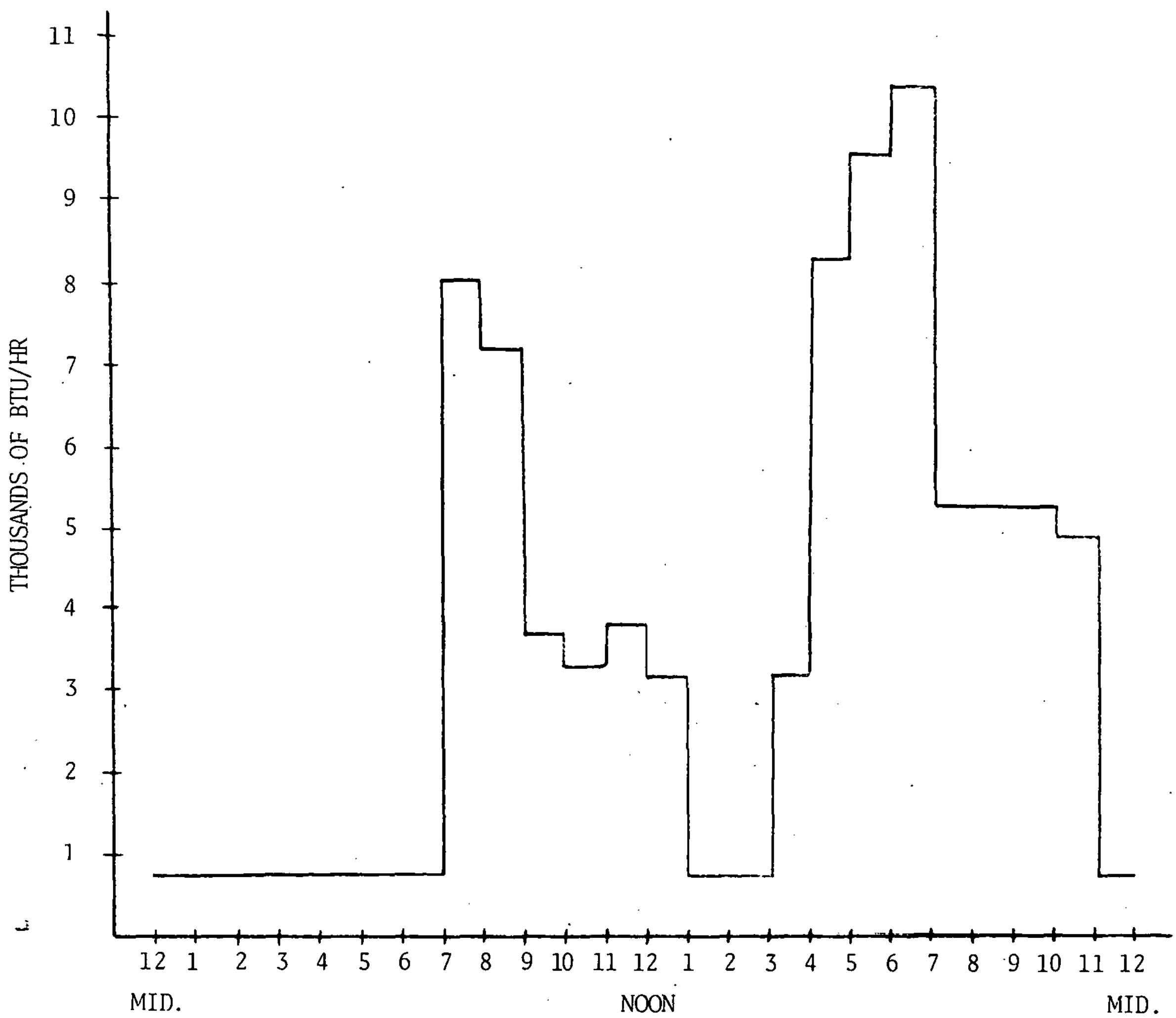
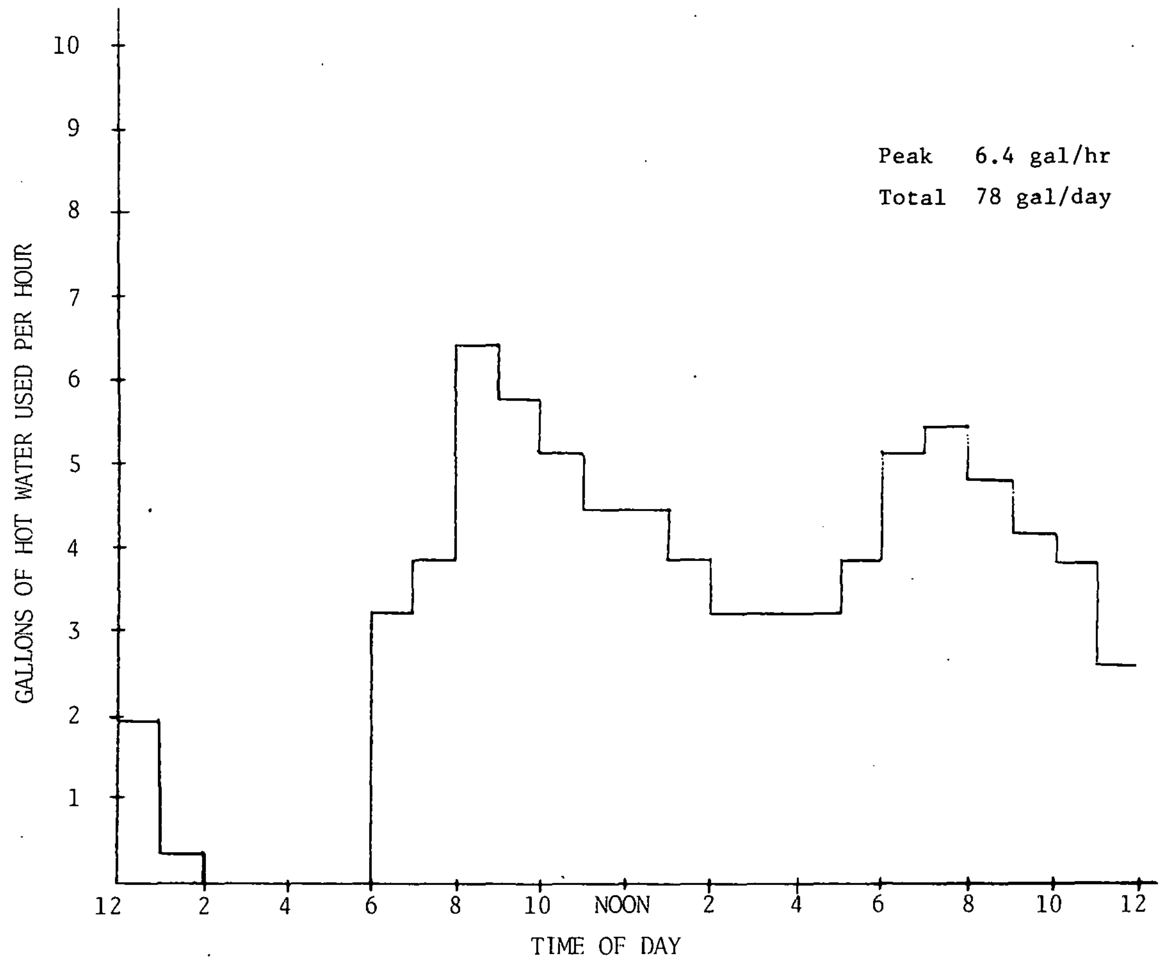


Figure C-6
RESIDENTIAL HOT WATER USE



D. Heat Pump Water Heater

Based on information from EUS, a COP of 3 was used in determining the average cooling load in each of the water heater locations.

$$COP = \frac{Q_H}{W} = \frac{Q_H}{Q_H - Q_C} = 3.0$$

$$COP - 1 = \frac{Q_C}{Q_H - Q_C} = 2$$

$$Q_C = \frac{2}{3} Q_H$$

In this case Q_H represents the thermal energy required to heat the water to 140°F and Q_C represents the cooling load in the heat pump water heater space. W represents the electrical requirements of the heat pump water heater.

$$Q_H = \dot{m} c p \Delta T$$

where,

$\dot{m}$ = lbm/hr of hot water requirements

cp = specific heat

$\Delta T = T_H - T_C \sim 90°F$

and then

$$Q_C = \frac{2}{3} \dot{m} c p \Delta T$$

The profile of Q_C is presented in Table C-3.

Table C-3

Cooling Due to Evaporator of Heat Pump Water Heater*

Hour of Day	(BTU/hr)
1	910
2	140
3	0
4	0
5	0
6	0
7	1530
8	1820
9	3060
10	2730
11	2440
12	2150
13	2150
14	1820
15	1530
16	1530
17	1530
18	1820
19	2440
20	2580
21	2300
22	2010
23	1820
24	1250
Total	37560 BTU/day

*Used for all cases–slightly less than implied by Fig. C-6 and a COP of 3.

E. Weather Profiles of Three Cities -

The three cities chosen for this study are:

- Jacksonville, Florida
- Pittsburgh, Pennsylvania
- Minneapolis, Minnesota

Seasonal ground temperatures and the heating, cooling, and net degree days used in this analysis are presented in Table C-4. The average maximum and minimum ambient temperatures for each month are presented in Table C-5.

These input data to the TRNSYS model were based on information from Ref. C-2, C-3, and C-4.

TABLE C-4

Degree Days (1975)

City	Heating Degree Days (HDD)	Cooling Degree Days (CDD)	Net Degree Days (HDD-CDD)	Latitude
Jacksonville	1188	2813	-1625	31°
Pittsburgh	5672	721	4951	41°
Minneapolis	8070	850	7220	45°

Ground Temperatures (°F)

City	Winter	Spring	Summer	Fall
Jacksonville	66	65	78	78
Pittsburgh	46	45	59	58
Minneapolis	42	40	57	56

TABLE C-5

Average Maximum and Minimum Daily Ambient Temperatures

Month	Jacksonville		Pittsburgh		Minneapolis	
	Max °F	Min °F	Max °F	Min °F	Max °F	Min °F
January	70	47	36	26	23	5
February	72	49	39	26	23	8
March	72	49	44	29	30	14
April	78	54	54	34	46	32
May	86	65	74	52	72	50
June	90	72	77	59	77	60
July	88	72	84	61	87	66
August	91	71	83	63	81	62
September	88	69	67	51	68	47
October	82	63	63	44	65	41
November	74	50	56	37	47	28
December	66	40	40	26	28	14

F. Results: Energy and Cost Savings

The data generated in Sections B, C, D and E were input to the TRNSYS heating and cooling simulation model (Ref C-1).

For each set of simulations, two runs were required. The first represents the base case without the heat pump water heater and the second with the heat pump water heater operating as previously described. The results of these analysis are presented in Tables C-6, C-7, and C-8 and Figures C-7 through C-19.

The parametric cost differentials associated with these energy differences are presented in Figures C-20 through C-26.

General Statements

The heat pump does not operate when the space temperature decreases below cutoff temperature. Conventional electric resistance heating is used when the space temperature is below the cutoff. Two cutoff temperatures (40°F and 50°F) were considered for different cases.

Internal heat gains (i.e. washer, dryer and furnace in same space as the heat pump water heater) were not included in most cases. They were specifically included for comparison in the one-story house, utility-room case.

Utilization efficiency for basement cases is a function of ground temperature. For cases considered, ground temperatures used were six months at 40-45°F and six months at 55-60°F.

Heating/Cooling percentage changes are plotted as a function of Net Degree Days (HDD - CDD) and as a function of HDD and CDD separately.

Typical best estimates of the house heating and cooling loads are:

	Load (10^6 BTU/yr) One-Story House		Two-Story House	
City	Heating	Cooling	Heating	Cooling
Jacksonville	.46	1.25	.49	1.48
Pittsburgh	1.75	.41	1.80	.51
Minneapolis	2.61	.46	2.67	.55

Utilization Efficiency of Heat Pump Water Heater In a One-Story House On a Slab

Cutoff Temperature = 50°F
No internal heat gains in heat pump space

City	1975 Heating Degree Days	1975 Cooling Degree Days	Net Degree Days	Utilization Efficiency (%) Utility Room	Garage
Jacksonville	1188	2813	-1625	85	96
Pittsburgh	5672	721	4951	50	55
Minneapolis	8070	850	7220	44	-

Table C-7

Utilization Efficiency of Heat Pump Water Heater In a Two-Story House with Basement

50°F cutoff temperature
No internal heat gains in heat pump space

City	Net Degree Days	Utility Room	Garage	Basement	Attic
Jacksonville	-1625	74%	96%	--	98%
Pittsburgh	4951	46%	58%	50%	--
Minneapolis	7220	41.5%	--	50%	--

Table C-8

Comparison of Heat Pump Water Heater Performance in Pittsburgh as a Function of Cutoff Temperature & Internal Heat Gain in Utility Room

(Single-Story House on a Slab)

Cutoff Temperature (°F)	Internal Heat Gain	Increase in Heating (%)	Decrease in Cooling (%)	Utilization Efficiency (%)
50	No	1.6	4.6	50
50	Yes	2.1	5.3	73
40	No	1.7	5.6	69
40	Yes	4.2	5.8	90

Figure C-7
UTILIZATION EFFICIENCY OF HEAT PUMP WATER HEATER
IN SINGLE-STORY HOUSE ON SLAB

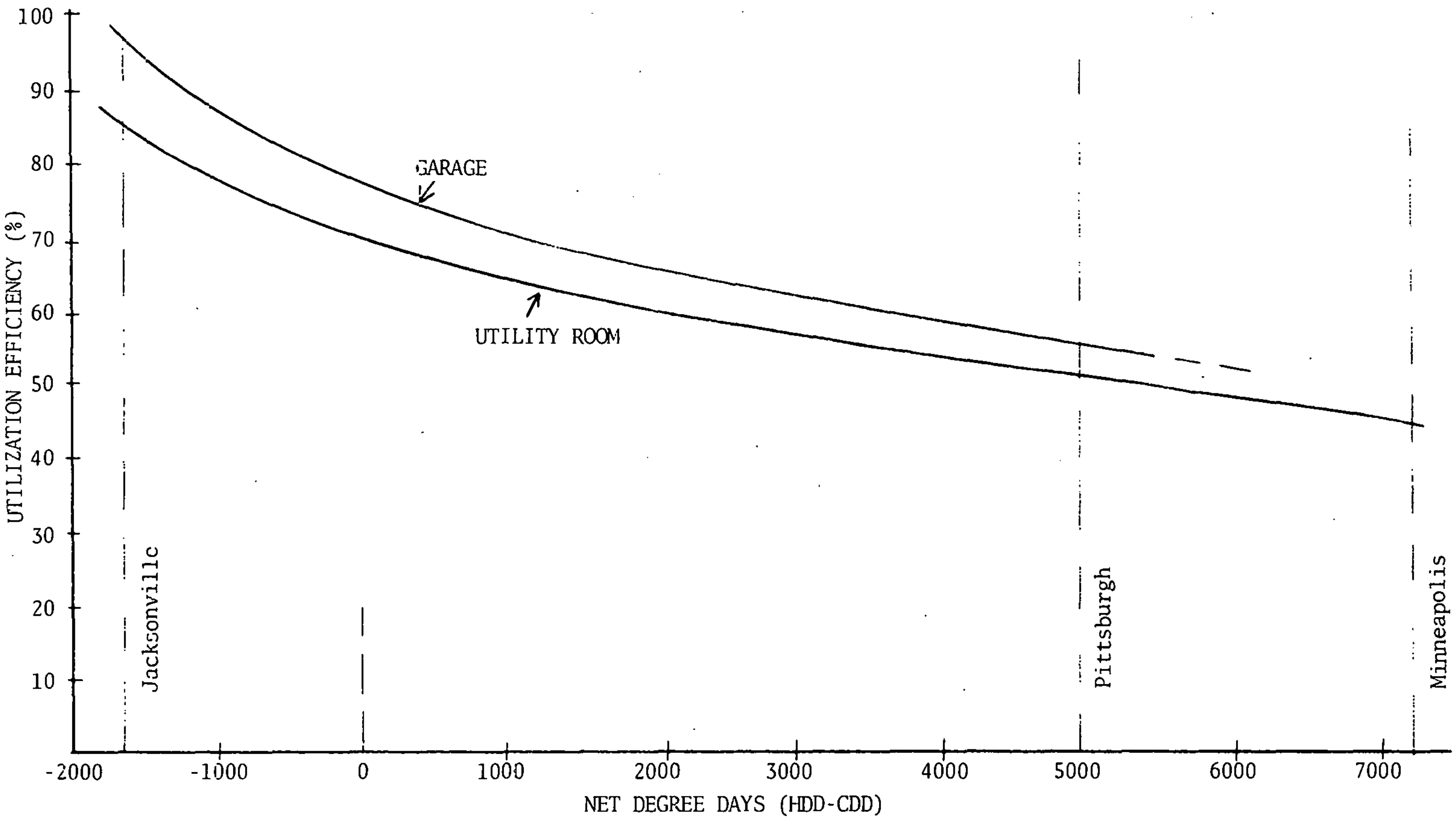


Figure C-8
INCREASE IN HOUSE HEATING REQUIREMENTS DUE TO
HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS
IN ONE-STORY HOUSE ON SLAB

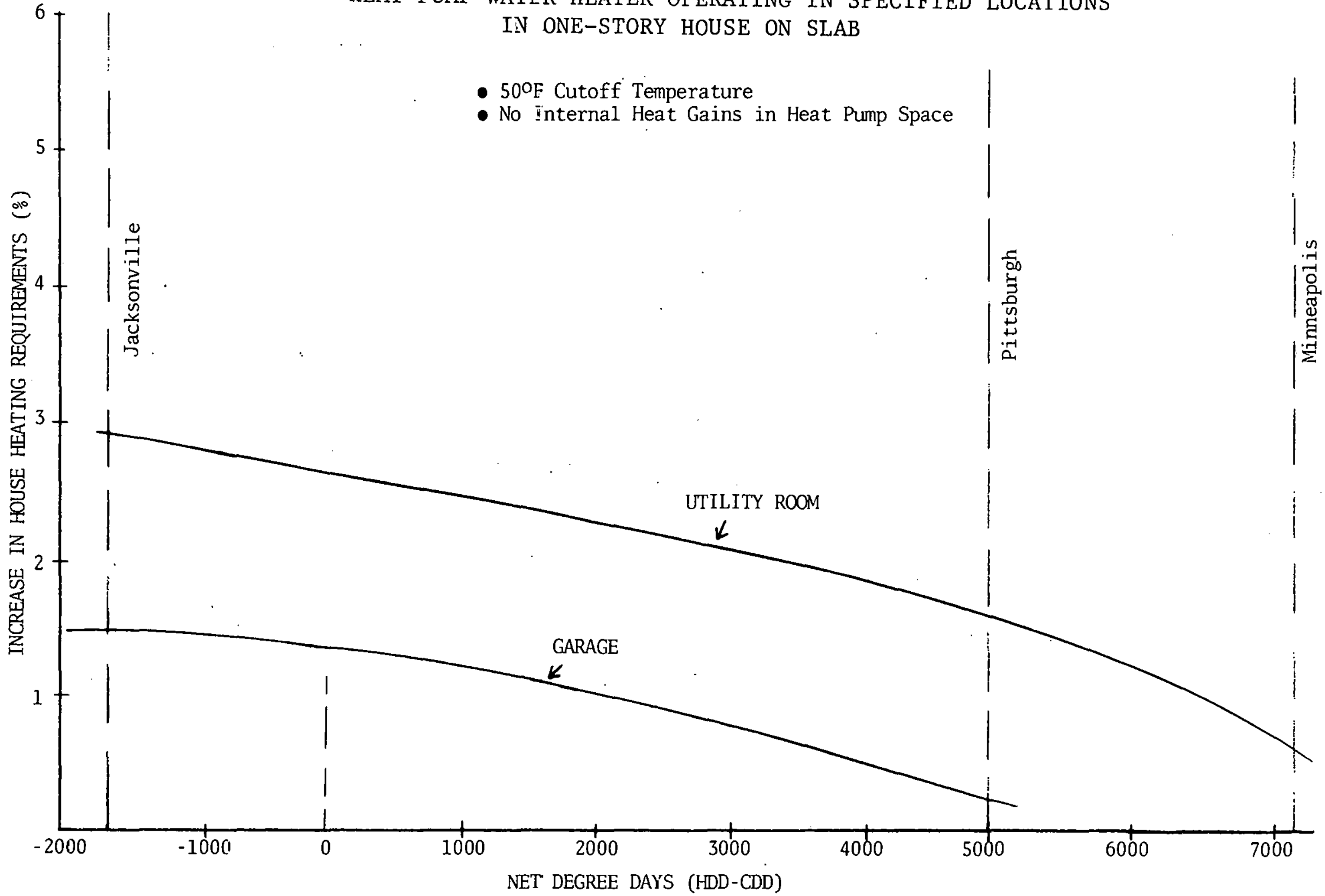


Figure C-9

INCREASE IN HOUSE HEATING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS IN ONE-STORY HOUSE ON SLAB

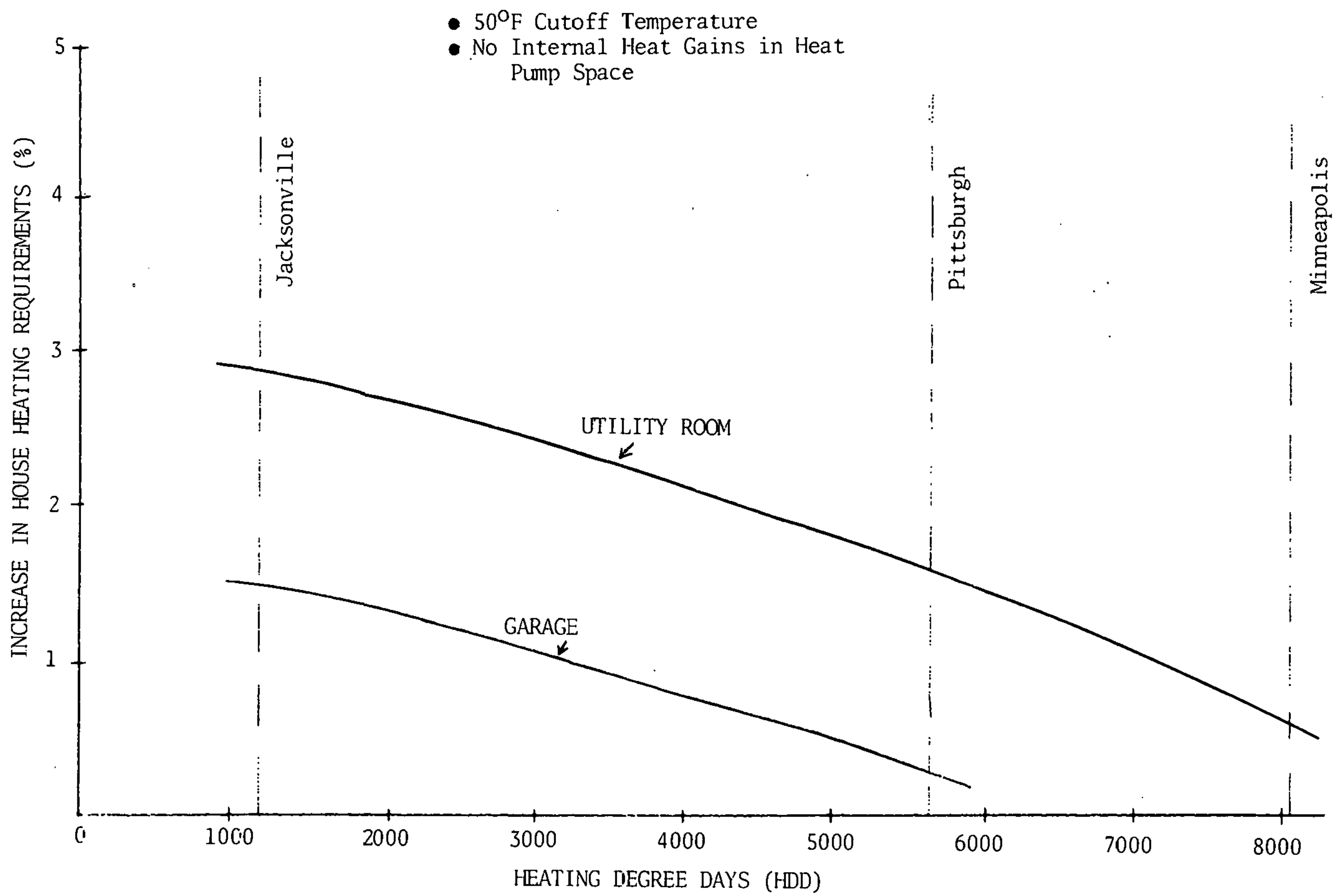


Figure C-10

DECREASE IN HOUSE COOLING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS IN ONE-STORY HOUSE ON SLAB

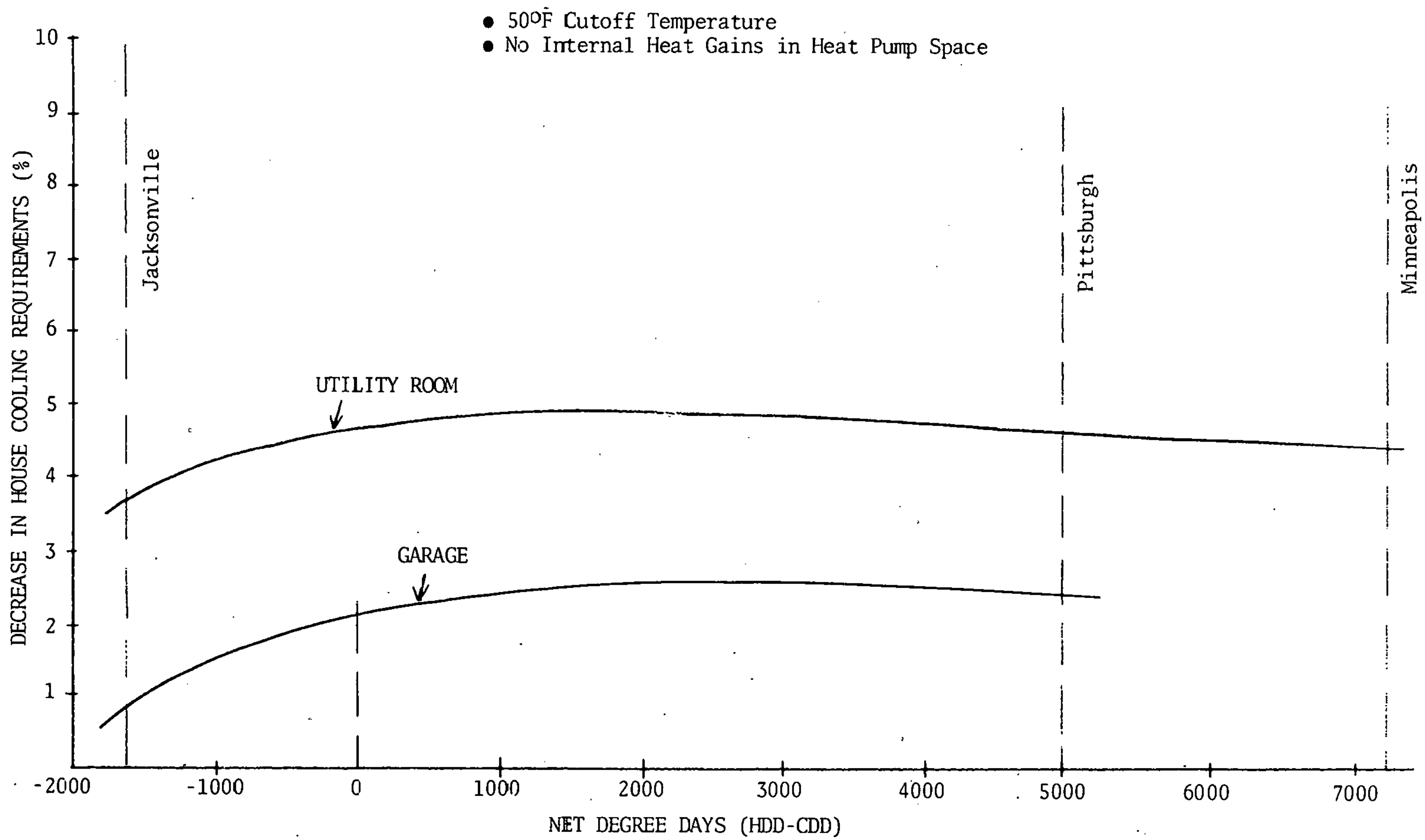


Figure C-11

DECREASE IN HOUSE COOLING REQUIREMENTS DOE TO
HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS
IN ONE-STORY HOUSE ON SLAB

- 50°F Cutoff Temperature
- No Internal Heat Gains in Heat Pump Space

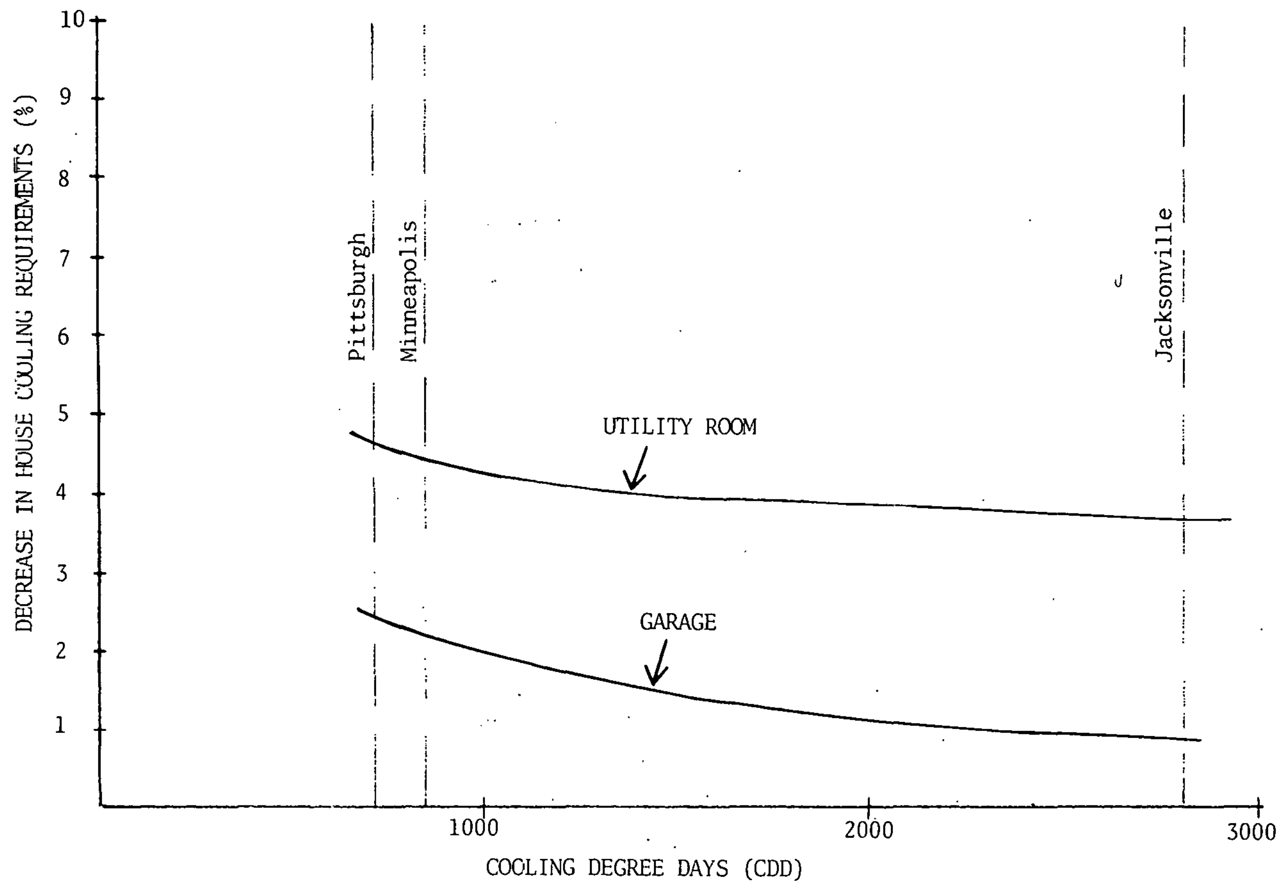


Figure C-12

UTILIZATION EFFICIENCY OF HEAT PUMP WATER HEATER
IN TWO-STORY HOUSE WITH BASEMENT

- 50°F Cutoff Temperature
- No Internal Heat Gain in Heat Pump Space

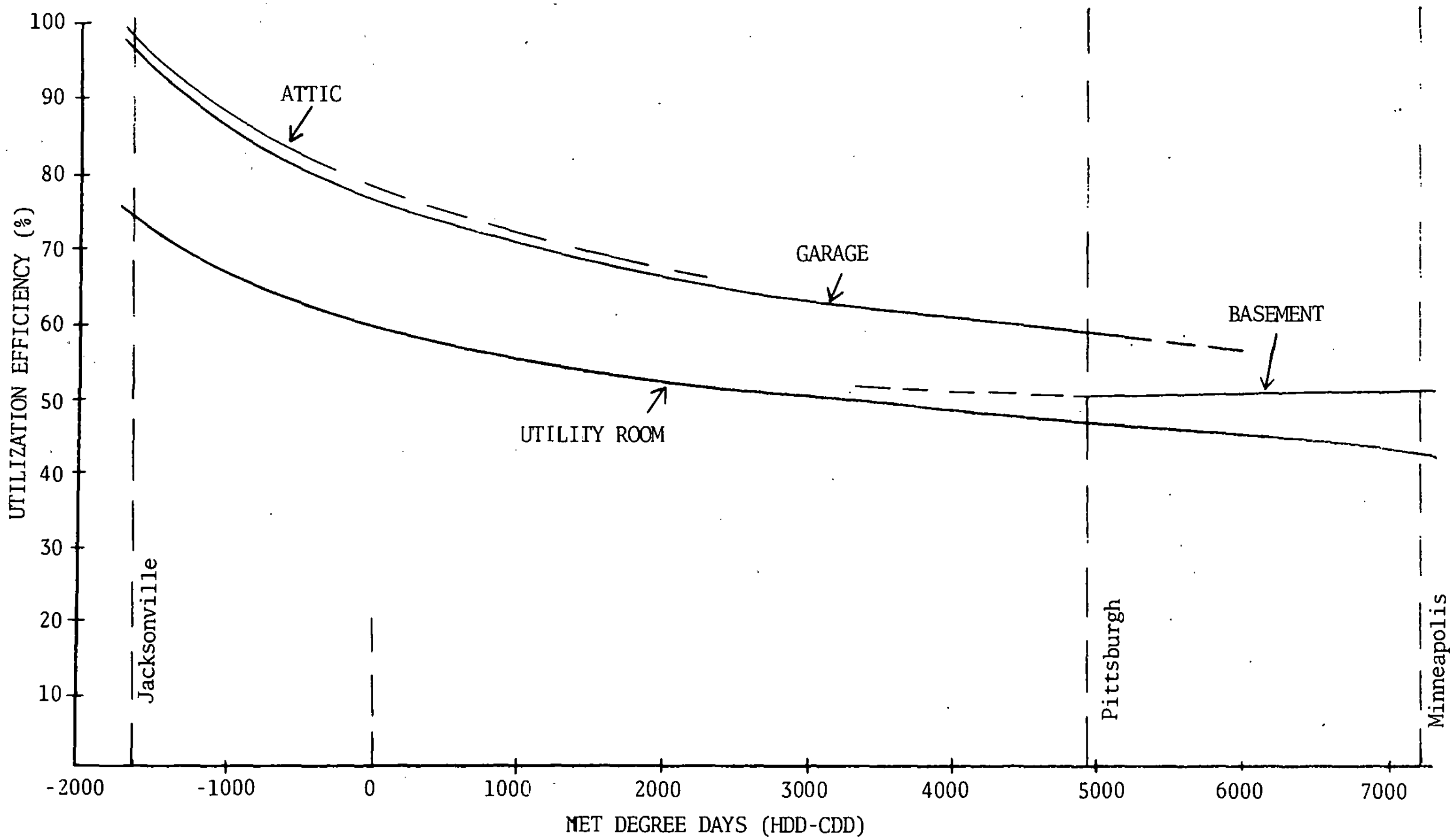


Figure C-13

INCREASE IN HOUSE HEATING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS IN TWO-STORY HOUSE WITH BASEMENT

- 50°F Cutoff Temperature
- No Internal Heat Gains in Heat Pump Space

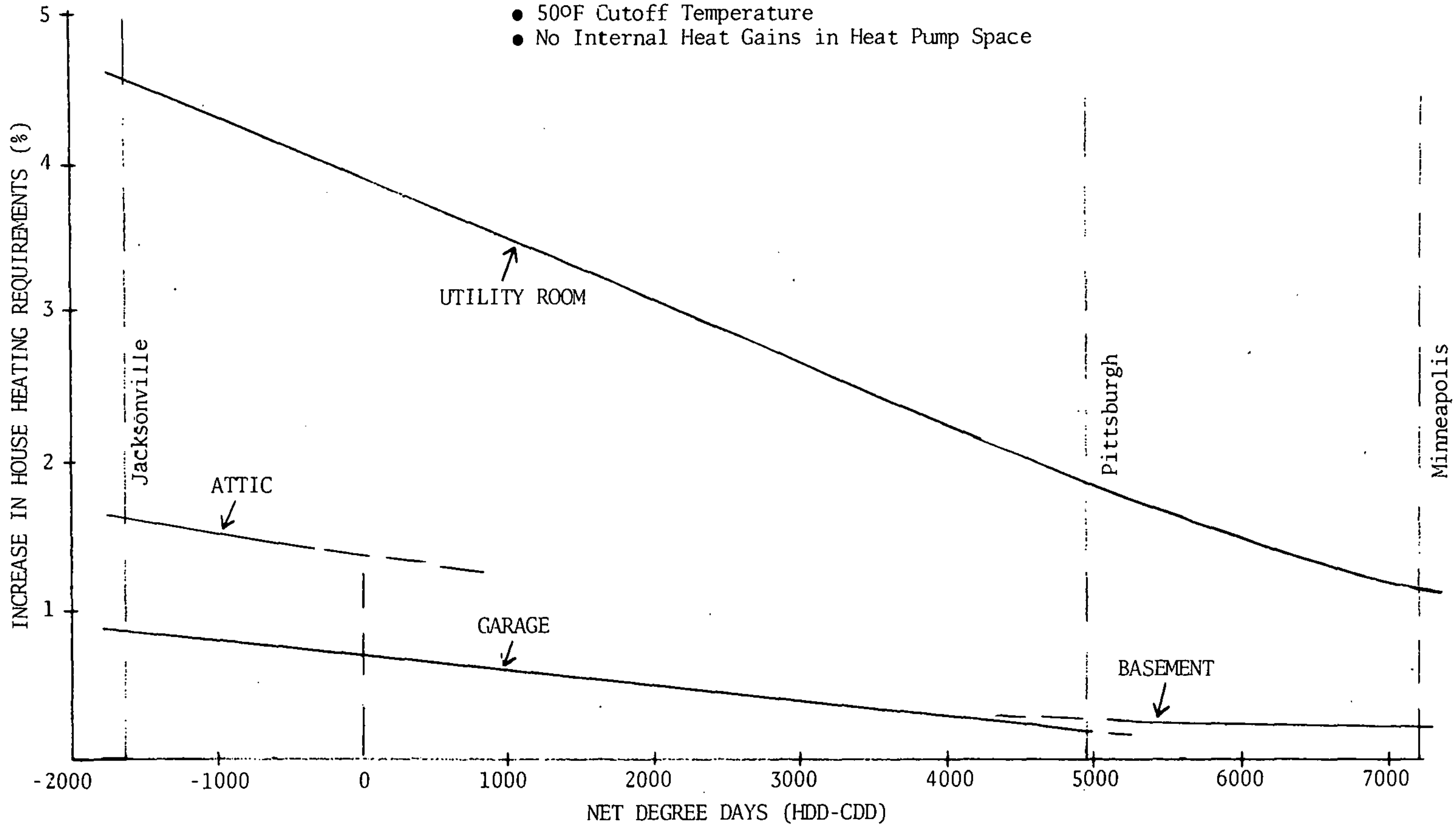


Figure C-14

INCREASE IN HOUSE HEATING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS IN A TWO-STORY HOUSE WITH BASEMENT

- 50°F Cutoff Temperature
- No Internal Heat Gain in Heat Pump Space

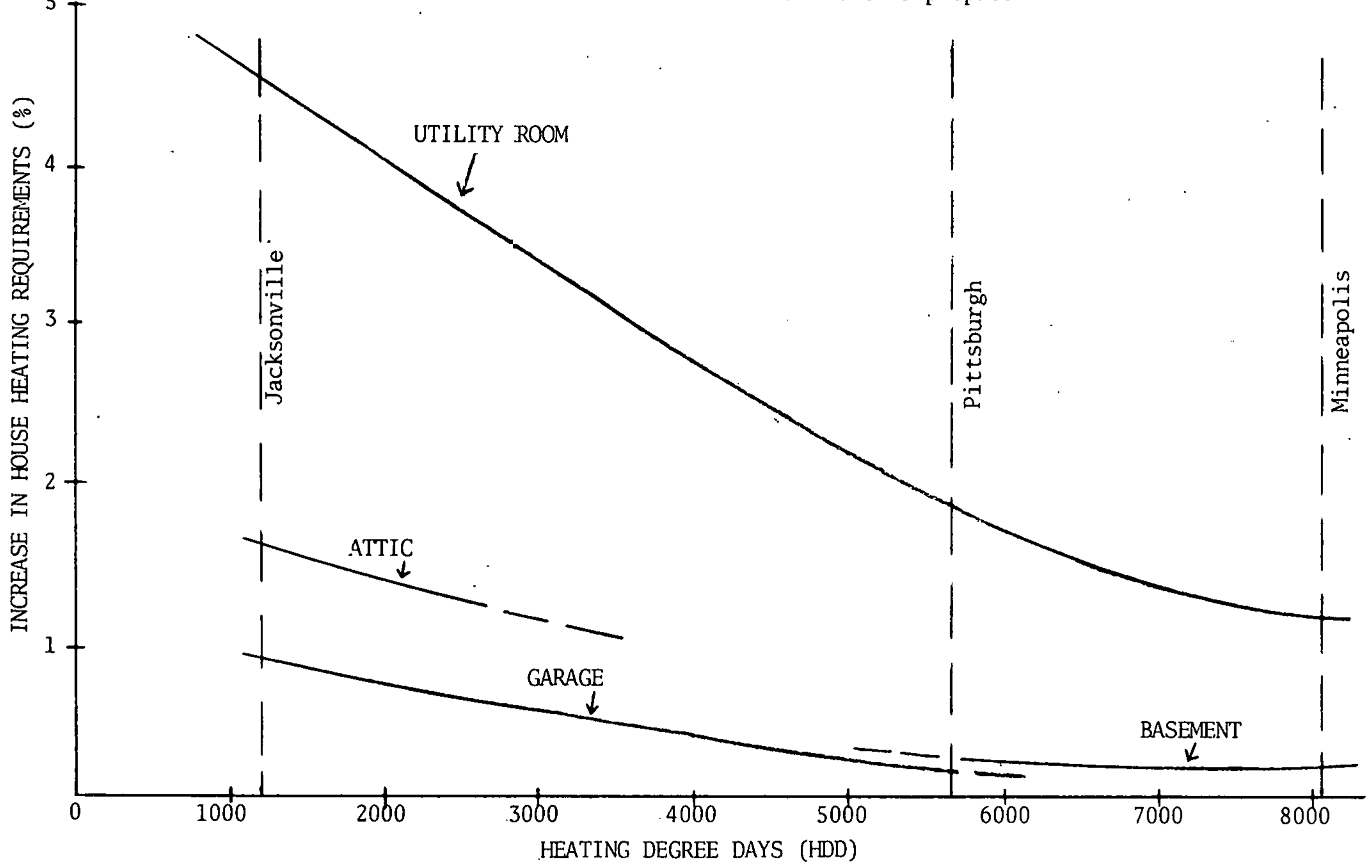


Figure C-15
DECREASE IN HOUSE COOLING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS IN TWO-STORY HOUSE WITH BASEMENT

- $50^{\circ}F$ Cutoff Temperature
- No Internal Heat Gain in Heat Pump Space

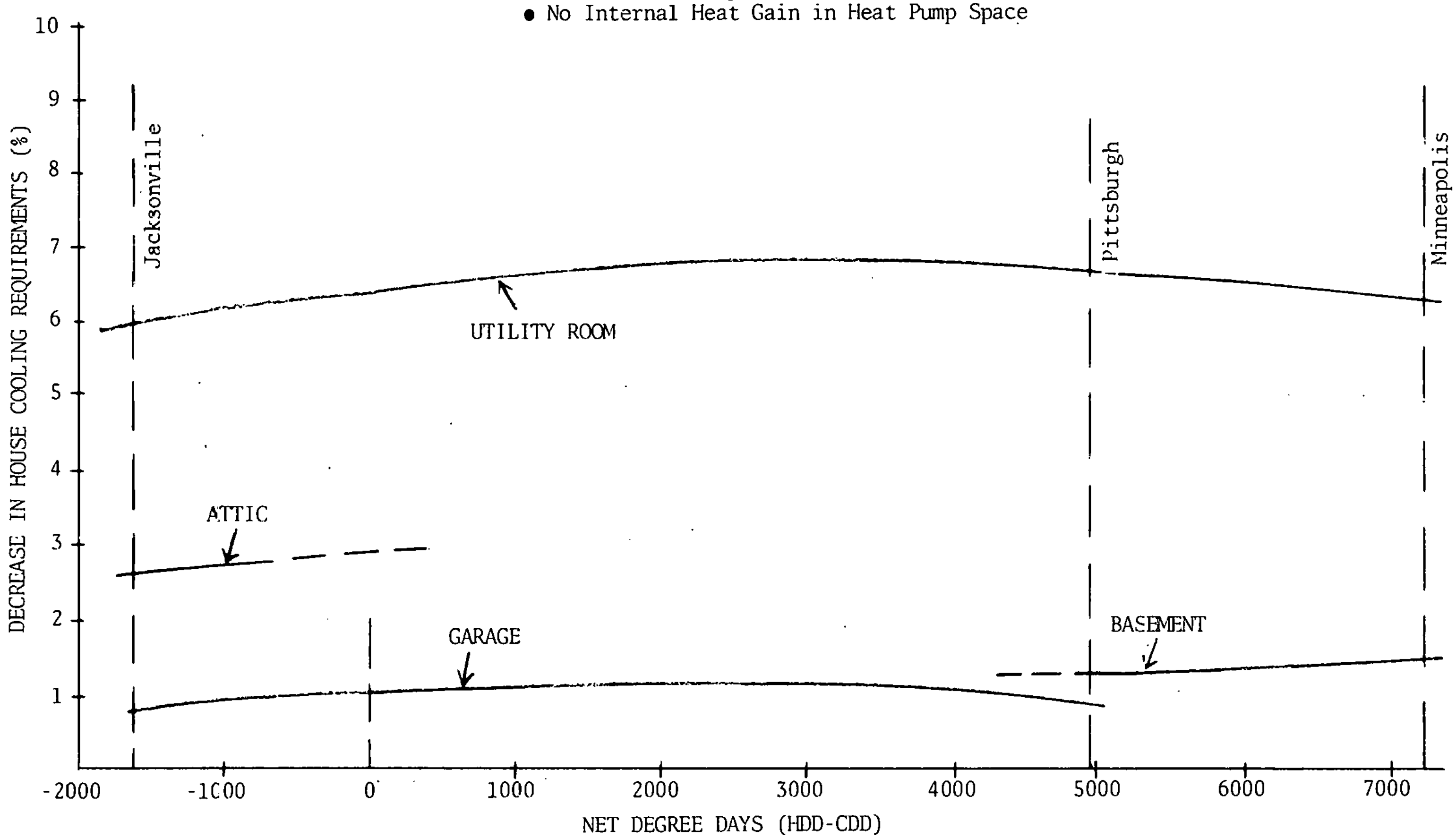


Figure C-16

DECREASE IN HOUSE COOLING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER OPERATING IN SPECIFIED LOCATIONS IN TWO-STORY HOUSE WITH BASEMENT

- 50°F Cutoff Temperature
- No Internal Heat Gain in Heat Pump Space

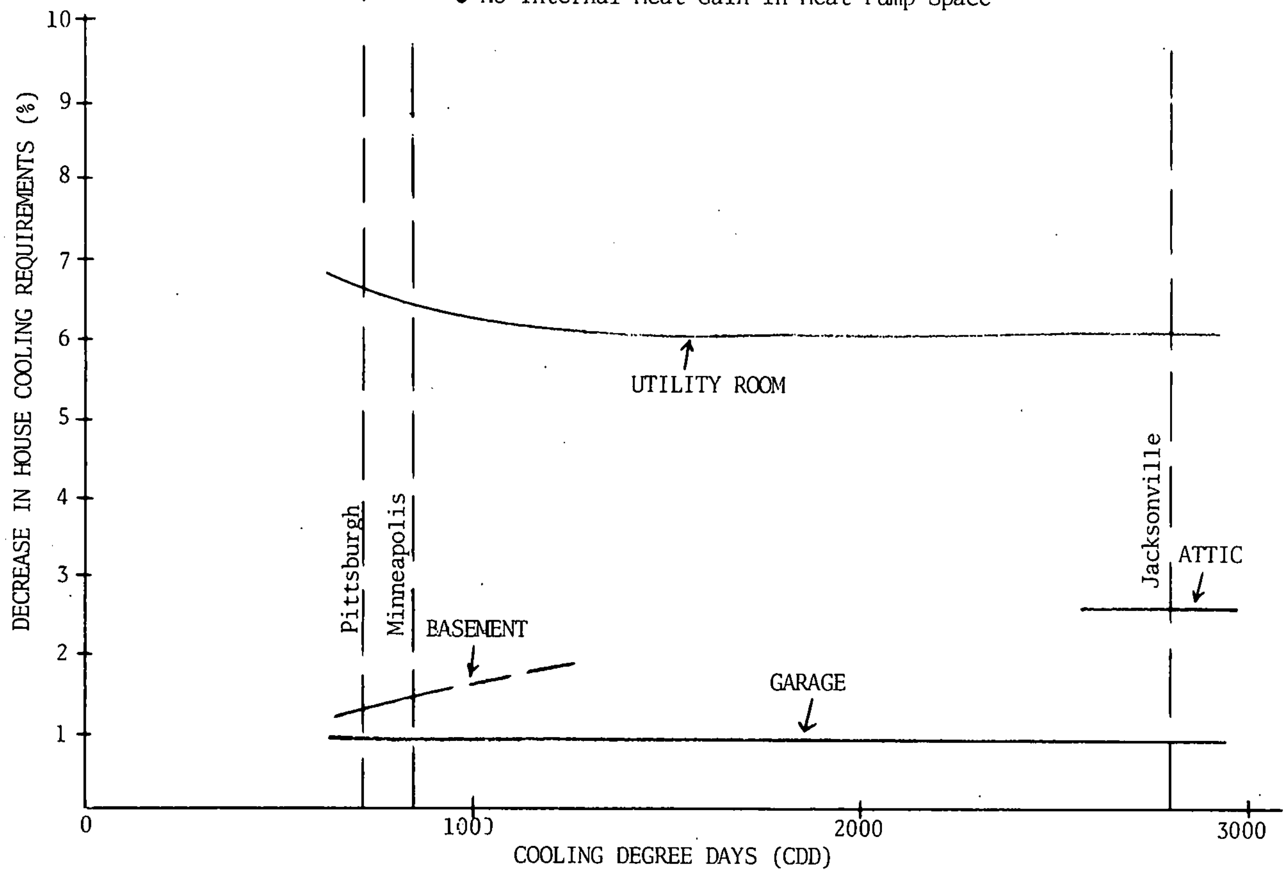


Figure C-17
UTILIZATION EFFICIENCY OF HEAT PUMP WATER HEATER LOCATED IN UTILITY ROOM

- Single-Story House on Slab
- $50^{\circ}F$ Cutoff Temperature

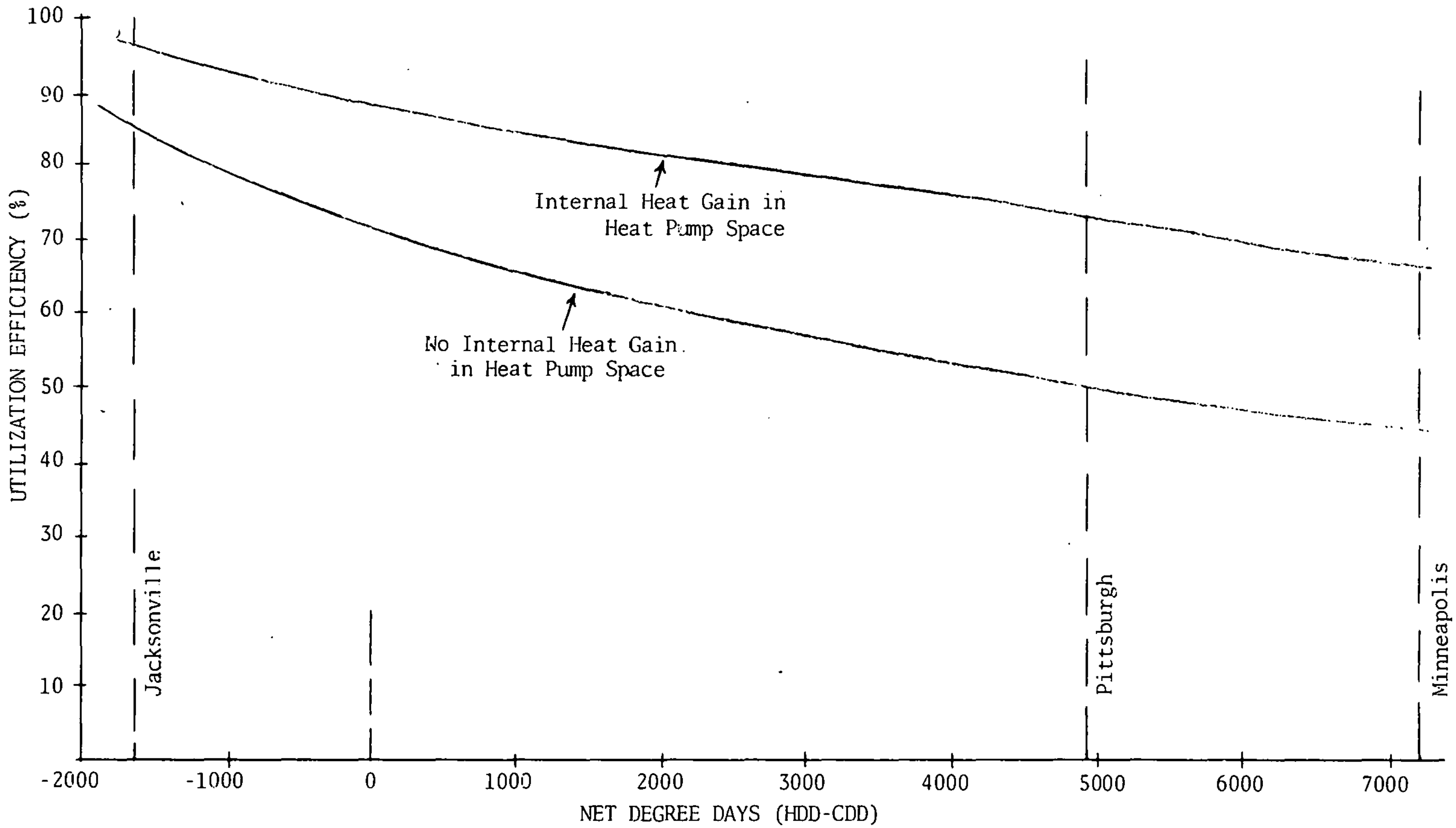


Figure C-18
INCREASE IN HOUSE HEATING REQUIREMENTS DUE TO HEAT PUMP WATER HEATER LOCATED IN UTILITY ROOM

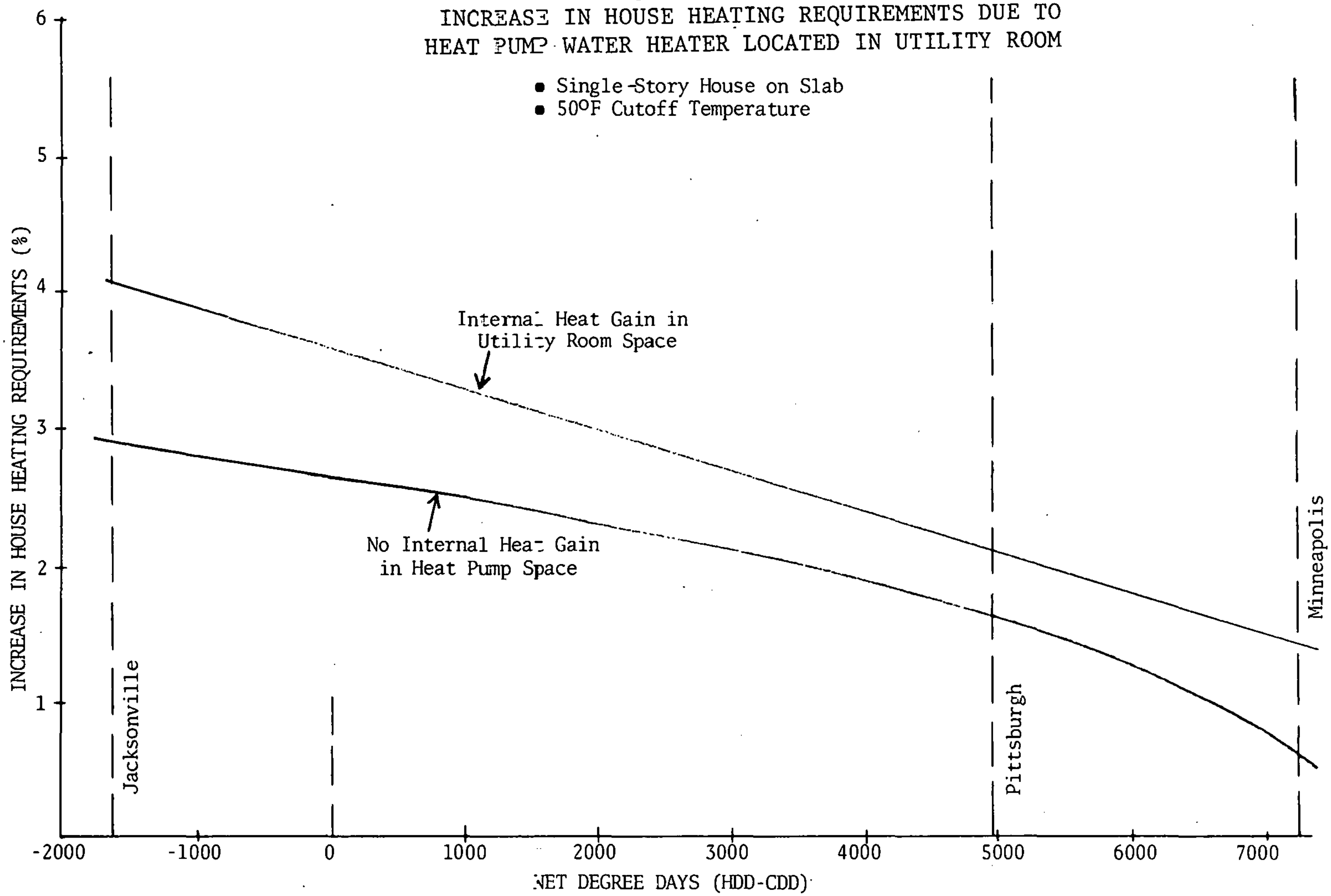
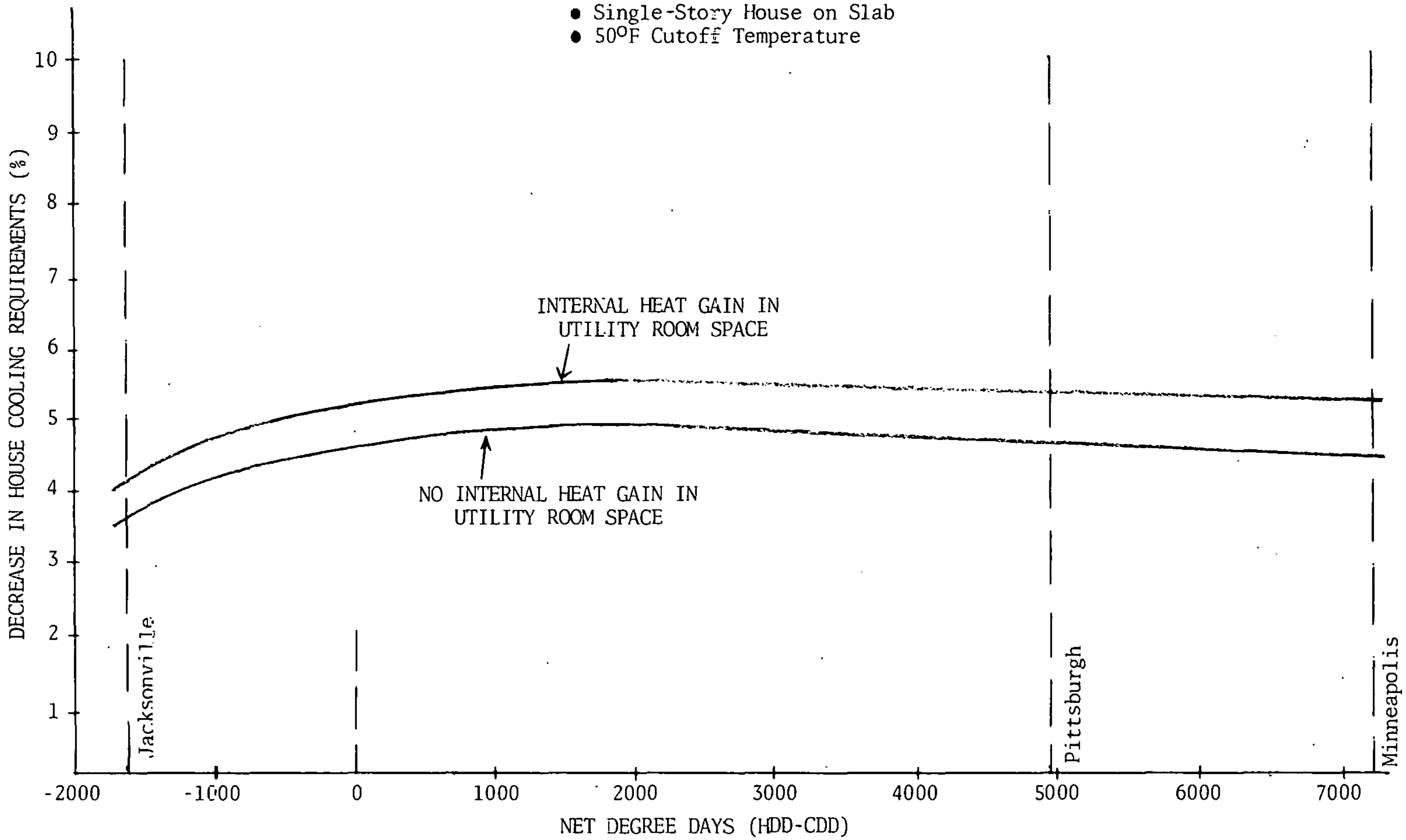


Figure C-19
DECREASE IN HOUSE COOLING REQUIREMENTS DUE TO
HEAT PUMP WATER HEATER LOCATED IN UTILITY ROOM

- Single-Story House on Slab
- 50°F Cutoff Temperature



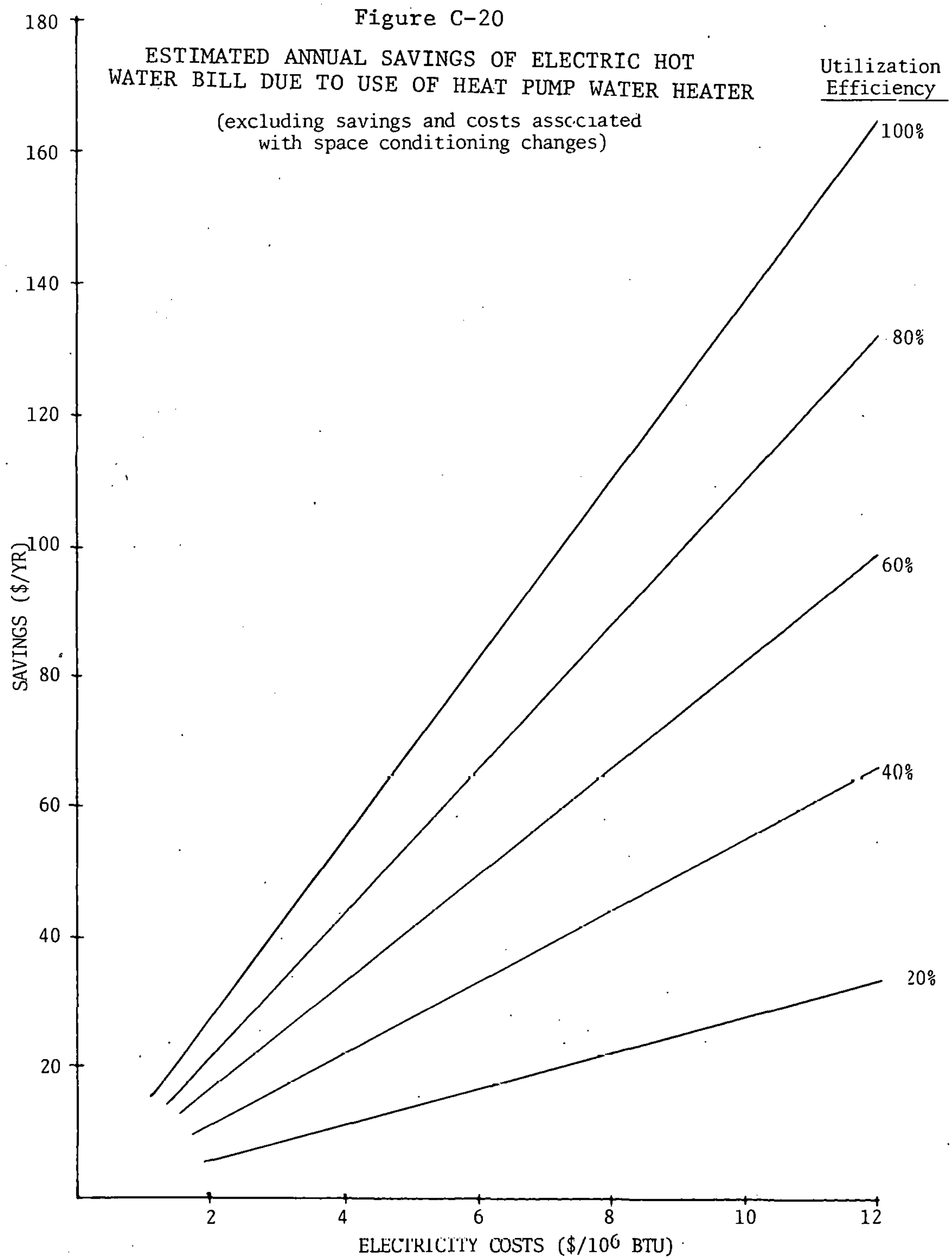


Figure C-20
ESTIMATED ANNUAL SAVINGS OF ELECTRIC HOT WATER BILL DUE TO USE OF HEAT PUMP WATER HEATER
(excluding savings and costs associated with space conditioning changes)
Utilization Efficiency
100%
80%
60%
40%
20%
SAVINGS ($/YR)
180
160
140
120
100
80
60
40
20
2
4
6
8
10
12
ELECTRICITY COSTS ($/10^6 BTU)

Figure C-21

ESTIMATED INCREASE IN HEATING BILL
DUE TO OPERATION OF HEAT PUMP WATER HEATER

- Jacksonville
- 1500 ft^2 Single-Story House

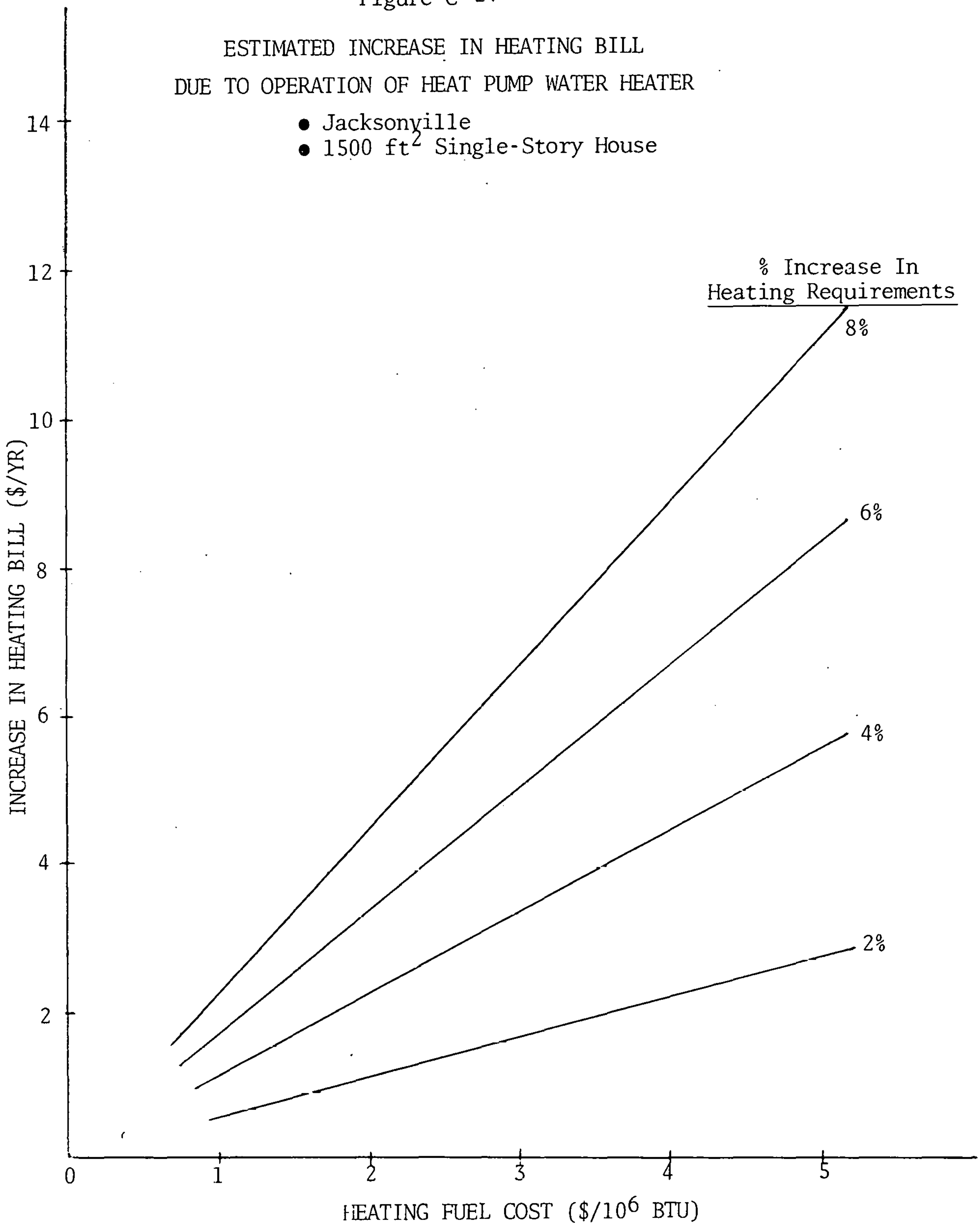


Figure C-22

ESTIMATED INCREASE IN HEATING BILL
DUE TO OPERATION OF HEAT PUMP WATER HEATER

- Pittsburgh
- 1500 ft^2 Single-Story House

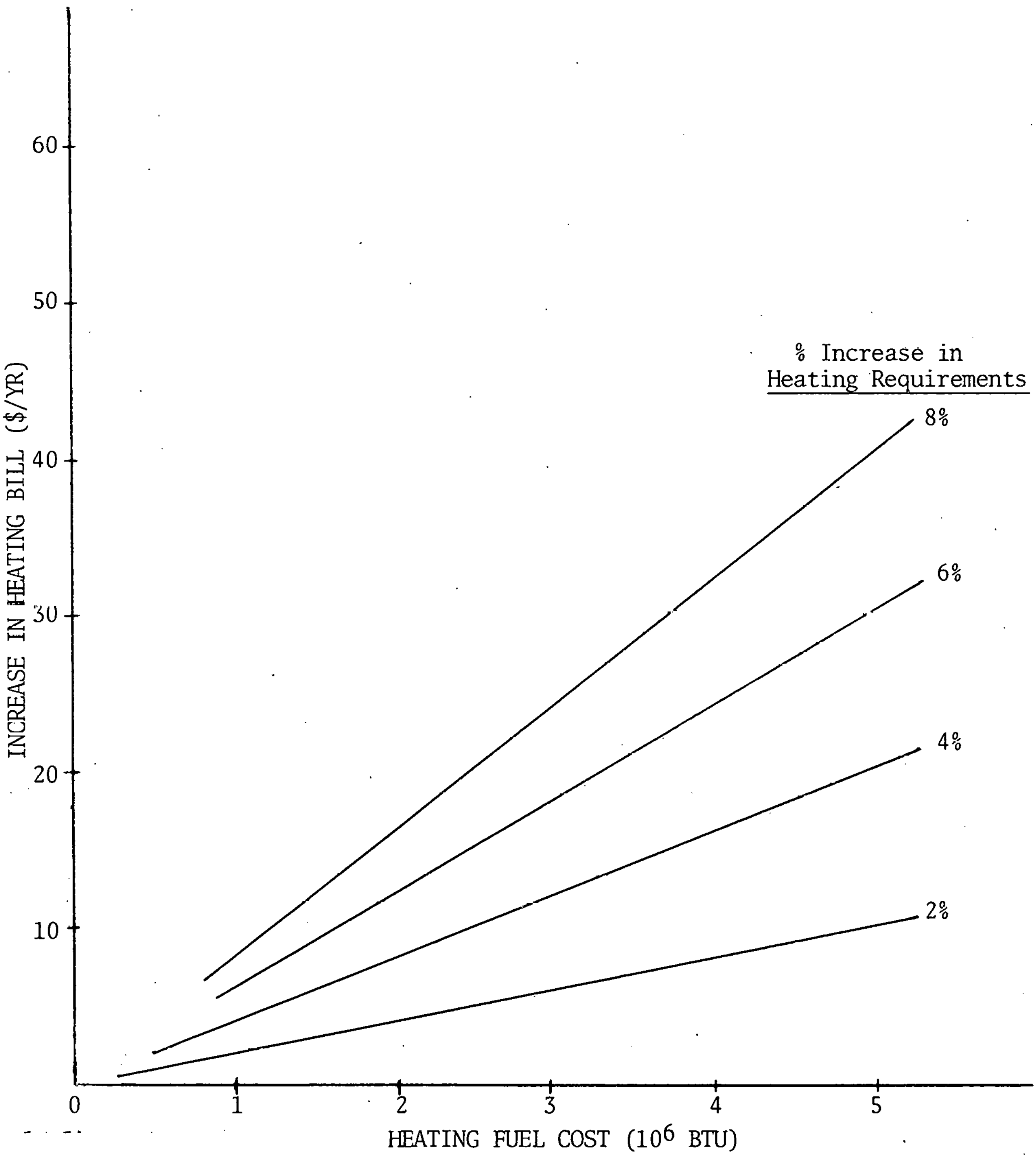


Figure C-23

ESTIMATED INCREASE IN HEATING BILL
DUE TO OPERATION OF HEAT PUMP WATER HEATER

- Minneapolis
- 1500 ft^2 Single-Story House

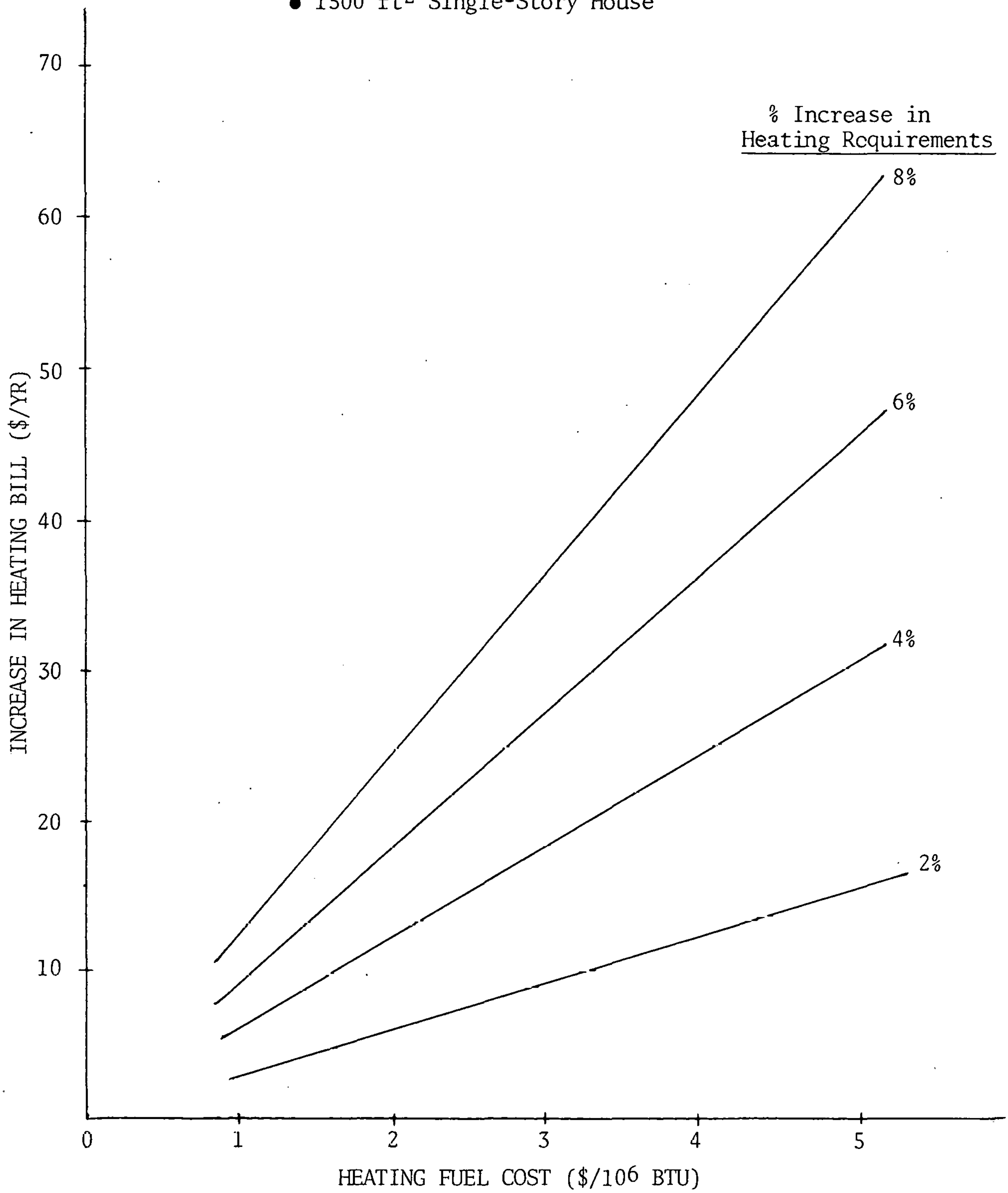


Figure C-24

ESTIMATED DECREASE (SAVINGS) IN A/C BILL
DUE TO OPERATION OF HEAT PUMP WATER HEATER

- Jacksonville
- 1500 ft^2 Single-Story House

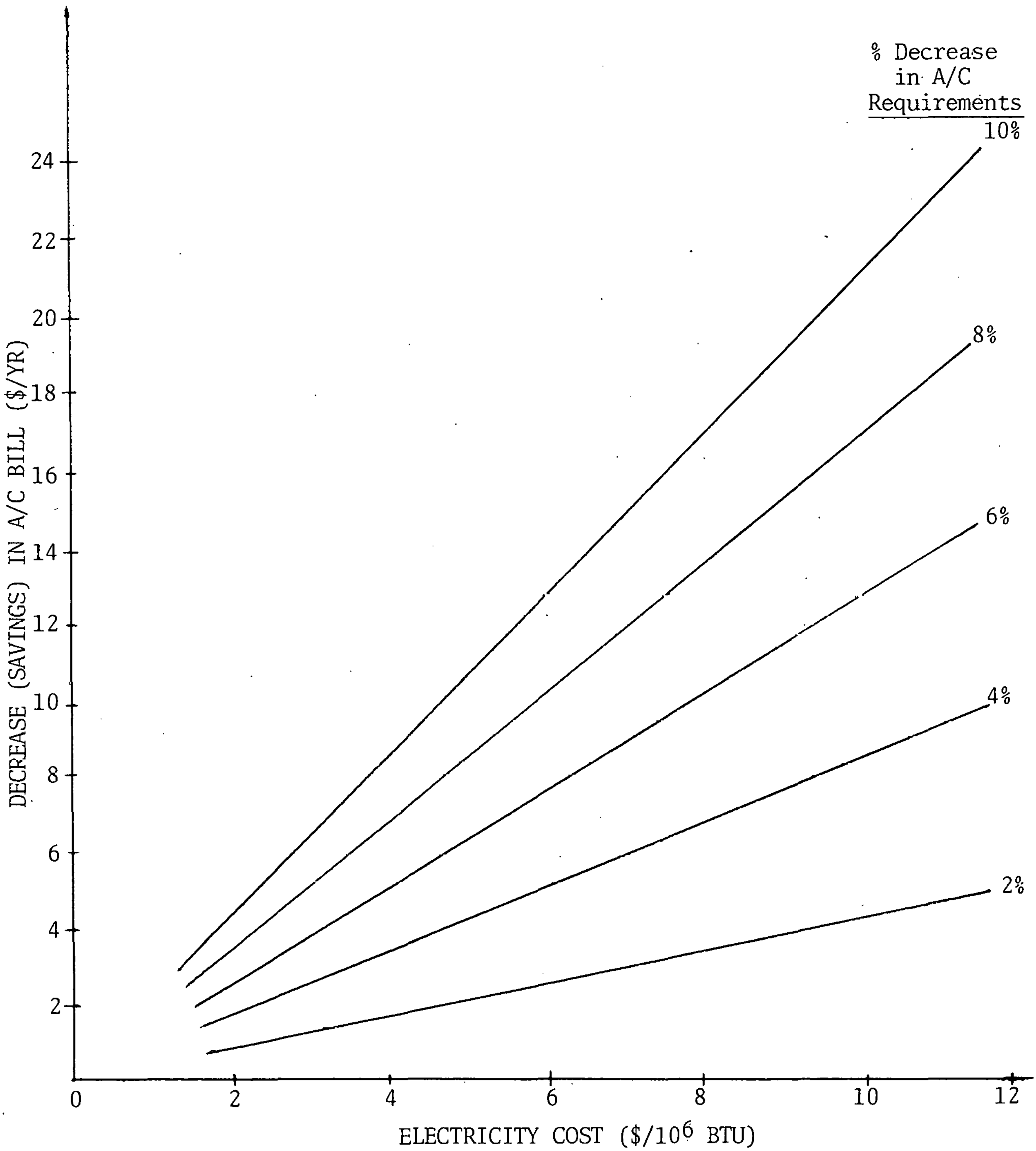


Figure C-25

ESTIMATED DECREASE (SAVINGS) IN A/C BILL
DUE TO OPERATION OF HEAT PUMP WATER HEATER

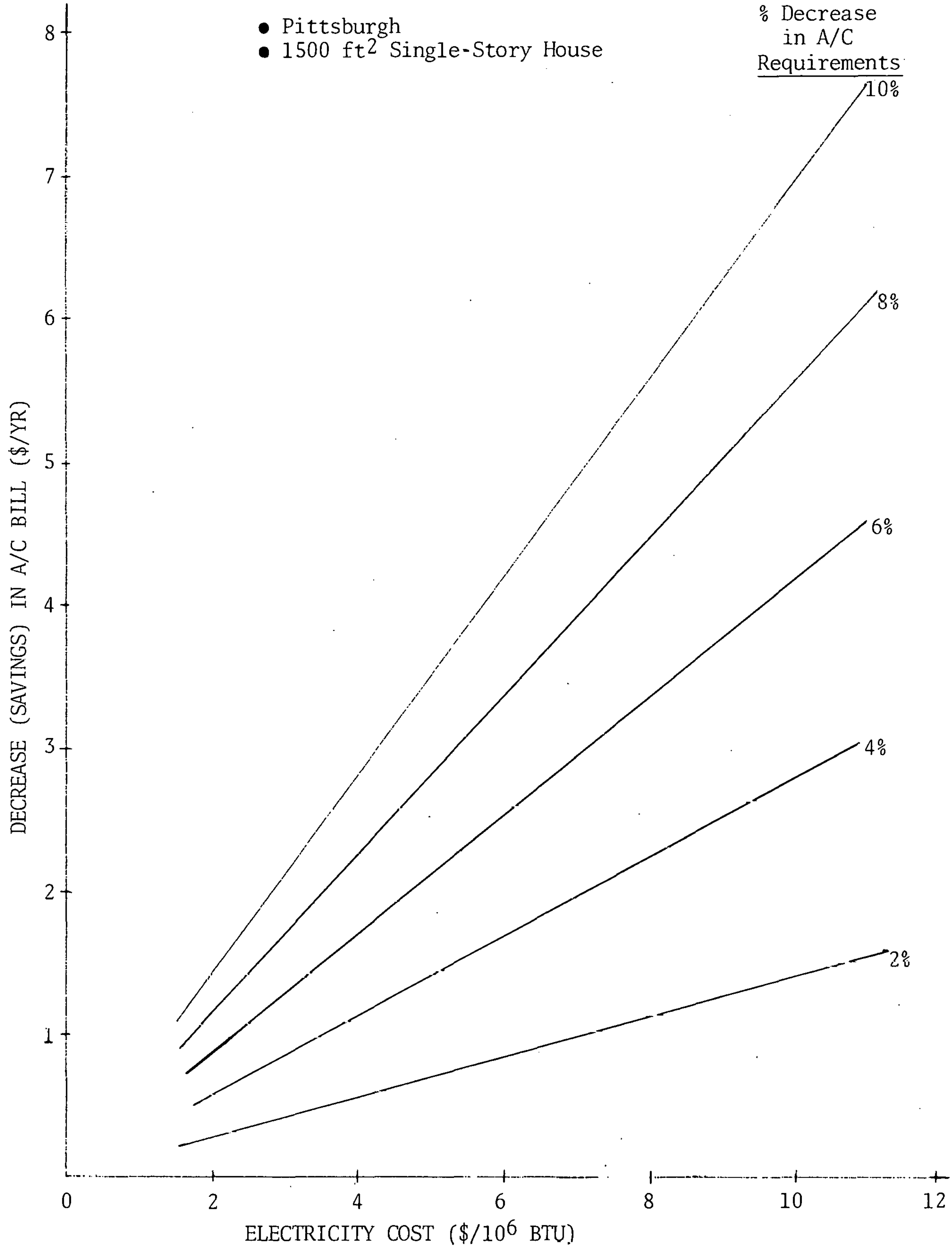
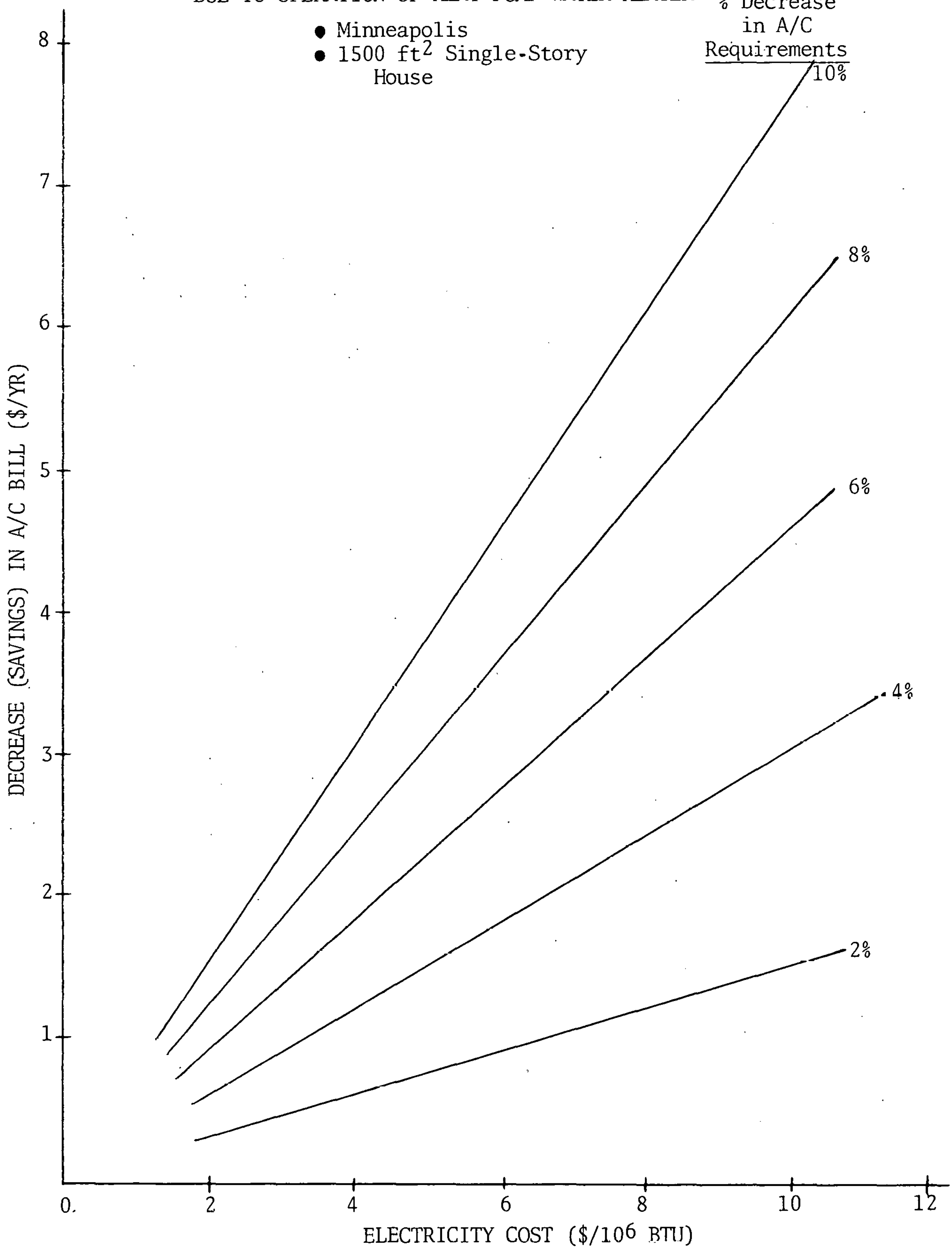


Figure C-26

ESTIMATED DECREASE (SAVINGS) IN A/C BILL
DUE TO OPERATION OF HEAT PUMP WATER HEATER



- Estimated Energy Savings and Costs due to Heat Pump Water Heater

Summary Comments

- Heating and Cooling (A/C) bills costed for 1500 ft^2 single story house. For 1700 ft^2 two story house the heating is increased ∿ 10% and the A/C is increased ∿ 20%.

- Sample Problem

For the heat pump water heater located in the Utility Room, in Pittsburgh (with a 50°F cutoff temperature and no internal heat gains), the savings estimate would be as follows.

	%	(\$/yr)*
Utilization	50	\$68.00
Heating	+1.6	-5.00
Cooling	-4.6	+3.35
	Net Saving	\$66.35/yr.

*Assumed fuel cost at $\$3.00/10^6$ BTU and electricity cost at $\$10/10^6$ BTU.

RESEARCH AND DEVELOPMENT

OF A

HEAT-PUMP WATER HEATER

TASK 4

FIELD DEMONSTRATION PLAN

JUNE 1978

Prepared under Subcontract No. 7321 by

Energy Utilization Systems, Incorporated
365 Plum Industrial Court
Pittsburgh, Pennsylvania 15239

for

Oak Ridge National Laboratory

Operated by

Union Carbide Corporation

Oak Ridge, Tennessee 37830

for

The United States Department of Energy

Contract No. W-7405-eng-26

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Abstract

This report covers the plan to manufacture and field test one hundred residential electric heat pump water heaters in single family homes. Tests will be located in the service areas of twenty climatically dispersed electric utilities. The project will be sponsored in part by the Department of Energy (DOE) through the Oak Ridge National Laboratories (ORNL); by twenty public and private utilities; and by two participating manufacturers, Energy Utilization Systems, Incorporated (EUS) and Mor-Flo Industries.

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CONTENTS

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LIST OF FIGURES AND TABLES

1. Introduction

Domestic hot water heating is usually the second largest user of energy in the home. In a typical suburban or rural house with an electric water heater, water heating uses between five and ten thousand kilowatt hours (kWh). With electricity costing between three and seven cents a kWh, the annual cost is between 150 and 700 dollars.

The heat pump type of water heater, with an average Coefficient of Performance (COP) of as much as three, would reduce the kWh use, and thus the cost, to one-third of these levels. At a projected installed cost of $200 more than for a conventional electric water heater, the payback period would be two years or less in most cases.

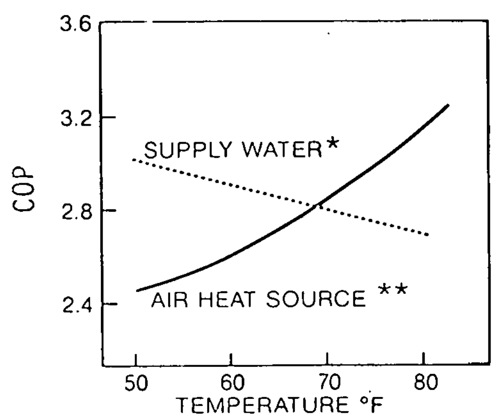
The idea of a heat pump water heater is not new. Two firms built them in the late fifties but due partly to the low and still declining cost of electricity, they did not sell well and were taken off the market. Now with the cost of electricity escalating rapidly, the heat pump water heater has become economically feasible in many, if not all, areas of the country.

The heat pump water heater works much like a window air-conditioner except that it pumps heat into the water tank rather than outdoors and is expected to pump it from otherwise waste heat in an unconditioned space such as the basement, garage, utility room or attic.

In the South, the device provides the fringe benefit of air-conditioning and dehumidification for most of the year. For installations providing this benefit this represents a saving of $75 a year or one-third the installed cost of the device in addition to saving two-thirds of the cost of water heating. The same principle would be true in the summer in the North. But in the winter, the heat pump would be taking heat from the house that would need to be replaced by the home heating system. That would be true if the unit is installed in a conditioned space such as a heated basement or utility room. But if the unit is installed in an unconditioned space the unit would use heat from the ground or outside air flowing through the basement walls and floor, losses from the furnace and/or ductwork or other interior heat gains. In a worst case analysis, where no waste heat is available, the water heater could be switched to resistance heating in the winter with its COP of one. Assuming equal seasons and a summer COP of 3, the annual average COP would then be 2. Part of the difference between the COP of 2 and the desired COP of 3 is offset by the value of dehumidifying the basement in the summer. A further consideration is that in the North, supply water is colder than it is in the South. This improves the efficiency of the heat pump and offsets some of the advantage of the warmer climate in the South. This is demonstrated in Figure 1 which shows curves of COP versus ambient air temperature and supply water temperature obtained in our laboratory tests. The colder water supply also requires more energy for water heating. This increases the potential for operating cost savings and thus for lowering the payback period. As shown in Figure 2, daily energy consumption is increased by 27% going from an average of 70°F in the South to 50°F in the North.

Figure 1

COP VS. SUPPLY WATER AND HEAT SOURCE TEMPERATURES

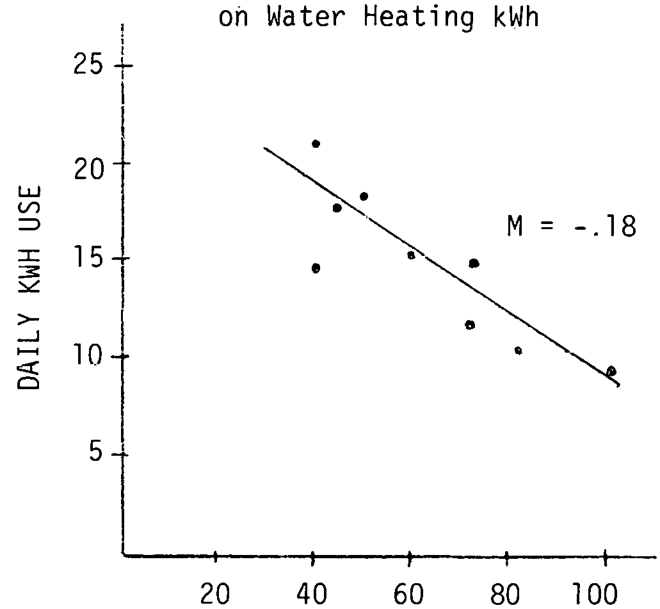


* Air source @ 70° F
** Supply water @ 70° F

Although the realtive value of the heat pump in different climates is not yet determined, the primary criteria, the payback period, is relatively insensitive to the COP. At least it is not as sensitive to COP as it is to the cost of electricity and to the amount of hot water consumed. (see Figure 3) In the case of the COP, the payback period (which is measured as the ratio of the additional installed cost to the monthly operating savings including maintenance costs

Figure 2

Effect of Supply Water Temperature
on Water Heating kWh



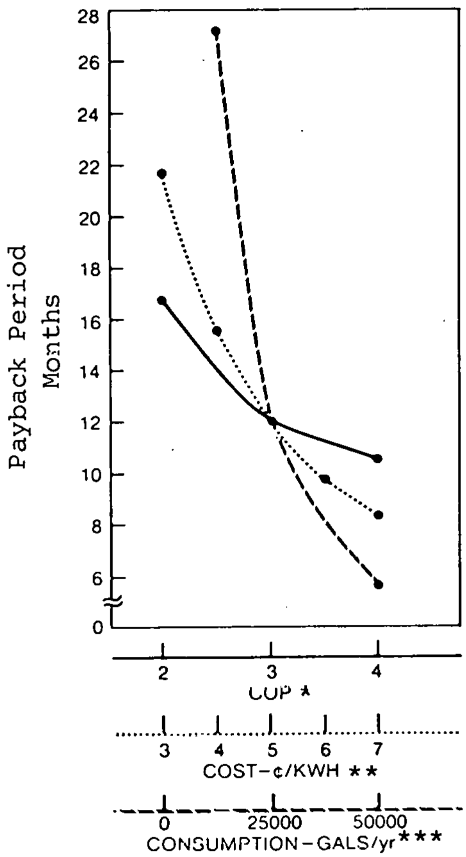
Source: Load Research Committee Report
Association of Edison Illuminating
Companies, 1963-64, Page L-137

compared to resistance type water heating) increases 57% between a COP of 4 and one of 2. These are the practicable limits for the device. In the case of the cost of electricity, payback period increases 161% between the limits of currently used prices of seven and three cents per kilowatt-hour. For hot water consumption, payback period increases by 110% between 50,000 gallons per year which is representative of a typical rural or suburban family and the national average of 25,000 gallons per year. Thus, many homes in the North where hot water consumption and the area electric rates are both above average will prove to have high potential even if the colder climate does reduce the COP.

With thirty million electric water heaters now in use there would appear to be a large market for a heat pump for retrofitting these existing units.

Figure 3

Payback Period vs COP, Cost of Electricity & Hot Water Consumption for New Installations



* Cost = 5¢/kWh; Consumption = 25000 gals/yr
** COP = 3; Consumption = 25000 gals/yr
*** COP = 3; Cost = 5¢/kWh

But the majority of these are either too old or too small to justify retrofitting or are eliminated by some accessibility or space limitation. Thus, the estimated potential for retrofitting existing water heaters is 2,400,000 units. This is derived from sixty percent being less than 52 gallons or too small to retrofit, fifty percent being more than five years old and sixty percent being located in inaccessible places.

Existing water heaters, gas, oil and electric, are being replaced at the rate of 3,5000,000 units per year. These represent a great potential for replacement with a heat pump type electric unit. Replacement of electric resistance type water heaters installed over the next five to ten years will all represent retrofit potential. That potential will be enhanced as the price of electricity continues to rise, shortening the payback period.

New units for new dwelling construction represent the greatest potential if large home builders adopt the heat pump water heater and use it as a selling point because of its low operating cost.

Aggregating the above market segments shows a potential market growing to seven million units by 1982, with initial market penetration in those areas served by merchandising utilities.

Realizing the maximum potential will take considerable time even in short payback period application because institutional constraints will impede the market acceptance. The most significant is the need to train plumbing installers who normally install water heaters to apply, install and service a refrigeration system. Without that training, customer complaints would be multiplied and the device would get a bad image at the start.

To reduce that possibility, electric utilities, each of whom will have people trained at the manufacturer's location, will participate in the demonstration program. From this program, it is hoped that enough field experience will be gained to create a comprehensive application, installation and service training program for the training of personnel throughout the normal channels of distribution of water heaters.

Engineering and test work performed by EUS and sponsored by DOE has succeeded in developing a residential electric heat pump water heater* which has thus far achieved a Coefficient of Performance (COP) of 2.8 in laboratory tests in an ambient of 70°F with 60°F supply water.

Units for the field test program will be 82 gallons because of the greater flexibility of control they offer. Six units have been on continuous life tests for periods up to eight months without a component failure. The next step in the development involves production for test in actual homes of a meaningful number of units to:

- determine the average performance of the device in different climates, locations in the house, types of houses, modes of control, and water characteristics.

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* Patents applied for

- determine application, installation and maintenance problems and costs
- determine actual payback period

The above information can then be used to:

- create an application, installation and maintenance manual
- promote the heat pump water heater as a cost and energy saving device
- encourage other manufacturers to manufacture and market the device
- provide publicity for electric utilities for their part in providing lower cost electric water heating

2. The Field Demonstration Plan

As envisioned, the plan would involve the final design and manufacture of one hundred and thirteen units by a combination of EUS and Mor-Flo.

Eighty-five units would be for new installations with tank and heat pump assembled, tested and shipped in one carton. The tank and shipping carton would be supplied by Mor-Flo. The tanks would be standard units but modified to use special covers to accomodate installation of the direct immersion condenser. EUS will build the heat pump units, assemble them to the tanks, test and ship via Mor-Flo's in-house fleet of trucks.

Twenty-five units would be the heat pump only for retrofit to existing tanks. They would be produced and shipped by EUS.

Three units would be new type for test by Underwriters Laboratory. Ten units (five of each type) would be held by EUS for replacement of defective units over the course of the test. Thus, eighty new units and twenty retrofit units will be installed and tested.

Five units will be shipped to each of twenty utilities. A preliminary list of utilities and the number and type they are expected to install is shown in Table 1. The list will be finalized after discussion with the utilities.

TABLE 1

Participating Utilities

Utility	Type	Installation Location	Quantity
Valley REC, Huntingdon, PA	New	Basement	5
Somerset REC, Somerset, PA	New	Basement	5
Indianapolis Power and Light	New	Basement	5
Public Service of Indiana	Retrofit	Basement	5
NRECA	New	Basement	5
Kansas Gas and Electric	New	Garage	5
Kentucky Assoc. of Electric Coops	New	Garage	5
Ohio Edison	Retrofit	Basement	5
Toledo Edison	Retrofit	Basement	5
Mississippi Power and Light	New	*	5
South Carolina Gas & Electric	New	Utility Room	5
Duke Power	New	Garage	5
Florida Power and Light	New	*	5
Southern California Edison	New	Outdoors	5
Gulf Power	New	Utility Room	5
Hawaiian Electric	New	Outdoors	5
Arizona Public Service	New	*	5
Portland General Electric	New	Utility Room	5
Eugene Water Board	New	Utility Room	5
Bonneville Power Admin.	Retrofit	Basement	5
		Total	100

* To be determined.

Installations will be monitored to measure the COP of the water heater including the quick recovery resistance element. Also measured will be the impact of the water heater on the heating and cooling systems of the house. Figure 4 is a schematic of the instrumentation to be used.

The heat pump water heater will be cycled -- on one day and off the next. The kWh used for water heating by the two systems are then compared over 12 months to establish the COP. The energy used for heating or cooling in alternate days will be compared to determine the effect of the heat pump on the environmental energy use. The COP will then be adjusted accordingly. An increase in heating requirement during heat pump use will lower the COP, a decrease in cooling requirement will increase the COP.

Houses selected for the demonstration will be limited to ones with central systems for space heating and air conditioning. Where air conditioning is not used, one meter will be eliminated. Where an oil furnace is used, one meter will be eliminated and space heating will be measured by the amount of oil used during the heating season. It is not anticipated that any house will have a gas furnace because that normally means a gas water heater. However, a separate gas meter could be substituted for the space heating kWh meter.

Specific parts of the plan are summarized in Table 2 and listed in detail in Section 3. Stated briefly, EUS and Mor-Flo will design and produce the units. Electric utilities will be expected to:

1.) purchase the heat pump water heaters at $200 for retrofit units and $371 for new units complete with tank; (estimated to be ultimate retail price)
2.) attend a training session in Pittsburgh
3.) install and service the water heaters and instrumentation

4.) collect data, including service-maintenance experience, each month and mail to EUS; and

5.) provide liability insurance or indemnify the contracting agencies.

DOE/ORNL will provide costs of:

1.) prototype units in excess of prices paid by utilities

2.) instrumentation

3.) data reduction and analysis; and

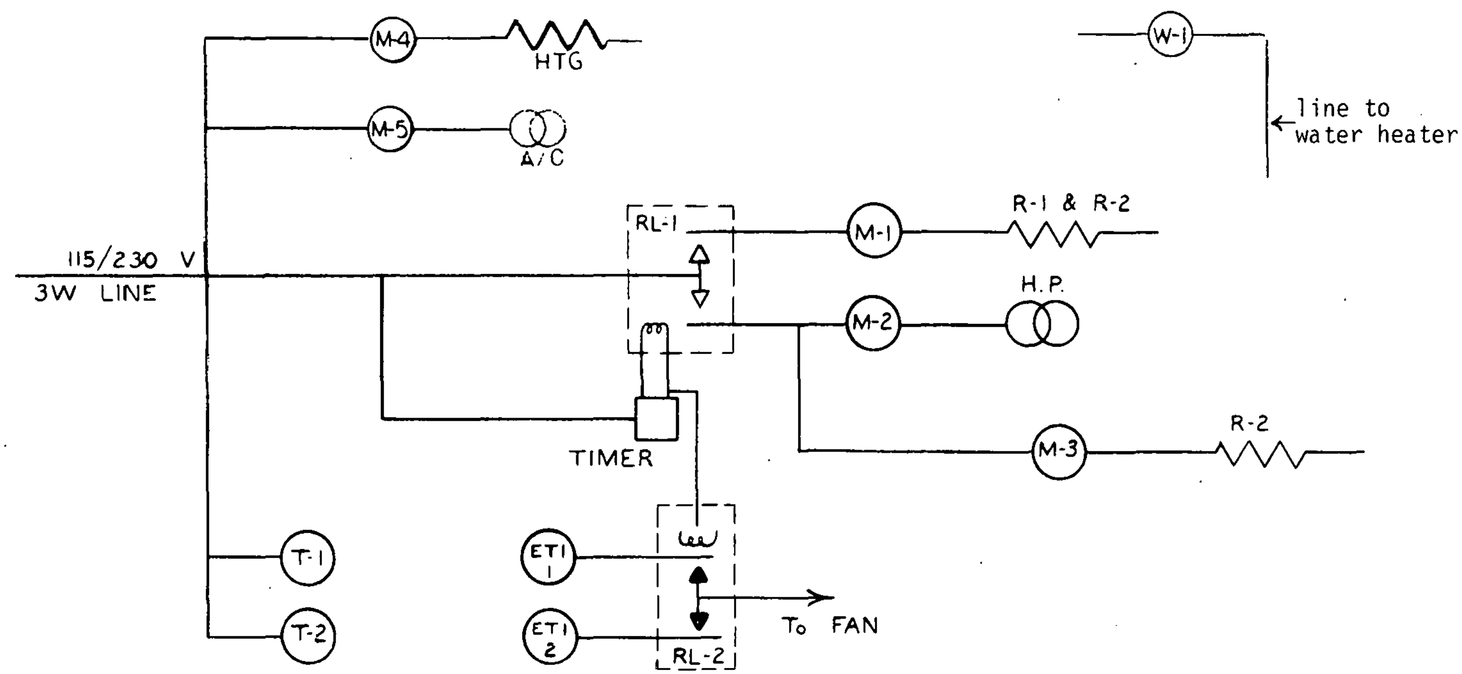
4.) spare parts or replacement units.

The test is scheduled to run for twelve months with a possible extension to be determined later.

FIGURE 4

Schematic

Instrumentation Package



ETI 1 + 2	Elapsed time indicators
RL - 2	Single-pole double-throw relay
RL - 1	Double-pole double-throw relay
Timer	14 day
M - 1	kWh meter for I^2R heater - R-1 and R-2
M - 4	kWh meter for space heating
M - 5	kWh meter for air conditioning
M - 2	kWh meter for heat pump water heater
M - 3	kWh meter for top element R-2 when heat pump is on
T - 1	Dual temperature recorder ambient air & delivery water temperature
T - 2	Supply water temperature recorder - 1 per city
W - 1	Water flow meter

TABLE 2

Electric Heat Pump Water Heater

Summary of Field Demonstration Plan

Organization	Participation	Objective of Organization	Methodology
EUS	Build 113 heat pumps Build and test 25 retrofit urits Build and test 88 new units Ship 3 units to UL for test Perform data reduction and analysis Prepare monthly, end of task and final reports Prepare literature and publicity material	Identify problems in manufacture, application, installation and servicing of heat pump water heater	Set up pilot production line
Mor-Flo	Supply water tanks Distribute units to utilities	Ultimately assemble and market new units	Use special tanks with large hole in top. Ship via house truck fleet
DOE/ORNL	Provide product costs above estimated ultimate retail price Provide cost of: instrumentation, renewal parts, data reduction and analysis, replacement at end of test, literature and publicity material	Expedite market acceptance of heat pump water heater and thus reduce energy use.	Publish equipment cost and operating performance report.
Utilities	Send representative to manufacturer for training Purchase 100 units @ 5 per utility Install unit and instrumentation Maintain unit and instrumentation Collect data and send to EUS Publish local publicity	Public Relations Lower consumer cost of electric water heating Reduce impact of water heating on system peak	Use of company or contracted personnel Local advertising and publicity

3.0 Outline of Work

Task 1 Develop final specifications and engineering design of the optimized heat pump water heater and pilot run manufacturing facility.

1.1 Complete final design and prepare drawings and specifications for heat pump water heater and tank

- 1.1.1 Prepare full scale assembly layout drawings
- 1.1.2 Prepare parts drawings with material specifications
- 1.1.3 Establish component performance and specifications
- 1.1.4 Prepare modified tank drawings for supplier (Mor-Flo)
- 1.1.5 Prepare bill of material for complete assembly and major sub-assemblies
- 1.1.6 Write installation instructions and service manual
- 1.1.7 Write utility training manual

1.2 Design temporary tools and fixtures for pilot run

- 1.2.1 Develop temporary tool and fixture designs
- 1.2.2 Prepare drawings of temporary tools and fixtures as required

1.3 Finalize design of instrumentation package for field installations

- 1.3.1 Prepare wiring diagram for instrumentation panel and test bench
- 1.3.2 Prepare component specifications
- 1.3.3 Prepare bills of material

1.4 Select suppliers for materials, parts, components, tools, fixtures and instrumentation

- 1.4.1 Secure at least two quotations on each item
- 1.4.2 Modify drawings and/or specifications as required after discussions with suppliers

1.4.3 Select lowest cost supplier assuming delivery schedules are reasonable and quality level is with specification

1.5 Prepare detailed cost estimates of final heat pump design in pilot run quantities and for instrumentation package

1.5.1 Prepare manufacturing labor spread sheets

1.5.2 Prepare facilities and equipment list for manufacture and test

1.5.3 Prepare manufacturing expense supplies per unit analysis

1.5.4 Allocate manufacturing overhead

a. Receiving and inspection
b. Space, utilities and equipment
c. Supervision
d. Final test and quality control
e. Tool and fixture allocation

1.6 Prepare pilot run facility layout

1.6.1 Locate equipment, tools, fixtures, benches, etc., in the pilot plant facility

1.6.2 Prepare manufacturing operations analysis and manufacturing schedule

1.6.3 Determine manpower type and quantity to produce pilot run within scheduled time frame at the rate of three per day

1.7 Submit draft of final Task 1 report and revise to include comments of ORNL TM. The report will include:

a description of the final design for pilot manufacture, description of the pilot manufacturing facility, design of instrumentation package, identification of major suppliers and detailed pilot run cost estimates.

Task 2 Prepare for pilot run

2.1 Purchase temporary tools and fixtures

2.2 Purchase equipment for pilot run

2.3 Order material, parts and components for heat pump water heater pilot run for 88 new type units and 25 retrofit units, and ten sets of service parts except tanks

2.4 Expedite one set of parts for prepilot run tool tryout, assembly and test of one complete unit prior to pilot run

2.5 Purchase supplies for pilot run

Task 3 Construct and test three pilot run prototypes

3.1 Using pilot run tools and fixtures, fabricate and assemble three prototype units

3.2 Test and record data from one prototype

3.2.1 Test unit per National Bureau of Standards performance test procedures

3.2.2 Test unit for performance at 50°F, 70°F and 90°F ambients

3.2.3 Test unit for performance at delivery temperatures of 120°F, 130°F, 140°F and 145°F

3.2.4 Test unit with simulated thermostat failure to tank safety cut off temperature of 175°F

3.2.5 Set unit up to run on life test

3.3 Report results of tests to ORNL TM with recommendations for changes, if any

3.4 Obtain UL approval

3.4.1 Send prototypes to UL for test and approval

3.4.2 Communicate with UL and revise units as required

Task 4 Manufacture and test heat pumps, instrumentation packages, and service parts (except tanks)

4.1 Fabricate and assemble 88 new type heat pumps

4.2 Set units on test tanks and run for 24 hours on NBS daily cycle

4.2.1 Revise or repair units if required

4.3 Assemble units to water heater tanks

4.3.1 Run standard hi-pot (high voltage) safety tests

4.4 Carton unit for shipment

4.5 Fabricate and assemble 25 retrofit heat pumps and separate condensers

4.6 Connect retrofit units to test tanks and test condensers and run for 24 hours on NBS daily cycle test

4.6.1 Revise or repair units if required

4.7 Run standard hi-pot (high voltage) safety tests

4.8 Disconnect heat pump from test tank and condenser, recharge and seal with field precharged line seal connectors

4.9 Carton heat pump and condenser in separate cartons in shipment

4.10 Fabricate and assemble 102 instrumentation packages and spare components

4.11 Operate each instrumentation panel on test bench to simulate field demonstration

4.12 Hi-pot (high voltage) test each panel

4.13 Package for shipment

4.14 Ship heat pump water heaters and instrumentation panels to designated electric utilities and invoice them for normal retail cost of heat pump water heaters

4.15 Fabricate, assemble, test and pack 10 sets of service parts

4.16 Submit draft report to ORNL TM summarizing the manufacturing experience and revise to include comments of ORNL TM. The report will include results of tests of pilot run units, description of any problems encountered and how they were overcome and a summary of pilot run product and instrumentation costs.

Task 5 Train utility service personnel for best methods of installation, servicing, data monitoring and collection

5.1 Make final utility selections and secure contractual agreements for purchase, installation service and monitoring of units and for acceptance of liability

5.2 Hold two training sessions at the pilot plant during the production period

5.2.1 Designated utilities will send their representatives to EUS at their expense

5.2.2 Provide prepared installation and service manuals to each trainee (See Task 1.1.6)

Task 6 Install heat pump water heaters and instrumentation packages in preselected home locations, monitor and service them

6.1 Install water heaters and instrumentation packages - Utilities or their contractors

6.2 Visit each installation to monitor performance and record data each month for one year - Utilities and EUS

6.3 Send monthly data, including service-maintenance experience, to EUS for data reduction and analysis - Utilities

6.4 Service and maintain field test units as needed for one year - Utilities

6.5 Return parts and components which have failed for analysis of failure cause - Utilities

6.6 Visit representative utility installations to monitor and assist with initial installations and verify procedures

6.7 Travel to other utility installations on request to help in problem solving

6.8 Visit selected utility installations to interview home owner and secure first hand knowledge of performance and reaction

6.9 Provide service parts to utilities as needed

6.10 Make recommendations to utilities for changes in heat pump operation based on data from field tests

6.11 Analyze monthly data from utilities and summarize for report (for twelve months).

6.12 Submit first report after all units are installed and operating relating installation problems encountered and the solutions used

6.13 Submit quarterly reports summarizing results and experience

6.14 Arrange for disposition or return of instrumentation packages -- Maintain heat pump water heaters in homes or restore original equipment - Utilities

Task 7 Analyze and evaluate results of twelve months field demonstration and prepare final report. Revise market analysis based on field experience and make recommendations for further work which might accelerate commercialization of the heat pump water heater.

7.1 Draft of final report

7.2 Submit final report after resolution of ORNL TM comments -- report to include camera-ready master and 200 copies

Task 8 Special presentations and monthly reports

8.1 Make special presentations as requested by ORNL TM

8.2 Submit monthly reports during life of contract (24)

4.0 Schedule

Months from Start