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DESIGN, DEVELOPMENT, AND DEMONSTRATION OF A PROMISING INTEGRATED APPLIANCE. PHASE I: DESIGN

Final Report

By
W. David Lee
W. Thompson Lawrence
Robert P. Wilson

September 1977
Date Published

Work Performed Under Contract No. EY-76-C-03-1209

Arthur D. Little, Incorporated
Cambridge, Massachusetts

U. S. DEPARTMENT OF ENERGY

Division of Buildings and Community Systems

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FINAL REPORT

W. David Lee, Program Manager
W. Thompson Lawrence
Robert P. Wilson

ARTHUR D. LITTLE, INC.
CAMBRIDGE, MASSACHUSETTS

Date Published - September 1977

PREPARED FOR THE
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Office of Conservation
Technology and Consumer Products Branch

Work Performed Under Contract No. EY-76-C-03-1209

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The program drew heavily from the Steering Committee for technical, economic, and marketing inputs. The core Steering Committee team members were:

Ralph Braden	Frigidaire Division, General Motors Corporation
Robert Cook	A. O. Smith Corporation
Gil Engholm	General Electric Company
Dieter Grether	Friedrich Air Conditioning & Refrigeration Co.
Virgil Haynes	Oak Ridge National Laboratory
John Mobarry	Honeywell Inc.
Kurt Riegel	Energy Research and Development Adminstration - Washington

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William Gibeaut	American Gas Association
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Herbert Phillips	Association of Home Appliance Manufacturers
Chappell Pierce	Gas Appliance Manufacturers Association
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The entire project team and Steering Committee express our gratitude to the one person who administered and coordinated the program and report production, my secretary Monica Mehigan.

1.0 SUMMARY AND RECOMMENDATIONS

1.1 SUMMARY

1.1.1 Program Purpose and Philosophy

Appliances and heating and cooling equipment consume a significant amount of energy. Of the 70 quads of primary energy per year consumed in the United States in 1970, 28.4 quads were consumed by residential and commercial appliances and heating and cooling equipment.

Furthermore, much of this energy is expended as waste heat. Of the 11 quads of primary energy consumed each year in single-family homes, only about 5 quads of the energy is not lost as waste while providing its service, i.e., heating water, food, or room air. A large portion flows to the outside as waste heat; 55% of the waste is exhausted through vents of flues, 22% is passed down the drain, and the remaining 23% is lost through the building walls.

Combination or integration of appliance functions was felt to offer an opportunity for the economical recovery of some of this waste energy so that it could be used in other functions. As a result, the Energy Research and Development Administration in July, 1976, initiated a program of accelerating consumer use of integrated appliances designed to save energy.

Through an ERDA-supported development and demonstration program of a (or several) promising energy-saving integrated appliance, it was felt that: the manufacture and consumer acceptance of, and realization of, the energy savings from an integrated appliance could be accelerated. This report presents the results of the work performed to identify the most promising integrated appliance candidates and to recommend a demonstration program most likely to accelerate the commercialization of the integrated appliances. The key tasks of the program are summarized below.

- Background information and data on conventional appliances, including: patterns of energy usage, interaction with other appliances, and appliance population was assembled.

- Criteria for identifying and evaluating potential integrated appliance candidates was developed. Included in the criteria were: the potential for national energy savings, cost effectiveness, and likely consumer acceptance.

- Promising integrated appliance candidates were identified and evaluated according to the criteria developed in the previous task.

- A demonstration plan for the most promising candidates was developed.

- A final report covering all work was prepared.

A basic assumption in this study is that rapid acceptance and commercialization requires the involvement of present appliance manufacturers in all stages of the program. An Industrial Steering Committee* consisting of representatives from major appliance and equipment manufacturing companies was established in order to guide the search for the promising integrated appliances. The members of the Steering Committee had sufficiently diverse product lines and interests that it was likely that one or several of the participating companies would actually carry the proposed product(s) into pilot manufacturing and ultimately into the open marketplace.

1.1.2 Screening of Candidate Integrated Appliances

Since hundreds of combinations of appliances and heating and cooling equipment are possible, a methodology was needed for identifying the most promising ones. Some 349 combinations were examined in terms of their nationwide energy-savings potential. Those offering a possible savings of greater than 10^{14} Btu/year were selected for further consideration. Continued re-examination in light of more stringent criteria gradually narrowed the number of promising candidates down to 18. Specific designs for these 18 remaining candidates were developed and likely consumer acceptance of these candidates was considered.

Ten candidates did not meet an additional criterion value that the energy savings should exceed the added cost within several years (3.5 years to payback in the residential sector and 5 years to payback in the commercial sector). One candidate fell short of the 10^{14} Btu/year criterion, and two individual candidates were merged into one. Six candidates remained:

1) Furnace/water heater,

2) Central air conditioner/water heater,

3) Commercial range heat recovery for water heating,

4) Refrigerator/water heater,

5) Drain heat recovery for water heating for residential buildings (gas and electric versions), and

6) Drain heat recovery for water heating for commercial buildings.

The first three candidates, discussed in the following section, were selected as the most promising energy-saving combinations and were considered for further ERDA support. The results of the analysis of these three final candidates are given in the following section.

---

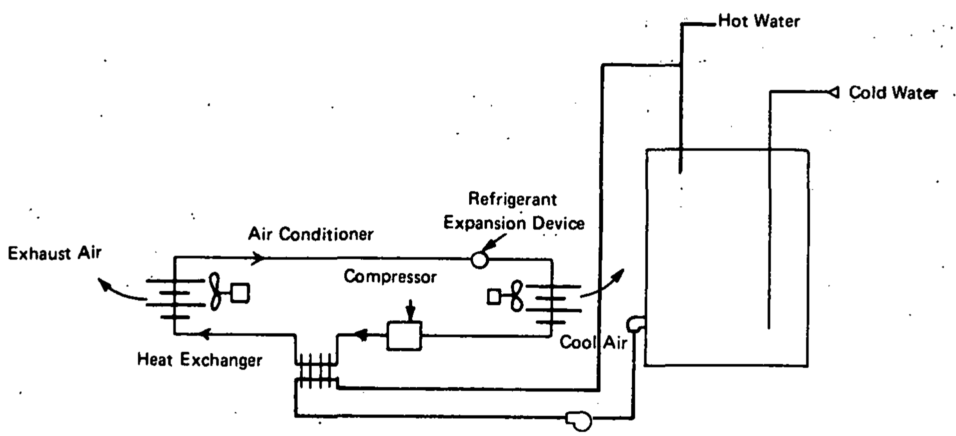
*See Acknowledgements.

1.1.3 Three Most Promising Integrated Appliances

Combined Central Air Conditioner Waste Heat Recovery System

Description

A heat exchanger recovers heat normally discharged to outdoor air for water heating. Water circulated from the storage water heater is heated in the heat exchanger by the hot refrigerant gas. The system concept is shown below.



Estimated Potential for Energy Savings

Based on the analysis of the system located in a number of climates in the United States, the predicted system energy savings is as follows:

Candidate	Energy Savings 10^6 Btu Primary Per Unit	Added First Cost Installed	Years to Payback	Max. 1990 Inventory Applicable 10^6 Units	1990 Annual National Potential Energy Savings - Primary 10^{14} Btu/year
Central Air Conditioner Heat Recovery for Electric* Water Heater	28.8**	$300***	3.5	10.6	3

*The technology is exactly the same for gas water heaters though the years to payback is beyond the 3.5 year level of acceptability.

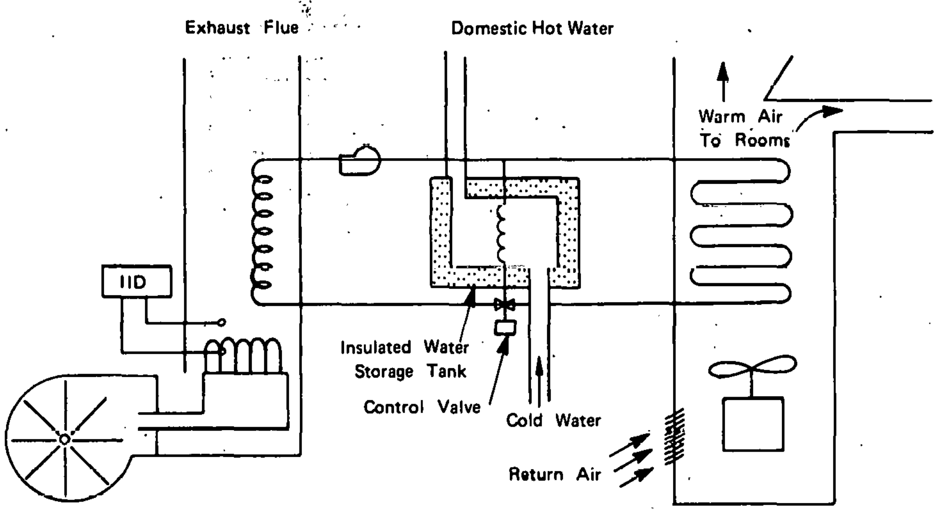
**The national average is weighted by the climatic distribution of central air conditioners projected to 1990.

***All costs are reported in 1975 dollars.

Combined Furnace/Water Heater

Description

A 120,000 to 140,000 Btu/hour forced-draft burner with an intermittent ignition device (IID) is combined with a 10 to 20 gallon hot water storage tank and appropriate controls. The single burner provides both water and space heating functions. A system schematic is shown below.



Estimated Potential for Energy Savings

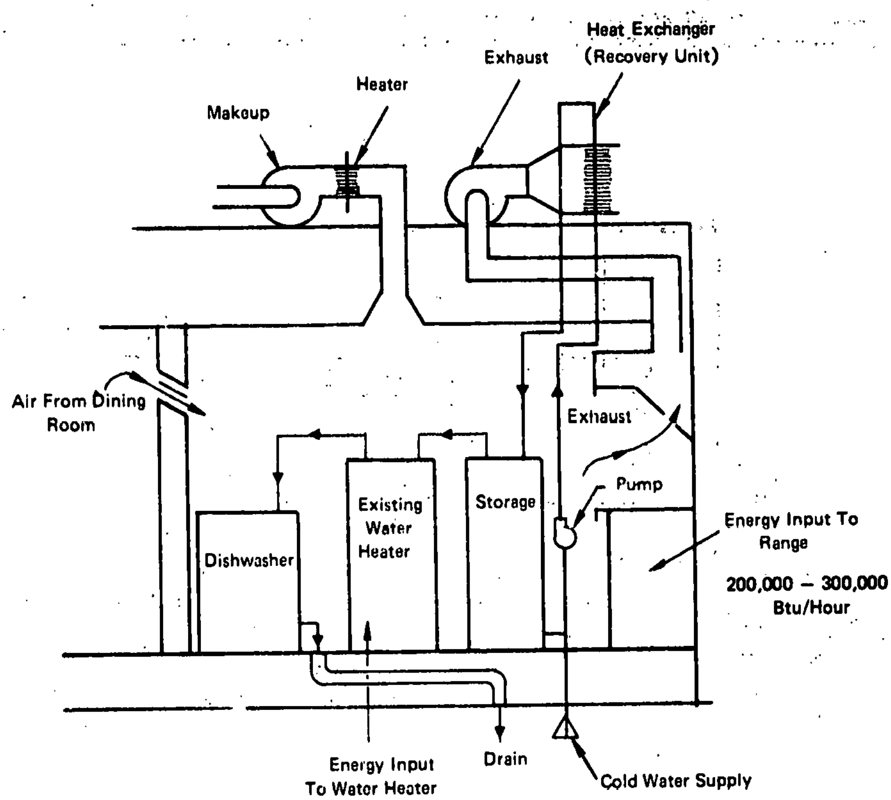
Based on the analysis of the above system in several U.S. climatic zones, the estimated energy savings of the combined furnace/water heater is as follows:

Candidate	Estimated Percent of Annual Sales	Energy Savings 10^6 Btu Primary Per Unit	Added First Cost Installed	Years Payback	Max. 1990 Inventory Applicable 10^6 Units	1990 Annual National Potential Energy Savings - Primary 10^{14} Btu/year
Furnace/Water Heater						
Gas Forced Air	64	38.0	$202	1.3	22.6	8.6
Gas Boiler	9	57.5	(38)	--	4.8	2.8
Oil Forced Air	17	30.5	82	1.9	3.2	0.3
Oil Boiler	10	30.0	(153)	--	1.7	0.5
Projected Sales Weighted Average	100	34.3	$124	1.0	32.3	12.2

Commercial Range Heat Recovery for Water Heating

Description

Located in the exhaust duct of the hood ventilation system, a heat exchanger recovers exhaust energy for water heating. The heated water can be used directly or can be boosted to a higher temperature for dishwashing. The system shown below returns the water heated by the exhaust to a holding tank connected to the existing water storage tank. In this fashion, water is preheated by the exhaust gas prior to entering the final water heater.



Estimated Potential for Energy Savings

Based on the analysis of the system, the estimated energy savings of the commercial range heat recovery for water heating system is as follows:

Candidate	Energy Savings 10^6 Btu Primary Per Unit	Added First Cost Installed	Years to Payback	Max. 1990 Inventory Applicable 10^3 Units	1990 Annual National Potential Energy Savings - Primary 10^{14} Btu/year
Gas					
Restaurant, School, Cafeteria, Institutions	92	$1,700	5.4	240	.22
Hospital, Hotel, Motel	1,250	5,330	1.2	11	.14
Electric					
Restaurant, School, Cafeteria, Institutions	224	1,700	2.5	84	.18
Hospital, Hotel, Motel	3,000	5,330	0.6	4	.11
				Total	.65

1.1.4 Comparison of Three Most Promising Integrated Appliances

Table 1.1 summarizes the estimated potential for energy savings assuming full market penetration into all applicable locations. Although the commercial range heat recovery integrated appliance is no longer above the 10^{14} Btu/year cut off level used in the screening, it is still considered to be a promising candidate. This is because realistic cost and design trade offs have been considered, and these are expected to enhance potential consumer acceptance, though lowering the national energy savings.

To account for consumer acceptance/energy-saving design trade offs, a new measure that combines energy-savings potential with probable consumer acceptance was used in the subsequent analysis. This measure, the cumulative energy savings based on estimated product market penetration (which is linked to the economic benefit to the consumer) over the ten-year period 1980 to 1990, is shown in Table 1.2.

1.2 RECOMMENDATIONS

1.2.1 Demonstration Plans

For each of the three final candidates, a demonstration plan was developed, designed to accelerate commercialization. Due to the difference in the state of development of the candidates, three different types of programs were required. The recommended estimated values of the programs and their emphases are given in Table 1.3. The estimate of the years that the ERDA-sponsored demonstrations would accelerate the commercialization of the product was based on judgments of the Steering Committee members and ADL staff.

The years by which the introduction and commercialization would be accelerated was then used to recalculate the cumulative energy-savings potential. The new projections are shown in Table 1.4 as the cumulative energy savings with ERDA support.

1.2.2 Benefit-Cost Ratio

The ratio of the value of the cumulative energy-savings potential in 1976 dollars to the estimated program cost to ERDA is shown in Table 1.5. Included in this estimate is a portion of the cost shared by manufacturers. This partitioning of cost reflects our (ADL) belief that the likelihood of successful commercialization of these candidates is quite high and the potential participating manufacturers might underwrite a portion of the Demonstration Program compatible with their expectation of successful commercialization of the product.

TABLE 1.1

COMPARISON OF THREE MOST PROMISING CANDIDATE INTEGRATED APPLIANCES

Candidate	Energy Savings 10^6 Btu Primary Per Unit	Added First Cost Installed	Years to Payback	Max. 1990 Inventory Applicable 10^6 Units	1990 Annual National Potential Energy Savings - Primary 10^{14} Btu/year
Central A/C-water heater (elec)	28.8	$300	3.5	10.6[1]	3.0
Furnace/water heater (gas/oil)	34.3	124	1.0	32.3[2]	12.2
Commercial Range - water heater (elec) small/large	224/3,000	1,700/5,300	2.5/0.6	0.33	0.65
(gas) small[3]/large[4]	92/1,250	1,700/5,300	5.4/1.2		

[1] Homes with central A/C (26 million) and electric water heating (40%) in the year 1990.

[2] New gas or oil furnaces installed between 1980-1990.

[3] Restaurant, schools, institutions.

[4] Hotels, hospitals, motels.

TABLE 1.2

CUMULATIVE ENERGY-SAVINGS POTENTIAL
(Without ERDA Support)

Integrated Appliance	Average Annual Primary Energy Savings 10^6 Btu per Unit	Average Nationwide Years to Payback	SALES: Percent Annual New Sales Captured	SALES: Max. Percent of In-Place Facilities Retrofitted	SALES: Average Sales Rate - 1985 (1,000's) OEM	SALES: Average Sales Rate - 1985 (1,000's) Retrofit	Cumulative National Energy Savings 1980-1990 in 10^{12} Btu of Primary Energy
Central A/C-water heater	20	4.3	19	2.5	162	81	159
Furnace/water heater	34.3	1.0	65	0	178	0	460
Commercial range/water heater							
(elec) small/large	224/3,000	2.5/0.6	45/70	24/50	1.3/.14	4.9/0.2	97
(gas) small/large	92/1,250	5.4/1.2	13/0	4/36	.3/0	4.9/1.0	

TABLE 1.3

DEMONSTRATION PLAN

Candidate	ERDA Acceleration: Years of Acceleration	Total Program Value $	EMPHASIS OF PROGRAM: Development	Demonstration	Public Information Dissemination
Furnace/Water Heater	3	500,000	50%	40%	10%
Air Conditioner HRS	2.5	200,000	20%	65%	15%
Commercial Range HRS	3	160,000	40%	50%	10%

TABLE 1.4

ESTIMATED EFFECT OF ERDA-SPONSORED DEMONSTRATION PROGRAM

Integrated Appliance	Average Annual Primary Energy Savings 10^6 Btu per Unit	Average Nationwide Years to Payback	SALES (With ERDA Support): Percent Annual New Sales Captured	SALES (With ERDA Support): Max. Percent of In-Place Facilities Retrofitted*	ERDA Acceleration: Average Number of Years of Acceleration	ERDA Acceleration: Cumulative National Energy Savings 1980-1990 in 10^{12} Btu of Primary Energy: Without ERDA	With ERDA	Effect of ERDA
Furnace/water heater	34.3	1.0	65	0	3	460	1,344	884
A/C heat recovery system	28.8	3.5	22	15	2.5	159	619	460
Commercial range/ water heater								
(elec) small/large**	224/3,000	2.5/.6	45/70		3	97	223	126
(gas) small/large	92,1,250	5.4/1.2	13/0	4/36				

*This grows at a linear rate from 1/10 of the value shown in 1980 to equal to the value shown in 1990.

**Restaurant, schools, institutions/hotel, hospital, motel.

TABLE 1.5

BENEFIT COST RATIO OF
ERDA-SPONSORED DEVELOPMENT AND DEMONSTRATION

Integrated Appliance	Program Cost: Total	Program Cost: Cost Shared Portion	Program Cost: ERDA Cost	Program Benefit 1980-1990 Cumulative Energy Savings: 10^{12} Btu	Program Benefit 1980-1990 Cumulative Energy Savings: Equivalent 10^6 Dollars*	Program Benefit / ERDA Cost
Furnace/ water heater	$500,000	$250,000	$250,000	884	3,094	12,400
A/C HRS	$200,000	$80,000	$120,000	460	1,610	13,400
Commercial range	$160,000	$64,000	$96,000	126	441	4,600

*Based on a uniform $3.50/primary mm Btu which is equivalent to $3.50/mm Btu gas and 4¢/kwh electric at the point of use.

Based on the high benefit to ERDA cost ratio of these candidates, we recommend that the air conditioner heat recovery system and the furnace/water heater demonstration program be implemented immediately and that the commercial range heat recovery system demonstration program be implemented now if funding permits or later if present funding is not available.

2.0 INITIAL SCREENING OF CANDIDATES

2.1 INTRODUCTION

2.1.1 Purpose and Philosophy

Appliances in residential and commercial buildings consume significant amounts of energy. Together with heating and cooling equipment, they account for 22 quads of primary energy consumption in 1970 representing 31% of the total 70 quads consumed in the U.S. Except for heat delivered for space heating, all of this energy flows through the house and is exhausted throughout the year as waste heat to the outside. Figure 2.1 following shows the energy flow pattern of a typical residence based on data given in Appendix A. By combining appliance functions, certain of these waste streams can be reused by another appliance, reducing the energy consumption.

In a single family home alone, these waste streams of energy are substantial, amounting to about 60% of the delivered energy into single family homes. Valued at the source of the energy (at the electric power plant), this amounts to nearly 6 quads of waste energy flow, or about 8.5% of the total national energy consumption. If only a quarter of this energy were recovered and reused, this could save about 1.5 quads of primary energy per year, or at the consumer level, a 5.8 billion dollar savings in energy cost per year.

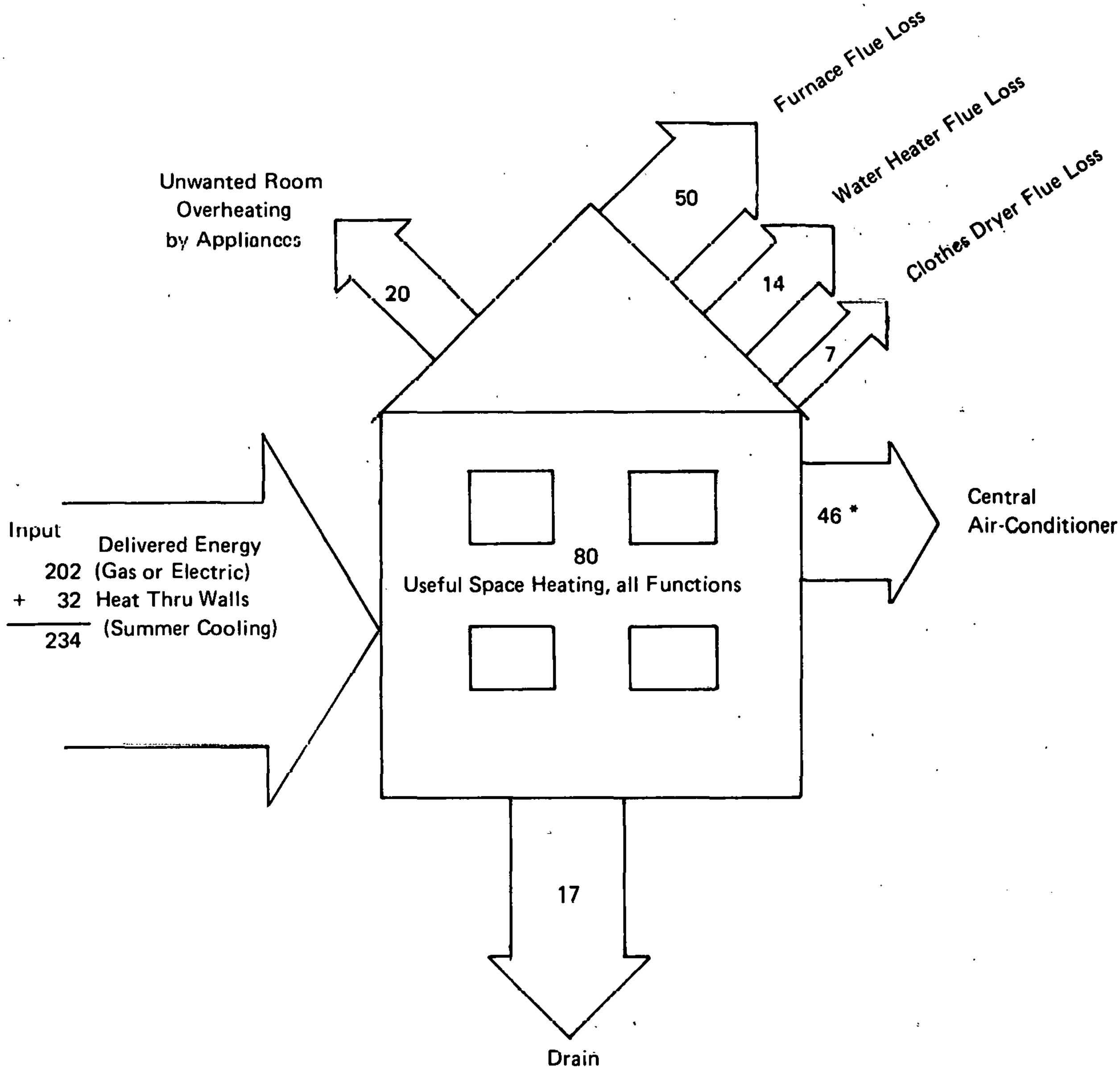
The purpose of Phase I of the Integrated Appliance Program is to identify the most promising combinations for saving the nation's energy and to prepare a plan to accelerate the commercialization of the product.

Through a screening process considering hundreds of possible combinations, those offering the greatest energy savings would be identified. Discussions with potential manufacturers for the integrated appliance (the Industrial Steering Committee Members) would then focus on the likelihood of commercializing these candidates and the potential benefit of an Energy Research and Development Administration (ERDA)-sponsored program to accelerate development, manufacturing, and marketing of energy-saving integrated appliances.

The discussions with manufacturers and the potential for energy savings would be used in evaluating the likely energy-savings benefit of ERDA-sponsorship of the development and demonstration of candidate integrated appliances.

2.1.2 Screening of Candidates

Given the dozen major residential and commercial appliances, there are hundreds of conceivable combinations which might be considered for energy conservation; and for each of these combinations, there are several



*Of the 46 x 10^6 waste heat energy, only 13.9 x 10^6 is electric or gas waste heat; the remaining energy is heat from the living space.

Source: See Summary of Appendix A.

FIGURE 2.1 TYPICAL ENERGY FLOW INTO & OUT OF A SINGLE-FAMILY HOME (UNITS IN 10^6 Btu/Year POINT OF USE ENERGY)

alternate configurations or arrangements. By means of the procedures outlined in this chapter, we selected six promising candidates for detailed analysis. They are:

- Air Conditioner/Water Heater
- Furnace/Water Heater
- Commercial Range/Water Heater
- Refrigerator/Water Heater
- Residential Drain Water Recovery*
- Commercial Drain Water Recovery

Although the merit of these candidates in terms of energy savings, market acceptability, etc., may seem obvious in retrospect, it was necessary to consider hundreds of other possibilities in order to avoid omitting equal or superior options. The purpose of the screening methodology was to compare systematically these hundreds of options on the basis of certain explicit criteria and to select a promising integrated appliance for further development and potential market introduction. The methodology established a path to the selection of integrated appliances with greatest potential, and also clearly identified those candidates that, for specific reasons, offer less promise.

The screening process is based essentially on the potential energy savings of the integrated appliance compared to the energy consumption of the appliances used separately. Several levels of screening were carried out, each intended to predict the potential energy savings of the candidate more accurately than the previous one.

The fundamental premise of the screening methodology stems from the objective of the entire program: to save energy in the U.S. by combining two or more appliance functions into one appliance. The potential for energy savings was the criterion for the winnowing-down process for hundreds of possible combinations of appliances for both residential and commercial uses. The comparison and screening process was guided by two primary assumptions:

- Upper limit to energy savings: Most of the combined appliances considered are such that waste energy from Appliance A is recovered to operate Appliance B. The energy savings cannot be greater than is currently "wasted" by A, nor can the savings exceed the energy usage of B.

- Waste heat rejection: During the heating season,** waste heat from individual appliances will contribute in part to heating the living space. If eliminated or reduced by an integration scheme, part of the potential energy savings must be replaced by an equivalent amount of space heating.

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*Drain water recovery refers to the use of waste drain water for preheating domestic water which is considered an integrated appliance.

**Defined in Appendix A.

2.2 INITIAL SCREENING OF CANDIDATES

Methodology

The starting point of the program was an investigation into the present level of energy consumption by appliances in the United States. First, ten prototypical buildings were identified in which appliances were used (mobile homes, apartments, hospitals, etc.). The study was limited to ten major appliances with greatest energy use, along with space heating and cooling equipment. For each appliance and building type, the annual unit energy consumption was estimated on the basis of:

- published data,
- manufacturer interviews, and
- assumed usage patterns.

The in-place 1970 population of each appliance in each building sector was estimated, and an appliance energy use inventory, believed to be the first comprehensive survey of its kind, was developed. A brief summary is given in Table 2.1. The complete residential and commercial appliance inventory is given in Appendix A.

As shown in Figure 2.2, the screening methodology begins with estimating the waste energy of the various appliances in the prototypical buildings in order to identify the maximum possible energy available for use by another function in the building. Waste energy available for another appliance function had to be precisely defined, as well as side benefits of the energy not used directly for specific appliance function. For instance, part of the heat given off by water heaters, refrigerators, ranges, and ovens to the living space during the winter is actually useful space heating, so long as the existing space heating system can adequately utilize this heat. Based on the analysis of thermal flow (Appendix B) in the house, we estimate that 80% of the waste heat given off inside the house by an appliance usefully contributes to space heating.

The first screening step was done by computer and consisted of matching of all possible waste energy-providing appliances with waste energy-accepting appliances, and estimating the resulting energy-savings potential. Two hundred residential combinations and 142 combinations in the commercial sector, or a total of 342 combinations, were examined.

In addition to the 342 heat recuperation concepts (waste heat utilization) considered, seven additional combinations of appliances were examined. These combinations rely on a single energy source to perform two appliance functions; the seven candidates are:

- Furnace/Water Heater
- Furnace/Range
- Furnace/Dryer

TABLE 2.1

SUMMARY OF APPLIANCE ENERGY USE
BY SECTOR (1970)
(All at the Point of Use)

Appliance Function	Residential (4 subsectors) 10^{14} Btu/year)	Selected Commercial (6 subsectors) 10^{14} Btu/year)
Hot water system (baths, showers, clothes washer, dishwasher)	15.2	2.3
Range/Oven	5.3	2.8
Refrigerator and Freezer	4.0	.6
Clothes Dryer	2.6	.8
Television	1.0	--
Lighting	1.4	1.9
Room Air Conditioner	1.1	--
Subtotal	30.6	8.4
Central Air Conditioner	.9	1.7
Space Heating	73.2	15.2
TOTAL	104.7	25.3

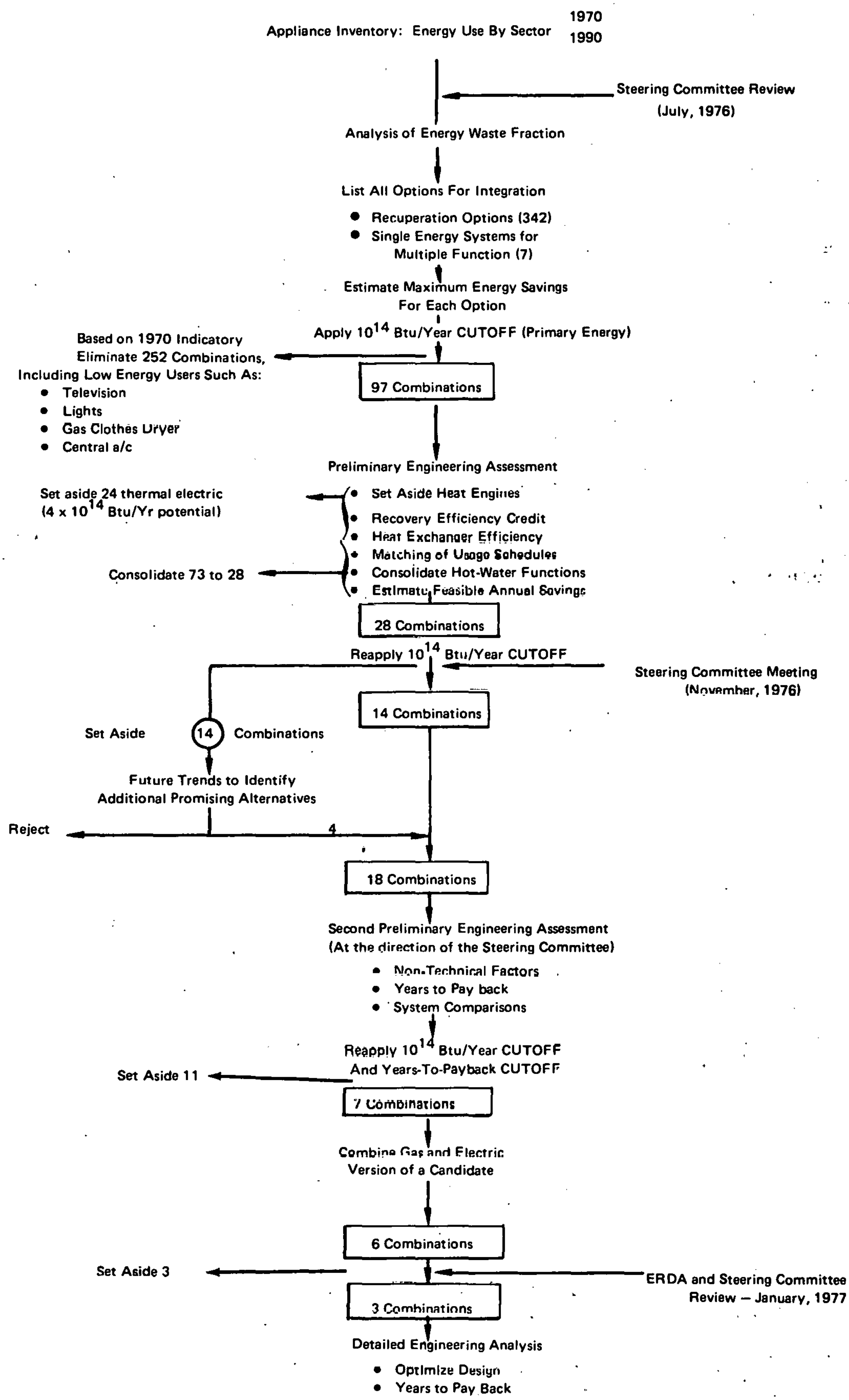

FIGURE 2.2 OVERVIEW OF SCREENING METHODOLOGY

- Range/Water Heater
- Range/Dryer
- Water Heater/Dryer
- Air Conditioner/Refrigerator

A single burner, refrigeration unit, or pilot acting as the energy converter for a number of appliances would replace the individual energy converters presently found in the various units. The possible energy savings from these devices are due to:

a) elimination of extra pilot lights,
b) reduction in standby energy losses, and
c) improvement in efficiency of the energy converter.

Candidates with a minimum potential of 10^{14} Btu/year of primary energy were chosen for further analysis. It was felt that a level of 10^{14} Btu/year potential savings, equivalent to saving about $330 million per year of electric and gas power*, could justify the anticipated public and private investment of money to develop a new product. Application of this 10^{14} Btu/year criterion eliminated all but 97 of the 349 combinations.

Further screening based on engineering feasibility was performed. The feasibility analysis included considerations such as matching recuperative energy supply and demand schedules and temperature requirement. Recuperative systems were made more realistic by eliminating that portion of the waste heat available at a temperature less than that needed by the heat user. Candidates requiring thermal-to-electric energy conversion were set aside as being not feasible. Candidates based on heat recuperation for specific hot water usages (such as dishwashing, clothes washing, showers, etc.) were consolidated into single candidates, and their energy savings were added. Practical heat exchanger effectiveness was introduced. After all of these refinements had been completed, the list was again subjected to the 10^{14} Btu/year cutoff, leaving only 28 combinations.

Fourteen of these 28 candidates were set aside for a number of reasons discussed in this chapter. Most of these candidates, after an engineering analysis, offered less energy savings than the final list of 14 candidates which satisfied the 10^{14} Btu/year criterion. This left 14 candidates for further analysis.

A projection of future trends for appliances and heating and cooling equipment (in 1990) was made in order to recognize other candidates whose importance will probably change in time as a result of:

- increased product saturation level,
- energy conservation measures,
- changing fuel availability and prices, and
- population shifts to the South and West

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*10^{14} Btu of primary energy evaluated at an even mix of gas primary ($3.50/mm Btu) and electric primary ($3.01/mm Btu) is $330 million.

Four additional candidates, making a list of 18 were identified by this future trend analysis.

A second engineering analysis was performed on these candidates based on their 1990 potential for energy savings and consisted of the following tasks:

a) System arrangements were developed and costed for each of the 18 candidates. The time to payback the added first cost by energy cost savings was estimated using expected 1985 energy prices (shown in Table 2.2 following).

b) Each candidate was screened for unacceptable requirements such as unavailability of materials; non-standard manufacturing techniques; failure to meet plumbing codes; excessive size, noise, or air pollution; and safety problems.

Eleven of the top 18 candidates which underwent the second engineering assessment were set aside primarily because they offered little economic benefit (less than 3.6 years for residential and 5 years for commercial applications). This left seven candidates offering large energy savings and acceptable payback periods. One candidate (residential drain recovery system) counted as both a gas and an electric configuration and was combined into a single candidate. So, the final list based on the initial screening was narrowed to six.

The final candidates were compared on the basis of likely consumer acceptance, potential cumulative 1980-1990 national energy savings, and likely affect of government support for development and product demonstration. Three of the candidates; namely:

- Air Conditioner/Water Heater
- Furnace/Water Heater
- Commercial Range/Water Heater

were selected (see Chapter 3) for consideration in a Phase II ERDA-sponsored development and demonstration program.

2.3 SUMMARY OF CRITERIA USED FOR SCREENING

2.3.1 Nationwide Energy-Savings Criterion

The primary criterion for accepting or rejecting a candidate was the level of possible nationwide energy savings which it is projected to offer. A level of 10^{14} Btu/year of primary energy was used as the energy

TABLE 2.2

PROJECTED 1985 ENERGY PRICES

Prices Used in This Study			Source Values		
Type	1975 Price	1985 Price (in 1975 Dollars)*	ERDA Low Forecast	ERDA High Forecast	ADL Medium Forecast
Electricity	3.21 ¢/kwh	3.43 ¢/kwh**	3.09	3.76	3.41
Gas	1.69 $/mm Btu	3.50 $/mm Btu	2.78	3.51	3.89
Oil (#2)	2.80 $/mm Btu	3.15 $/mm Btu	2.89	3.41	--

*1975 Dollars means that the cost shown here is exclusive of expected inflation. Therefore, added first costs of product will be on a 1975 dollar basis, i.e., without inflation.

**Equivalent to $3.01/mm Btu of primary energy based on 11,366 Btu/kwh.

Sources:

ERDA Forecast from: ERDA Working Documents, 1976

ADL Forecast from: ADL Working Document, Baseline ERDA Projections, August, 1976, as part of "New Technology Assessment Study."

cutoff criterion. Any candidate integrated appliance with potential energy savings of less than 10^{14} Btu/year after total replacement of existing separate appliances would be set aside. The approach taken was to apply the energy criterion repeatedly, after more detailed analysis has been performed on the candidates. The first application was applied to all 349 possible candidates (pairs of appliances) based on maximum energy savings, and the second application of the cutoff was after a preliminary engineering assessment of the surviving 97 candidates. A third application of this criterion was after the field had been narrowed to 28 candidates. Further screening was based on the 10^{14} Btu per year criteria and economic and consumer acceptance considerations.

The 10^{14} Btu/year criterion reflects the following considerations:

- 10^{14} Btu/year represents a little over 1% of the annual primary energy consumption (70×10^{15} Btu) by residential appliances (space heating not included). This makes it a practical lower limit for significance.

- It was felt that the energy cost savings equivalent to 10^{14} Btu/year, namely $330 million per year*, was consistent with the desire to have a large enough energy cost savings that the private and public investment requires to develop and commercialize the produce is a fraction of the potential cost savings.

- A reasonable number of integrated appliance (97 out of 349) candidates exceeded the 10^{14} Btu/year cutoff. A 10^{15} Btu/year criterion would have left no candidates, and 10^{13} Btu/year would have left an unmanageable number of candidates still to be considered.

2.3.2 Cost-Payback Criterion

A cost-payback criterion was used in the screening of the top 15 candidates. This is simply the number of years of energy savings required before the consumer recovers the added first costs of the new appliances. This was set at no more than 3.6** years to payback added costs of a residential appliance and 5** years for a commercial

*Based on an even mix of gas ($3.50/mm Btu) and electric ($3.01/mm Btu) in 1985.

**3.6 years to payback is equivalent to a 25% rate of return for a homeowner, while 5 years corresponds to a 15% rate of return for a commercial investor.

appliance. Recent industry testimony at the Federal Energy Administration hearings on appliance efficiency targets indicates that a practical payback criterion is about 3 to 4 years at the consumer level. Energy prices for the year 1985 were used in computing years to payback.

For screening of the top six candidates, a refinement of the years to payback criteria was needed and a relationship between years to payback and likely consumer acceptance was developed as discussed below.

For new or replacement situations where the consumer is purchasing an appliance, the fraction of purchasers who would accept an integrated appliance in its place was characterized by the expression

$$\text{New, replacement acceptance fraction} = \alpha + \frac{\beta}{\text{YTPB}}$$

where YTPB is years to payback and α and β are constants as follows:

Sector	Acceptance Parameters α	β	Restrictions
Residential	.05	.60	for YTPB >1
	.65	0	for YTPB <1
Commercial	.00	1.0	for 5> YTPB >2
	.35	.30	for 1< YTPB <2
	.70	0	for YTPB <1

For retrofit application, the number of purchasers of the retrofit device was thought to increase with the number of years the product is on the market. The fraction of the retrofittable population (retrofit for central air conditioners is possible for the 13 million residences with central A/C) which will purchase the device is:

$$\text{Retrofitting fraction of population} = \frac{\gamma}{10}^{*} \text{ (years the product has been on the market)}$$

where

$$\gamma = \alpha_{Ret} + \frac{\beta_{Ret}}{\text{YTPB}} .$$

---

*Therefore, the average fraction retrofitting over 10 years is

$$\frac{1}{10} \sum_{i=1}^{10} \left(\frac{\gamma}{10} \right) i = .29\ \gamma .$$

The following values of $\alpha_{Ret} + \beta_{Ret}$ were used as guidelines in the final stages of the detailed analysis (Chapters 4, 5, and 6):

Sector	Acceptance Parameters α_{Ret}	β_{Ret}	Restrictions
Residential	.0	.50	for YTPB >1
	.50	.0	for YTPB <1
Commercial	.05	.35	for YTPB >1
	.30	.20	for 1> YTPB >0

2.3.3 Acceptability Criteria

Throughout the screening process, candidates were considered for their acceptability in the residential and commercial markets. This broad category of criteria includes, but is not limited to, the following limitations:

- Marketability;
- Violation of building codes;
- Excessive appliance size, noise, or pollution;
- Climatic sensitivity or limitations; and
- Requirements for unconventional manufacturing methods.

Primary among these considerations were marketability, building codes, and ease of manufacture.

2.3.4 Appropriateness for ERDA Support

Another criterion is what is called an appropriateness criterion. This criterion separates those developments which would normally be undertaken by the private sector without any government support from those which are a higher risk and lower rate of return which would benefit from ERDA support. This criterion states that those projects which are likely to be undertaken by the private sector should not be considered for ERDA support unless use of the improved appliance is accelerated. Between the zone of unnacceptable risk projects and those that will naturally be undertaken by the private sector are candidates which will benefit from ERDA support. In our screening methodology, we consider which candidates are likely to be developed by the private sector without ERDA support and set them aside.

2.3.5 ERDA Investment Efficiency

ERDA also requires an evaluation of the cumulative discounted energy savings resulting from ERDA support of an energy-saving product. This is reflected in the "efficiency of total investment" (ETI) criterion as follows:

$$ETI = \frac{\sum_{t=1}^{N} \frac{\$E_i}{(1 + r)^{t - 1}}}{\sum_{t=1}^{N} \frac{\$I_i}{(1 + r)^{t - 1}}}$$

where

E_i is the annual energy cost savings in year 1,

r is the discount rate of invested money, and

I_i is the investment in year 1.

For an investment in the first year of I, the criterion simplifies to:

$$ETI = \frac{\sum_{t=1}^{N} \frac{\$E_i}{(1 + r)^{t - 1}}}{I}$$

A simplified version (r = 0) of this criteria was used in the final analysis of the top six candidates. The top candidates had very high ETI's based on the Arthur D. Little, Inc., estimates of the ERDA investment required to accelerate commercialization of the top concepts.

2.4 SCREENING - MAXIMUM POTENTIAL ENERGY SAVINGS

The amount of energy which can be saved by integration of appliances is limited to that portion of the existing energy consumption that is wasted. This is an assumption based on the premise that use habits, i.e., amounts of hot water used, food cooked, etc., are fixed and could not be changed with the introduction of any new appliances. The energy that goes into useful functions (heating food, water, etc.) and its waste that contributes to space heating during the heating season is not available for energy savings through integration of appliances. An analysis of each appliance in the residential and commercial sector was made in order to identify the following components of annual energy consumption:

- Energy contributing to the useful function of the appliance.
- Energy given off to the living space, some of which is useful during the heating season.

- Energy lost to the exterior through drains or vents.

During the heating season, a fraction of the energy given off by the appliance to the living space contributes to useful space heating. Depending on the location of the appliance and the level of air mixing in the building, the "jacket heat" from the appliance will either be well distributed and thus reduce the heating load, or be concentrated in a non-thermostatted room and cause "overheating." Essentially, this heat contributes usefully to the heating of the living space only if its contribution is felt at the thermostat. This analysis (see Appendix B) showed that in most homes, about 80% of this heat contributed usefully to the space heating requirements during the heating season by lowering the demands seen at the thermostat.

The heating season in the U.S. is about 230 days, representing the cumulative, population-weighted total of days in the U.S. each year with the temperature below 65°F. Therefore, the fraction, β, of summer operation is:

$$\beta = \frac{135}{365} = 0.37.$$

Waste heat (E_{waste}) is:

$$E_{stack} + E_{drain} + E_{interior} \; [\beta + (1 - \beta) \; (1 - \alpha)]$$

where

α is the waste heat usefully contributed to heating the living space during the heating season (80%)

and

E_{stack}, E_{drain}, and $E_{interior}$ represent the heat lost through stack, drain, and interior, respectively.

Table 2.3 presents the results of the appliance energy use and waste energy analysis. (Appendix A contains the detailed appliance inventory and sources.)

The energy waste fraction for each of the appliances in each of the various building sectors was matched to the heat demand of each of the other appliances in order to estimate the maximum conceivable energy savings from integrating each pair of appliances. A total of 342 combinations* of appliances were analyzed in this fashion.

*Of 200 possible combined appliances for residential use, 110 are possible for a gas-heated house (six gas-fired products combined with five electric-fired ones) and 90 are applicable to the electric-heated house. For commercial use, 142 possible combinations were identified.

TABLE 2.3

SUMMARY OF APPLIANCE ENERGY USE
AND WASTE ENERGY IN SINGLE FAMILY HOMES (1970)
(All at the Point of Use)

Appliance Function	Single Family Homes* 10^{14} Btu/year	Waste Energy* 10^{14} Btu/year
Hot water system (baths, showers, clothes washer, dishwasher)	14.3	12.6
Range/Oven	4.3	2.2
Refrigerator	3.4	1.6
Clothes Dryer	1.3	1.2
Television	.9	.4
Lighting	1.2	.6
Room Air Conditioner	.7	2.3
Subtotal	26.1	20.9
Central Air Conditioner	.7	2.3
Space Heating	52.9	21.6
TOTAL	79.7	44.8

*Based on point of use energy in Table A.4 of Appendix A.1.

Table 2.4 presents the results of matching of waste energy providing appliances with energy-receiving appliances. Possible nationwide energy savings that might be achieved through each combination are given in descending order, with the cut off point for promising candidates indicated. Note that the cut off is slightly below the 10^{14} primary energy level between those candidates separated by more than a few percent difference.

Electric-heated residences were assumed to have all electric appliances, and gas-heated homes were assumed to have both gas and electric appliances in proportion to market saturation indices. The appliance combinations and energy-savings potential for these two cases are listed separately.

In Table 2.5 are the seven additional combinations of appliances that could offer energy savings through use of a single energy source for two appliance functions. The single energy source can be viewed as a "parallel" use of energy in an integration scheme, whereas the 342 other combinations were "series" configurations.

Table 2.6 gives the ten commercial candidates found to approach the 10^{14} Btu/year cut off. The commercial opportunities for integrated appliances are substantially fewer than for residential applications for the following reasons:

- Nationally there is considerably less waste heat available from commercial buildings (about 40% of the amount in residential buildings).

- The number of commercial establishments with a waste heat user compatible with the waste heat source is far fewer than in the residential sector. For instance, waste A/C heat can provide water heating, but few commercial buildings with large waste A/C heat (such as office buildings) have comparable water heating needs. With few exceptions, the integrated appliance opportunities occur in commercial kitchens where the largest variety of appliances, both waste heat source and user type, are found.

The appliance inventory partitioned the hot water usage into its three primary functions: dishwashing, clothes washing, and baths and showers. Any integrated appliance involving one of these functions would cost the same and recover more energy if it involved all three functions; for example, heat recuperation could be coupled to either the water heater or to a common hot water drain. For this reason, the separate integrated appliances were combined where they involved water heating. This consolidation reduced by 45 the number of separate combinations that have to be considered to 52.

TABLE 2.4

POTENTIAL ENERGY SAVINGS BY MATCHING ELECTRIC APPLIANCES*
(Btu/Year)

HEAT USER	HEAT SOURCE	BTU/YR	
REFRIG-FREEZER	HOT WATER-BATH	1.447E 14	
LIGHTS	HOT WATER-BATH	1.097E 14	
HOT WATER-BATH	REFRIG-FREEZER	1.043E 14	
LIGHTS	REFRIG-FREEZER	8.741E 13	
TV	HOT WATER-BATH	8.526E 13	
RANGE-OVEN	HOT WATER-BATH	8.340E 13	
HEATING	HOT WATER-BATH	7.938E 13	
TV	REFRIG-FREEZER	7.708E 13	
RANGE-OVEN	REFRIG-FREEZER	7.630E 13	
HW/DISH WASHER	REFRIG-FREEZER	6.240E 13	
HW/DISH WASHER	HOT WATER-BATH	5.916E 13	
LIGHTS	HW/DISH WASHER	5.454E 13	
TV	HW/DISH WASHER	5.454E 13	
REFRIG-FREEZER	HW/DISH WASHER	5.454E 13	
RANGE-OVEN	HW/DISH WASHER	5.454E 13	
HOT WATER-BATH	HW/DISH WASHER	5.170E 13	
ROOM A/C	REFRIG-FREEZER	4.961E 13	
REFRIG-FREEZER	ROOM A/C	4.961E 13	
LIGHTS	TV	4.677E 13	
REFRIG-FREEZER	TV	4.677E 13	
REFRIG-FREEZER	LIGHTS	4.630E 13	
RANGE-OVEN	TV	4.598E 13	
LIGHTS	RANGE-OVEN	4.574E 13	
TV	RANGE-OVEN	4.574E 13	
REFRIG-FREEZER	RANGE-OVEN	4.574E 13	
HW/CLOTHS WASHR	REFRIG-FREEZER	4.437E 13	
HOT WATER-BATH	TV	4.433E 13	
HOT WATER-BATH	LIGHTS	4.389E 13	
HOT WATER-BATH	RANGE-OVEN	4.337E 13	
LIGHTS	HEATING	4.317E 13	
TV	HEATING	4.317E 13	
REFRIG-FREEZER	HEATING	4.317E 13	
RANGE-OVEN	HEATING	4.317E 13	
HW/CLOTHS WASHR	HOT WATER-BATH	4.206E 13	
HEATING	REFRIG-FREEZER	4.111E 13	
HOT WATER-BATH	HEATING	4.092E 13	
LIGHTS	HW/CLOTHS WASHR	3.984E 13	
TV	HW/CLOTHS WASHR	3.984E 13	
REFRIG-FREEZER	HW/CLOTHS WASHR	3.984E 13	
RANGE-OVEN	HW/CLOTHS WASHR	3.984E 13	
HW/DISH WASHER	HW/CLOTHS WASHR	3.984E 13	
HW/CLOTHS WASHR	HW/DISH WASHER	3.878E 13	
HOT WATER-BATH	HW/CLOTHS WASHR	3.777E 13	
HW/DISH WASHER	HEATING	3.744E 13	
TV	LIGHTS	3.597E 13	
HW/DISH WASHER	TV	3.575E 13	
HW/DISH WASHER	RANGE-OVEN	3.552E 13	
RANGE-OVEN	LIGHTS	3.519E 13	10^{14} Btu/year Primary Energy
HEATING	HW/DISH WASHER	2.958E 13	
HW/CLOTHS WASHR	HEATING	2.662E 13	
HW/CLOTHS WASHR	TV	2.542E 13	
HW/CLOTHS WASHR	RANGE-OVEN	2.525E 13	
HW/DISH WASHER	LIGHTS	2.496E 13	
HOT WATER-BATH	ROOM A/C	2.365E 13	
ROOM A/C	HOT WATER-BATH	2.365E 13	
HEATING	HW/CLOTHS WASHR	2.210E 13	Cut off
HW/CLOTHS WASHR	LIGHTS	1.775E 13	
LIGHTS	ROOM A/C	1.676E 13	
ROOM A/C	LIGHTS	1.676E 13	
CENTRAL A/C	REFRIG-FREEZER	1.507E 13	
REFRIG-FREEZER	CENTRAL A/C	1.507E 13	
ROOM A/C	TV	1.302E 13	
TV	ROOM A/C	1.302E 13	
ROOM A/C	RANGE-OVEN	1.274E 13	
RANGE-OVEN	ROOM A/C	1.274E 13	
HEATING	TV	1.079E 13	
HEATING	RANGE-OVEN	1.056E 13	
HW/DISH WASHER	ROOM A/C	9.036E 12	
ROOM A/C	HW/DISH WASHER	9.036E 12	
ROOM A/C	CENTRAL A/C	8.053E 12	
CENTRAL A/C	ROOM A/C	8.053E 12	
CENTRAL A/C	HOT WATER-BATH	7.186E 12	
HOT WATER-BATH	CENTRAL A/C	7.185E 12	
CLOTHES DRYER	TV	7.121E 12	
CLOTHES DRYER	HW/DISH WASHER	7.121E 12	
CLOTHES DRYER	REFRIG-FREEZER	7.121E 12	
CLOTHES DRYER	RANGE-OVEN	7.121E 12	
CLOTHES DRYER	HOT WATER-BATH	6.750E 12	
LIGHTS	CLOTHES DRYER	6.693E 12	
TV	CLOTHES DRYER	6.693E 12	
HW/DISH WASHER	CLOTHES DRYER	6.693E 12	
REFRIG-FREEZER	CLOTHES DRYER	6.693E 12	
RANGE-OVEN	CLOTHES DRYER	6.693E 12	
HW/CLOTHS WASHR	ROOM A/C	6.425E 12	
ROOM A/C	HW/CLOTHS WASHR	6.425E 12	
HOT WATER-BATH	CLOTHES DRYER	6.345E 12	
LIGHTS	CENTRAL A/C	5.093E 12	
CENTRAL A/C	LIGHTS	5.093E 12	
CLOTHES DRYER	HW/CLOTHS WASHR	5.063E 12	
HW/CLOTHS WASHR	CLOTHES DRYER	4.759E 12	
CLOTHES DRYER	HEATING	4.272E 12	
CENTRAL A/C	TV	3.957E 12	
TV	CENTRAL A/C	3.957E 12	
CENTRAL A/C	RANGE-OVEN	3.871E 12	
RANGE-OVEN	CENTRAL A/C	3.871E 12	
HEATING	CLOTHES DRYER	3.845E 12	
CLOTHES DRYER	LIGHTS	2.848E 12	
HW/DISH WASHER	CENTRAL A/C	2.746E 12	
CENTRAL A/C	HW/DISH WASHER	2.746E 12	
HW/CLOTHS WASHR	CENTRAL A/C	1.952E 12	
CENTRAL A/C	HW/CLOTHS WASHR	1.952E 12	
CLOTHES DRYER	ROOM A/C	1.031E 12	
ROOM A/C	CLOTHES DRYER	1.031E 12	
CLOTHES DRYER	CENTRAL A/C	3.133E 11	
CENTRAL A/C	CLOTHES DRYER	3.133E 11	

*Single family detached

TABLE 2.4

POTENTIAL ENERGY SAVINGS BY MATCHING GAS APPLIANCES*
(Btu/Year)

HEAT USER	HEAT SOURCE	BTU/YR
HOT WATER-BATH	HEATING	4.022E 14
RANGE-OVEN	HOT WATER-BATH	3.226E 14
HEATING	HOT WATER-BATH	3.217E 14
HW/DISH WASHER	HOT WATER-BATH	2.421E 14
RANGE-OVEN	HW/DISH WASHER	2.232E 14
HOT WATER-BATH	HW/DISH WASHER	2.116E 14
RANGE-OVEN	HEATING	2.042E 14
RANGE-OVEN	REFRIG-FREEZER	1.772E 14
HW/CLOTHS WASHR	HOT WATER-BATH	1.721E 14
HOT WATER-BATH	REFRIG-FREEZER	1.689E 14
HOT WATER-BATH	RANGE-OVEN	1.678E 14
HW/DISH WASHER	HW/CLOTHS WASHR	1.619E 14
RANGE-OVEN	HW/CLOTHS WASHR	1.619E 14
HW/CLOTHS WASHR	HW/DISH WASHER	1.587E 14
HOT WATER-BATH	HW/CLOTHS WASHR	1.535E 14
HW/DISH WASHER	HEATING	1.532E 14
HW/DISH WASHER	REFRIG-FREEZER	1.432E 14
HW/DISH WASHER	RANGE-OVEN	1.430E 14
HEATING	HW/DISH WASHER	1.210E 14
HW/CLOTHS WASHR	HEATING	1.089E 14
HW/CLOTHS WASHR	REFRIG-FREEZER	1.018E 14
HW/CLOTHS WASHR	RANGE-OVEN	1.016E 14
HEATING	HW/CLOTHS WASHR	8.932E 13
HOT WATER-BATH	ROOM A/C	6.940E 13
RANGE-OVEN	ROOM A/C	4.928E 13
HW/DISH WASHER	TV	4.677E 13
RANGE-OVEN	TV	4.677E 13
HW/DISH WASHER	LIGHTS	4.630E 13
RANGE-OVEN	LIGHTS	4.630E 13
HOT WATER-BATH	TV	4.433E 13
HOT WATER-BATH	LIGHTS	4.389E 13
HEATING	REFRIG-FREEZER	4.111E 13
HEATING	RANGE-OVEN	4.084E 13
HW/DISH WASHER	ROOM A/C	3.697E 13
HW/CLOTHS WASHR	TV	3.325E 13
HW/CLOTHS WASHR	LIGHTS	3.292E 13
HOT WATER-BATH	CENTRAL A/C	2.949E 13
HW/CLOTHS WASHR	ROOM A/C	2.629E 13
RANGE-OVEN	CENTRAL A/C	1.497E 13
HW/DISH WASHER	CENTRAL A/C	1.123E 13
HEATING	TV	1.079E 13
HW/CLOTHS WASHR	CENTRAL A/C	7.988E 12
CLOTHES DRYER	TV	5.918E 12
CLOTHES DRYER	HW/DISH WASHER	5.918E 12
CLOTHES DRYER	REFRIG-FREEZER	5.918E 12
CLOTHES DRYER	RANGE-OVEN	5.918E 12
CLOTHES DRYER	HOT WATER-BATH	5.611E 12
HW/DISH WASHER	CLOTHES DRYER	5.492E 12
RANGE-OVEN	CLOTHES DRYER	5.492E 12
HOT WATER-BATH	CLOTHES DRYER	5.207E 12
CLOTHES DRYER	HW/CLOTHS WASHR	4.208E 12
HW/CLOTHS WASHR	CLOTHES DRYER	3.905E 12
CLOTHES DRYER	HEATING	3.551E 12
HEATING	CLOTHES DRYER	3.125E 12
CLOTHES DRYER	LIGHTS	2.367E 12
CLOTHES DRYER	ROOM A/C	3.570E 11
CLOTHES DRYER	CENTRAL A/C	2.604E 11

10^{14} Btu/year Primary Energy

Cut off

*Single family detached

TABLE 2.5

SINGLE ENERGY SOURCE COMBINATIONS

1 Common gas-fired water heating/space heating unit:

Designed to provide instantaneous (minimal storage) heating of domestic water, eliminating standby losses.

2 Common gas-fired range/space heating unit:

Designed to reduce the number of pilot lights necessary and to increase the recovery efficiency of range and furnace.

3 Common gas-fired range/dishwasher:

Designed to use a common burner for the range and for heating water for the kitchen dishwasher.

4 Furnace/dryer combination:

Would be the same in principal as the furnace/range combination, eliminating one pilot.

5 Range/dryer:

Would be the same as the furnace/dryer combination, eliminating several pilots

6 Water heater/dryer:

Would utilize a common burner for both functions, eliminating one pilot.

7 Room air conditioner/refrigerator:

Would utilize a common condensing unit.

TABLE 2.6

COMMERCIAL INTEGRATED APPLIANCES

		Potential Energy Savings 10^{14} Btu/year
1	Water heater → heater	1.6
2	Water heater → range	.3
3	Water heater → space heater	1.0
4	Range → Water Heater	1.0
5	Range → space heating	.6
6	Central A/C → Water Heater	.4
7	Central A/C → Range	.35
8	Space Heater → Water Heater	.26
9	Space Heater → Range	.51
10	Refrigerator → Heating	1.6

2.5 SCREENING - CONSIDERING ENGINEERING FEASIBILITY

2.5.1 Recovery Efficiency Credit

The recuperation of 100 Btu of waste energy to provide heat directly to the appliance function of Appliance A will actually result in an energy-savings greater than 100 Btu, because of the inefficiency of Appliance A. Figure 2.3 demonstrates the fuel-saving credit given to recuperative heat recovery providing the desired appliance function. The benefit ratio of waste heat savings in terms of input energy savings is the recuperative efficiency divided by the recovery efficiency of the original appliance.

2.5.2 Heat Exchanger Effectiveness

Many of the combination integrated appliances require a heat exchanger to recover heat from the waste stream from an appliance. Two limitations on the amount of heat that can be recovered by this mechanism were considered and applied in the preliminary analysis of the promising candidates.

The first limitation comes from second law of thermodynamics considerations of the level of temperature of the waste stream available for recovery. If the waste stream is at a temperature, T_w, lower than the heat recipient final operating temperature, T_a, and the waste stream fluid capacity rate $\dot{m}(C_p)_w$ is larger than the acceptor capacity fluid rate $\dot{m}(C_p)_a$, the maximum fraction of the heat which can be recovered, $\dot{q}_r$, is limited to:

$$\dot{q}_r = \dot{m}(C_p)_a\ (T_w - T_0)$$

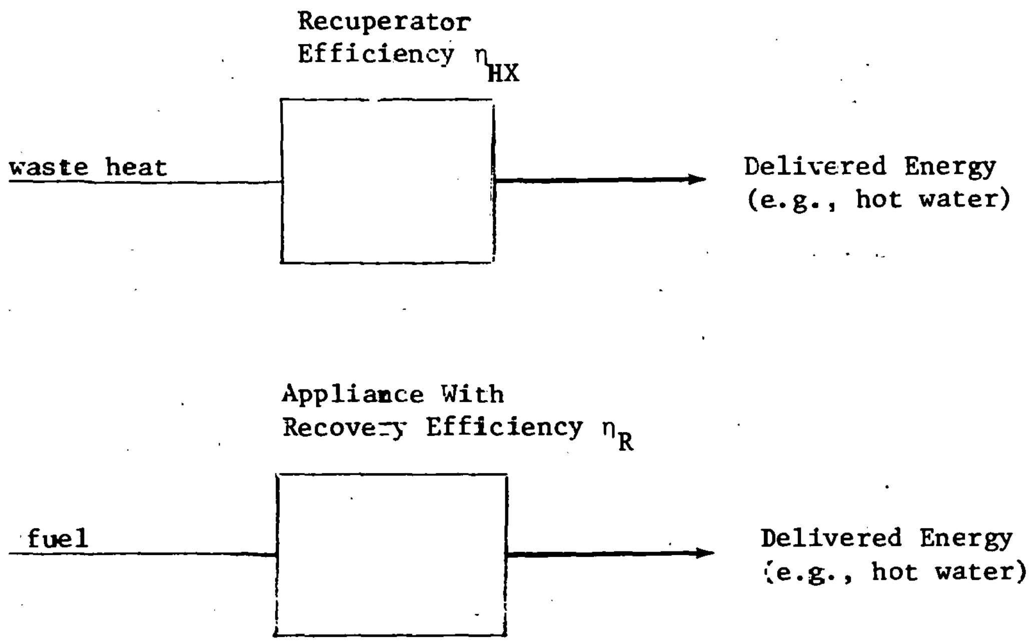
where

T_0 is room temperature, as illustrated in Figure 2.4,

whereas the maximum heat that can be accepted is

$$\dot{q}_a = \dot{m}(C_p)_a\ (T_a - T_0).$$

The second limitation arises from mismatching of the waste stream to the input stream. The maximum amount of heat recovery is limited by the stream with the minimum fluid capacity rate $\dot{m}(C_p)$. In a number of instances, the mass flow rate of either the waste heat recovery stream or the recipient stream is limited by external constraints. For instance, in heat recovery from a range through heat reclamation of the exhaust air stream, the amount of heat transfer possible is limited by the heat capacity and mass flow rate of the exhaust air.



Benefit of waste heat (in terms of fuel savings)

$$= \text{(Btu of waste heat)} \times \frac{\eta_{HX}}{\eta_R}$$

FIGURE 2.3 CREDIT DUE TO RECOVERY EFFICIENCY

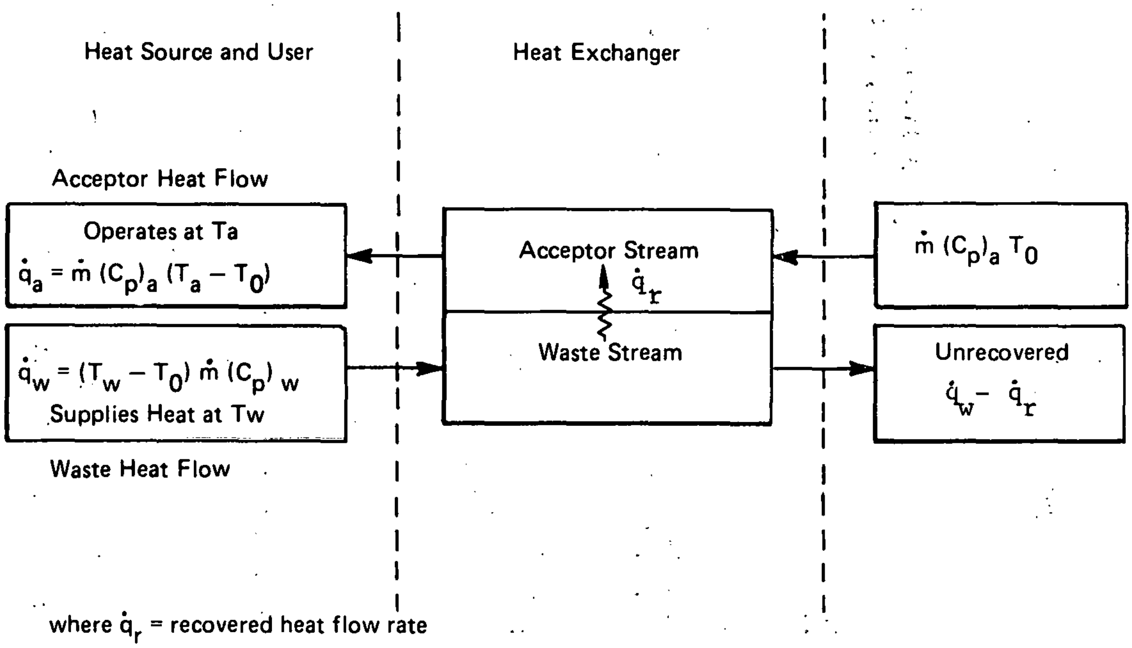


FIGURE 2.4 HEAT EXCHANGER EFFICIENCY

The heat exchanger capacity itself is limited by practical constraints on its size. With a large enough heat exchanger surface area, nearly all of the waste heat available from the mininum fluid capacity rate stream can be recovered. However, this is generally impractical, and depending on the flow rate configuration--whether cross flow or counter flow--and types of fluid used, the ability of the heat exchanger to transfer heat from one stream to another has some practical limit. This limitation is expressed as the effectiveness, or the ratio of the actual heat transfer to the maximum that could be transferred from the minimum fluid capacity rate stream. Effectiveness is symbolically η.

The manner in which these limitations were analyzed was based on standard heat transfer principles. For each appliance combination, the heat of incoming and outgoing streams for both acceptor stream and waste stream were calculated in terms of the change in temperature ($T_a - T_0$ for the acceptor stream, and $T_w - T_0$ for the waste stream), mass flow rate ($\dot{m}$), and heat capacity of the stream [$(C_p)_a$ for the acceptor and $(C_p)_w$ for the waste stream]. This is represented schematically in Figure 2.4.

Depending on the relative temperatures and the relative mass flow rate times heat capacity of the streams, four combinations of limitations are possible.

Thus, when $T_w < T_a$,

(1) and $\dot{m}(C_p)_w > \dot{m}(C_p)_a$

$$\boxed{\dot{q}_r = \eta\, \dot{q}_a \left(\frac{T_w - T_0}{T_a - T_0}\right)}$$

(2) and $\dot{m}(C_p)_w \leq \dot{m}(C_p)_a$

$$\boxed{\dot{q}_r = \eta\, \dot{m}(C_p)_w\, (T_w - T_0) = \eta \dot{q}_w}$$

when $T_a < T_w$

(3) and $\dot{m}(C_p)_w > \dot{m}(C_p)_a$

$$\eta_{max} = \frac{\dot{m}(C_p)_a\ (T_a - T_0)}{\dot{m}(C_p)_a\ (T_w - T_0)} = \frac{T_a - T_0}{T_w - T_0}$$

$$\boxed{\dot{q}_r = \dot{m}(C_p)_a \; (T_a - T_0) = \dot{q}_a}$$

Note that in this case:

$$\eta_{max} = \frac{T_a - T_0}{T_w - T_0}$$

(4) and $\dot{m}(C_p)_w < \dot{m}(C_p)_a$

$$\boxed{\dot{q}_r = \eta \; \dot{m}(C_p)_w \; (T_w - T_0) = \eta \; \dot{q}_w}$$

η is the design point heat exchanger efficiency for the various heat transfer media. These design points were selected from the performance curves of heat exchangers given in Appendix C and reflect a cost versus energy recovery tradeoff aimed at giving the highest energy for reasonable size (NTU) heat exchangers. The values used in the screening are:

	Air to Air	Water to Air	Water to Water	Refrigerant to Water	$T_a < T_w$ and $\dot{m}(C_p)_a < \dot{m}(C_p)_w$
η_{max}	.7	.7	.7	.8	$\frac{T_a - T_0}{T_w - T_0}$

2.5.3 Matching of Seasonal Use Schedules

In order to obtain a feasible energy savings, the amount of recoverable energy was estimated on the basis of a one-day storage limit. An example of the recoverable savings for an air conditioner/water heater is shown in the cross-hatched area of Figure 2.5. The annual heat rejection by the air conditioner may equal or exceed the annual heating requirement of the water heater; however, energy storage large enough to carry excess waste heat from the summer into the winter would be required. This seasonal type storage was felt to be beyond the scope of the integrated appliance program, belonging more to the Annual Cycle Energy Storage (ACES) program of ERDA.

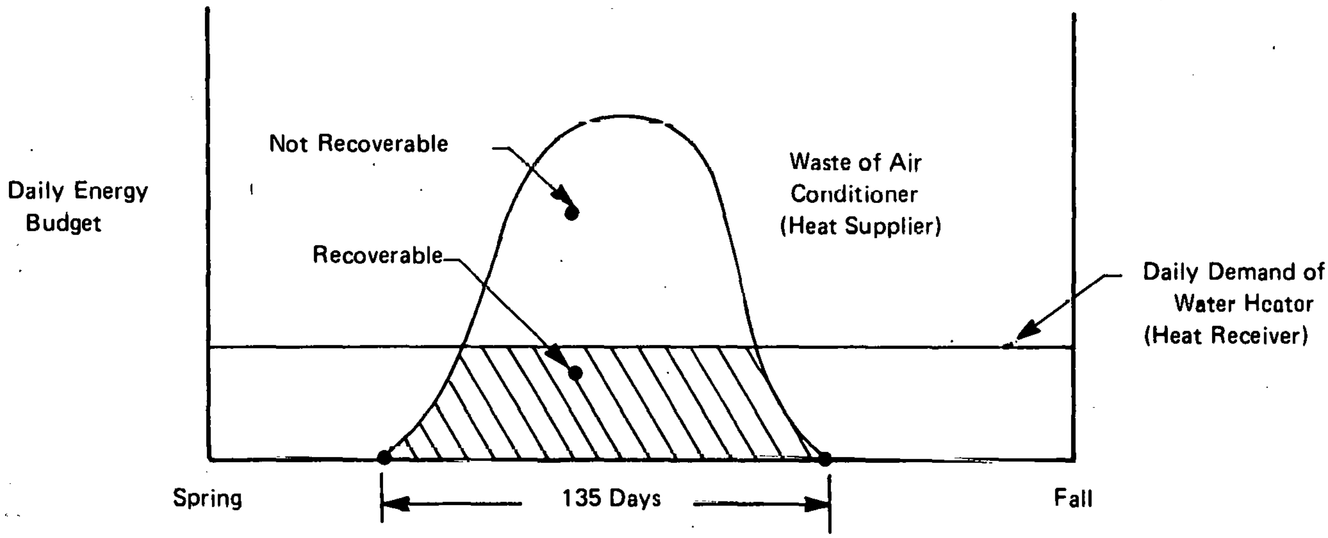


FIGURE 2.5 EXAMPLE OF MATCHING SEASONAL USAGE SCHEDULES FOR AN AIR CONDITIONER (WASTE HEAT SUPPLIER) AND A WATER HEATER (WASTE HEAT RECEIVER)

2.5.4 Residential Heat Engines

In a number of instances, waste heat is available at temperatures high enough to suggest thermal-to-electric conversion. This conversion would be accomplished by a number of thermodynamic cycles which in principle have maximum conversion efficiencies of between 10% and 20%, based on waste streams with temperatures in a range of 200°F to 400°F. While these conversion efficiencies are about 1/3 to 1/4 that of a direct heat-exchanger recuperative system, in 1985 (when an integrated appliance would be making significant market penetration) the value of the electric power delivered is projected to be between three and four times more valuable than the gas or oil used to produce the waste heat system. The major limitation in the consideration of thermal-to-electric conversion is the initial first cost, which is at present quite high. It is unlikely that the $1,000 to $1,500 per kilowatt installed cost for such a device will make it a practical near-term alternative for the residential sector.

As can be seen in Figure 2.6, the opportunities for thermal-to-electric conversion are more promising in the commercial sector where equipment duty cycles are longer. The duty cycle, or fraction of time (η_{duty}) which the equipment is on, determines the cost per kwh as:

$$\frac{\$}{\text{kwh}} = \frac{\text{Installed cost per kw}}{(8{,}760)\ (\eta_{duty})\ (\text{years to payback})^{*}}$$

Because of the speculative nature of the cost-to-savings ratio for these systems, the thermal-to-electric combinations of integrated appliances were judged to be outside of the scope of this program and probably more appropriate to a program focusing on longer range technologies. Removing the 24 heat engine (thermal-to-electric) concepts from the list reduces the number of candidates to the 28 shown in Table 2.7 .

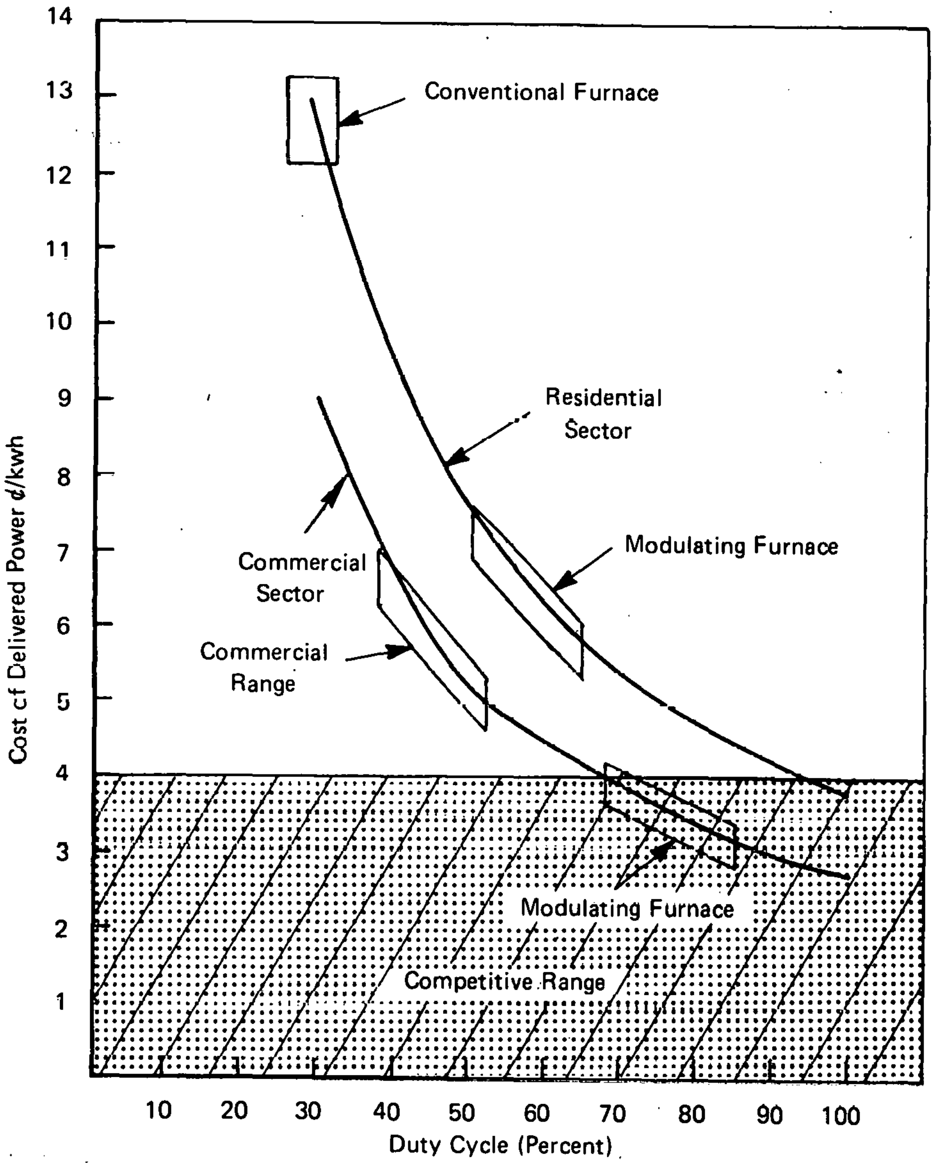
A preliminary assessment, taking into account:

Factors for Preliminary Assessment

1) Recovery efficiency credit
2) Heat exchanger effectiveness
3) Seasonal usage matching

was then performed on these 28, resulting in the estimated energy savings shown in Table 2.7.

*Section 2.3.2 gives acceptable years to payback, which is 3.6 years for the residental sector, and 5 years for the commercial sector.



$$\text{Basis of Curve \$/kwh} = \frac{\text{\$1200 per installed Kw}}{\text{8760 (duty cycle) (acceptable years to payback)}}$$

Where acceptable years to payback is 3.6 residential
5 Commercial

FIGURE 2.6 HEAT ENGINE FEASIBILITY

TABLE 2.7

TWENTY-EIGHT CANDIDATE INTEGRATED APPLIANCES
(Energy at the Point of Use)

Sector	Code	Option	Possible Savings 10^{14} Btu/yr	Savings After Preliminary Assessment 10^{14} Btu/yr	Top 14
Residential (gas)	G1	Heating → WH*	6.6	4.2+	•
	G2	WH → Range	3.3	0.7	
	G3	WH → Heating	7.2	4.3++	•
	G4	WH/WH** (Drain only)	6.0	3.4	•
	G5	Heating → Range	2.1	1.0	•
	G6	Refrig → Range	1.1	0.0	
	G7	Range → WH	1.7	1.7	•
	G8	Refrig → WH	1.1	1.5	•
	G9	Room A/C → WH	2.0	2.3	•
Residential (Electric)	E1	Refrig → WH	0.5	0.5	•
	E2	WH (Drain)→ Range	0.8	0.1	
	E3	Refrig → Room A/C	0.6	0.0	
	E4	WH/WH (Drain)	2.2	0.9	•
	E5	Dryer → WH	0.5	0.35	•
	E6	Dryer → Range	0.5	0.05	
	E7	Refrig → Dryer	0.5	0.15	
	E8	Range → WH	0.4	0.3	
	E9	Refrig → Heating	0.12	0.1	
	E10	Dryer → Heating	0.2	0.2	
	E11	WH → Heating	1.3	1.3	•
	E12	WH → Dryer	0.6	0.2	
	E13	Room A/C → WH	0.3	0.25	
Commercial	C1	Range → WH	1.0	1.0	•
	C2	WH → Heating	1.0	1.0	•
	C3	WH → Range	0.3	0.06	
	C4	Range → Heating	0.6	0.4	
	C5	WH/WH (same as C1)	1.6	0.9	•
	C6	Refrig → Heating	1.6	1.3	Already developed

*WH is water heating, including dishwashing, clothes washing, etc.

**WH/WH includes waste water heat recovery.

+Limited by maintaining flue gas temperatures above 240°F.

++Reflects the fact that only 30°F of the 50°F is actually recoverable for space heating.

The top 14 of this list were selected for further analysis with the exception of the commercial refrigerator/space heating combination, which was found to be well developed and commercially available (heating controls and appropriate hardware are presently available to handle waste heat from supermarket or restaurant refrigeration for the space heating system). The 13 marginally-promising candidates were set aside for the reasons given in Table 2.8.

2.6 SCREENING - CONSIDERING PROJECTED 1990 APPLIANCE INVENTORY

Likely shifts in consumer products over the next 15 years could significantly affect the efficacy of integrated appliances. The screening methodology has been based on a snapshot view of the appliance inventory in 1970, and many of the integrated appliance concepts will not impact the marketplace until the mid-1980's. At that time, the integrated appliance may be competing against improved individual appliances, which reduce their attractiveness. Also, shifts in the relative number of appliances may result in the kind of reordering of promising candidate integrated appliances shown in Figure 2.7. This ordering is expected to result from changes noted below.

a) Changes in relative saturation of appliances (e.g., increased use of clothes dryers, number of air conditioners).

b) Alterations in the use pattern or energy consumption of the equipment (e.g., efficiency improvements or cold-water washing).

c) Shifts in the types of energy used for appliances or heating and cooling equipment (e.g., less oil, more electricity).

Fifteen year projections for these factors, particularly for consumer products, are inherently difficult and as such should only be expected to provide a view to major trends with rather large margins of error.

Shifts in the relative appliance population will be governed by increases in the housing stock and shifts in the relative saturation of different appliances. Projections for this program were based on three studies, one by Arthur D. Little, Inc.[2.1], the other by Oak Ridge National Laboratories[2.2] and unpublished Arthur D. Little projections. These projections are summarized in Table 2.9.

Although some 2.8 million new gas or oil furnaces are expected to be sold each year, the total inventory is not expected to change very much. This is because about 1/2 of the sales are going to replacement systems and the other 1.4 million to new homes. But, this 1.4 million new home units is just offset by the annual removal of homes which are all assumed to have gas or oil heating since only 8% of existing housing stock is electrically heated.

TABLE 2.8

THIRTEEN MARGINALLY PROMISING INTEGRATED APPLIANCES

Sector	Option	Type of Fuel Used	Reasons Discarded	Estimated Max. 10^{14} Btu/year
Residential	Drains HRS for Range	Gas and Electric	Range requires 300-600° F	0.7
	Refrigerator HRS for Range	Gas	Same as above	0.0
	Range HRS for Hot Water	Electric	Stand alone too costly; combine with Refrigerator HRS?	1.4
	Room A/C for Hot Water	Electric	Inadequate national energy savings	0.25
	Refrigerator/Room A/C	Electric	Location and usage schedule	0.0
	Dryer HRS for Range	Electric	Same as above	0.05
	Refrigerator HRS for Dryer	Electric	Refrigerator waste would produce excessive drying times	0.015
	Refrigerator HRS for Space Heat	Electric	Would overheat kitchen	0.1
	Dryer HRS for Space Heating	Electric	Dryer waste is in latent heat	0.2
	Drain HRS for Dryer	Electric	Temperature too low	0.1
Commercial	Drain HRS for Range	Gas	Temperature too low	0.06
	Range HRS for Heating	Gas	Inadequate national energy savings	0.4
	Refrigerator HRS for Space Heat	Gas	Already developed	1.3

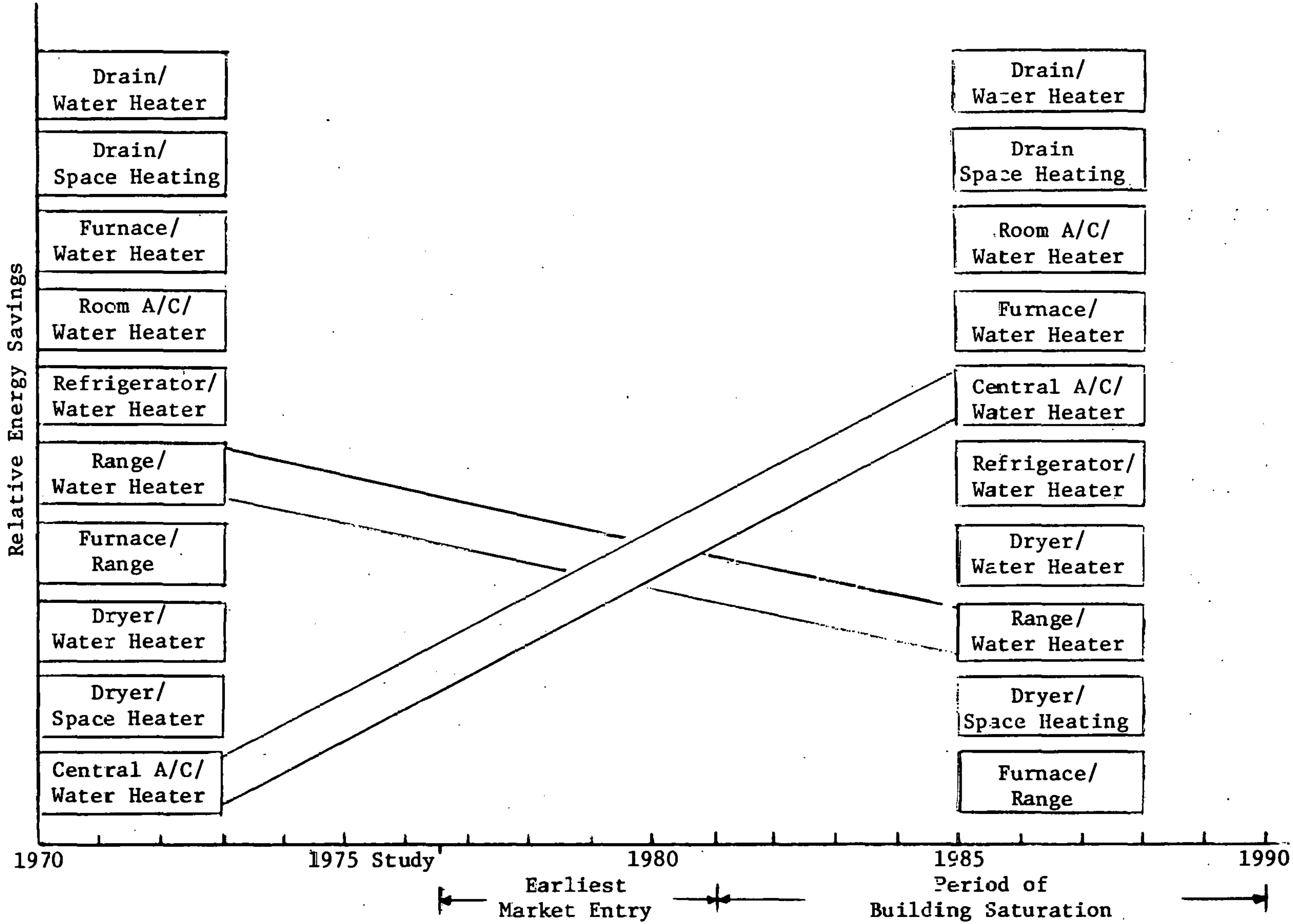


FIGURE 2.7 POSSIBLE EFFECTS OF FUTURE TRENDS

TABLE 2.9

PROJECTED SATURATION OF APPLIANCE MARKET

	Saturation of Appliances (%)			
	1970	1990		Value Adopted For 1990
	ADL-CEQ (1974)	ADL-CEQ (1974)	ORNL (1976)	
Housing stock Single family res.*	46.3	60	62	61
Space heating				
gas/oil	87	70	73	72
electric	8	22	22	22
other	5	8	5	6
Water heating				
gas/oil	65	61	60	60
electric	25	37	37	37
other	6	2	3	3
Refrigerator & freezer	120	160	160	160
Range				
gas	59	25	36	34
electric	41	74	64	66
Air Conditioner				
room	25	35	43	35
central	11	55	34	38
Clothes Dryer				
gas	17	12	--	12
electric	39	60	--	60

*Not including mobile homes

Nearly all of the gas-heated homes have gas water heaters, but only 27% of the oil-heated homes have oil-heated hot water, the remainder use electric water heaters. This accounts for the higher penetration of electric water heaters than electric space heating systems.

Growth in the refrigerator-freezer area is due to the increased number of stand-alone freezer population.

The decline in gas-fired ranges expected for the 1980 to 1990 time period results from the present buying trend of electric ranges and microwave units over gas ranges.

Based on recent Arthur D. Little, Inc., unpublished projections of air conditioner saturations, the saturation of central air conditioners is expected to rise. Nearly one half of all new homes built in the United States recently have central A/C, and sales of central A/C to older homes is substantial.

Clothes dryers are expected to rise from 56% saturation to 72%, and these are expected to come mostly as electric units. Recent sales history of clothes dryers confirms the small market share (20%) of gas clothes dryers.

Individual levels of appliance energy consumption are expected to change as a result of increased fuel prices and government pressures. Federal Energy Administration improvement targets will boost efficiency by about an average of 20% by 1980. From 1980 to 1990, a further increase in consumer product efficiency is expected. Estimates of product efficiency improvements are given in Table 2.10.

The predicted housing stock appliance saturations and individual unit energy consumption have been combined to produce the Projected 1990 Energy Consumption Levels shown in Table 2.11. Major energy consumption changes in electric water heating, electric ranges, central air conditioners, and electric clothes dryers are projected to occur, and these changes will affect the list of promising candidates.

Based on the projected 1990 appliance inventory, new figures for candidate potential energy savings were developed and are given in Table 2.12 which shows a total of 18 candidates with potential savings above the 10^{14} Btu/year (0.3×10^{14} for electric type) cutoff for the year 1990. The changes in the list from those candidates selected on the basis of the 1980 inventory are noted; a net of four additions to the list of 14 resulted.

Several designs for each of the 18 candidates were developed (Second Preliminary Assessment) and their years to pay back the added first cost were estimated. The most attractive designs were further analyzed for likely consumer acceptance and possible ten year nationwide cumulative energy savings.

TABLE 2.10

INDIVIDUAL RESIDENTIAL APPLIANCE ENERGY CONSUMPTION PATTERNS PROJECTED TO 1990 (At Point of Use)

Appliance	Annual Consumption Per Unit[1] (10^6 Btu/Year) For 1972	For 1990	Other Estimates of Percent Reduction ORNL	ADL-CEQ/FEA	Percent Reduction Used In This Study
Water Heater					
gas	37.2	27.2 (24)[2]	35	25	27 (35)[2]
electric	21.9	18.4 (17)[2]	14	20	16 (22)[2]
Dishwasher (Auto)					
gas	7.8	6.2	No estimate	22	20
electric	4.6	3.6		22	20
Dishwasher (Man)					
gas	7.8	7.8		No estimate	0
electric	4.6	4.6		↓	0
Clotheswasher					
gas	7.8	6.2		10	20
electric	4.6	3.6		10	20
Bath/Shower					
gas	21.6	21.6	↓	0	0
electric	12.7	12.7	0	0	0
Refrigerator	5.6	3.6	42	40	35
Range/Oven					
gas	13.8	7.2	42	26	48[3]
electric	4.09	3.0	14	25	26[3]
Clothes Dryer					
gas	8.2	6.5	No estimate	10	20
electric	4.3	3.9	↓	10	8
Air Conditioning[4]					
central	13.9	10.4	28	40	25
room	4.4	3.3	28	No estimate	25
Space Heating					
gas	120.0	90.0	35	50	25
electric	46.0	23.0	15	45	50[5]

[1] Supporting data for 1972 energy consumption is presented in Appendix A.

[2] Includes an 8% reduction in water usage.

[3] Includes 50% of ovens replaced by microwave.

[4] Includes improved insulation.

[5] Heat pump.

TABLE 2.11

PROJECTED NATIONAL 1990 ENERGY USAGE LEVELS
BY APPLIANCE FUNCTION
(Primary Energy Used by Single Family Residences)*

	1970			1990			
	Used in This Study	CEQ/ADL	ORNL	ORNL	CEQ/ADL	Value Used in This Study	% Growth
Housing Stock (In Millions)	46.3	46.75	43.99	62.09	60.0	61	32
Space Heater							
electric	4.9	4.6	6.2	24.8	12.0 all heat pump	18.0 ± 6.0 (heat pump included)	270
fuel**	51.4	52.0***	64.4	52.1	74.5	63.0 ± 11.0	23
Water Heater							
electric	9.3	9.0	6.4	12.0	14.0	13.0 ± 1.0	40
fuel	11.5	11.0	8.1	6.4	13.0	10.0 ± 3.0	-13
Refrig & Freezer	11.4	11.0	8.3	12.3	17.0	15.0 ± 2.0	32
Range							
electric	2.9	2.9	3.0	5.7	5.9	5.8 ± 0.1	100
fuel	3.4	3.0	2.0	1.1	1.0	1.1 ± 0.1	-68
Air Conditioner							
room	2.4	3.1	3.1	7.2	3.6	5.4 ± 2.0	125
central	2.3	2.4	2.5	10.3	17.4	14.0 ± 3.0	500
Clothes Dryer							
electric	2.4	2.0	No estimate	--	6.0	6.0	150
gas	0.6	0.35	"	--	.8	.65	8

*In units of 10^{14} Btu/year.
**gas, oil, and other
***neglected additional fuel use by oil-fired units included in ORNL

TABLE 2.12

POTENTIAL ENERGY SAVINGS OF CANDIDATE INTEGRATED APPLIANCES
(For Comparative Screening Purposes Only)

Option	Type of Energy Saved	Potential National Energy Saving* (10^{14} Btu/year) For 1970 Inventory	For 1990 Inventory	Changes to the List
Residential Application				
Furnace/Water Heating	Gas	4.2	3.9	
Drain HRS+ For Space Heating	Gas	4.3	4.8	
	Electric	1.3	1.3	
Drain HRS For Water Heating	Gas	5.0	5.0	
	Electric	0.9	1.4	
Room A/C HRS For Water Heating	Gas	2.3	3.6	
	Electric	Below Cut-Off	0.5	← Added
Range HRS For Water Heating	Gas	1.7	1.1	
	Electric	Below Cut-Off	0.5	← Added
Refrigerator HRS For Water Heating	Gas	1.5	1.1	
	Electric	0.5	0.6	
Furnace HRS For Range	Gas	1.0	Below Cut-Off	← Subtracted
Central A/C HRS For Water Heating	Gas	Below Cut-Off	2.7	← Added
	Electric	Below Cut-Off	1.0	← Added
Dryer HRS For Water Heating	Electric	0.4	1.1	
Dryer HRS For Space Heating	Electric	Below Cut-Off	0.8	← Added
Commercial Applications				
Range HRS For Water Heating	Gas/Electric**	1.0	1.5	
Drain HRS For Space Heating	Gas/Electric	0.8	1.0	
Drain HRS For Water Heating	Gas/Electric	0.9	1.4	

*Assuming complete replacement of two individual appliances by new integrated appliance. Taken at point of use.

+HRS stands for Heat Recovery System

**Gas and electric versions of commercial appliances are treated as one.

For a constant sales rate s, in sales per year of retrofit and new integrated appliances, the cumulative energy savings for the ten years 1980-1990 of the product is:

$$\text{Cumulative Energy Savings} = \sum_{i=1980}^{1990} (s)\ (1990 - i)\ (\text{energy savings per unit})$$

$$= 55\ (\text{energy savings per unit})\ (s)$$

The added first cost, acceptance rates, sales rates, and cumulative potential energy savings for the remaining candidates is given in Table 2.13.

Eleven of the remaining gas and electric integrated appliance candidates were set aside for the reasons given in Table 2.14, leaving seven final candidates, which includes a gas and electric version of the residential drain HRS. As will be seen in the following section, the gas and electric version are the same, and the drain HRS is (like the commercial candidates) treated as one candidate, leaving six final candidates.

TABLE 2.13

CUMULATIVE NATIONAL ENERGY SAVINGS
(18 Candidates, By Fuel Type)
(1980-1990)

Final Candidates	Name of Candidate	Added Cost ($)	Payback Years	Fuel Type	Consumer Acceptance Fraction (α)	Potential Sales (10^6/Year) Replacement and New	Potential Sales (10^6/Year) Retrofit	Expected 10 Year (1980-1990) Cumulative Energy Savings (10^{12} Btu)
•	Furnace/Water Heater	146[1]	2.0	G	36%	2.3	NA	956
•	Central A/C HRS	113[1]	3.5/1.5	G/E	45%	0.7	0.7	700
•	Commercial Range HRS	1800[1]	3.5/1.2	Both	.22/.54%	.011	NA	532
•	Refrigerator HRS	110	8.0[2]/3.5	G/E	22%	2.5	2.9	435
•[3]	Residential Drain HRS	800	2.0/0.8	G/E	.35/.72%	0.53	NA	934
•	Commercial Drain HRS	700	11.0[2]/4.7	Both	17%	1.4	0.3	725
	Dryer/Space Heating	70	2.4	E	32%	.10	.63	298
	Dryer HRS	227	4.5	E	19%	.42	.36	148
	Room A/C HRS	113	10.8[2]/4.6	G/E	18%	.14	.12	102
	Residential Drain/Space	227	21.0[2]/9.0	G/E	12%	.14	.07	72
	Residential Range HRS	50	14.0	G/E	2%	.21	0	12
	Commercial Drain/Space	→ See Table 2.14 Following ←						

[1] These preliminary estimates were refined in the course of the detailed analysis; revised cost estimates are reported in Tables 4.8, 5.7, and 6.4.

[2] Years to payback beyond cut off, only electric configuration considered.

[3] Gas and electric version combined into one candidate.

TABLE 2.14

REASON FOR SETTING ASIDE
11 OF REMAINING 18 CANDIDATES

Candidate	Reason
Room A/C for Water Preheat (Gas and Electric)	Years to payback estimated to be 4-6 years for electric water heaters and 11 years for gas and cumulative 10-year savings about 1/5 of the top six.
Commercial Drain HRS for Space	Water preheat is a better use of drain energy and they are not compatible with one another.
Drain HRS for Space Heating (Gas and Electric)	Its 9-year (electric) and 21-year (gas) payback period is unacceptable.
Range HRS (Gas and Electric)	Its 14-year payback period is unacceptable. New electric ranges and microwaves with reduced waste energy replace conventional units, reducing national potential for energy savings below cutoff.
Dryer HRS (Electric)	In order to meet 10^{14} criteria, latent heat recovery is necessary which means that a drain is required. Estimated years to payback is 4.5 years.
Dryer HRS for Electric Space Heating	Lint removal question a problem. Applicable to forced-air electric heating installations, reducing its energy-savings potential to close to the 10^{14} Btu/year level.
Gas Configurations for: Refrigerator HRS, Central A/C HRS	Payback period for recovery for a gas-fired water heater is beyond acceptable range.

3.0 FINAL SCREENING OF CANDIDATES

3.1 INTRODUCTION

Chapter 2 developed the methodology and gave the process by which some 349 candidate integrated appliances were examined, compared, and finally screened down to six final candidates which offered promising nationwide energy-savings potential.

This chapter presents the selection of the three candidates (described in Chapters 4, 5, and 6) to be considered for an ERDA-sponsored development and demonstration program, and a discussion of the merits and problems of the other three candidates not selected. The key to the final selection was to evaluate the likely effects of an ERDA-sponsored development and demonstration program on achieving the cumulative energy-savings potential projected for each candidate in Table 2.13 of the previous chapter. The rationale and results of this evaluation follow.

3.2 SELECTION OF FINAL THREE CANDIDATES

The remaining six candidates were screened on the basis of the likelihood of achieving the cumulative energy savings (Table 2.13) through an ERDA-sponsored development and demonstration program. This was a qualitative assessment done partially through a poll taken of the Steering Committee and partially by interviews with the Steering Committee members. Three of the candidates were judged to be more likely to benefit from ERDA support than the other three. These three are:

- Central Air Conditioner HRS*
- Furnace/Water Heater
- Commercial Range HRS

Table 3.1 summarizes the assessment of the final six candidates by the Steering Committee following their review of the Arthur D. Little, Inc., assessment given in Table 2.13. Their estimate of the consumer acceptance was used to generate the cumulative potential energy savings shown in the last column.

Though the commercial drain HRS looks to be an attractive commercial candidate, it was not selected for further consideration because it was felt that ERDA support of its commercialization would not enhance its success. As discussed in this chapter, this candidate has substantial precedence and was judged likely to proceed without government help.

---

*Denotes Heat Recovery System

TABLE 3.1

STEERING COMMITTEE POLLING
(Average of Six Evaluations)

Candidate	Estimated Added 1st Cost[1] ($)	Consumer Acceptance (%)	Risk to Manufacture (0 to 5)[2]	Acceleration of Penetration by ERDA Participation in Phase II (Years)	Cumulative 1980-1990 Energy Savings[3] (10^{12} Btu)
Residential					
Refrigerator HRS	157	17	3.5	6.25	336
Central A/C HRS	124	40	2.25	5.5	622
Drain HRS	762	6	4.5	6.5	255
Furnace/WH HRS	130	48	2.5	4.25	1,274
Commercial					
Range HRS	300	35	2.0	3.3	532
Drain HRS	633	34	2.3	3.7	593

[1] Installed cost with quantity production.

[2] 0 - low risk; 5 - high risk.

[3] These figures were calculated from those of Table 2.13 reduced or increased by the Steering Committee estimate of the consumer acceptance.

The other two candidates set aside were the residential drain HRS and the refrigerator HRS. Both were set aside because the risk of failure was considered much higher than for the top three candidates and because their potential for energy savings is lower. All three candidates set aside are discussed in this chapter. This is done to encourage future work on these candidates by providing background data.

3.3 COMMERCIAL DRAIN HEAT RECOVERY SYSTEM

3.3.1 System Configuration

This system, shown in Figure 3.1, consists of a heat exchanger connected to the drain for dishwashers and sinks in commercial kitchens or from washing machines in commercial laundries. Recovery from other hot water uses such as showers and baths are discussed separately later. The heat exchanger, shown in Figure 3.2, is a double-walled, fail-safe construction with cold water entering at line pressure and passing through it to an existing hot water heater. The outer shell of this heat recovery unit can be made from a thin gauge steel since the drainwater side is at atmospheric pressure. The heat exchanger section is made from two concentric tubes. The outer tube is welded to a flanged end plate. The inner tubes are loose within this outer tube section. Thus, if a leak develops either in the tube in contact with the drain water or on the tube carrying the supply water, water will begin to run out of the space between the two tubes at the end plate, thereby indicating the presence of a leak that needs to be repaired. The flanged end can be removed to permit cleaning of the system. A complete unit would be well insulated to prevent standby losses. The unit size for a typical restaurant would contain approximately 30 gallons of drain water. For larger applications, larger tanks would be used, or multiple units could be placed in series or in parallel.

Drainage from the washing machines or dishwashers pass through at atmospheric pressure to the drain. The inlets and outlets on the drain side are configured so that the heat exchanger will always have water in it. This accomplishes two functions; first, it maintains a reservoir of the drain water to preheat incoming water to accommodate the mismatch of draw and drain functions of batch appliances, i.e., dishwasher, sink, clothes washer, and second, it tends to prevent the build up of materials in the heat exchanger by preventing them from drying out between uses. The heat exchanger can be cleaned by pouring a chemical solvent in through the dishwasher or through a special opening to remove deposits. Since the heat exchanger is constructed so that it does not drain, these chemicals can be allowed to sit over a period of time.

3.3.2 Precedence

Hatco Corp., Milwaukee, Wisconsin, is a manufacturer of equipment for the food service industry. They currently have a line of electric booster

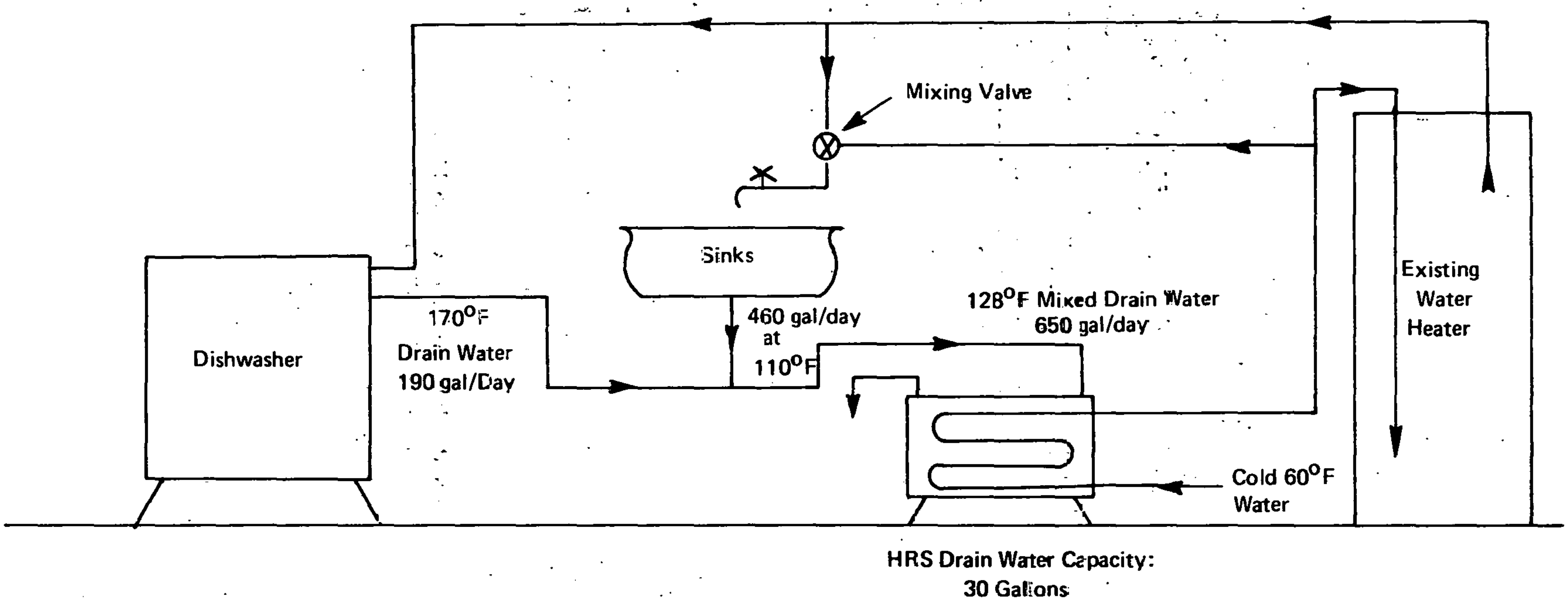


FIGURE 3.1 HEAT RECOVERY SYSTEM FOR TYPICAL RESTAURANT SHOWING AVERAGE OPERATING VALUES

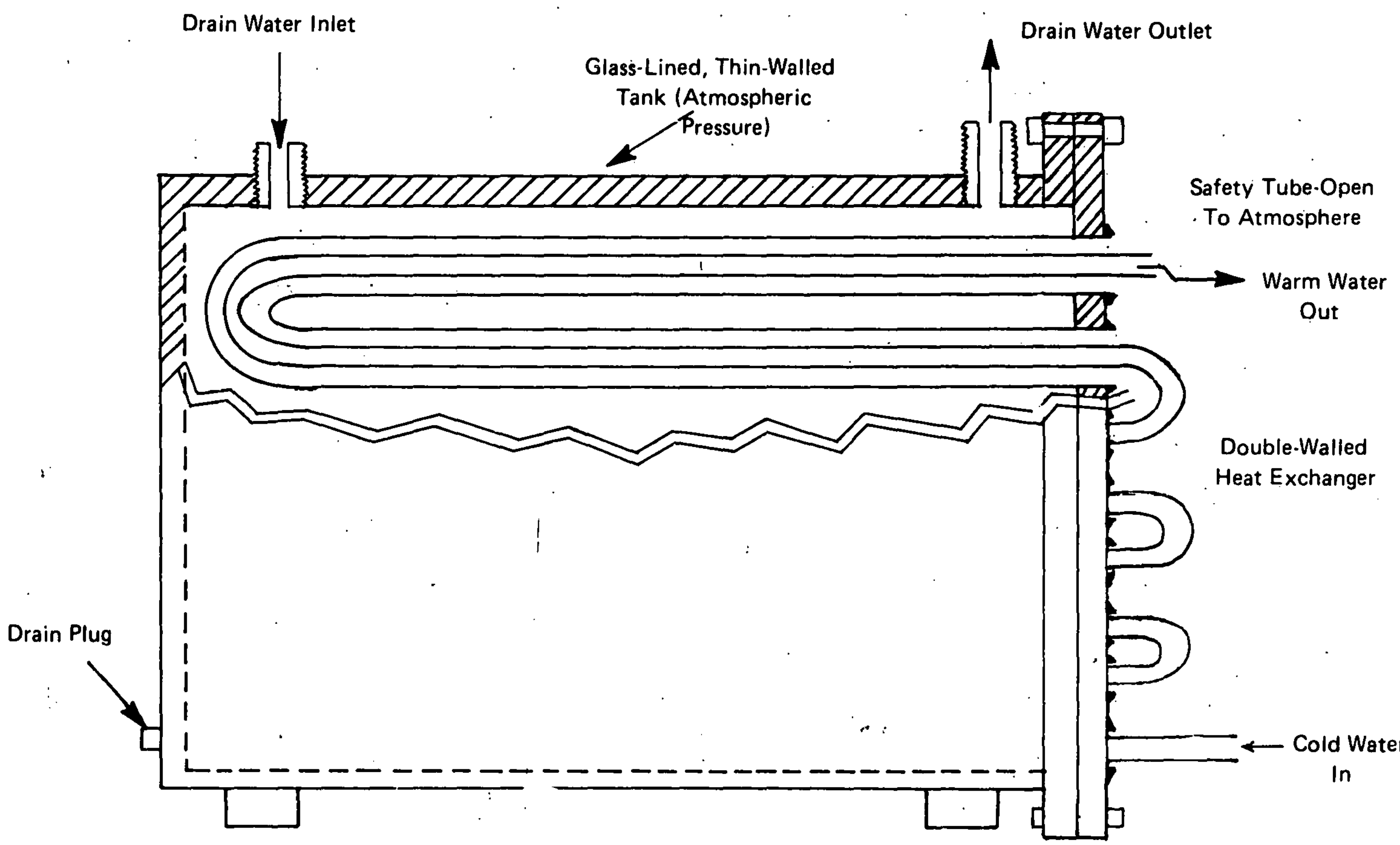


FIGURE 3.2 HEAT EXCHANGER DESIGN FOR COMMERCIAL DRAIN WATER HEAT RECOVERY SYSTEM

water heaters for use with dishwashers, primary electric water heaters, food warmers and heaters, etc. Over the last year, they have been working on the development of a heat recovery system similar to the one just described above. This unit would be connected to the drain of the dishwasher and used to preheat the water coming into the dishwasher. The unit that they are developing employs a double-walled fail-safe design. They anticipate that this unit may completely replace primary water heaters for this application. Currently, a unit with an effectiveness (effectiveness is $Q_{actual}/Q_{maximum}$; see Section 2.5.2) of approximately .5 is being offered through their normal sales channels to commercial kitchen operators, both as a retrofit system and as a replacement part for existing water heaters.

Another manufacturer of equipment for the restaurant industry, Elsters, Inc., of California, has developed several versions of heat recovery systems for dishwashing. They are presently field testing a unit in conjunction with ERDA at Colonie, New York. Their system is an in-line recovery unit operating off the drain water from the dishwasher. These tests are just getting underway and operational data is not available at this time.

A change in the dishwashing techniques which has developed over the last five years may have an adverse effect on drain water recovery systems in the future. A number of manufacturers of dishwashers are currently offering a chemical rinse system which enables the dishwasher to satisfactorily sanitize the dishes using a lower rinse temperature. Hobart, the primary supplier of dishwashers for commercial markets, has developed several smaller units employing a chemical wash and is currently working on developing larger systems. The difficulty with these systems comes from a build-up of the chemical in the rinse water unless a significant percentage of the water used is wasted by allowing it to run down the drain. Even though the temperature of the water, 120°F, is lower than the normal 180°F drain water, there still is a significant amount of energy wasted. For small units, the amount of hot water lost is not important, but on a large system, the cost to heat the water even to 120°F is major. Although temperatures below 120°F may be used, Hobart feels the dishes do not dry satisfactorily, even with the use of wetting agents to promote better drying. Hatco reports that the sales of booster heaters for dishwashers by other companies are down 75 to 80 percent due to the use of chemical rinses.

The drain water energy recovery projections have all been based on the assumption that water usage for dishwashing will continue at about the same level. If the use of chemical rinses grows, these estimates will be too high.

3.3.3 National Potential Energy Savings

In commercial buildings such as hospitals, hotels, and motels, hot water is used for purposes other than dishwashing and laundry. In these buildings, a substantial amount of hot water is used for showers and baths; however, the drainage systems in these building are mixed with cold water from toilets. The combined effect of the cold water and the solid waste makes recovery from a mixed drain a formidable problem. A technique for heat recovery from a mixed drain system is examined in the following section, with the conclusion that it is only marginally cost effective for the electrically-heated hot water system. We believe that the same conclusions apply to the commercial mixed drain heat recovery system and omit it from further discussion.

Table 3.2 presents data showing the estimated machine washing hot water usage for a variety of commercial buildings. Shown are the temperatures of the drain water, possible rates of heat recovery, number of days of operation for each building, and the potential energy recovery per year.

The cost estimates for the systems are given in Table 3.3 based on 100 gallons per hour water recovery flow rate. The costs for the larger system were scaled on the basis of the maximum hourly flow rate. The years to payback were figured on the basis of these costs and are shown in Table 3.4, along with market acceptance estimates and the projected cumulative energy savings. Based on this analysis, a value of 761×10^{12} Btu cumulative* energy savings is estimated for this candidate.

3.4 RESIDENTIAL SYSTEM FOR RECOVERING WASTE HEAT FROM DRAIN WATER

3.4.1 System Configuration

Similar to the commercial drain water HRS, the systems described in this section are designed to recover the heat that is wasted in the drain water. Sources of this waste energy are dishwashers, washing machines, baths and showers, and normal hot water usage for sinks. In most homes, drains, including toilet water (black water), are joined throughout the house and a heat exchanger in the cellar or crawl space is employed. Since black water containing significant amounts of solid waste will pass through the system, there can be no batch heat exchanger; the drain must flow freely, otherwise sedimentation may occur, creating a health hazard.

When separate gray (shower, dishwasher, clothes washer) water drain systems exist, a batch heat exchanger similar to those discussed in the previous section for use in commercial applications may be used. As will be shown later, the batch heat exchanger is simpler, less expensive, and more cost effective than the flow through drain recovery system.

3.4.2 Precedence

At this time no systems for recovering waste energy from the drain are commercially available. Several demonstration-type installations have been made in the United States and Europe. In Europe, Philips[3.1] has built a house which incorporates a number of energy-saving systems. The house is based on the use of solar energy for space heating and for water heating.

*1980-1990.

TABLE 3.2

COMMERCIAL DRAIN WATER RECOVERY

	WATER USAGE							
	DISHWASHING			LAUNDRY				
		Waste Water Temp.			Waste Water Temp.			
	Gallons Per Day	Before HRS (°F)	After HRS (°F)	Gallons Per Day	Before HRS (°F)	After HRS (°F)	Days of Operation Per Year	Point of Use Energy Recovered (10^6 Btu/year)
Restaurant	650*	128	80	-		-	312	81
Health Care	650	128	80	-		-	365	95
Schools	760**	128	80	-		-	180	54
Cafeterias	1,730	128	80	-		-	312	215
Hospitals, Hotels, Motels	1,950	128	80	5,400***	105	79	365	1,003
Institutions (business cafeterias)	1,950	128	80	-		-	260	202
Laundries	-		-	5,184	105	82	365	503

*Total water demand of 790 gal/day (see Table 6.1) less unrecoverable 140 gallons used for hood wash and steam production.

**950 gallons of the 1,710 gal/day (Table 6.1) were considered unrecoverable.

***Weighted average (6,000 gpd, 4,275 gpd, 5,400 gpd respectively).

TABLE 3.3

HEAT RECOVERY UNIT COST
(Restaurant/Health Care Facility Size*)

Components of HRS	Component Cost
200 ft. 3/8-inch tubing	$ 55
200 ft. 5/8-inch tubing	150
Shell with flanged end	75
Insulation and jacket	20
Assembly	100
Subtotal	$400
Manufacturers' markup	300
Shipping and installation	100
Total Cost	$800

---

*Costs for other units are scaled on the basis of estimated maximum hourly flow, as follows:

	GPH
Restaurant	100
Health Care	700
Schools	125
Cafeterias	200
Hospitals, Hotels, Motels	620
Institutions	240
Laundries	400

TABLE 3.4

SUMMARY OF COMMERCIAL DRAIN HRS

Type	Point of Use Energy Recovered Per Unit (10^6 Btu/Year)	Cost ($)	Years to Payback: Gas	Years to Payback: Electric	Potential 1990 Primary Savings (10^{14} Btu/year)	% Market Penetration, New Sales (α): Gas	% Market Penetration, New Sales (α): Elec	% Market Penetration, Retrofit (γ): Gas	% Market Penetration, Retrofit (γ): Elec	Cumulative Energy Savings 1980-1990 (10^{12} Btu): Gas	Cumulative Energy Savings 1980-1990 (10^{12} Btu): Elec	Cumulative Energy Savings 1980-1990 (10^{12} Btu): Total
Restaurant	81	800	2.3	0.8	.17	42	67	2.1	4.2	30	36	66
Health Care	95	800	1.9	0.7	.06	48	69	2.6	4.6	13	13	26
Schools	54	1,000	4.2	1.6	.11	21	56	.7	3.0	8	19	27
Cafeterias	215	1,600	1.7	0.6	.09	54	70	2.8	4.9	19	19	38
Hospitals, Hotels, Motels	1,003	5,000	1.1	0.4	.23	64	71	3.9	5.3	65	58	123
Institutions	202	1,900	2.1	0.8	.04	46	67	2.4	4.2	8	8	16
Laundries*	503	3,200	1.5	0.5	1.03	57	71	3.2	5.2	250	215	465
												761

*Inventory: 1980: 88,000 gas, 10,000 electric
1990: 95,000 gas, 31,000 electric

In addition, it incorporates several heat recovery systems. One system, shown in Figure 3.3, recovers heat from drain water. The system utilizes a central tank through which the gray waste water passes. Contained within this tank is a coil fed by cold inlet water and a freon-water heat exchanger which is connected to a heat pump. When hot water is required, it enters the building, passes through the coil in the waste heat recovery tank, then passes through a water heater which is heated by a heat pump or by resistance heaters and then to the point of use. The heat pump takes heat from the waste heat tank and transfers it to the hot water tank.

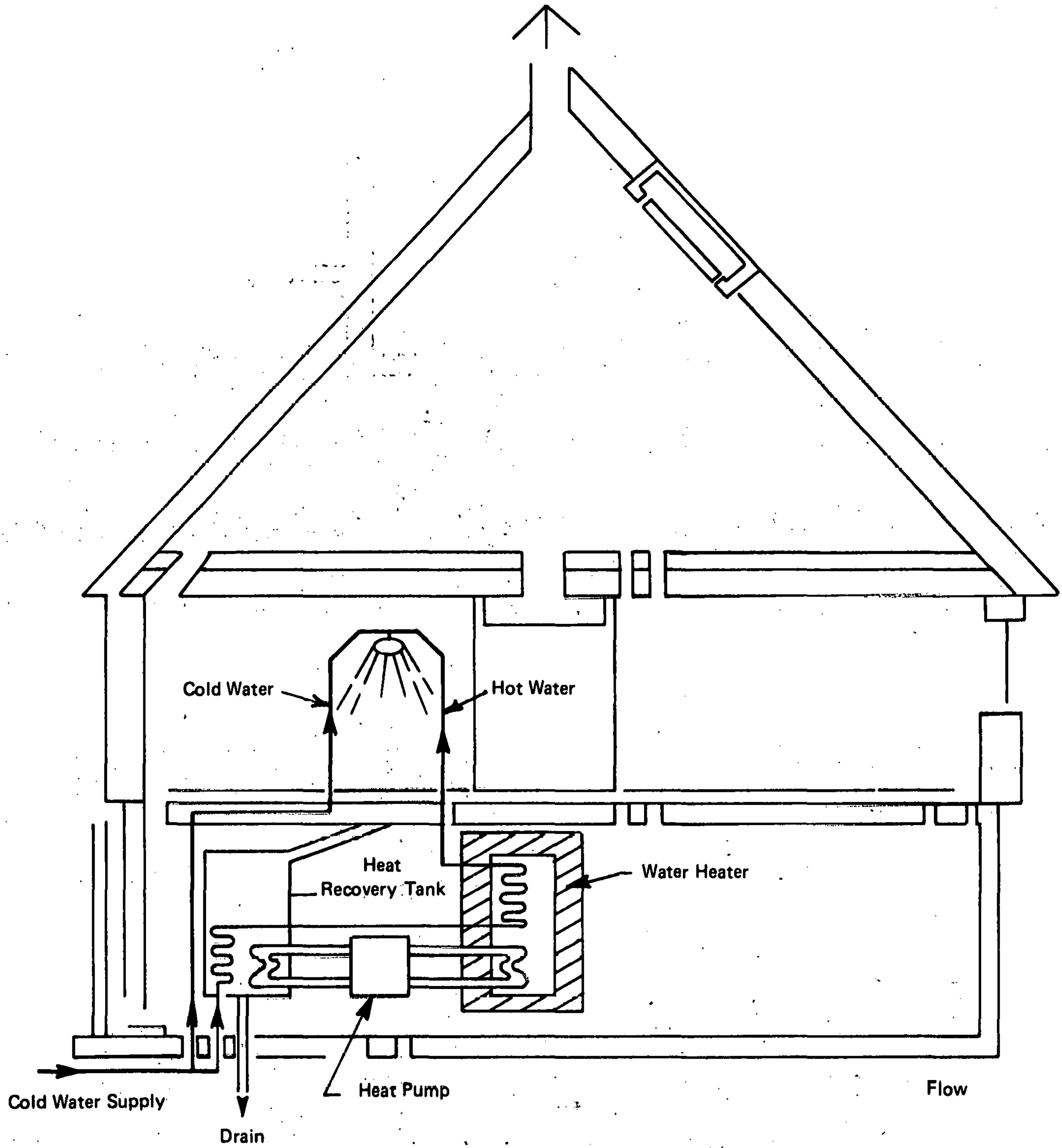
In the United States, a demonstration home, Habitat 2000[3.2], shown in Figure 3.4 includes two different drain water heat recovery systems. In one system a counterflow heat exchanger is specifically used for the showers, while the second system, similar to the Philips' system, gathers waste water from a number of sources.

In the shower recovery unit, hot water comes directly into a mixing valve from the standard water heater. The cold water enters the mixing valve after passing through the heat exchanger. When the shower is first turned on, only cold water and hot water enter the mixing valve. As the heat exchanger fills up with warm drain water, the temperature of the cold water begins to increase and the mixing valve must be adjusted manually to compensate for the change in water temperature. Automatic valves are available to compensate for these temperature changes; however, the cost of the automatic valve is quite significant relative to the cost of the overall recovery system. The heat exchanger in this system has been patented by Hobart.

This house utilizes a system similar to the Philips' system for recovering additional energy from the drain water. After leaving the drain, all of the water from the laundry, the sinks, and the showers pass through a secondary heat exchanger for preheating the domestic water. When hot water is drawn, the incoming cold water passes through the secondary heat exchanger and then into a preheat tank. Connected to the preheat tank is a heat pump. This heat pump adds energy to the water before it passes into the final water heater.

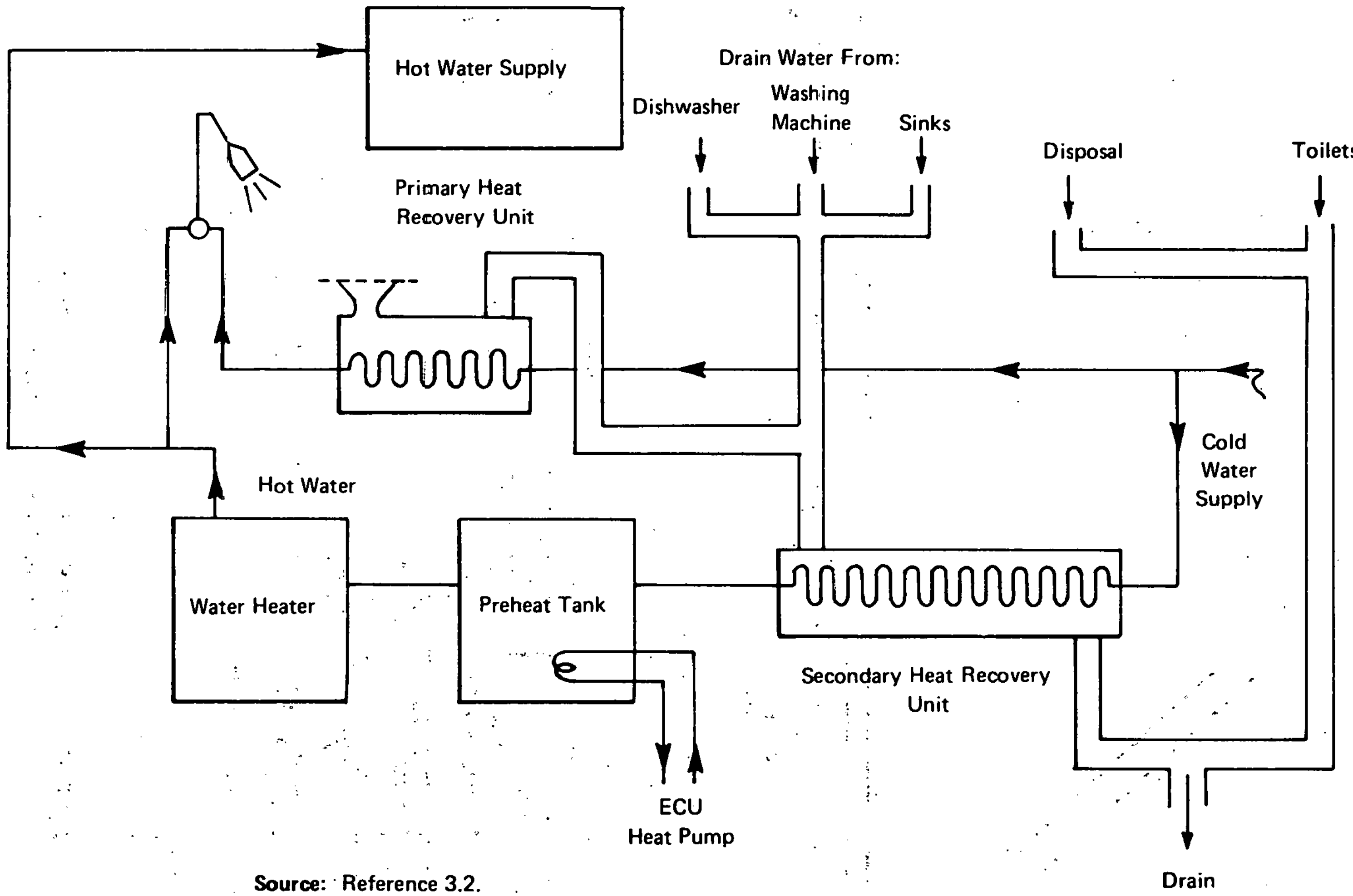
3.4.3 National Potential Energy Savings

It was felt that it was advantageous for a drain HRS to be adaptable to existing dwellings as well as new buildings with special, separate, non-mixed drains in order to obtain a significant nationwide energy savings. This requires that the system use existing drain pipe configurations, and furthermore, discussions with Steering Committee members made it apparent that it was essential to have a double-walled, fail-safe type heat exchanger design. A design which meets these requirements in shown in Figure 3.5, along with estimated system parameters.



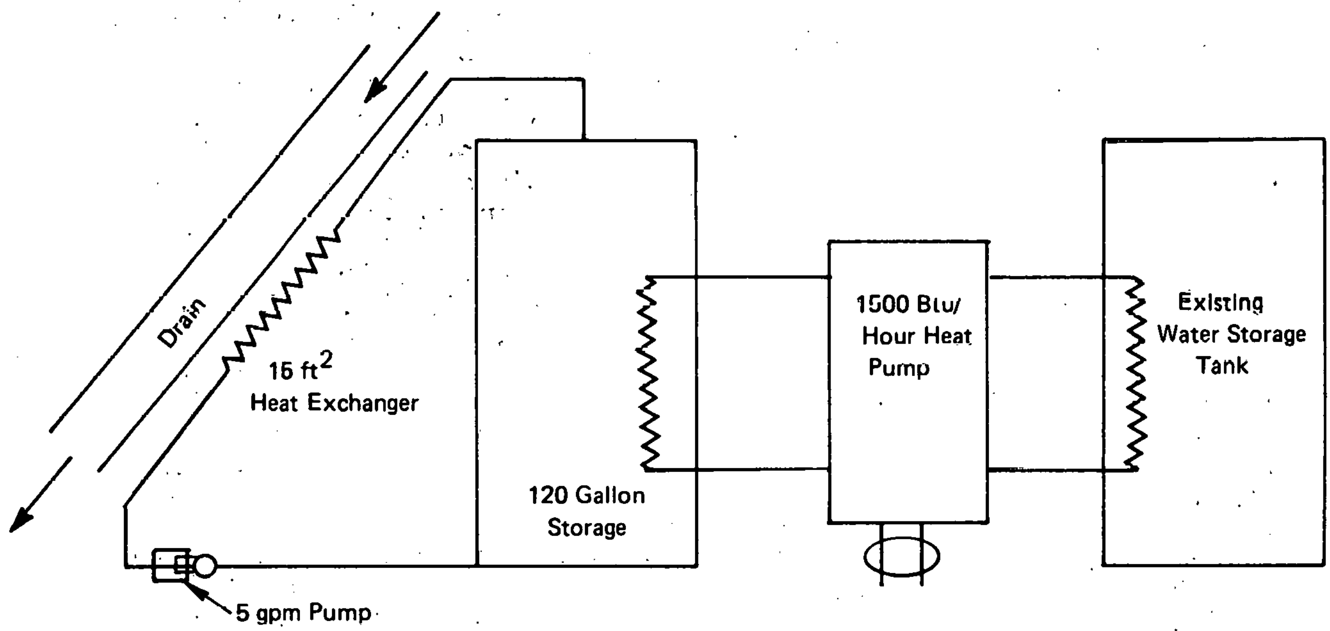
Source: Reference 3.1.

FIGURE 3.3 PHILIPS' SYSTEM FOR WARM WATER PREPARATION FROM DRAIN WATER WITH HEAT PUMP



Source: Reference 3.2.

FIGURE 3.4 HABITAT 2000 DRAIN WATER HEAT RECOVERY SYSTEM



Estimated Parameters	Gas	Electricity
Annual Primary Energy Savings (per unit) (Btu/yr)	17×10^6	46.3×10^6
Value of Annual Energy Savings (1985)	$ 60	$ 136
Added First Cost	$ 700	$ 700
Years to Payback	11.7	5.1
Nationwide Inventory (1970)	33	11
(1990)	35	22
Annual Energy Savings (Nationwide, primary, 100% penetration, 1990)	5.95	9×10^{14}
Retrofit population (1990)	44	22

FIGURE 3.5 DRAIN HEAT RECOVERY SYSTEM FOR WATER PREHEAT

In this design, a heat exchanger is clamped to a section of the drain pipe. A glycol solution is circulated through the heat exchanger and into an insulated storage tank. Heat is extracted from the storage tank by a heat pump and transferred into a large hot water heater. In operation, a flow temperature sensor mounted in the drain pumps turns the heat pump on in the presence of warm flow through the pipe. As this tank is somewhat larger than the standard water heater and contains the heat exchanger, it would have to be installed to replace the existing water heater. *A 1 gpm limit on the hot or warm drain water flow rate is necessary to keep the size of the heat exchanger within reasonable limits.* Even with this constraint, the heat exchanger requires about 15 feet of the drain pipe for the heat exchange. Maintaining this 1 gpm limit will require major changes in the drain rates of dishwashers, clothes washers, and drain restrictors in sinks and tubs.

The estimated added first cost of the system is given in Figure 3.5, along with the estimated energy-savings potential. Based on these figures, an expected payback period of 5 years is found for electrically heated water heaters.

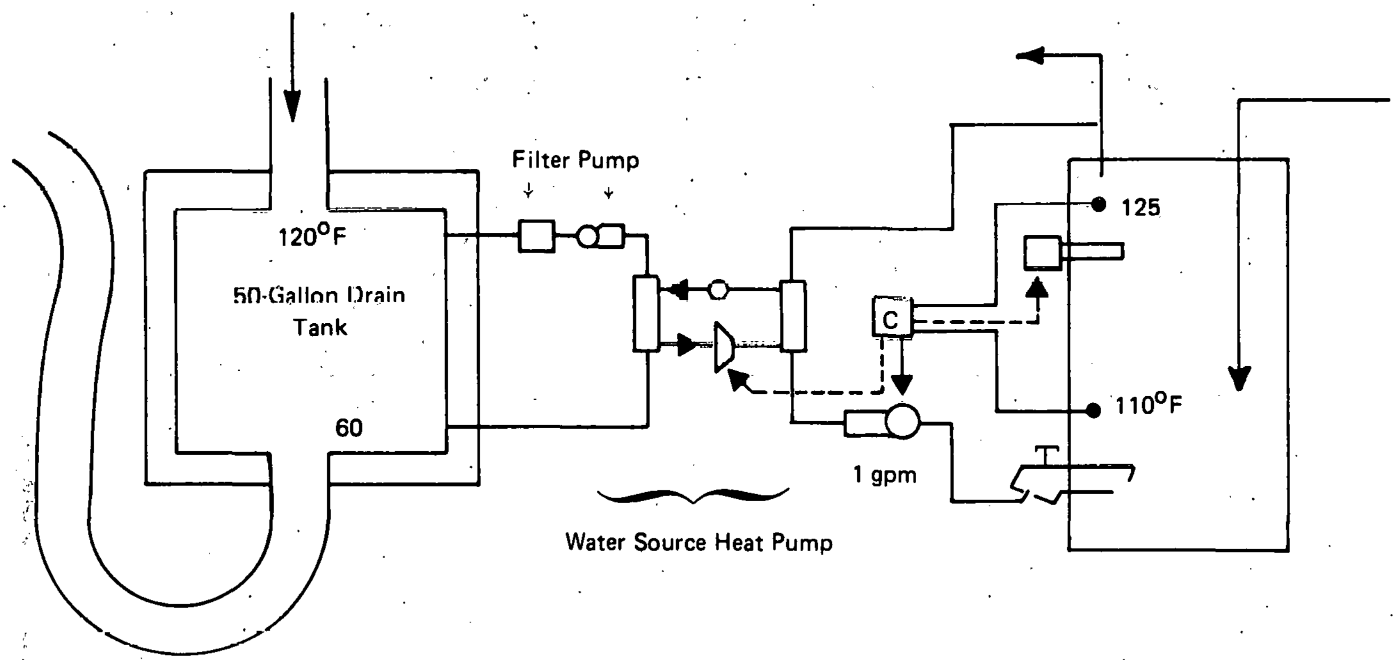
The required limit on drain flow rate and the poor payback makes this an impractical design, and so it was concluded that the retrofit drain HRS was not practical and attention was given to a system explicitly designed for new homes with special separated drains, as in the Philips' house and the Habitat 2000 house. This design, shown in Figure 3.6, is based on the Philips' house discussed earlier. It is applicable only to new houses with electric water heaters, but it does have the potential for saving a sizable amount of energy in the ten year period, 1980-1990.

3.5 REFRIGERATOR/WATER HEATER HEAT RECOVERY SYSTEM

The electric power into the refrigerator and the heat withdrawn from the cabinet by the refrigeration function all end up as waste heat in the room. This heat may be rejected to a cold water supply, providing free water heating while improving the refrigeration efficiency at the same time. During the summer months, all of this waste heat can be recovered, while during the heating season we estimate that about 80% of this heat from the refrigerator contributes usefully to the space heating functions of the house (see Appendix B for further explanation) and cannot be recovered.

Any heat recovery system for the refrigerator must be designed to guarantee that the refrigerator is able to reject heat to either the room air or to the cold water. Total replacement of the existing refrigerant-to-air heat exchanger by the water heat recovery heat exchanger causes several problems. These are:

- A supply of cold water must be provided regardless of the need for heated water. During extended periods of low or no water usage, storage of the heated water will



- - - - ► Denotes: Thermostatic-Controlled Function

C Denotes: Temperature Controller

Estimated Parameters	Gas	Electricity
Average Primary Energy Savings (10^6 Btu/yr)	17.2	46
Value of Annual Energy Savings (1985)	$ 60.2	$ 138
Added First Cost	$ 440	$ 440
Years to Payback	7.3	3.1
Nationwide Inventory Applicable (1990) 10^6 Units*	8	12
Expected Acceptance Rate	0%	25%
Cumulative National Energy Savings Potential 1980–1990 (10^{12} Btu)	0	759

*2 Million New Housing Units Per Year Between 1980–1990, 60% with Electric Water Heaters.

FIGURE 3.6 DRAIN WATER RECOVERY IN NEW CONSTRUCTION

be impractical and the heated water must be disposed of so long as the refrigerator runs.

- After long periods of use, mineral deposits will build up in the water side of the heat exchanger, reducing the heat transfer capability and jeopardizing the performance of the refrigerator.

For these reasons, the air-cooled heat exchanger remains in-line with the water recovery heat exchanger so that a full heat rejection capability is always available independent of the water heat exchanger.

The amount of heat that is recoverable from the refrigerator depends on the temperature of the cold water entering the water heat recovery unit. The effect of the entering water temperature and the potential energy savings from a standard refrigerator/freezer combination is shown in Figure 3.7. Clearly, the greatest heat recovery with practical heat exchangers is achieved with the lowest available water temperature. A system designed to take advantage of this is shown in Figure 3.8. In this system, a low mixing (stratified) storage tank is used in series with the existing water heater. This may be a small tank containing a diaphragm separating the cold water supply from the hot water return or as depicted here, it may be a conventional insulated tank with special, yet to be developed, inlet and outlet diffusers. A small pump is used to circulate the cold water from the bottom of the stratified tank through the heat recovery coil to the top of the stratified tank. In the event of a hot water draw through the exsiting water heater, the incoming cold water displaces hot water in the upper part of the stratified tank, providing preheated water to the existing water heater.

An estimate of the energy-savings potential for one of these units is also given in Figure 3.8. Only a fraction (about 1/2) of the available actual heat recovery is credited to the system, for it is estimated that approximately 80%* of the heat during 60% of the year (heating season) is useful space heating. This reduces the net energy savings to about 48% (80% x 60% = 48%). Based on these energy savings and the added first costs of the system, the years to payback the added first cost ranges between two and three for electric water heaters and ten years for gas water heaters. The two-year payback occurs with oil-fired space heating units where an electric water heater is used. In this case, the refrigerator waste heat displaces electric water heating, while the loss in space heating during the heating months is made up by oil space heating rather than electric space heating. The ten-year payback is associated with a gas-fired water heater and a gas-fired furnace and is out of the range of acceptability.

The anticipated nationwide inventory based on the sales of new refrigerators are also summarized in Figure 3.8.

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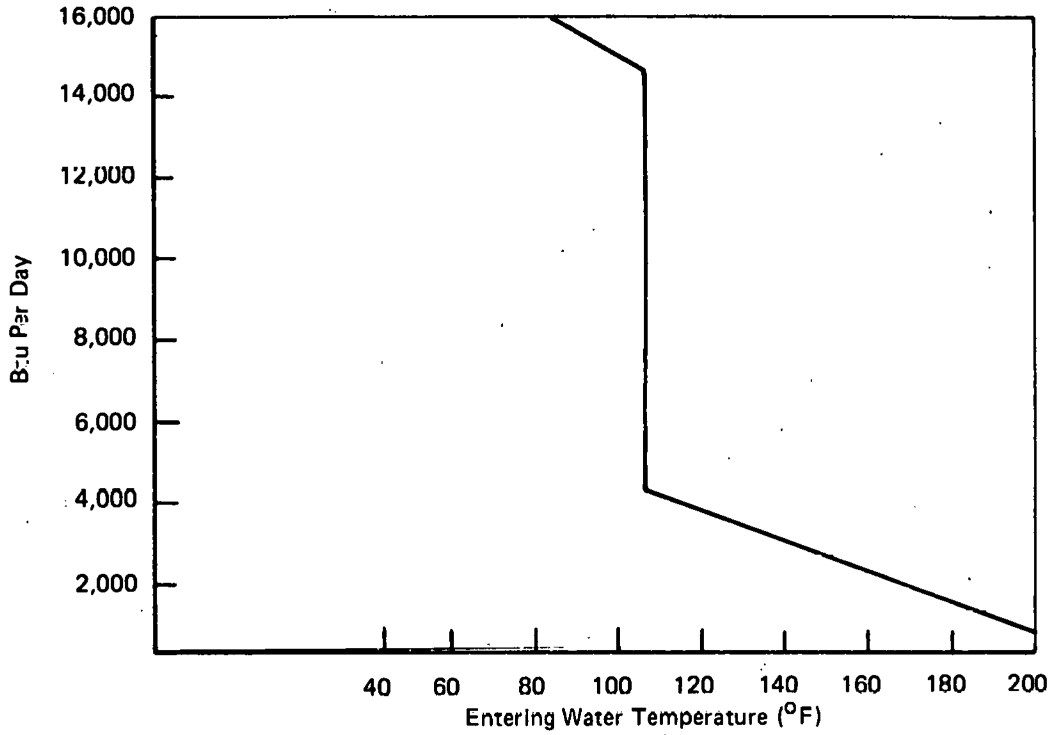
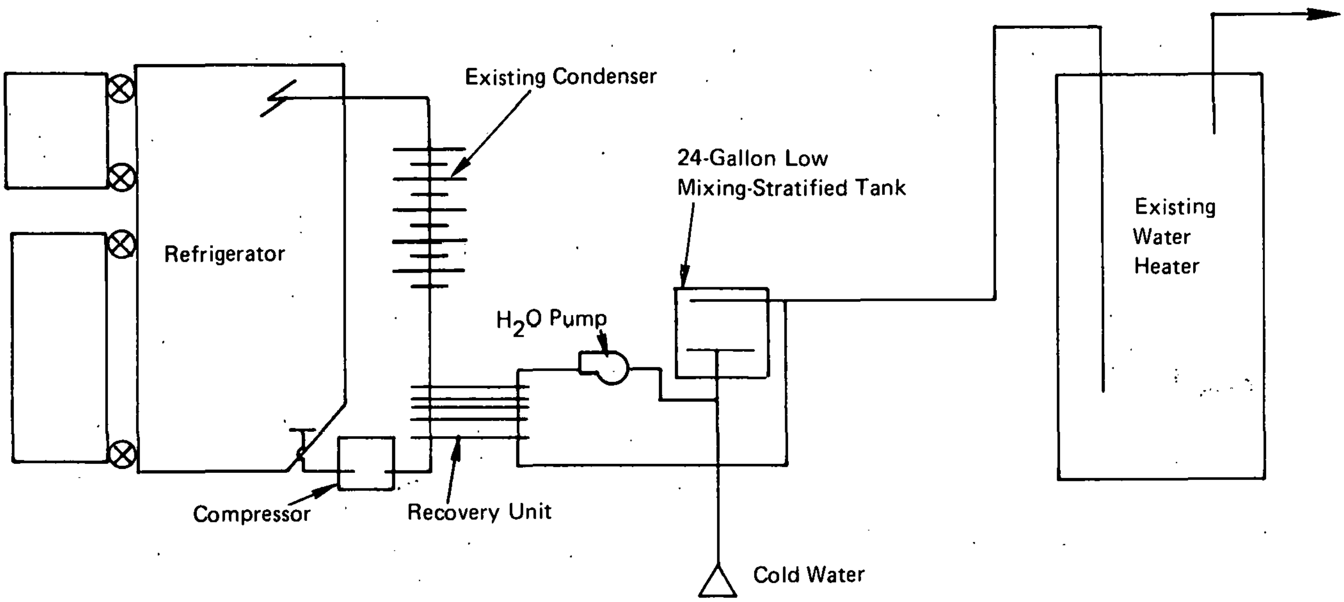
*See Appendix B.



Assumed Heat Exchanger Effectiveness – 80%
(See Section 2.5.2 For an Explanation of Effectiveness)

FIGURE 3.7 **MAXIMUM HEAT RECOVERY POTENTIAL FROM A REFRIGERATOR AS A FUNCTION OF INLET WATER TEMPERATURE**



Estimated Parameters	Gas	Electricity
Annual Primary Energy Savings (10^6 Btu/year)	5.0	13.8/21.1
Value of Annual Energy Savings (1985)	$ 17.50	$ 47/72*
Added First Cost	$ 142	$ 142
Years to Payback	8	3.4
Acceptance (new refrigerators purchased)	0%	25%/37%
Average Sales in Millions (1985)**	0	.5/.4
Cumulative 1980–1990 Energy Savings (10^{12} Btu)	0	844

*In an oil-heated home with electric water heating, replacement of electric water heating by refrigerator waste energy causes increased space heating, but with lower primary energy or fuel (oil) use resulting in increased primary savings; annual energy cost savings includes the summertime electric savings, plus the wintertime effective replacement of electric with oil, resulting in an annual savings of $72 compared to the $47 for the all electric.

**Based on 8 million refrigerator sales, 24% of sales to electric water heaters and 14% to homes with electric water heaters and oil furnaces.

FIGURE 3.8 REFRIGERATOR-WATER HEATER

The problems with the refrigerator/water heater recovery system are related to uncertainties in the reliability of the system and uncertainties of the impact on the product warranty of the refrigerator.

4.0 AIR CONDITIONING HEAT RECOVERY FOR WATER HEATING

4.1 DESCRIPTION OF CONCEPT

4.1.1 Overview

Nearly all residential central air conditioners used in the United States operate by rejecting waste heat to the outside air. The concept under consideration in this section is that of recovering a portion of this waste heat for domestic water heating. In certain locations of the United States and certain applications, the heat is rejected to water available from some nearby source, but the warm water is not recovered for domestic use. Air conditioners or heat pumps with water cooling are few in number since large volumes of cool water are required.

The air conditioner heat recovery system (A/C-HRS) for water heating still relies on outside air as the heat sink to which the heat not used for water heating is rejected. The lower temperature portion of the heat goes to the air, while the higher temperature portion of the heat is used for heating of the domestic water. When water heating is not required, all of the waste heat goes to the air. The portion of the waste heat recovered depends on the water heating needs of the user; larger heat recovery units are needed for larger water users.

If the A/C-HRS is installed in a heat pump unit, the recovery unit may be operated during the heating mode. The unit will draw heat exactly the same as in the case of the air conditioner and hence is withdrawing heat which could be usefully provided to the space heating function. In this manner, about 1 Btu of water heating is provided by about .3 to .4 Btu of electric energy when the outside temperature is above the balance point. The balance point is the temperature below which the heating demand exceeds the heat pump capacity and electrical auxiliary heating is used. The HRS should not be used below the balance point.

4.1.2 Precedents

The heat recovery system is not a new concept, having seriously been investigated in the early 1960's by Florida Power and Light[4.1]. A system was conceived which is being marketed today in which water is circulated from the storage hot water tank to a heat exchanger placed between the compressor and condenser of the air-cooled air conditioner. When the air conditioner is operating and the water storage tank temperature is below the upper limit, the circulation pump, shown in Figure 4.1, is activated and heat is extracted from the refrigerant, thereby providing water heating. Typically, the refrigerant enters the water heat exchanger at 200-250°F, providing ample temperature for achieving useful water temperatures for domestic purposes.

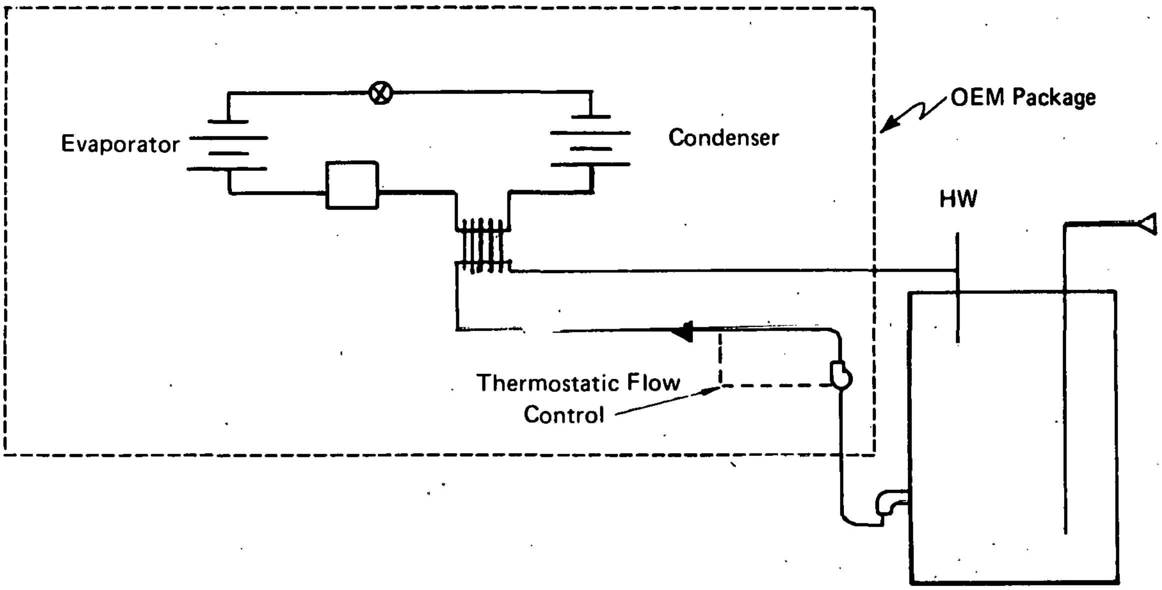


FIGURE 4.1 A/C – WATER HEATER UNIT INSULATED WATER TRANSFER LOOP

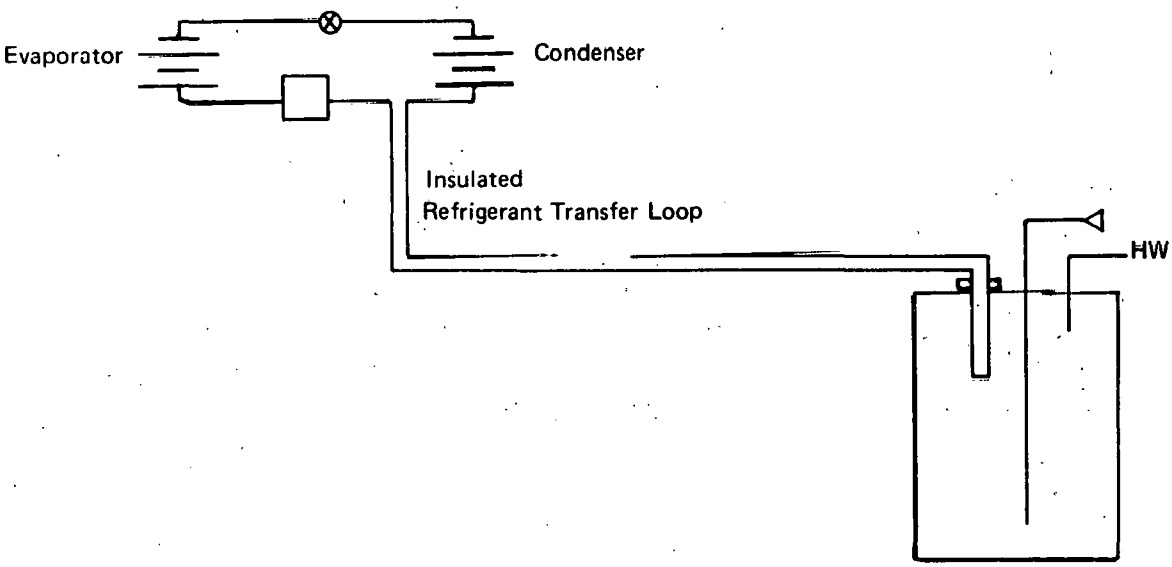


FIGURE 4.2 A/C – WATER HEATER UNIT INSULATED REFRIGERANT TRANSFER LOOP

Several devices for new and in-place air conditioners using this principle are available. The devices manufactured, shown in Table 4.1, except for Clark Energy Saver are based on this principle. The Clark Energy Saver is shown schematically in Figure 4.2 and consists of a bayonet-type heat exchanger in the water heater which is fed by refrigerant circulated from the air conditioner.

Marketing of these devices for use in heat recovery from central air conditioners has taken place primarily in Florida and a few other southern states where the annual hours of air conditioning compressor on-time exceed 2,000 hours a year. These systems are also used on commercial refrigeration units which operate year-round independent of climatic conditions. Because of the large amount of heat rejection from commercial refrigeration units (in supermarkets, dairy processing farms, and meat processing plants), commercial applications of these devices have exceeded the residential applications since energy operating costs have a larger effect on profits, and the commercial operator is normally more sensitive to the life-cycle costs of operating equipment and is prepared to purchase energy-saving devices with reasonable payback periods.

The refrigerant circulating bayonet-in-tank type configurations is shown schematically in cross section in Figure 4.3. A well filled with heat-conducting fluid separates the walls which contact the water and the refrigerant. The purpose of the well is to provide a vent if either the refrigerant-filled lines or the bayonet in the tank form a leak. This prevents the possibility of contaminating the water supply with the refrigerant-oil mixture contained in the refrigeration unit in the event that a leak in both walls occurs.

The heat exchanger configurations for the water circulating systems use a tube-to-tube, tube-in-tube, or a tube-in-shell configuration. The tube-in-tube configuration has but a single wall separating the refrigerant oil mixture from the potable water and therefore may not prevent crossover in the event of a leak. The tube-to-tube configuration bonds the two tube walls, one containing refrigerant, the other containing the potable water, with a heat conductive solder which has been shown by the manufacturer to provide a vent to atmosphere in the event of a leak in either of the two.

The tube-in-shell is a conventional water-refrigerant heat exchanger design typically used for water-cooled condensers and does not afford any crossover leak protection since the refrigerant circulates on the outside of the water-filled tubes.

A summary of the units presently available is given in Table 4.2. The capacity of the device is given in Btu's of water heating per ton hour of air conditioning. Scaling refers to the formation of solid deposits in the flowing water passages, causing reduced performance. The other columns are self-explanatory.

While the performance of the A/C-HRS is relatively easy to establish in laboratory conditions with fixed air conditioning and water heating parameters, the performance of the system in the field under variable

TABLE 4.1

MANUFACTURERS OF THE
AIR CONDITIONER HEAT RECOVERY UNITS

Firm	Type of Product	Approximate Number Installed: Commercial Application	Approximate Number Installed: Residential Application
Energy Conservation Unlimited, Inc. Longwood, Florida	Retrofit and OEM*	1,000	200
Friedrich Air Conditioning & Refrigeration Co. San Antonio, Texas	Retrofit and on new heat pumps	NA	First year of production
Clark Energy Saver, Inc. Miami Beach, Florida	Retrofit	20	100
Sun-Econ, Inc. Ballston Lake, New York	Retrofit and OEM	1,000	200
Carrier Air Conditioning Co. Syracuse, New York	New A/C units	NA	First year of production
Refrigeration Research, Inc. Brighton, Michigan	OEM	Unknown	

*OEM means Original Equipment Manufacturer and refers to units sold as components to manufacturers of HVAC equipment.

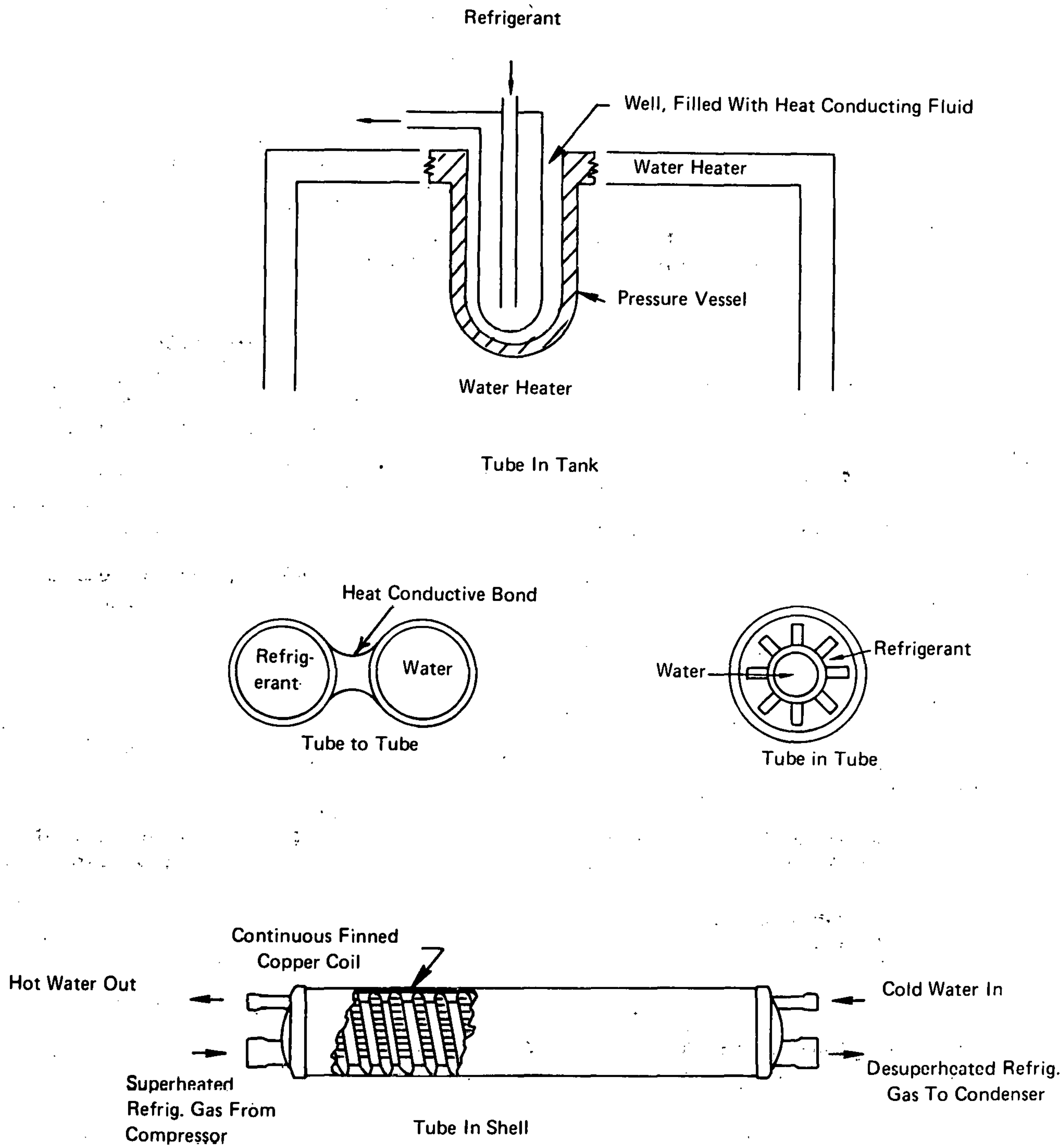


FIGURE 4.3 REFRIGERANT/WATER HEAT EXCHANGERS

TABLE 4.2

PRESENTLY AVAILABLE UNITS

Manufacturer	Configuration	Presently Available Form	Safety	Scaling and Freezing	Installation	Typical Btu/ton-hour of 130°F Hot Water	Device Cost	Installed Cost
Clark Energy Saver, Inc.	Tube-in-Tank	Available only as retrofit.	"Fail-safe" design. Refrigerant and potable water separated by double wall and atmosphere well.	Freezing not a problem. Scaling not likely to be any more severe than for water heater itself.	Installation limited to applications where compressor and water heater are close to one another. Cutting, connecting to refrigerant system requires evacuating and recharging refrigeration system.	1000*	$80	$200
Friedrich Air Conditioning and Refrigeration Co.	Tube-to-Tube	Built-in new A/C units and as retrofit.	"Fail-safe" design.	Freezing a problem when used with outdoor compressor. Scaling also likely to be a problem in certain areas.	Minimal installation problems with units built in to A/C.	1700**	Built in $250	$280
Energy Conservation Unlimited Sun-Econ, Inc.	Tube-in-Tube	Retrofit.	Not a fail-safe design since a single wall separates refrigerant/oil from potable water.	Freezing a problem when used with outdoor compressor in Configuration B. Scaling also a problem.	Field connection to refrigerant system a problem.	3500* 4600**	 $225	$400 $500
Refrigeration Research, Inc.	Tube-in-Shell	OEM heat exchanger only.	Not a fail-safe design.	Freezing and scaling are a problem.	Field connection to refrigerant system a problem.	2400**		

*Based on raising a stored volume of water from 60°F to 130°F.

**Based on a constant flow of water with 60°F inlet temperature.

weather conditions and use patterns is quite another subject. The field test data is found primarily for the commercial applications of large refrigeration units. Little, if any, residential field test data exist. The only data available is summarized in Table 4.3. Shown here are the claimed energy savings and the supporting field test data which was provided by the manufacturers. The reasons for not having the raw test data were many and varied; but for the most part, manufacturers said that little field test data was ever taken.

Of note is the air conditioner energy savings which occur as a result of the increased heat transfer provided by the HRS. The A/C-HRS effectively adds condenser heat transfer area, reducing the condenser temperature and increasing the A/C unit performance.

4.1.3 System Designs

At present, no package unit for new central air conditioner units exists, and no design optimizing the overall system (A/C and water heater) performance has been developed. Figure 4.4 shows a number of system concepts designed to evaluate the range of possible energy savings from the air conditioner heat recovery system. Rather than being looked upon solely as attempts to improve the existing designs, these schematics were also developed for the purposes of understanding the present systems and their limitations.

The first two schematics are based on the bayonet-in-tank system. The vertical bayonet characterizes the design presently available, while the horizontal bayonet replaces one of the electric resistance elements. The other three schematics are variations on the water circulation systems. The first is typical of the system concept presently in use.

In the system shown by Figure 4.4.b., a thermostatically-operated flow control may be used to maintain the outlet water temperature above a predetermined temperature. This avoids the possibility of returning cool water to the upper part of the water heater, reducing the temperature of the stored delivery water. Other designs do not use a thermostat control flow and rely on having ample water heating between periods of water use, such that the entire tank is raised to the normal use temperature.

Figure 4.4.c. uses a preheat or holding tank. The purpose of the preheat tank is to provide added capacity of water storage and increased stratification of hot and cold water over that which would be achieved in a single tank. In this system, an optional bypass three-way valve could be used to direct heated water when it is above the desired final delivery temperature to the final delivery tank, rather than mixing it with the preheat water storage tank. And finally, in Figure 4.4.d. an additional heat exchanger placed after the air-cooled condenser is incorporated for those applications with low temperature supply water. This additional heat exchanger provides little water heating, but is

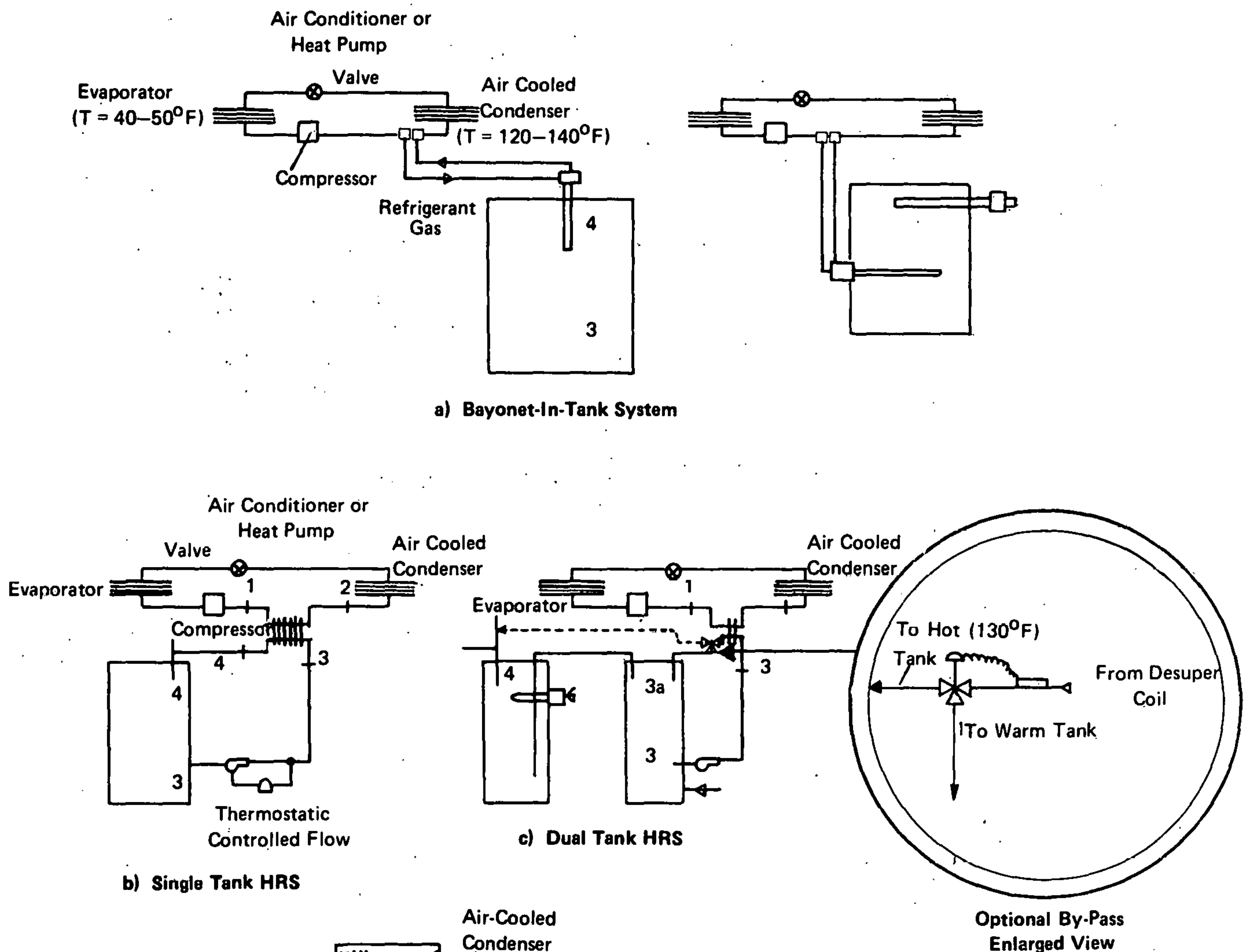
TABLE 4.3

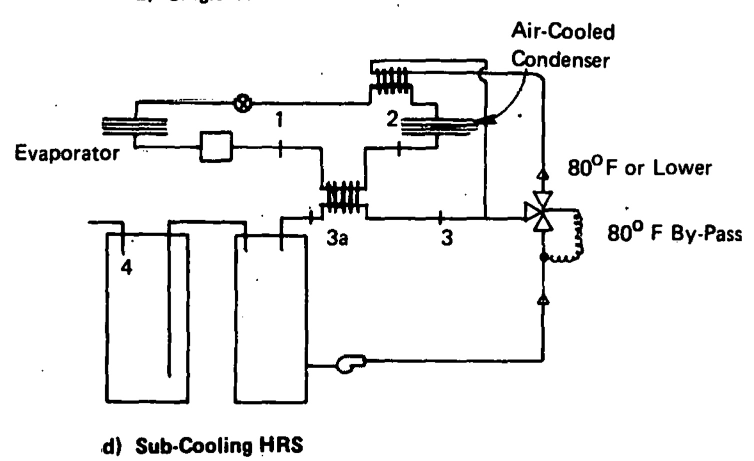
RESIDENTIAL FIELD TEST DATA

			FIELD TEST DATA				
Manufacturer	Configuration	Capacity Btu/hour Water Heating / Ton Air Conditioning	Location	Duration	Water Savings	A/C Savings	Estimated Annual Savings (in kwh)
Clark Energy Saver	Bayonet-in-Tank	1000	Miami, Florida	1 day	2.6 kwh	6.7 kwh	2600*
				1 month			1824
Friedrich Air Conditioning & Refrigeration Co.	Tube-to-Tube	1700	NO FIELD TEST DATA				
Energy Conservation Unlimited	Tube-in-Tube	3500	Anniston, Alabama	6 months winter	1600 kwh	Unknown	-
			Akron, Ohio	6 months	860 kwh	Unknown	1200**
			Lakeland, Florida	7 months	3500 kwh	Unknown	3500**
Sun-Econ	Tube-in-Tube	4600	NO FIELD TEST DATA				

*Estimated by using 2,000 hrs/year of air conditioning on-time

**Water heating savings only.





Typical Temperatures (°F)

Location	No Water Draw	After Water Draw
1	200–220	200–220
2	120–150	100–120
3	100–120	60–80
4	130–140	130–140
3a	110–130	70–90

FIGURE 4.4 SYSTEM SCHEMATICS FOR THE A/C OR HP – HRS

designed to improve the efficiency of the air conditioning unit itself. Additional cooling of the refrigerant at this point in the refrigeration cycle has a substantial effect on improving the system performance. Because the refrigerant is in a liquid state below its condensing temperature at this point and is "subcooled," the additional heat exchanger is called a subcooler. Entering water temperatures below 90°F would be necessary to improve the air conditioner performance. A flow control valve monitoring the cold water from the preheat tank decides whether the cold water is sufficiently low in temperature to provide adequate subcooling to improve the air conditioning efficiency.

4.1.4 Major System Component Designs

The heat exchanger designs conceived for these systems are similar to those presently used by certain manufacturers, with the exception that additional heat transfer surface is used. Schematics of the heat exchanger cross sections are given in Figure 4.5. Copper tubes relying on solder-dipped bonds constitute the major elements of the tube-to-tube heat exchanger used in the water circulation designs. The schematic of the bayonet-in-tube is also shown and differs from the designs presently used by the addition of some tube finning.

4.2 ANALYSIS OF DIFFERENT CONFIGURATIONS

4.2.1 Analytic Approach

Two computer models were developed for the analysis of the energy-savings potential of the air conditioner and heat pump recovery system for water heating. One model is of the air conditioner or heat pump system and relates the effect of the HRS heat removal to the performance of the air conditioner. The model is exercised over a wide range of conditions and the results are subsequently condensed into simple equation form and represent the state equations for the refrigeration unit. The refrigeration unit state equations are used in the second computer model--the A/C-HRS model--which links the water heater with the air conditioner or heat pump and exercises the system on an hourly basis throughout a selected year in a particular city using existing hourly weather tapes.

The two programs were separated, rather than having the refrigeration unit* as a subroutine in the larger heat recovery model. This was done so that the state equations (curves in Figure 4.8, 4.9, and 4.10) are developed once and are not redeveloped each hour requiring tremendous additional computer time.

The air conditioner/heat pump model is a complete heat exchanger and compressor computer model, shown schematically in Figure 4.6.
The link between the condenser and evaporator is typically a thermostatically-controlled expansion valve or a capillary tube. In this analysis, the system balance is achieved with a fixed 10°F superheat

*A complete description of the model is given in Appendix E.

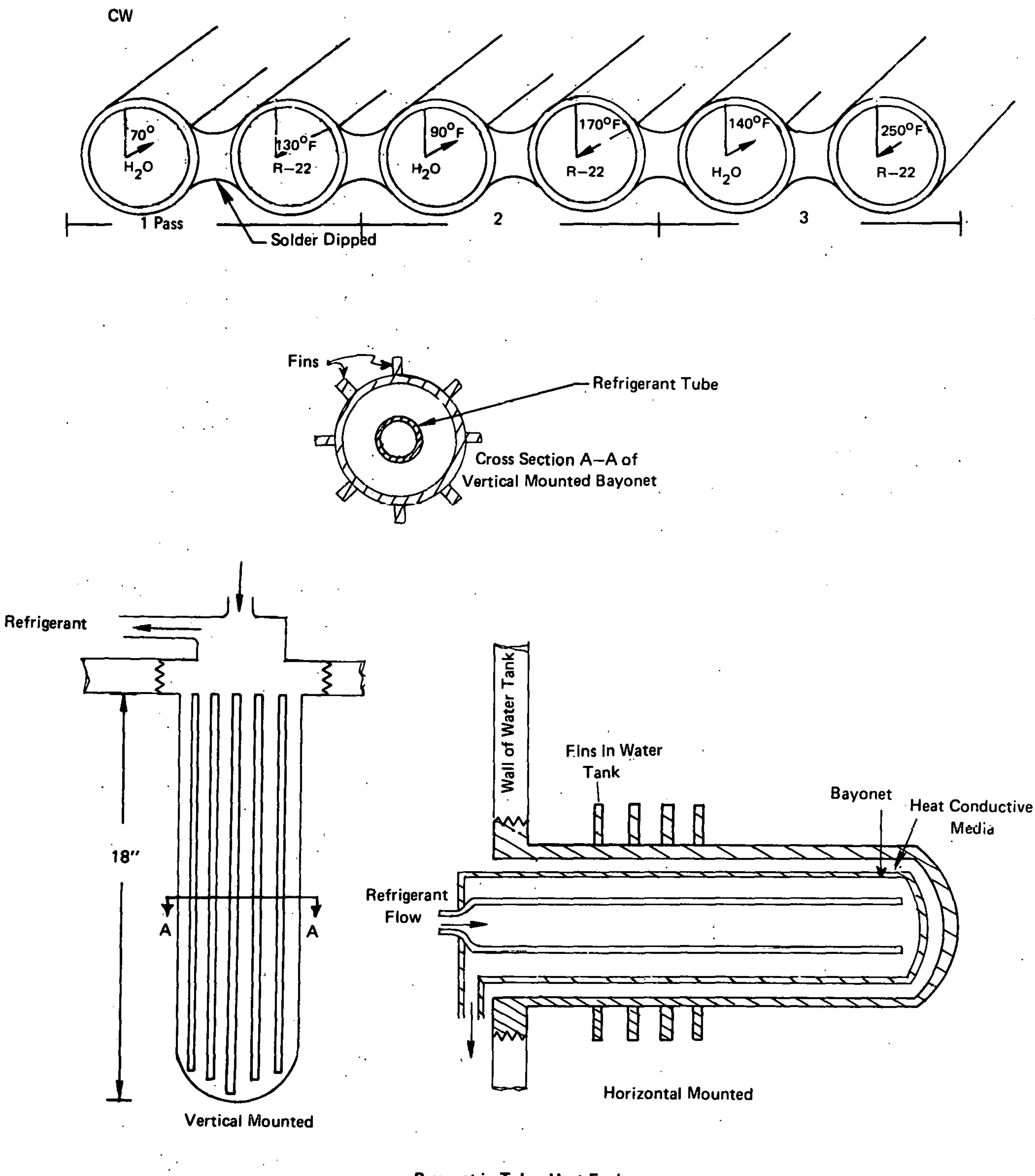


FIGURE 4.5 HEAT EXCHANGER DESIGNS FOR A/C-HRS

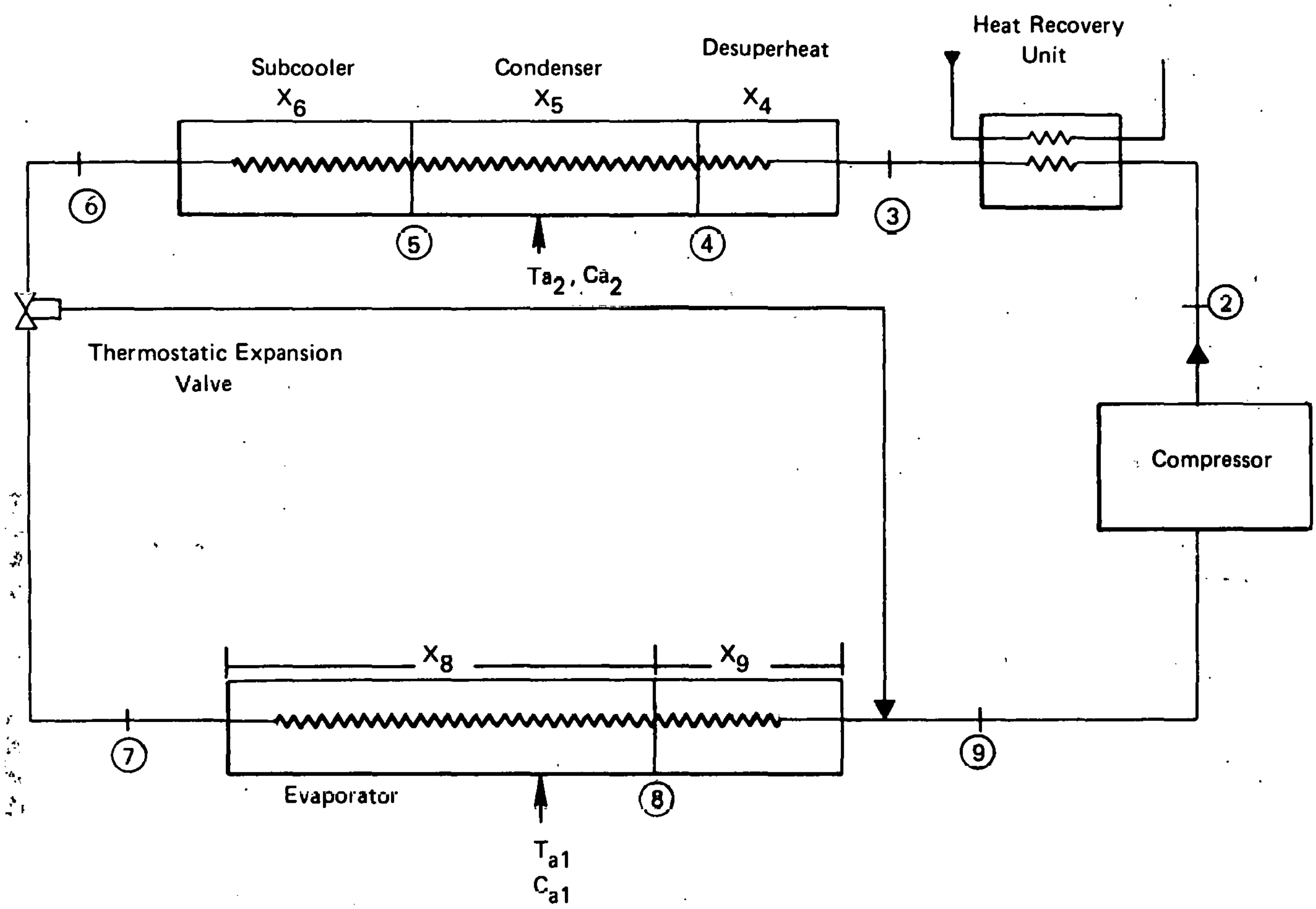


FIGURE 4.6 AIR CONDITIONER/HEAT PUMP MODEL

from the evaporator, reproducing the control effect of a thermostatic expansion valve. In actual practice, the thermostat control valve is able to control the evaporator superheat (amount by which the gas is heated above the evaporating temperature of the refrigerant) quite close to 10°F by modulating the throttle opening. For fixed outdoor air and indoor air conditions, the program exercises the system over most conditions that satisfy the coil and compressor characteristics. The results of a run with an 85°F outdoor condition and 75°F indoor condition are shown in Figure 4.7.

The comparative analysis of the heat recovery systems could best be made by assuming that the expansion device provides the highest permissible EER (ratio of cooling capacity in Btu/hr to total unit input in watts) at any condition both with and without the heat recovery system. Figures 4.8, 4.9, and 4.10 are graphs of the conditions of maximum EER as a function of the inlet temperature, T3, to the condenser and represent the state equations used to characterize the air conditioner in the second computer model--the A/C HRS model.

It is highly unlikely that the capillary expansion tube or thermostatic control valve presently used for residential air conditioners will seek the optimal condition assumed here when operated with a HRS. Discussions with manufacturers indicate that further analysis and engineering will be required to achieve this kind of control function.

The A/C-HRS model used to analyze the heat recovery unit is shown in Figure 4.11. Two water storage tanks are connected so that they may be used for analysis of both preheat and single-tank configurations. When used as a single tank, the flow schematic is that shown in Figure 4.12. Here the separation of the tank approximates the stratification that would take place in a single tank.

When heat recovery water delivery temperatures are equal to or exceed the water temperature contained in the upper portion of the single tank, the water from the heat recovery system is put into the final tank. This approximates the interchange between the stratified layers of a single storage tank.

When using a preheat tank, the flow is as shown previously in the schematic of the model, Figure 4.11.

4.2.2 Parametric Analysis of System

The purpose of the following parametric analysis is to evaluate the potential energy savings and first cost of the air conditioner/water heater concept, so that a design with the minimum years to payback can be identified and used as a target for a National Demonstration of the concept. The approach was to select a climatic zone based on air conditioning hours representative of the United States and examine the energy savings and life-cycle cost of a variety of system designs discussed earlier. With air conditioner population breakdowns by climatic zone projected to 1990, an estimate of the potential energy savings for the nation was developed.

Condenser Temperatures (°F)		Fraction of Condenser Area			Evaporator Temperatures		Evaporator Fraction of Area		Refrigerant Mass Flow		Refrigerant Temperature	
Condensing T4	Subcooling T6	Superheat X4	Condensing X5	Subcooling X6	Evaporating T7	Superheat F9	Evaporating X8	Superheat X9	#/hr W	System Watts EIN	Into Condenser T3	EER
117.3540	115.5624	0.1171	0.8750	0.0079	42.5001	52.5001	0.9685	0.0315	543.9543	4669.6528	200.00	7.654
117.4495	114.4425	0.1165	0.8700	0.0135	42.4001	52.4001	0.9687	0.0313	542.5521	4668.7207	200.00	7.680
117.5509	113.3137	0.1160	0.8647	0.0193	42.3001	52.3001	0.9689	0.0311	541.3385	4667.9224	200.00	7.705
117.6627	112.1756	0.1154	0.8592	0.0254	42.2001	52.2001	0.9691	0.0309	540.0128	4667.3599	200.00	7.730
117.7838	111.0259	0.1147	0.8535	0.0318	42.1001	52.1001	0.9693	0.0307	538.6733	4667.0107	200.00	7.764
117.9166	109.8655	0.1141	0.8474	0.0385	42.0001	52.0001	0.9695	0.0305	537.3181	4666.9243	200.00	7.778
118.0605	108.6925	0.1134	0.8409	0.0457	41.9001	51.9001	0.9697	0.0303	535.9449	4667.0898	200.00	7.802
118.2212	107.5061	0.1127	0.8341	0.0532	41.8001	51.8001	0.9698	0.0302	534.5516	4667.6401	200.00	7.825
118.3976	106.3047	0.1119	0.8268	0.0613	41.7001	51.7001	0.9700	0.0300	533.1343	4668.5439	200.00	7.847
118.5945	105.0863	0.1111	0.8190	0.0699	41.6001	51.6001	0.9702	0.0298	531.6894	4669.9150	200.00	7.869
118.8144	103.8486	0.1102	0.8106	0.0793	41.5001	51.5001	0.9704	0.0296	530.2112	4671.8071	200.00	7.890
119.0632	102.5886	0.1092	0.8014	0.0894	41.4001	51.4001	0.9706	0.0294	528.6931	4674.3545	200.00	7.909
119.3471	101.3019	0.1081	0.7914	0.1005	41.3001	51.3001	0.9708	0.0292	527.1253	4677.7026	200.00	7.927
119.6751	99.9828	0.1069	0.7803	0.1128	41.2001	51.2001	0.9710	0.0290	525.4946	4682.0527	200.00	7.944
120.0642	98.8225	0.1056	0.7678	0.1267	41.1001	51.1001	0.9712	0.0288	523.7816	4687.7974	200.00	7.958
120.5336	97.2074	0.1040	0.7533	0.1426	41.0001	51.0001	0.9714	0.0286	521.9548	4695.3726	200.00	7.969
121.1220	95.7134	0.1022	0.7362	0.1616	40.9001	50.9001	0.9716	0.0284	519.9595	4705.6709	200.00	7.975
121.9115	94.0925	0.0999	0.7146	0.1855	40.8001	50.8001	0.9718	0.0282	517.6854	4720.5776	200.00	7.974
123.1194	92.2161	0.0965	0.6843	0.2193	40.7001	50.7001	0.9721	0.0279	514.8229	4745.0996	200.00	7.966

FIGURE 4.7 PERMISSIBLE CONDITIONS OF THE A/C COMPONENTS
OUTSIDE AIR: 85°F INSIDE AIR: 75°F

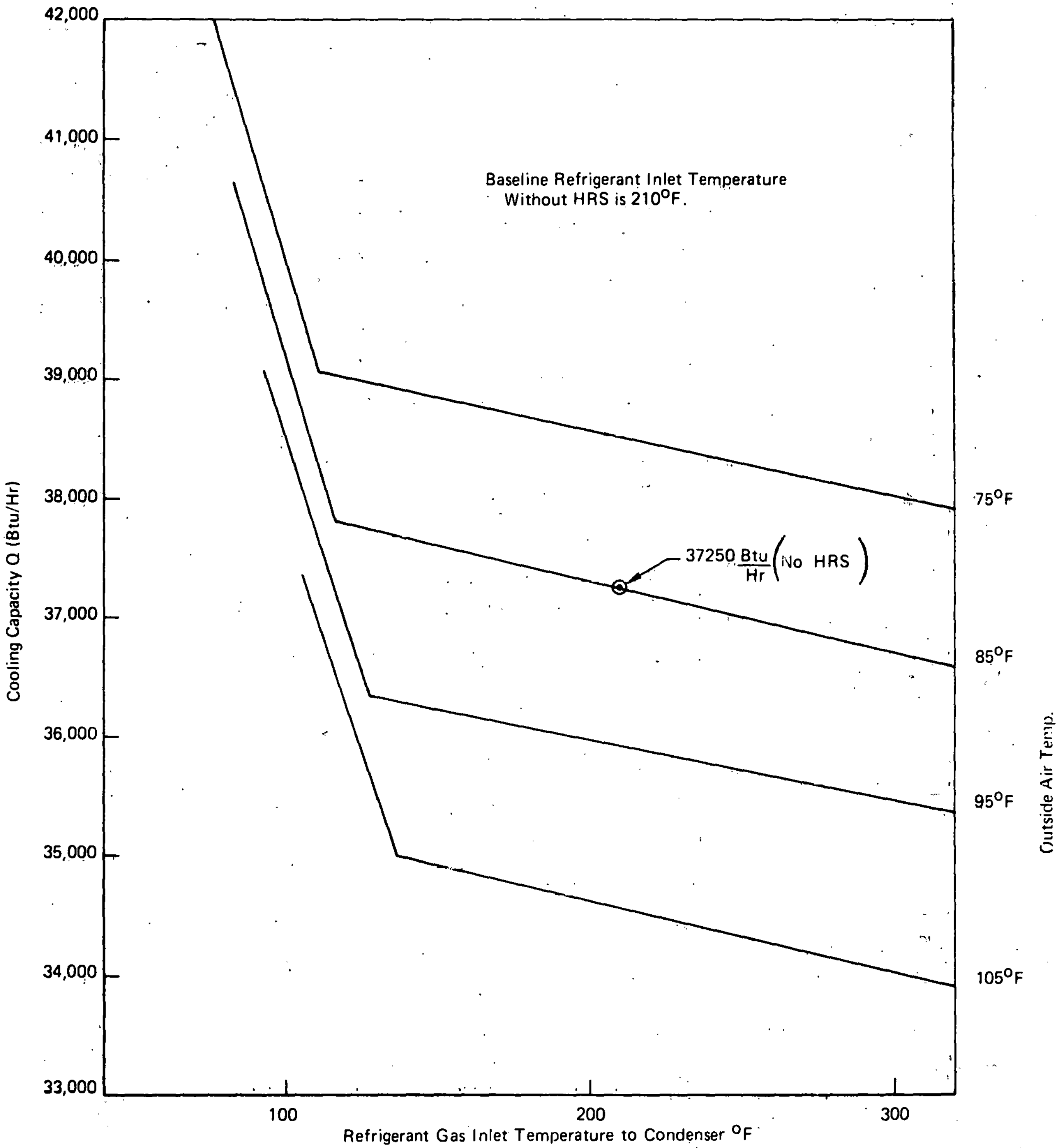
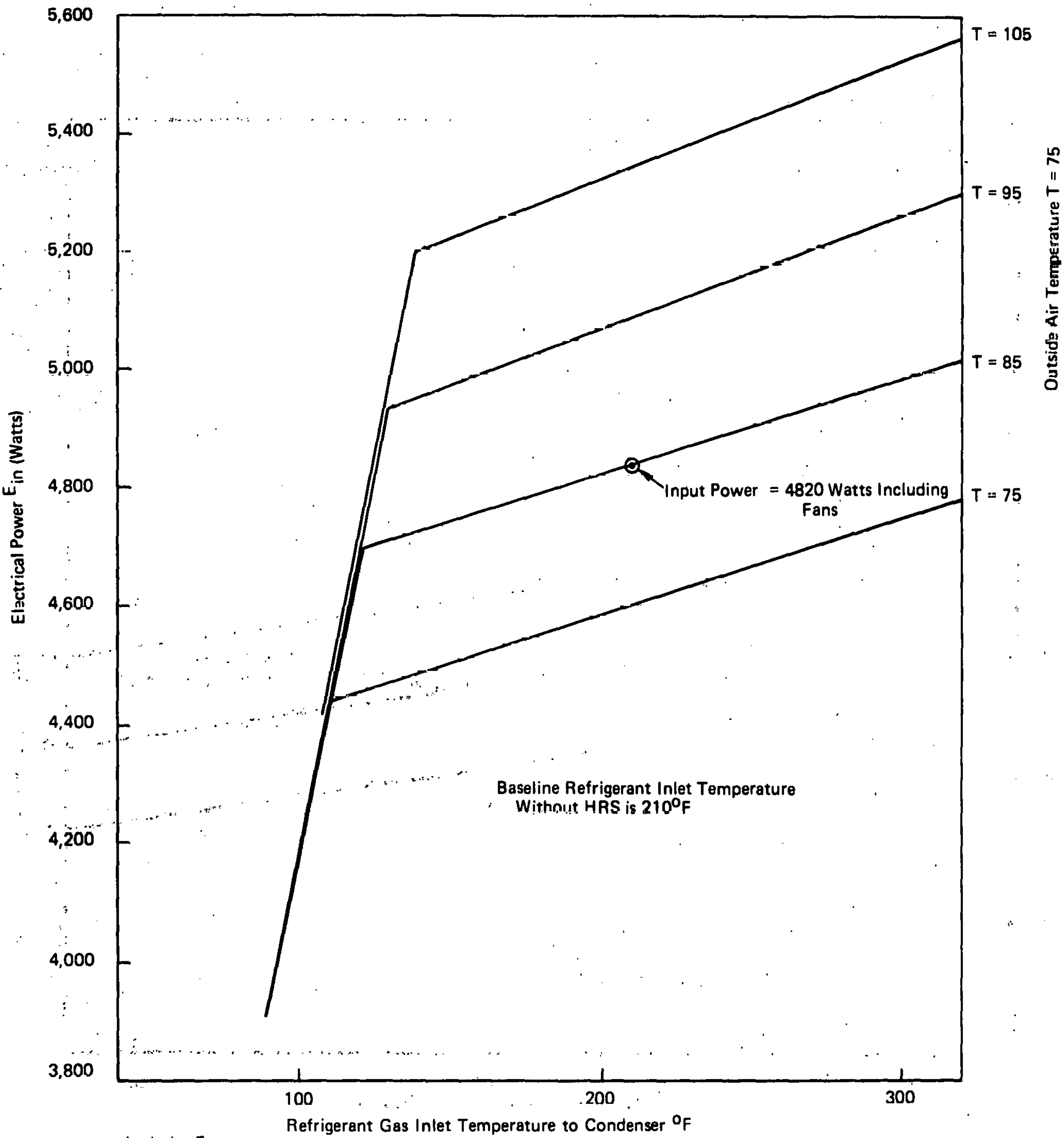


FIGURE 4.8 COOLING CAPACITY VS. REFRIGERANT INLET TEMPERATURE TO CONDENSER



+Includes Fans

FIGURE 4.9 ELECTRICAL POWER* VS. REFRIGERANT INLET TEMP. TO CONDENSER

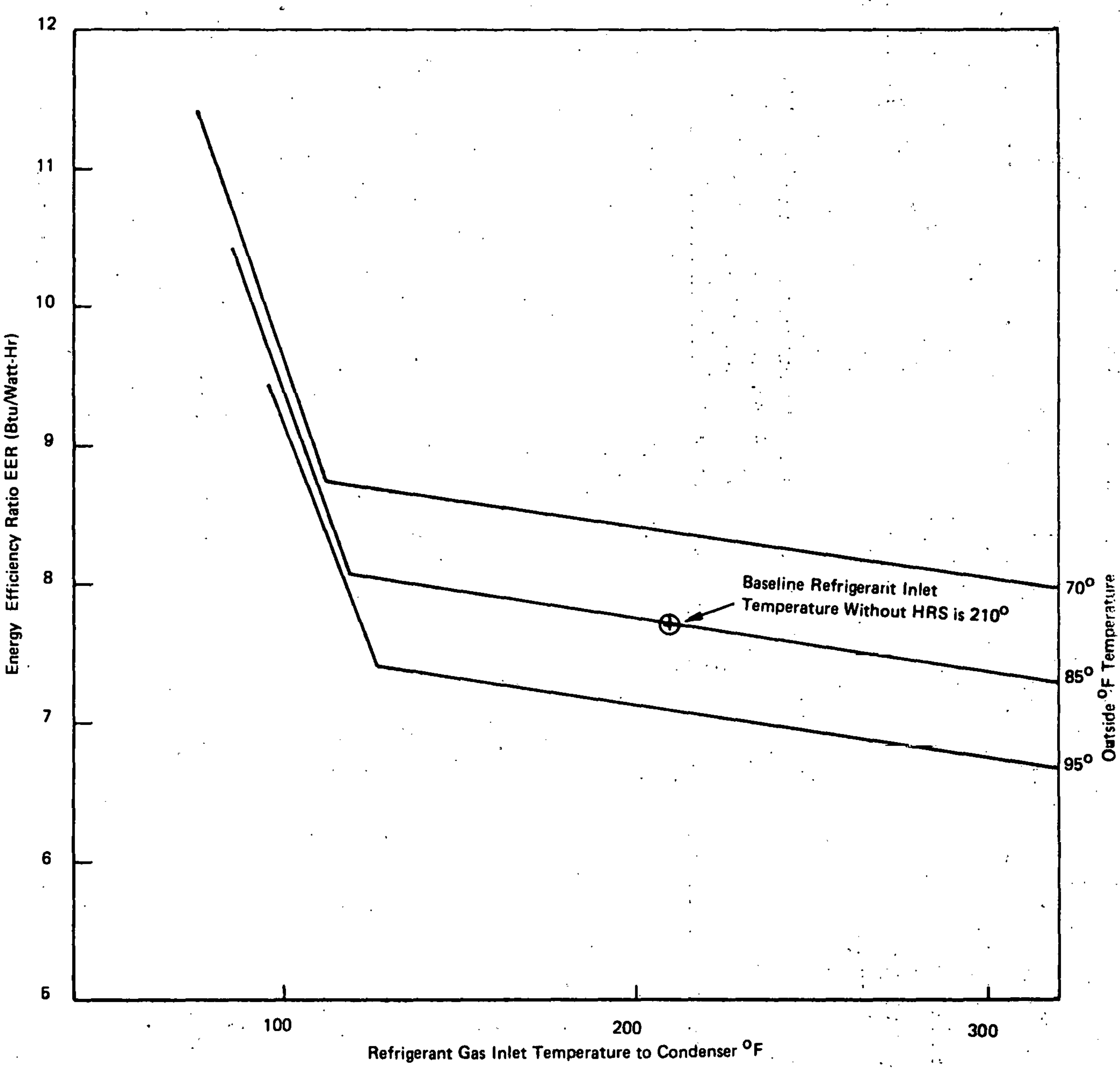


FIGURE 4.10 ENERGY EFFICIENCY RATIO VS. REFRIGERANT INLET TEMPERATURE TO CONDENSER

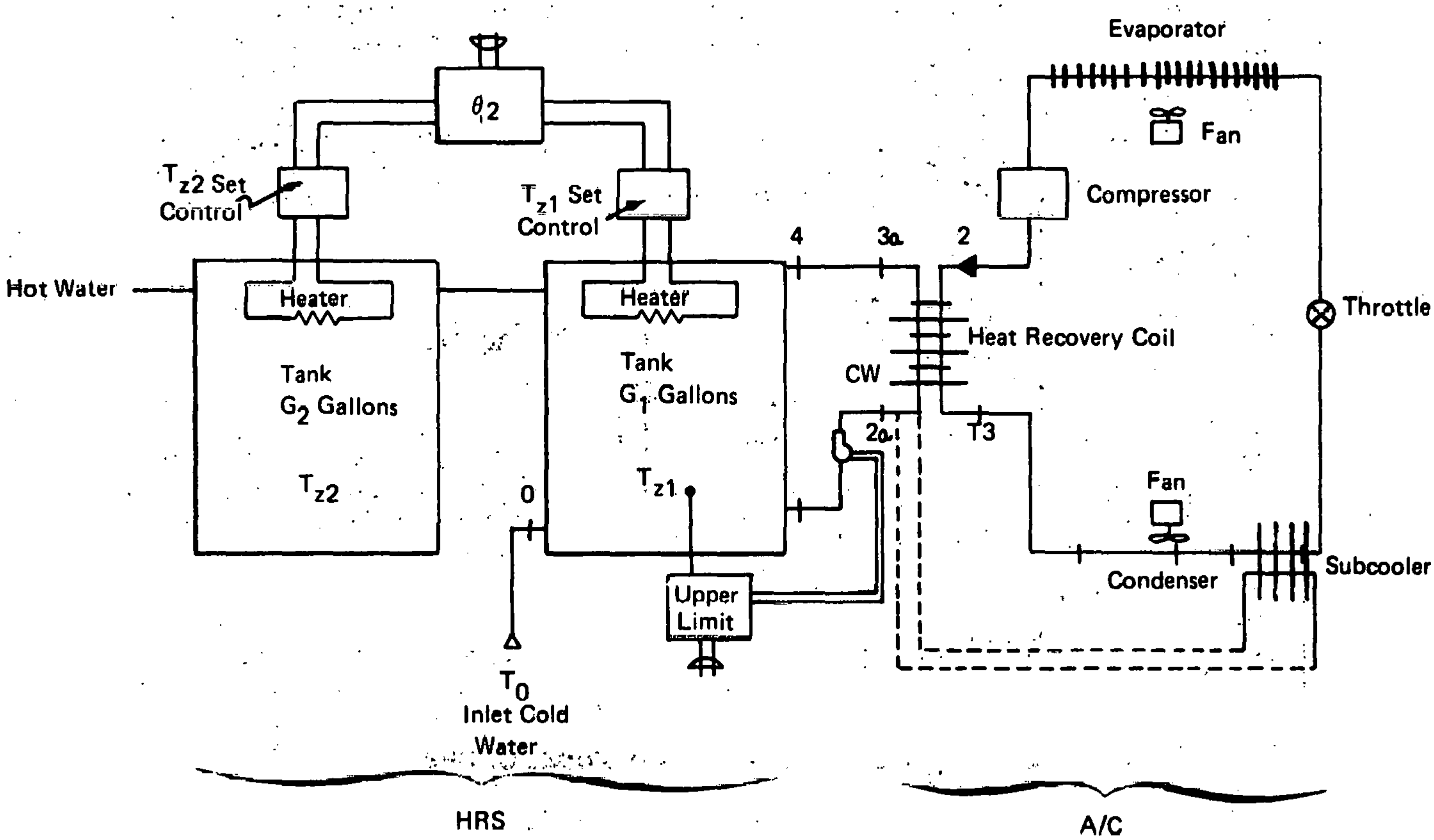


FIGURE 4.11 A/C–HRS SYSTEM MODEL

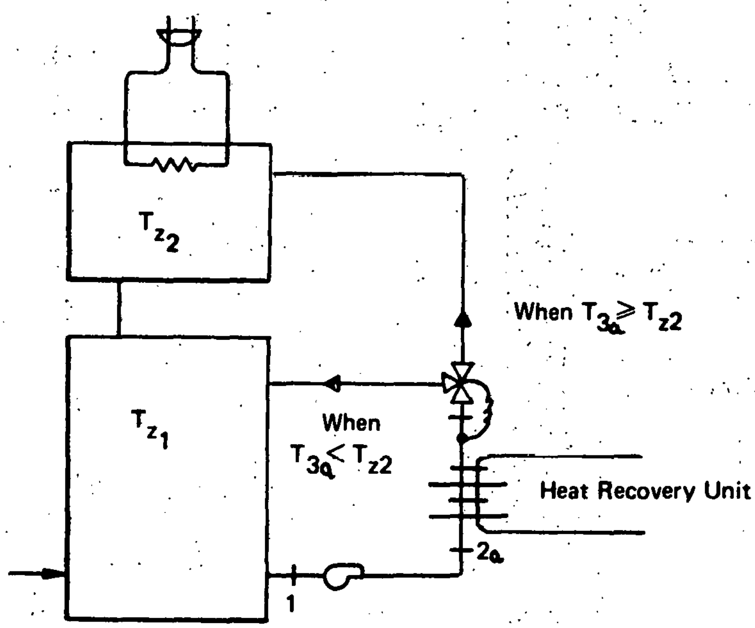


FIGURE 4.12 TWO ZONE SINGLE TANK MODEL OF HRS

The breakdown of population of air conditioners by climatic region is shown in Figure 4.13.

The costs of systems discussed in Section 4.2 (see Figure 4.4) are shown in Table 4.4. These are based on a minimum production rate of 500,000 units per year of OEM models and 50,000 units per year of retrofit unit models. Several distribution paths to the user were considered; the extremes for one of the systems are shown in Table 4.5, suggesting the great uncertainty of the final cost to the users. Depending on the distribution channels, the cost to the consumer may vary by as much as $160. The cost used in this analysis is based on the factory cost times a 2.5 markup plus installation costs. The same formula was used to analyze the costs of the other promising candidates (Chapters 5 and 6).

While the bayonet-in-tank system has merit, it was not considered further in the parametric analysis because it could not be equipped as OEM equipment on an air conditioner. The goal of the parametric analysis was to establish an energy-saving target for a National Demonstration Plan, and it was felt that the design optimization of the system, and hence the Demonstration, should begin with an OEM package which would provide a better proving ground for the concept than a retrofit as discussed in Section 4.3, National Demonstration Plan.

Because of the complexity of the system with subcooling, it was not analyzed, but rather deferred to the Demonstration Phase where more detailed design analysis could be done in conjunction with a manufacturer.

The parametric analysis concentrated on the Single Tank System (Figure 4.4.b.). Heat exchanger and tank sizes were the most notable variations, and their effect on the system cost effectiveness (years to payback) is shown in Table 4.6 following.

The Single Tank System with different heat exchanger sizes has an optimum around a UA of 165 Btu/hr °F for the 3.5 ton air conditioner used in this analysis. The reason for this optimum can be seen in Figure 4.14 which shows the effect of increased UA on the recovered heat. Clearly, heat exchanger area beyond 150 to 200 is not worthwhile. A more comprehensive treatment of this section is given in Appendix D.5.

The Dual Tank System was found to be less attractive than anticipated as a result of its increased surface-to-volume ratio over the Single Tank System, resulting in greater standby loss*. It also had a slightly higher installed cost than the Single Tank System.

The Single Tank System with a UA = 165 was used in the computer model in different climatic zones so that a nationwide average energy-savings potential of the concept could be estimated. The predicted performance in different regions is shown in Table 4.7.

*This may not be the case for a tank with greater insulation. These further design questions are left to the Demonstration Phase.

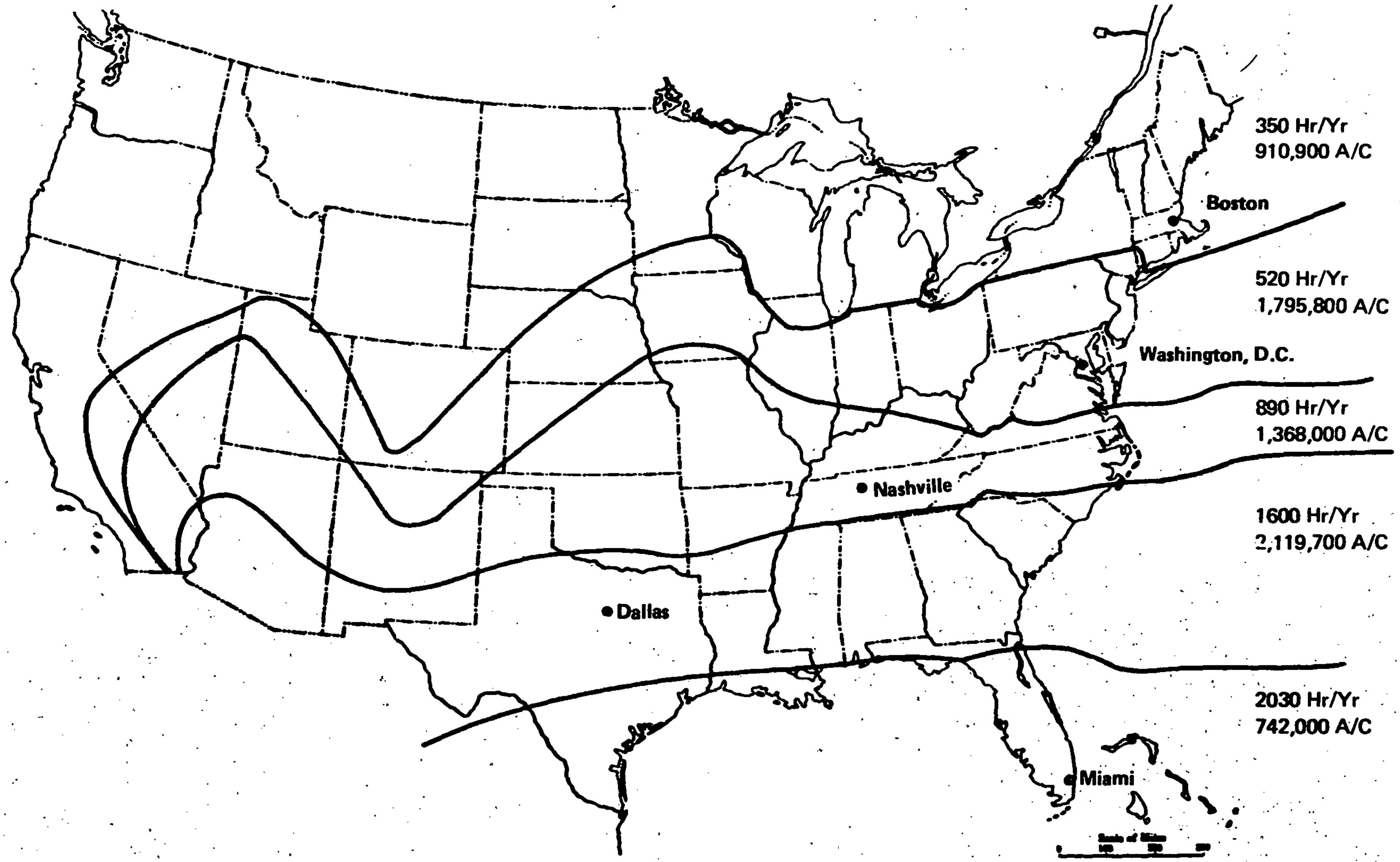


FIGURE 4.13 POPULATION OF CENTRAL A/C BY REGION

TABLE 4.4

ESTIMATED ADDED FIRST COST
(Based on 500,000 units/year OEM;
50,000 units/year Retrofit)

Configuration	Bayonet-in-Tank Figure 4.4.a.	Single Tank Figure 4.4.b.	Two Tank Figure 4.4.c.	Single Tank Figure 4.4.b
Type of Installation	Retrofit	New	New	Retrofit
Extra Factory Cost	$ 45	$106	$116	$101
Installed Extra Cost to the Consumer (2.5 x factory cost + installation)	$160	$300	$334	$316

TABLE 4.5

RANGE OF ADDED FIRST COST OF A/C-HRS (SINGLE TANK SYSTEM) (Based on 500,000 units/year OEM; 50,000 units/year Retrofit)

Installed Cost Used in this Study	$300
Extra Factory Cost	$106
Cost Installed by:	
Large Builder	$181
HVAC Contractor	$221
Small Builder	$252
Local Installer	$340

TABLE 4.6

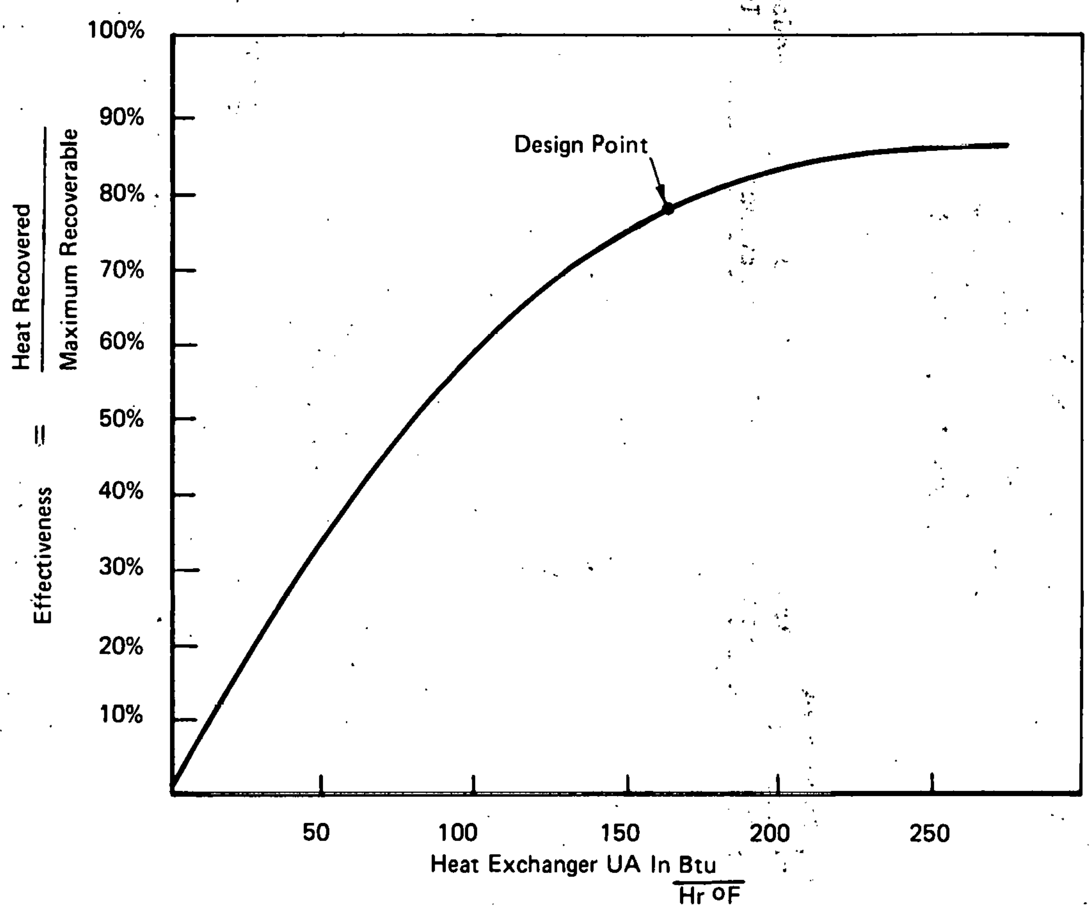
PARAMETRIC ANALYSIS FOR WATER CIRCULATING SYSTEMS IN NASHVILLE, TENNESSEE

	Heat Exchanger Size UA in Btu/hr °F	Tank Size in Gallons	Annual kwh/Year Savings	Added First Cost Installed	Years to Payback
Single Tank	100	60	1945	$285	3.6
	165	60	2340	300	3.2
	165	120	1720	475	6.9
	220	60	2464	330	3.3
	500	60	2553	465	4.5
Dual Tank	165	Two 30	1056	326	7.7

Baseline Model Prediction
(No Heat Recovery)

	Water Heater	Air Conditioner
Size	60 gal. electric	3.5 ton
Duty	70 gal/day drain	883 hrs/year compressor operation
Energy Consumption*	24.8×10^6 Btu/year	14.0×10^6 Btu/year

*The reader should note that the predicted energy consumption for the baseline water heater is about 15% higher than the national average value used in the energy appliance inventory, Table A.1 used in the screening. The predicted air conditioner energy consumption is about 8% higher. The discrepancy between the model predicted values and the average national values is within the accuracy margin (± 20%) of the national average figures.



Basis: $\dot{m}C_p$ (Water Side) = 440 $\frac{\text{Btu}}{\text{Hr}\ ^\circ\text{F}}$

$\dot{m}C_p$ (Refrigerant) = 110 $\frac{\text{Btu}}{\text{Hr}\ ^\circ\text{F}}$

FIGURE 4.14 VARIATION OF EFFECTIVENESS OF HEAT TRANSFER AREA

TABLE 4.7

A/C-HRS ENERGY SAVINGS IN
DIFFERENT CLIMATIC ZONES -
(Cooling Season Only)

Zone Number	Representative City	Baseline Total Annual Kwh A/C and Water Heating Without A/C HRS[1]	Cooling Compressor Hours	% 1990 Population of A/C Units[2]	Annual Point of Use Savings (kwh/year) Total	A/C Portion	Annual Primary Savings (mm Btu/year)	Years to Payback (Added First Cost: $300) Electric Water Heating	Gas Water Heating[3]
1	Maimi	16,930	2,030	15	4,271	300	48.5	1.7	4.2
2	Ft. Worth	15,230	1,700	35	3,067	100	34.9	2.4	7.6
3	Nashville	11,400	890	16	2,340	70	26.6	3.2	10.0
4	Washington	9,700	520	26	1,330	60	15.1	5.6	17.0
5	Boston	8,893	350	8	1,300	34	14.77	5.7	16.5
1990 Inventory Weighted Average		13,000			2,538[4]		28.8	3.5	10.6

[1] Electric water heating is 7,260 kwh/year.

[2] Arthur D. Little, Inc. estimates based on projected trends in new housing starts and historical data on sales to existing homes.

[3] Based on the same amount of water heated by air conditioner as the electric water heater plus the 25% credit for the gas recovery efficiency of 80%.

[4] Using heat pumps (heating and cooling), the annual savings could be raised to 4,200 kwh (65% increase), and the years to payback reduced to 1.8 (a 50% reduction). It is anticipated, though, that only 1 out of 5 air conditioners will be heat pumps in 1990.

The potential nationwide savings of the concept based on the cooling mode only is shown in Table 4.8.

Additional savings could be obtained if reduced size air conditioners were used. With lower outdoor balance points (95°F was assumed here), the compressor run times would increase and the heat recovery would increase. This would be particularly important in the transition months of spring and fall.

4.3 NATIONAL DEMONSTRATION PLAN

4.3.1 Introduction

A number of barriers to rapid commercialization of the air conditioner heat recovery system (A/C-HRS) exist which, if successfully removed by this proposed development and demonstration plan, could accelerate the energy savings of this concept.

The key barriers can be grouped in the major categories of technical, institutional, and promotional. Technical barriers to widespread commercialization of the concept fall in two areas. These are: the absence of definitive field test studies demonstrating the energy-savings potential under real use pattern conditions, and limited analysis and development in the following areas:

1) Optimum refrigeration unit (air conditioner or heat pump) control (throttle; condenser fan speed) to maximize overall system efficiency.

2) Automatic freeze protection systems for outdoor water-filled lines.

3) Selection of HRS size for different climatic zones, water use patterns, compressor/water heating sizing.

4) Trade offs for selecting temperature below which the water heating unit is discontinued during heat pump operation.

Institutional factors inhibiting the commercialization of the concept stem from the absence of accepted test procedures for the heat recovery unit. Rating and specification of the units is not possible without a standard test procedure, and this retards the widespread use of the device. In addition, Federal and state energy conservation regulations do not recognize the large energy-savings potential of the A/C-HRS as part of A/C efficiency standards. This fact alone severely limits the acceptance of the product.

A government-sponsored and monitored demonstration program in cooperation with the Electric Power Research Institute, could enhance the credibility of the A/C-HRS.

TABLE 4.8

OVERVIEW OF INTEGRATED A/C-HRS APPLIANCE
(Cooling Season Only)

Candidate	Energy Savings 10^6 Btu Primary Energy Per Unit	Added First Cost Installed	Years to Payback	Max. 1990 Inventory Applicable 10^6 Units	1990 Annual National Potential Energy Savings - Primary 10^{14} Btu/year
Central A/C Heat Recovery for Electric Water Heaters	28	$300	3.5	10.6	3

Basis for Projections

	1980	1990
Residential units (single family, mobile home, low density condo)	57,500,000	68,000,000
• With Central A/C	13,300,000	25,814,000
• With Central A/C and Electric Water Heater	4,300,000	10,557,000
Annual Sales of Central A/C		
• To All Residential	2,100,000	Unknown
• To Single Family	1,173,000	2,346,000
• To Single Family with Electric Water Heater	511,000	1,103,000

4.3.2 Recommended Demonstration Plan

The major features of the recommended National Demonstration are shown diagrammatically in Figure 4.15. A brief discussion of the Plan follows.

The focus of the National Demonstration Plan should be on the heat recovery system as OEM equipment on air conditioners and heat pumps for the following reasons:

1) The OEM system (see Figure 4.16) minimizes the risk of field contamination or incorrect charging of the refrigerant system since only the water lines are field connected.

2) The OEM system allows design trade offs involving the air conditioning system. The air conditioner size, controls, and heat exchanger can be chosen to take advantage of the A/C-HRS.

3) The OEM system is the one which falls into the jurisdiction of efficiency standards and is the version that will gain from being incorporated into the air conditioner standards.

4) The OEM system can be commercialized through the existing A/C distribution channels (2 million sold per year), whereas the retrofit distribution and advertising channel will take years to build to the level of the OEM market.

Based on the analysis in Phase I, designs for OEM package heat recovery units (see Figure 4.16 following) which maximize energy savings and have payback of 2-3 years should be developed in Task 1 for different climatic zones. Control schemes, the impact of component sizing, and system designs for extending the system applicability should be considered.

Tests of the promising designs should be made and after satisfactory performance from the units has been achieved in the laboratory, field demonstration units manufactured and installed in Task 2.

Field surveillance (Task 3) of the performance of the devices should be conducted and analysis of the data, including comparisons with predicted performance, should be made. At the end of the surveillance period which will last approximately one calendar year, an electric utility workshop should be held, designed to transfer the findings of the demonstration and to encourage utility promotion of the A/C-HRS.

Recommended test procedures for the A/C-HRS should be developed in Task 4 and a co-sponsored EPRI-ERDA workshop for electric utilities should be undertaken in Task 5.

Key areas of the National Demonstration Plan are discussed below.

TASK 1

Development of Unit Design

- OEM A/C-HRS package
- By climatic zone
- Largest energy savings within 2-3 year payback
- Laboratory test

TASK 2

Field Installation

- Fabricate field test units; assemble test instruments
- Utility customer contract
- Install units

TASK 3

Field Surveillance and Data Analysis

- Monthly meter reading
- Data Analysis

TASK 4

Recommended Tests and Standards

- Test procedure for A/C HRS
- Efficiency standards for FEA-NBS programs

TASK 5

ERDA-EPRI Utility Workshop

- Results of field demonstration
- Recommended advertising

FIGURE 4.15 DEMONSTRATION PLAN

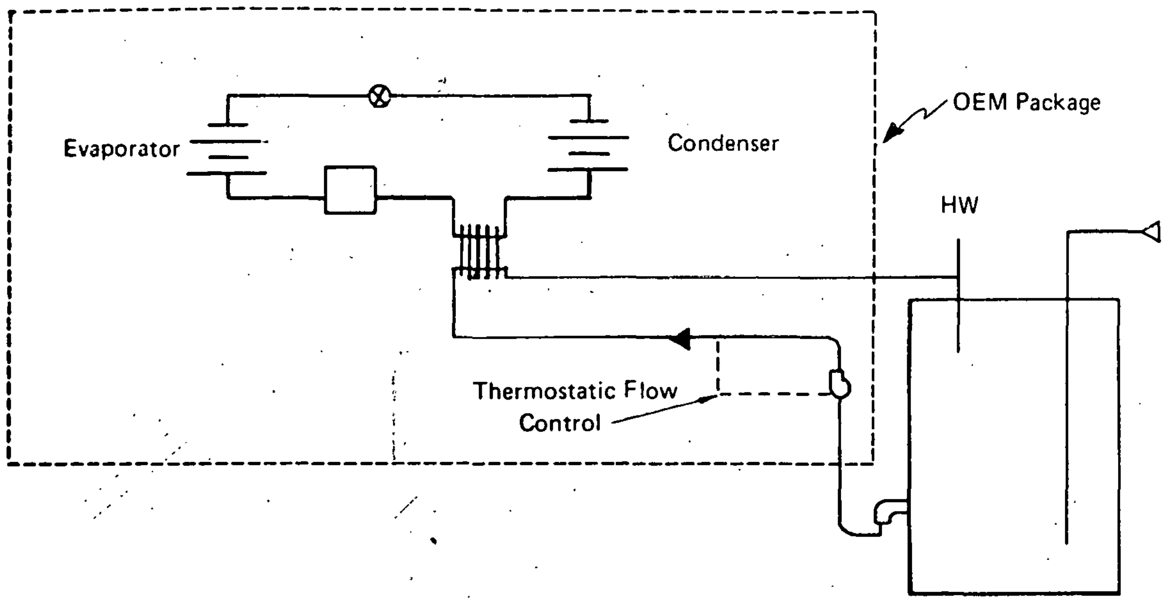


FIGURE 4.16 OEM PACKAGE OF THE A/C-HRS (INSULATED WATER TRANSFER LOOP) TYPE

Field Data

There are no definitive field tests of the available devices which document the energy savings and conditions under which the savings were achieved. In discussions with manufacturers, we are led to believe that they are unlikely to undertake field demonstration at their own expense for several years. Discussions with utilities does not indicate much further independent work. Without federal or state-sponsored tests, field demonstration to quantify the actual energy savings of the devices is unlikely to be undertaken by industry and/or utilities for several years. Although laboratory tests under ideal conditions can be used for estimating the performance of the device under limited conditions, reputable manufacturers recognize that actual field performance of the devices may differ widely from the laboratory test taken at fixed climatic and use pattern conditions.

Improved Designs

The following improvements appear to have some promise and should be explored in the Demonstration Phase:

- Optimal controls - Condensor fan speed control, throttle control, water pump speed control. Designed to take advantage of the additional condensing capacity resulting from the HRS. These controls will improve the overall system efficiency.

- Subcooling heat exchanger - For improving air conditioner efficiency.

- Component sizing trade offs - For different climatic zones and hot water usage patterns.

System Costs

Our cost analysis of the system components and system designs examined earlier indicates substantial cost reduction through increased manufacturing scale. Additional cost reductions can be achieved if streamlined distribution chains can be implemented whereby the final user/homeowner is able to purchase the device from a utility or other regional entity. A summary of the installed cost of new devices was shown in Table 4.4. The possibility of setting up product distribution chains through local utilities at reduced markup should be investigated.

Industry Standards

There are no industry-wide accepted standards for testing and rating of the heat recovery units. We believe that this inhibits the use of the device, particularly as it relates to Federal and state efficiency standards for new air conditioner and heat pump efficiencies. Since the purpose of the efficiency standards are to save the nation energy and since the annual energy consumption in the home will be reduced through the use of a heat recovery unit, an HRS system equipped as an OEM package should be part of the test procedure and efficiency relation.

Schedule

The National Demonstration Plan could take place as shown in Figure 4.17.

Estimated Costs for Demonstration Plan

Based on discussions with manufacturers and our own estimates of manpower requirements, we judge that the total program cost (industry and ERDA) to complete the Demonstration Program will be $200,000 with the emphasis of the program effort broken down as follows:

Task	Recommended % of Program Effort
Development	20
Demonstration	65
Public information dissemination	15

4.4 POTENTIAL BENEFIT OF NATIONAL DEMONSTRATION PLAN

The potential benefit of the National Demonstration Plan comes in two areas. The first is that it will accelerate the development and manufacture of optimum A/C-HRS units with higher energy savings than would be achieved without ERDA support. We estimate a 40% increase in energy savings over units likely to be made available without ERDA support. Secondly, the Demonstration Plan could accelerate the distribution and sales of the units by two to three years by managing to incorporate the HRS as part of the air conditioner efficiency test under Federal standards for efficiency. The benefit of the ERDA-sponsored program can be measured in terms of the Cumulative 1980 to 1990 National Energy Savings due to an ERDA-sponsored demonstration. The calculation of this savings is as follows:*

*The same relationship is used in Chapters 5 and 6 to evaluate the other two candidates.

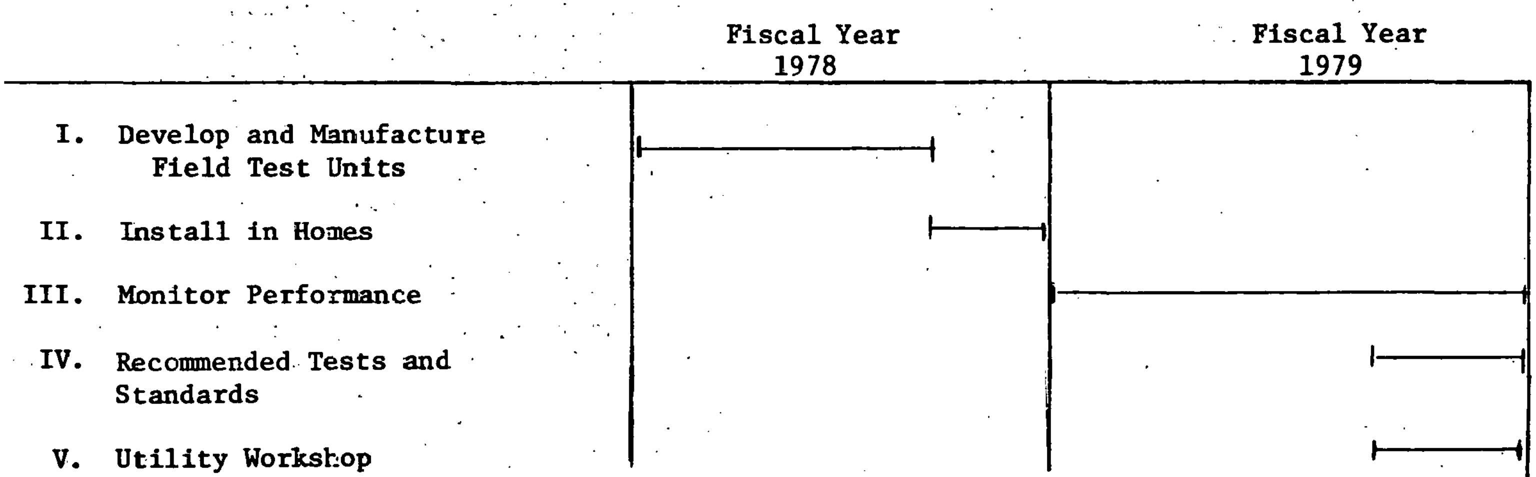


FIGURE 4.17 SCHEDULE OF NATIONAL DEMONSTRATION PLAN

Cumulative National Energy Savings =

$$\sum_{j=0}^{5} \; \sum_{i = \text{first year introduced}}^{1990} \begin{array}{l} (\text{OEM sales} + \text{retrofit sales}) \\ \times\ (1990 - i)_j \\ \times\ (\text{per unit energy savings})_j \end{array}$$

where:

i = year

j = climatic zone 1, 2, 3, 4, or 5

OEM sales = $(\alpha + \frac{\beta}{\text{YTPB}})$ x air conditioner sales x acceleration

Retrofit sales $= \frac{\gamma}{10}$ x (years from start of sales)
x (total number of air conditioners and heat pumps in place without a HRS)
x (acceleration)

α and β are constants (typically, $\alpha = .05$ and $\beta = .60$) see Section 2.3.2 for complete definition of α, β, and γ)

YTPB is Years to Payback

Acceleration is the fraction of the full manufacturing capacity of the product for each year

γ is a fraction reflecting the fraction of consumers that will purchase the retrofit product after ten years of being on the market. ($\gamma = \alpha_{ret} + \frac{\beta_{ret}}{\text{YTPB}}$ as shown in Section 2.3.2.)

The constants α, β, and γ are measures of the consumer acceptance of the device. The "acceleration" is a measure of the ability of manufacturers to produce the devices in a specific year. This is paced by engineering and manufacturing lead times and may be shifted or accelerated by government support.

Of the fraction of air conditioners offered with OEM fitted recovery units, consumers will purchase a fraction f based on the years to payback where

$$f = \alpha + \frac{\beta}{\text{YTPB}} \qquad \text{YTPB} \geq 1$$

$$f = \alpha + \beta \qquad \text{YTPB} < 1$$

Alpha (α) represents the innovation market--consumers willing to purchase the new device independent of the economics--and is typically .05; $\beta + \alpha$ is the maximum consumer acceptance of those purchasing air conditioners that will purchase one with heat recovery if the payback is one year or less.

The estimate of the acceleration profile with and without an ERDA-sponsored program is a judgment based on discussions with potential manufacturers. The acceleration profile used, along with the profile of the total A/C sales to homes with electric water heaters, is given in Table 4.9.

In our judgment, an optimistic scenario would be that an ERDA-sponsored program would encourage 20% of the owners of A/C units in the year 1990 (2% in 1980) to purchase retrofit devices, and 22% of those buying A/C units in 1990 to purchase OEM-fitted devices at an added first cost of \$300. A more conservative scenario would have only 15% retrofit sales in 1990 (1.5% in 1980). The most conservative estimate assumes little market enhancement except for the 2.5 year acceleration of production lead time.

The average sales rate and resulting cumulative energy savings with and without the ERDA Demonstration is given in Table 4.10.

We estimate that in the late 1980's, approximately 20% of air conditioner sales could be heat pumps. The payback period for the HRS-heat pump is only about 1.8 years which result in a 68%* increase in consumer acceptance over the 3.5 year payback for the A/C-HRS. This would add about 15% more HRS sales, and these units would have, on the average, an annual energy savings of 48 mm Btu per year or a 65% increase over the A/C-HRS. With the inclusion of the heat pump-HRS into future sales, the cumulative nationwide energy savings could increase by 30-40%.

We judge that the medium scenario, shown graphically in Figure 4.18, is a likely projection of the possible effects of an ERDA-sponsored demonstration.

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* The heat pump-HRS will operate at the average COP (2.5) of the heat pump above the balance point (35°F). Once the balance point is reached, it is switched out. In Atlanta, Georgia, about 53,300 degree-hours are above the 35°F balance point and below the 65°F outdoor temperature. Assuming the percentage on-time of the compressor is $\frac{65 - T_{outside}}{65 - 35}$, then the compressor hours above 35°F outside temperature is $\frac{53,300}{30} = 1,800$ hours.

This would result in about 3,000 kwh of water heating at a COP = 2.5 or a savings of 1,800 kwh/year. We assume that the average number of compressor hours above the balance point in other cities of the U.S. is about the same.

TABLE 4.9

ACCELERATION PROFILE FOR A/C-HRS

Year	A/C Sales	Acceleration Without ERDA	Acceleration With ERDA
1980	511,000	0.100	0.400
1981	552,000	0.150	0.500
1982	596,000	0.200	0.700*
1983	643,000	0.400	1.000
1984	695,000	0.700*	1.000
1985	751,000	0.900	1.000
1986	811,000	1.000	1.000
1987	875,000	1.000	1.000
1988	946,000	1.000	1.000
1989	1,021,000	1.000	1.000
1990	1,103,000	1.000	1.000

*2.5 year acceleration

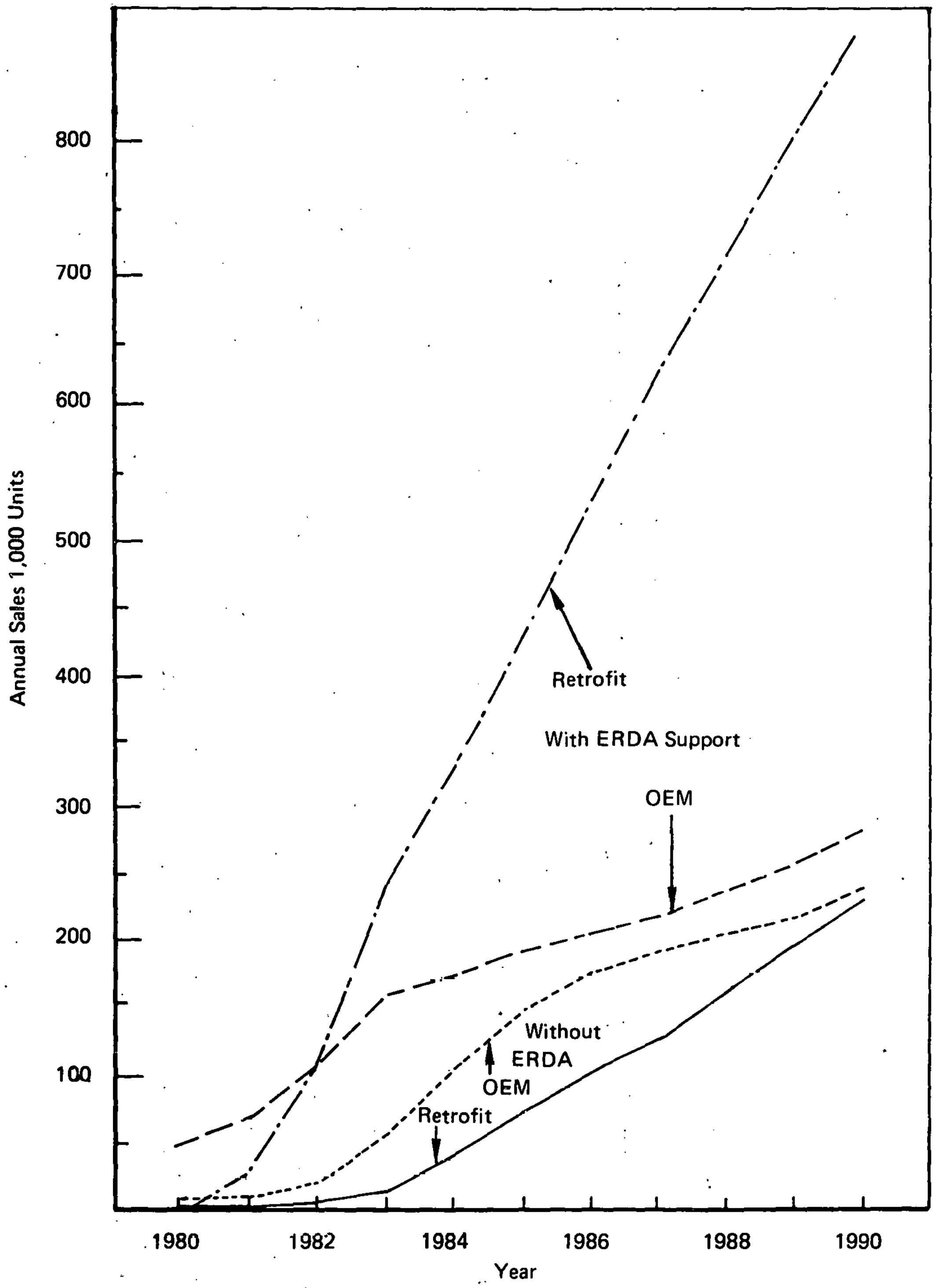
TABLE 4.10

ESTIMATED EFFECT OF
ERDA-SPONSORED DEMONSTRATION PROGRAM
(Cooling Season Only)

	Average Annual Primary Energy Savings (10^6 Btu Per Unit)	Average Nationwide Years to Payback*	Max. Percent Annual OEM Sales Captured	Max. Percent of In-Place Facilities Retrofitted**	Average Sales Rate 10^3 (1985) OEM	Average Sales Rate 10^3 (1985) Retrofit	Percent of 1990 In-Place A/C with HRS	Cumulative Energy Savings 1980-1990 10^{12} Btu Primary	Cumulative Energy Savings 1980-1990 Effect of ERDA
Maximum									
With ERDA	28.8	3.5	22	20	190	550	32	718	560
W/O ERDA	20.0	4.3	19	2.5	146	73	10	159	
Likely									
With ERDA	28.8	3.5	22	15	190	430	26	619	460
W/O ERDA	20.0	4.3	19	2.5	146	73	10	159	
Minimum									
With ERDA	28.8	3.5	22	10	190	300	17	508	350
W/O ERDA	20.0	4.3	19	2.5	146	73	10	159	

*Based on $300 added first cost in all cases.

**This grows at a linear rate from 1/10 of the value shown in 1980 to equal the value shown in 1990.



The Retrofit Sales Rely on The Success of OEM Market Penetration To Gain Exposure And Distribution.

FIGURE 4.18 PROJECTED SALES OF OEM AND RETROFITS A/C-HRS WITH AND WITHOUT ERDA SUPPORT

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5.0 INTEGRATED WATER AND SPACE HEATING SYSTEM

5.1 DESCRIPTION OF CONCEPT

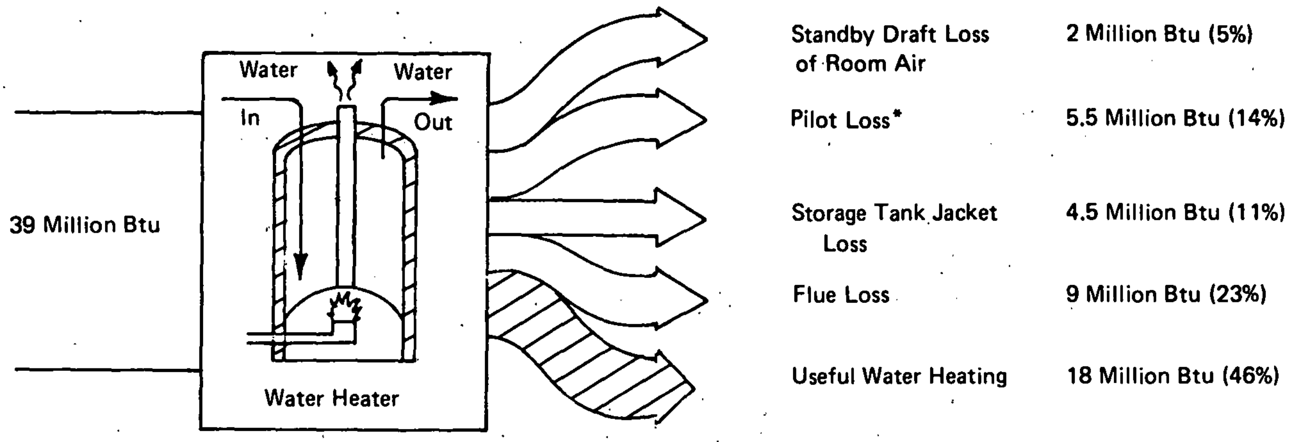
5.1.1 Overview

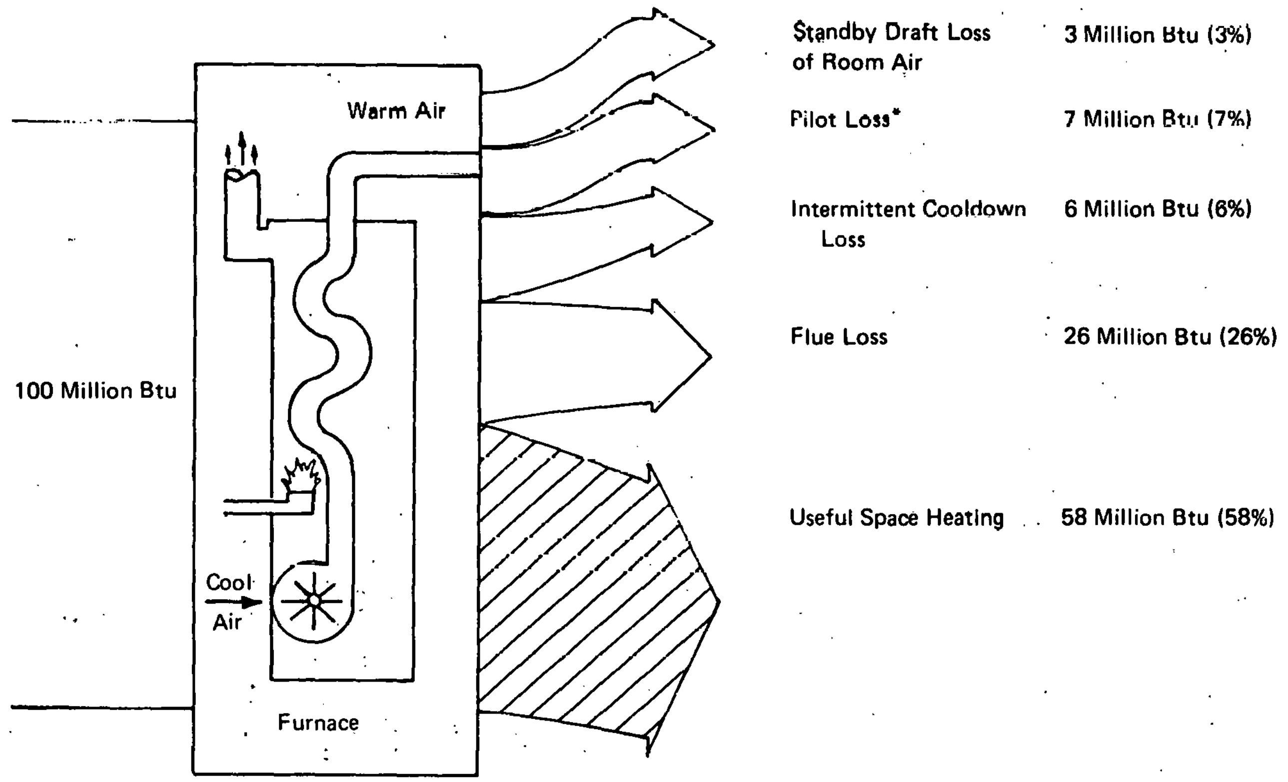
In order to evaluate the possibilities for an integrated heating system for hot water and comfort heating, it is useful to describe the systems as they function separately. Residential water heating and space heating are currently provided by separate devices, with the few exceptions discussed below. In a typical gas heated 1500 ft^2 house, one will find a forced-air furnace of 100,000 Btu/hr capacity and a storage water heater of 45,000 Btu/hr input (the stored hot water represents approximately 27,000 Btu). Both devices have steady state heat recovery efficiencies of about 72%, which is well below the 85% minimum practical value for operating without condensation or unsafe levels of CO (set by 25% excess air and 300°F flue temperature). During standby operation, both devices have standing pilot losses and draft hood losses to which the water heater adds a storage tank jacket loss of about 500 Btu/hr (continuous). The furnace adds 40-80 Btu/cycle intermittent cool-down loss, and the boiler adds 100-300 Btu/cycle intermittent cool-down loss. Typical yearly energy budgets of these devices are depicted in Figure 5.1. The combined load for water and space heating is typically about 76 million Btu/year. The current energy use by separate appliances is almost double this value (139 million Btu/year).

The concept to be developed and analyzed below is that of a single more efficient combustion device to be used for both functions. The objectives of the system are itemized in Table 5.1. For the same typical demand of 76 million Btu/year for water and space heating, the target for the integrated heating system (gas-fired) is about 100 million Btu/year. A schematic of the system is shown in Figure 5.2. The system may be thought of either as a high capacity, compact water heater used for space heating or as a boiler or furnace with "piggyback" water heating.

Energy-saving improvements in the individual water heater and furnace or boiler are obviously possible, and are being vigorously pursued by manufacturers. "Energy-conserving" furnaces and water heater models are currently offered which together achieve savings of about 20 million Btu/year, which is a substantial part of the target noted in Table 5.1. However, several factors would encourage the consumer to select an integrated device over improved separate devices:

- Reduction of the water-heater storage volume (lower standby losses) cannot be accomplished without increasing burner size up to the input rate typical of a furnace, a feature automatically achieved in the combined unit.





*These figures are for gas-fired system; for oil-fuel system, the cool-down loss is larger but the pilot loss is eliminated by an Intermittent Ignition Device (I.I.D.). The pilot loss figures are simply the pilot input rate times the standby hours, assuming no pilot energy usefully recovered. Water heater pilot rate 700 Btu/hr; furnace pilot rate 1,000 Btu/hr.

Based On: 0% Pilot Recovery for Gas-Fired Water Heater, Ref. pg. 77, Arthur D. Little, Inc., Study of Energy-Saving Options for Refrigerators & Water Heaters, Vol. 1 – Refrigerators, Prepared for FEA, May, 1977.

15% Pilot Recovery for Gas-Fired Furnaces and Boilers, Gelinas, et. al., "Automatic Ignition of Residential Gas Appliances," State of Calif. Contract 4010, Dec., 1976.

FIGURE 5.1 ENERGY USE FOR REPRESENTATIVE SEPARATE WATER HEATER AND FURNACE

TABLE 5.1

OBJECTIVES FOR INTEGRATED FURNACE-WATER HEATER

Type of Space Heating Unit	Typical Annual Energy Savings (mm Btu/year)* Gas Boiler	Gas Furnace	Oil Boiler	Oil Furnace
Eliminate both standing pilots with a single Intermittent Ignition Device (IID) (based on Figure 5.1)	12.5	12.5	0	0
Reduce both of the flue losses to about 3/5 of their current levels, by increased heat exchange effectiveness (UA), complemented by a forced draft burner	14	14	8	8
Reduce the draft losses (warm room air) of both devices by either a vent damper or forced-draft indirect heating	5	5	0	0
Reduce the storage tank size and minimize heat exchanger weight, thereby cutting jacket and intermittent cool-down losses*	24	4.5	22	2.5
Total Energy Savings Target	57.5	38	30	10.5

*Based on:

1) Geographic average of 4500 degree days, and
2) Comparison with separate water heater (electric water heater for oil-fired furnace and boiler) and furnace or boiler functions.

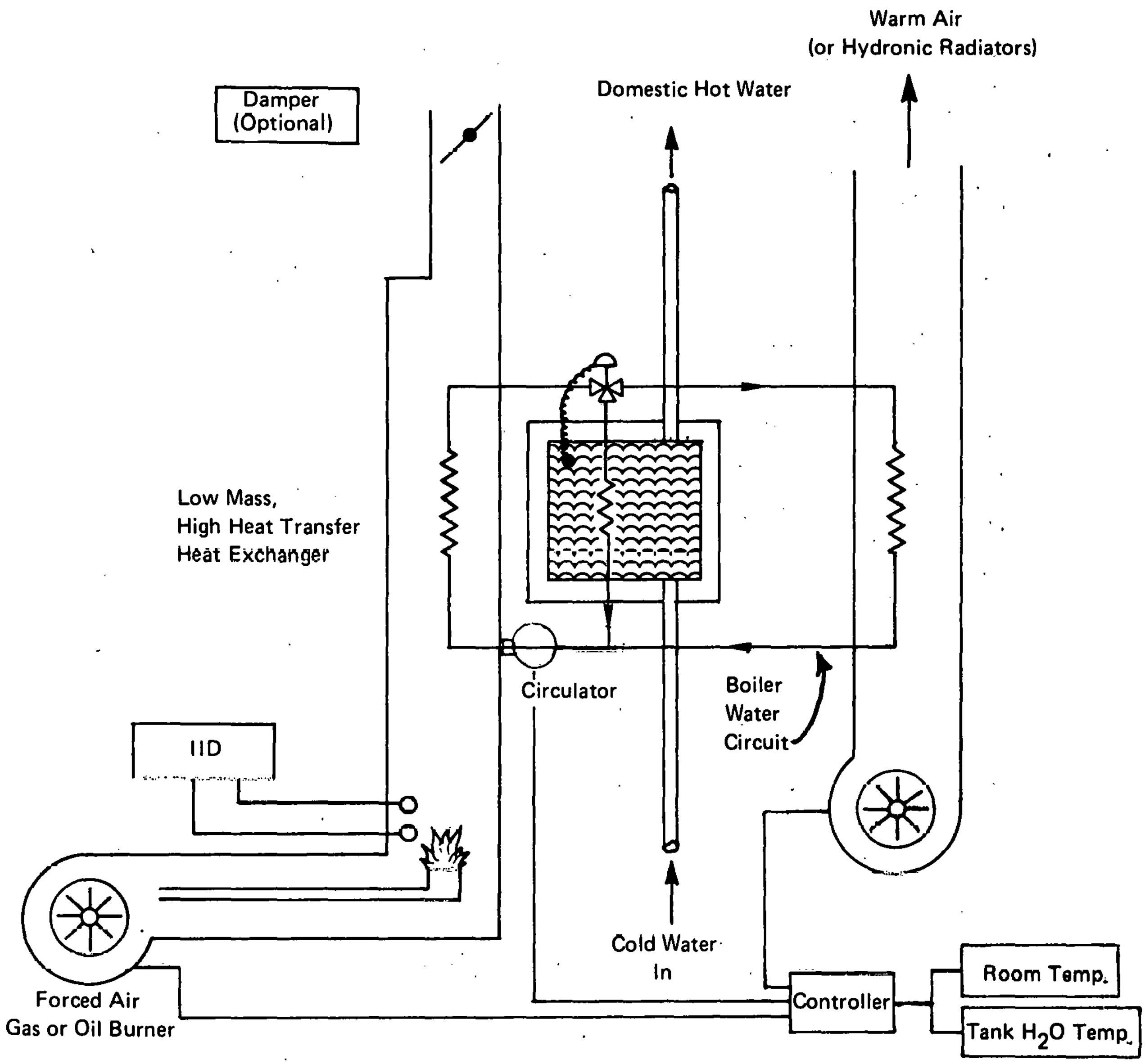


FIGURE 5.2 SCHEMATIC OF INTEGRATED WATER AND SPACE HEATING SYSTEM

- The owner of an oil-fired heating system could eliminate the costly-to-operate electric water heater.

- The conventional water heater at a current installed cost of only about $135 will not soon be equipped with Intermittent Ignition Device*, vent damper*, or forced draft because of the added expense of an electrical connection and the need for substantial overhaul of the manufacturing methods and pricing structure.

- Separate energy-conserving models of furnaces and water heaters have a combined added first cost over conventional units of $175-$225, whereas we project that a single combination device can save more energy and sell for an added first cost of under $200 (see Table 5.7, p. 133).

It is these factors which lead us to consider the system in more detail. These factors, particularly the first two, have also motivated past developments of combination water and space heating systems, particularly for oil-fired systems.

5.1.2 Precedents

For several years, residential fuel costs in Japan and Europe have been approximately double those in the U.S., and it is in these locations that the instantaneous water heater/space heater has been highly developed and marketed. In the United States, piggyback water heating has been done on oil-fired boilers, not so much to save fuel but rather to avoid the initial cost and operating expense of an electric or oil-fired water heater. We will briefly review the following systems which are precedents:

- "Over-sized" water heater: The high input rate storage water heater with internal coil heat exchanger used for hydronic space heating (Japan; 100,000 Btu/hr, about 40-gallon storage @ 140°F).

- "Tankless" boiler: The gas-fired or oil-fired boiler with "tankless" coils for water heating (Germany, United States; 80,000-150,000 Btu/hr, internal coil).

- Boiler with external tank: An auxiliary hot water tank of 10-40 gallons stores domestic hot water and is heated by a coil from the boiler.

- Low-capacity indirect-fired boilers: Indirect-fired boilers for combined space and water heating are available from four German manufacturers: Viessman, Rousch, Oechssler, and Rekord. The typical specifications of these units are 5 gallons storage at 175°F, 80% efficiency, 120,000 Btu/hr input. The Viessman and Oechssler units employ fin-tube heat exchangers.

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*Gas-actuated vent dampers are available at this writing; also battery-operated IID's have become available. However, the forced-draft burner option would require line voltage.

- Experimental indirect-fired boilers: Experimental indirect-fired high-efficiency units for either water or space heating have been built by Amana, Raytheon, A.O. Smith, Econotherm, and Scientific Energy Systems (SES); 120,000-240,000 Btu/hr, 84-85% E_r, IID, low mass heat exchangers.

Table 5.2 compares the precedent systems on the basis of how closely they meet the objectives of Table 5.1. Figures 5.3, a, b, and c depict the first three of these systems; the indirect-fired system is depicted in Figure 5.2.

In all of these precedent systems (an estimated 8 million in use in the U.S.) the boiler water and domestic hot water are separated by only a single wall heat exchanger. Whether this is a universally accepted practice should be explored in the National Demonstration Phase. For the purposes of this analysis we have assumed that single walled heat exchangers between boiler water and domestic water are acceptable.

5.2 ANALYSIS OF DIFFERENT CONFIGURATIONS

5.2.1 System Design for Variable Load

The instantaneous water heating loads of 1/2 to 3 gpm hot water represent a range of energy drain rates of 20,000 to 120,000 Btu/hr. This variation can be handled in three ways:

Option 1. An instantaneous water heating system (80% efficiency) can modulate from 35,000 to 150,000 Btu/hr.

Option 2. A storage tank can be designed with adequate capacity for the largest hourly hot water draw, so that burner size is irrelevant except for space heating.

Option 3. Combinations of storage volume and burner size are considered which meet the maximum hourly draw. The burner cycles on only for the larger hot water draws.

We select Option 3 for the proposed concept. The problem with Option 2 is that the required tank size would be about 52-gallons; a tank of this size would have unnecessarily large jacket losses and not be a compact single unit. The problems with Option 1 (instantaneous, fully modulated burner) are as follows:

- The large number of burner cycles (400,000 to 500,000 over the unit lifetime) would require sophisticated and costly controls.

- The air and fuel would have to be modulated in parallel, which requires an additional costly control usually found only on commercial boilers with \$2,000-\$5,000 burners.

TABLE 5.2

FEATURES OF PRECEDENT SYSTEMS FOR SPACE/WATER HEATING

Feature	"Oversized" Water Heater	"Tankless" Boiler (Internal Coil)	Boiler with External Tank	Low Capacity Indirect-Fired Boiler	Experimental Indirect-Fired Units
Fuel type	Gas/Oil	Gas/Oil	Gas/Oil	Gas/Oil	Gas/Oil
Input (range)	100,000 Btu/hr	130,000 ± 20,000 Btu/hr	130,000 ± 20,000 Btu/hr	80,000 - 150,000 Btu/hr	180,000 ± 60,000 Btu/hr
Water storage (volume at temperature)	40 + 10 gallon @ 140°F	8 ± 4 gallons @ 180 ± 20°F	10 gallons @ 180°F or 40 ± 10 gallons @ 140°F	5 gallons @ 175°F	20 ± 10 gallons @ 140 ± 20°F depending on burner input
Additional energy stored in heated metal sections	Unknown, estimate 2,000 Btu	Approximately 5,000 Btu	Approximately 5,000 Btu	Approximately 2,000 Btu	Approximately 1,000 Btu
Total stored energy	28,000 ± 7,000 Btu	14,000 ± 6,000 Btu	15,000 - 30,000 Btu	7,000 Btu	5,000 - 15,000 Btu
Heating of stored water	Direct (some heat loss during standby)	Direct (heat loss during standby)	Indirect	Indirect	Indirect
Vent Damper	Partial (forced draft)	Partial (forced draft) on oil-fired	Partial (forced draft) on oil-fired	No	No
Recovery efficiency	79%	75%	75%	80%	84-87% (95% with condensation)
Intermittent Ignition Device	Yes	Yes on oil-fired	Yes on oil-fired	Yes	Yes
Net added cost over furnace plus water heater costs	-	\$100-\$200 depending on added storage	\$200-\$250	-	Not in mass production
Period of experience	6 years	20-30 years	10-15 years	5-10 years	3 years
Status	Luxury option (for central heating system	Widely used option for cast iron boilers	Luxury option for boilers	Luxury option for large apartments	Experimental prototype only
Number of units in service	Unknown	7 million in U.S.	1 million in U.S.	Unknown	Approximately 10
Manufacturers	Tada-Smith, Others	Peerless, A.O. Smith, Crane, American Standard, Weil-McLain, Burnham	ACE Tank & Heater, Everhot, Petroleum Engineering	Viessman, Oechssler	A.O. Smith, Econotherm, Raytheon, SES
Locations	Japan	New England and North Central U.S.	New England and North Central U.S.	Germany, Holland	--
Limition	Coil for space heating is designed for only 47,000 Btu/hr	Relatively high losses for summer hot water	Same as column 3, but less liming	Capacity low by U.S. standards	Not yet optimized Long term performance not known
Annual energy savings	28 million Btu	8 ± 4 million Btu	9 ± 5 million Btu	30 ± 5 million Btu	40 ± 5 million Btu
Estimated total energy use for U.S. demand of 58 mm Btu space and 18 mm Btu water heating	111 million Btu	131 ± 4 million Btu	130 ± 5 million Btu	109 ± 5 million Btu	99 ± 5 million Btu

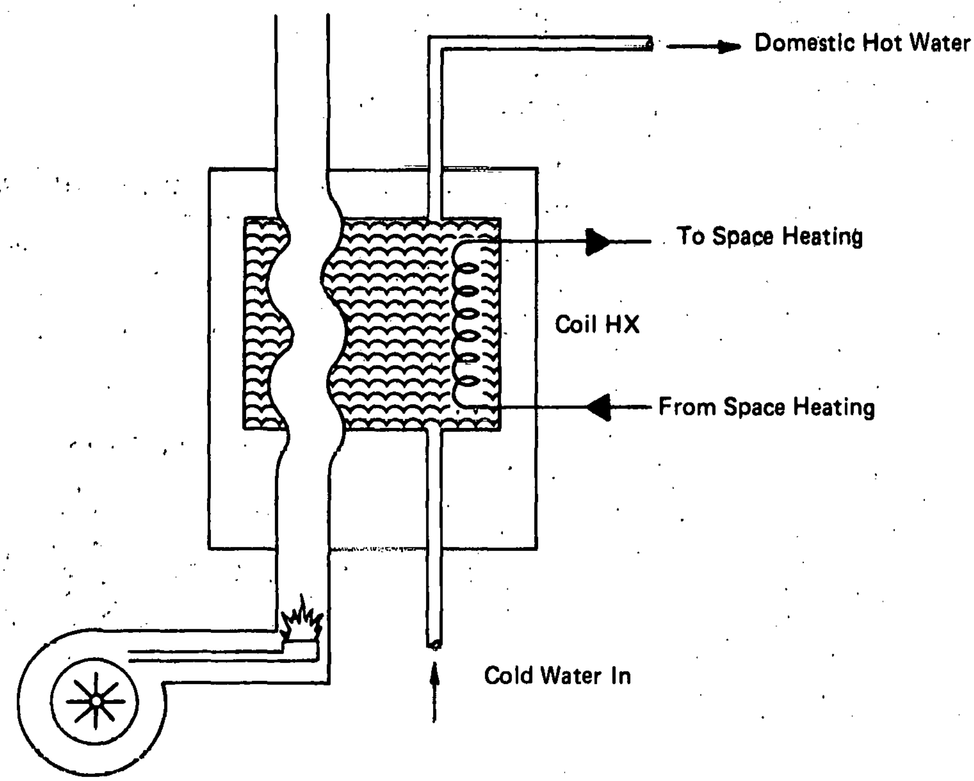


FIGURE 5.3 SCHEMATICS OF PRECEDENT SYSTEMS
(a) OVERSIZED WATER HEATER

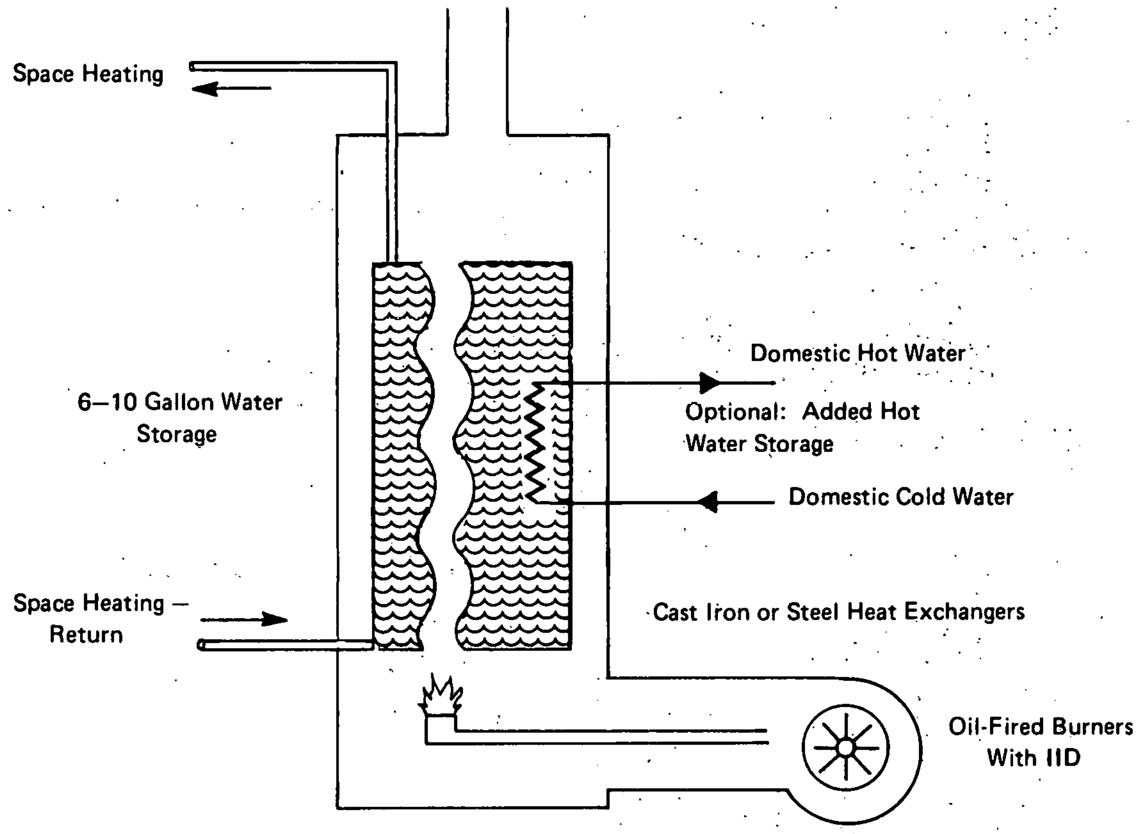


FIGURE 5.3 (b) "TANKLESS" BOILER (INTERNAL)

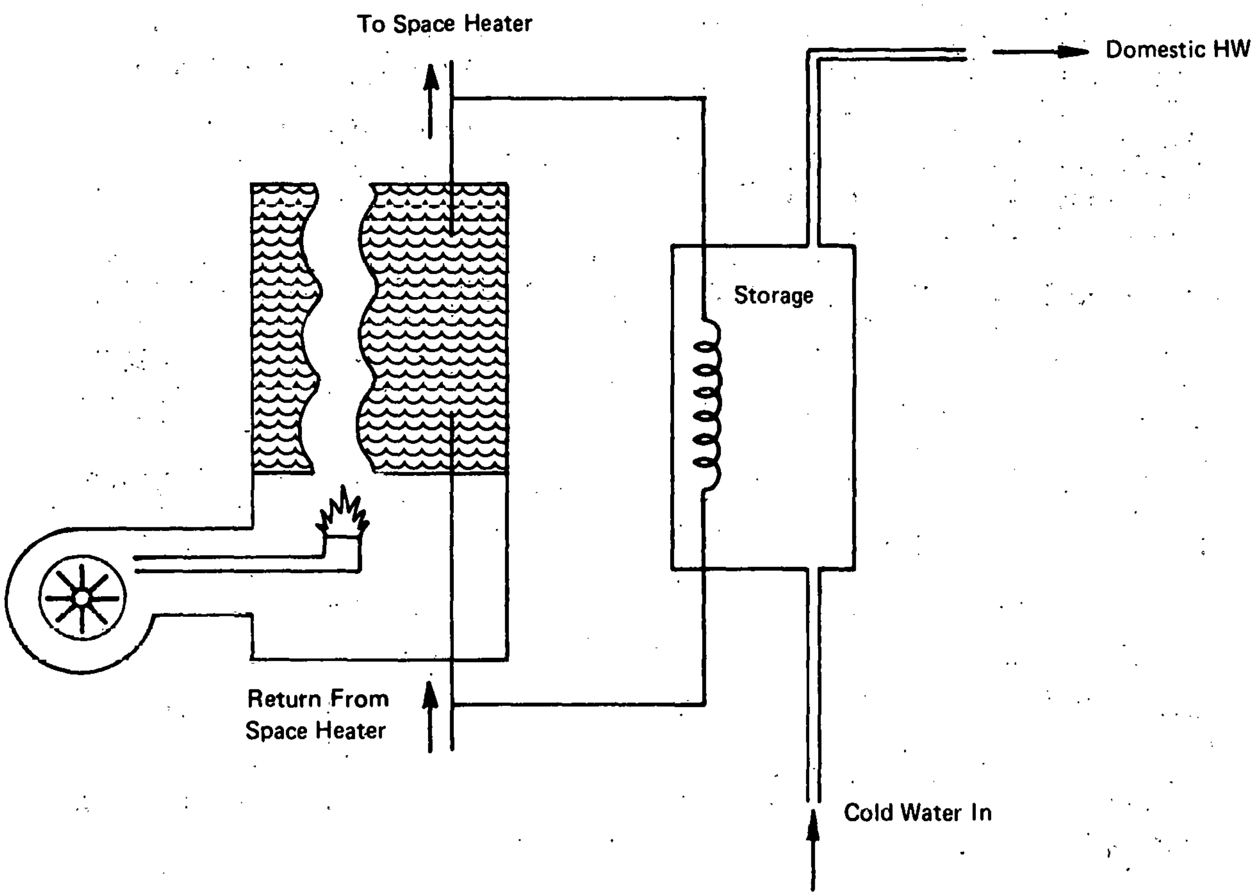


FIGURE 5.3 (c) "TANKLESS" BOILER (EXTERNAL)

- The heat exchanger would not function efficiently over the full-range of modulation.

- Undesirable condensation might occur at low firing rates.

- Burner overheating and ignition problems might occur at low firing rates.

- Oil-fired burners cannot modulate over a times 6 range without multiple nozzles.

- The burner must pass ANSI standards for CO emissions at all firing rates, which is quite difficult over a times 6 range.

For these reasons we adopt Option 3 for the proposed system. The question of input rate and storage volume will be addressed after first examining the space heating load behavior.

The variable space heating load is usually handled by fixing the maximum burner input (approximately 80,000 to 160,000 Btu/hr) to satisfy the lowest expected outdoor temperature. In most residential heating systems, the burner is then allowed to cycle on/off, with the standby time fraction increasing as load diminishes. Since this cycling increases standby losses, some residential heating systems have two-stage burners, although this is relatively rare. The proposed system is not designed with two-stage firing simply because the standby losses are quite small due to indirect firing, low thermal mass, and IID.

5.2.2 Burner Input Rate and Water Storage Volume

Based on the above considerations, we wish to consider burner input rates of 80,000-150,000 Btu/hr with storage tank sizes from 55 to 5 gallons. The formula relating input rate $\dot{Q}$ to minimum volume V is derived from the maximum hourly water heating load Q_W and space heating load Q_S:

$$Q_W = V\, C_p\, (T_w - T_0) + \alpha\, E_r\, \dot{Q}\, (1\text{ hr} - t_{sh}) \tag{1}$$

$$Q_S = E_r\, \dot{Q} t_{sh} \tag{2}$$

where:

α is a stratification coefficient (unity for perfect stratification) and accounts for the small fraction of hot water which is not hot enough to be useful,

E_r is the recovery efficiency,

T_w is the hot water temperature, and

C_p is the water heat capacity (Btu/gal °F).

Solving Equation (2) for the space heating time t_{sh} and substituting Equation (1), we arrive at an expression for V:

$$V = \frac{Q_w + \alpha\ Q_s}{C_p\ (T_w - T_0)} - \frac{\alpha\ E_r\ \dot{Q}}{C_p\ (T_w - T_0)} \tag{3}$$

Under the assumptions:

Q_w = 58,000 Btu/hour
(Equivalent to 40 gallon storage tank at 140°F (26,500 Btu) plus 45,000 Btu/hr burner at 70% efficiency (31,500 Btu) gas-fired water heater)

Q_s = 30,000 Btu/hour
(Based on overall house conduction of U = 500 Btu/hr °F and 65°F indoor and 5°F outdoor temperature)

T_w = 140°F
(Gives compact tank)

T_0 = 60°F
(Average for NYC)

α = 0.80
(Typical for vertical cylinders)

E_r = 0.80

we obtain

$$\frac{(Q_w + \alpha\ Q_s)}{C_p\ (T_w - T_0)} = 123 \text{ gal}$$

and

$$\frac{\alpha\ E_r\ \dot{Q}}{C_p\ (T_w - T_0)} = 9.6 \times 10^{-4}\ \dot{Q} \text{ gal}$$

in Equation (3).

When the burner rate exceeds 123,000 Btu/hr this expression implies that the storage volume is picked for reasons other than 1-hour demand. We have stated that excessive cycling is one reason to have adequate storage volume (the burner should not necessarily come on for hot water used in shaving). Table 5.3 gives minimum tank size for various input rates. Also listed is the volume required in the case of Q_s = 50,000 Btu/hour, which is typical of the northern United States.

TABLE 5.3

TANK SIZE AND BURNER INPUT
TO MEET TYPICAL MAXIMUM LOADS

Burner Input (Btu/hr)	TANK SIZE: 30,000 Btu/hr Space Heating	TANK SIZE: 50,000 Btu/hr Space Heating
80,000	46 gallons	70 gallons
100,000	27 gallons	51 gallons
120,000	8 gallons	32 gallons
140,000	Minimum (5 gallons)	13 gallons
160,000	Minimum (5 gallons)	Minimum (5 gallons)

Another consideration in selecting burner input rate is flue size, which must increase with exhaust gas flow rate. A 4-inch diameter vent would be adequate for forced draft gas-fired units up to 150,000 Btu/hr, provided the flue temperature is low enough and fan sized properly. Conventional oil-fired units are fitted with larger flues because of corrosion and higher flue temperatures. Although further study is needed, there seems to be no universal flue size limitation for the input rates of interest (up to 150,000 Btu/hr).

One final consideration is the artificial price structure of boilers and furnaces on the market today. Prices tend to increase with burner input rate disproportionately to actual manufacturing cost; that is, the price of a 150,000 Btu/hr - input model may be 20% higher than a 120,000 Btu/hr model, when only a low-cost burner part is substituted and the same heat exchanger used. This means that a 150,000 Btu/hr integrated appliance will meet resistance as a substitute for an 80,000 Btu/hr forced air furnace, but be much more cost competitive with a 120,000 Btu/hr furnace. The basic point is that the selected burner input should be minimum, if possible, subject to other considerations.

Based on Table 5.3 we recommend a burner size in the 120,000-140,000 range with 10-20 gallons of water storage at 140°F. This will permit 2.5 gpm instantaneous draws, satisfy the maximum 1-hour demand, and have a compact tank suitable for an attractive single cabinet.

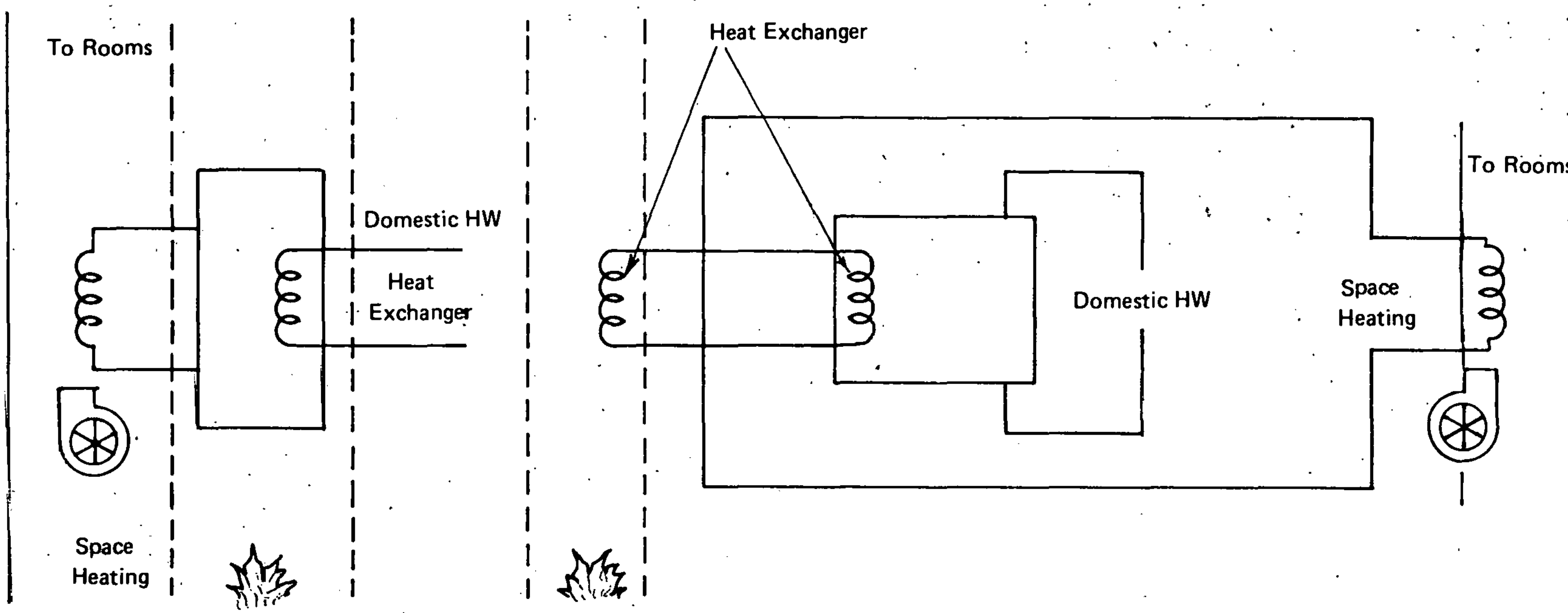
5.2.3 Direct or Indirect Heating

The combustion gases may either impinge on the storage tank directly, or impinge on a heat exchange loop connected to the storage tank (see Figure 5.4). We recommend the indirect heating option because it separates the flue from the storage volume, eliminating the need for a vent damper. During the standby periods, thermal siphoning between the storage tank and the flue is prevented by proper control of the three-way valve shown in Figure 5.2.

5.2.4 Selection of Heat Transfer Fluid

The domestic water, boiler water, ethylene glycol, steam, or air may be considered as a heat exchange fluid for absorbing heat from the burner gases. We recommend that boiler water or a non-toxic heat transfer fluid such as DowFrost* be considered, so as to avoid corrosion or liming under the extreme conditions of high temperature and water hardness. Air should not be selected because a) forced circulation is needed and a pump is less costly, and b) an air-to-domestic water heat exchanger would be quite costly. This implies the use of a duct-coil in forced air replacement application. It also implies that boiler and water heater manufacturers may be in a better position to design and manufacture the integrated appliance than forced air furnace manufacturers. The system will be "wet."

*Tradename for Dow Chemical non-toxic heat transfer fluid of propylene glycol.



(a) Direct

(b) Indirect

FIGURE 5.4 HEATING OPTIONS

As mentioned earlier, single-wall heat exchangers between boiler and domestic water are presently used. The acceptibility of such system will need to be investigated.

5.2.5 Flue-Gas Condensation as an Energy-Saving Option

The advantages of condensation (8 million Btu/year savings based on increasing recovery efficiency from 85 to 93 percent*, easier venting) are offset by the following disadvantages:

- Added cost of additional surface area to accomplish the condensation heat transfer.
- Need for a low temperature heat sink, such as the incoming preheat water.
- Modifications and additions to codes for installation and certification.
- Need for a special vent material which can withstand corrosion of saturated gases and condensate.
- Small pump and drain line to handle condensate.

The consumer cost of these items is estimated to be about $100 based on a:

- condensate pump ($25),
- heat exchanger ($30),
- special vent ($10), and
- installation ($35).

The years to payback at $3/mm Btu is about 4 years, which is marginal.

5.2.6 Selection of Preferred System

The preferred system is essentially that of Figure 5.2 without the vent damper. It consists of:

- 120,000 to 140,000 Btu/hour input forced draft burner (oil or gas) with IID
- 10 to 20-gallon domestic water storage tank at 140°F, lined, with a 3-inch fiberglass insulation

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*Inherent losses remaining at 120°F include 3% for uncondensed water vapor, 2% jacket loss, 2% unrecovered sensible heat.

- 85% recovery efficiency achieved by a heat exchanger designed for 300°F, 25% excess air flue gases.

- Additional heat exchanger for boiler water to domestic water (copper coil) of the single-wall type.

- Reverse acting aquastat for preferential water heating to control the valves which direct water flow, and tank temperature controller to activate burner.

This system would save approximately 38 million Btu yearly for the U.S. typical 139-million Btu budget. For specific representative cities, the savings would vary as shown in Table 5.4.

The system cost to the consumer (less installation) is estimated at $572, as detailed in Table 5.5. This is the cost for hot-air replacement systems where a coil must be provided. Otherwise, the cost is approximately $472. These costs are based on production in quantities of 100,000 per year or more. Due to uncertainties in component costs, these estimates are only reliable to 20%. The largest uncertainty is in the cost of the coil, which is not currently a mass-produced item. Our estimated $100 for the coil could be in error by ± 50%. System components are examined in Table 5.6.

The added costs are determined for gas and oil-fired systems as shown in Table 5.7 where we have assumed that there is no significant installation cost difference between the units installed separately or as an integrated appliance.

The potential energy savings of the preferred system is given in summary form in Table 5.8.

5.3 NATIONAL DEMONSTRATION PLAN

5.3.1 Introduction

The principal goal of the Demonstration is to accelerate the development and commercialization of an energy-saving product which combines furnace and water heater functions together in one unit. This plan reflects our belief that an ERDA-sponsored development and demonstration program, undertaken in 1977, could accelerate the final commercialization of this product by nearly 4 years.

Key areas of this program for accelerating the commercialization of the product are:

- the development of designs for minimum life-cycle costs to the consumer,

TABLE 5.4

ENERGY SAVINGS FOR DIFFERENT CLIMATIC REGIONS

Region	Degree Days[1]	Annual Target Savings* in 10^6 Btu/year: Gas Boiler	Gas Furnace	Oil Boiler	Oil Furnace	National Sales Weighted Average
NE - Norwalk, Conn.	6470	59.3	39.8	32.0	12.5	36.1
NC - Detroit, Mich.	6345	62.3	42.8	33.5	14.0	38.7
S - Pine Bluff, Ark.	2795	50.1	30.6	27.1	7.6	28.0
W - Roswell, N. M.	3515	52.7	33.2	28.5	9.0	30.3
U.S. Sales Weighted Average	4361	57.5	38.0	30.0	10.5	34.3
Estimated Percent of Sales by Type of Heater[2]	--	9	64	10	17	--

* The energy savings is calculated by using the nominal savings given in Table 5.1 in the following equation:

$$\text{Savings} = Q_1 + Q_2 \left[\frac{\text{Degree Day}}{4361}\right]$$

where: Q_1 = flue loss of water heater + pilot loss + intermittent cool down loss

Q_2 = flue loss of furnace + draft losses

Sources: [1] Reference 5.1

[2] Reference 5.2

TABLE 5.5

ESTIMATED COSTS FOR
INTEGRATED BOILER/WATER HEATER
(120,000 Btu/hr)

	Cost to Manufacturers	Consumer Cost
Burner Heat Exchanger		
Heat Coil in Combustion Chamber (copper)	$16.70	
Fin Tube in Combustion Chamber	13.00	
Combustion Chamber	8.70	
Chamber Insulation (1 inch for 1,000°F)	.84	
Chamber Cover	4.35	
Circulating Pump	8.35	
Connection to Storage Tank	1.95	
Material Subtotal	$53.89	
Assembly Labor	5.36	
Total Combustion Chamber/Heat Exchanger	$59.25	$137.00
Tank with Heat Exchanger		
Storage Tank - 20 gallons, glass-lined	$18.00	
Tank Insulation	1.10	
Heat Exchanger	12.50	
Tank Cover and Dip Tubes	7.90	
Material Subtotal	$39.50	
Assembly Labor	2.00	
Total Storage Tank	$41.50	$102.00
Combustion System		
Forced Draft Blower with Electric Ignition and Safety Controls	$70.00	
Reverse Acting Aquastat	5.00	
Three-Way Valve	10.00	
Material Subtotal	$85.00	
Assembly Labor	8.00	
Total Combustion System	$93.00	$233.00
SUBTOTAL (without forced-air coil)		$472.00
Forced Air Coil for Duct Installation		100.00
TOTAL (with forced-air coil)		$572.00

TABLE 5.6

MAJOR SYSTEM COMPONENTS

Forced Draft Burner

Midcontinent Metal Products offers a forced draft gas-fired burner which is of the type required.

Heat Exchanger (Comb. Chamber)

An off-the-shelf heat exchanger, copper coil with fin tubing is recommended. These are currently used by Rheem, A.O. Smith, and others for swimming pool heaters.

Insulated Storage Tank

An off-the-shelf, 15-gallon tank, glass-lined, with 3-inch fiberglass insulation is required.

Controls

A reverse-acting aquastat (available from Honeywell) is recommended for giving preference to the water heating function over space heating.

IID

Several electric ignition devices are currently being sold for retrofit and/or OEM, including Penn-Baso, White-Rodgers, and Carborundum.

Hot Air Coil

A coil such as that available from Trane (Type T) is recommended. This is a simple, serpentine copper tube with aluminum fins, designed for 40°F air temperature rise at 2550 CFM with water dropping from 180°F to 158°F at 10 GPM (face velocity: 570 ft/minute).

TABLE 5.7

ADDED COSTS FOR INTEGRATED WATER AND SPACE HEATING SYSTEM

Type of Heating System	Estimated Percent of Annual Sales	Integrated System Cost* ($)	Separate Water Heater Cost ($)	Conventional Heating System Cost* (120,000 Btu/hr) ($)		Total Costs of Separate Devices ($)	Added Cost ($)
Gas Forced Air	64	572	130	240	(315)	370	202
Gas Boiler	9	472	130	380	(560)	510	(38)
Oil Forced Air	17	572	130	360	(400)	490	82
Oil Boiler	10	472	130	500	(850)	630	(158)
Projected Sales Weighted Average		553				429	124

*All costs less installation. These estimated costs are at the lower end of the range of quotations received by ADL and were adopted in order to be conservative in our estimates of added cost for the integrated system. The average quotations for conventional systems were approximately 30% higher as shown in parentheses. Installation of a replacement unit adds $200-300 for furnaces and $400-500 for boilers to the heating system cost. For example, G.A.M.A. quoted an average installed cost of $560 for a gas forced air furnace.

TABLE 5.8

OVERVIEW OF INTEGRATED APPLIANCE

Candidate	Estimated Percent of Annual Sales	Energy Savings 10^6 Btu Primary Per Unit	Added First Cost Installed ($)	Years to Payback ($3.50/mm Btu)	Max. 1990 Inventory Applicable 10^6 Units*	1990 Annual National Potential Energy Savings - Primary (10^{14} Btu/year)
Furnace/ Water Heater						
Gas Forced Air	64	38.0	202	1.5	22.6	8.6
Gas Boiler	9	57.5	(38)	—	4.8	2.8
Oil Forced Air	17	10.5	82	2.3	3.2	0.3
Oil Boiler	10	30.0	(158)	—	1.7	0.5
Projected Sales Weighted Average	100	34.3	124	1.0	32.3	12.2

Basis for Projections

	1980	1990
In-Place Residential Units	51,000,000	58,000,000
• With Gas or Oil	43,400,000	42,000,000
Average Heating Unit Sales		
• Furnaces (Gas/Oil)	2,800,000	3,600,000
• Boilers (Gas/Oil)	250,000	250,000

*Reflects the average annual furnace and boiler sales projection of 3.2 million sales per year 1980 to 1990.

- government-sponsored field testing and dissemination of energy-saving information on the product, and

- recommended test procedures and standards for the combined product in light of Federal and State efficiency programs on the two separate products.

5.3.2 Recommended Plan

The recommended plan for the development and demonstration program is shown graphically in Figure 5.5 following. A brief discussion of the plan follows.

In Task 1, a design analysis focusing on developing a combined furnace-water heater with the minimum number of years to payback should be undertaken. Extensive use should be made of previous research and development by the manufacturers on the combined water heating furnace. As a minimum, the size of the storage volume, burner rating, firing configuration, heat transfer fluid and level of condensation of combustion products should be examined. Further examination of European and Japanese systems should be made in light of the targets of the preferred system discussed in Section 5.2.6.

Two development programs are envisioned in Task 2. One is a modification to existing boiler technology designed to achieve an 85% recovery efficiency, while the other focuses on an advanced design with fin-tube heat exchangers and combustion product development. The advanced design will achieve recovery efficiencies of upwards to 95%. The 85% system will be a modified boiler, typical of the swimming pool type coil boilers presently used in the United States.

The motive for having parallel developments is to evolve a first stage design which could be most readily adapted by existing boiler manufacturers concurrently with a more advanced system with greater savings that would be introduced after market acceptance of the modified design (80-85%).

In Task 3, field test units should be fabricated giving consideration to the ultimate product manufacturing techniques. The units should be tested in the laboratory for uniformity and then installed in homes for field testing. A minimum of 20 field demonstration units are recommended.

Surveillance of the energy savings and ability to meet water and space heating demand should be monitored for a year in Task 3, and the results of the demonstration summarized in a final report for public dissemination in Task 4. In addition, a workshop with gas utilities and oil service companies should be held to pass along the findings of the Demonstration Program and encourage utility programs designed to accelerate the use of the furnace-water heater.

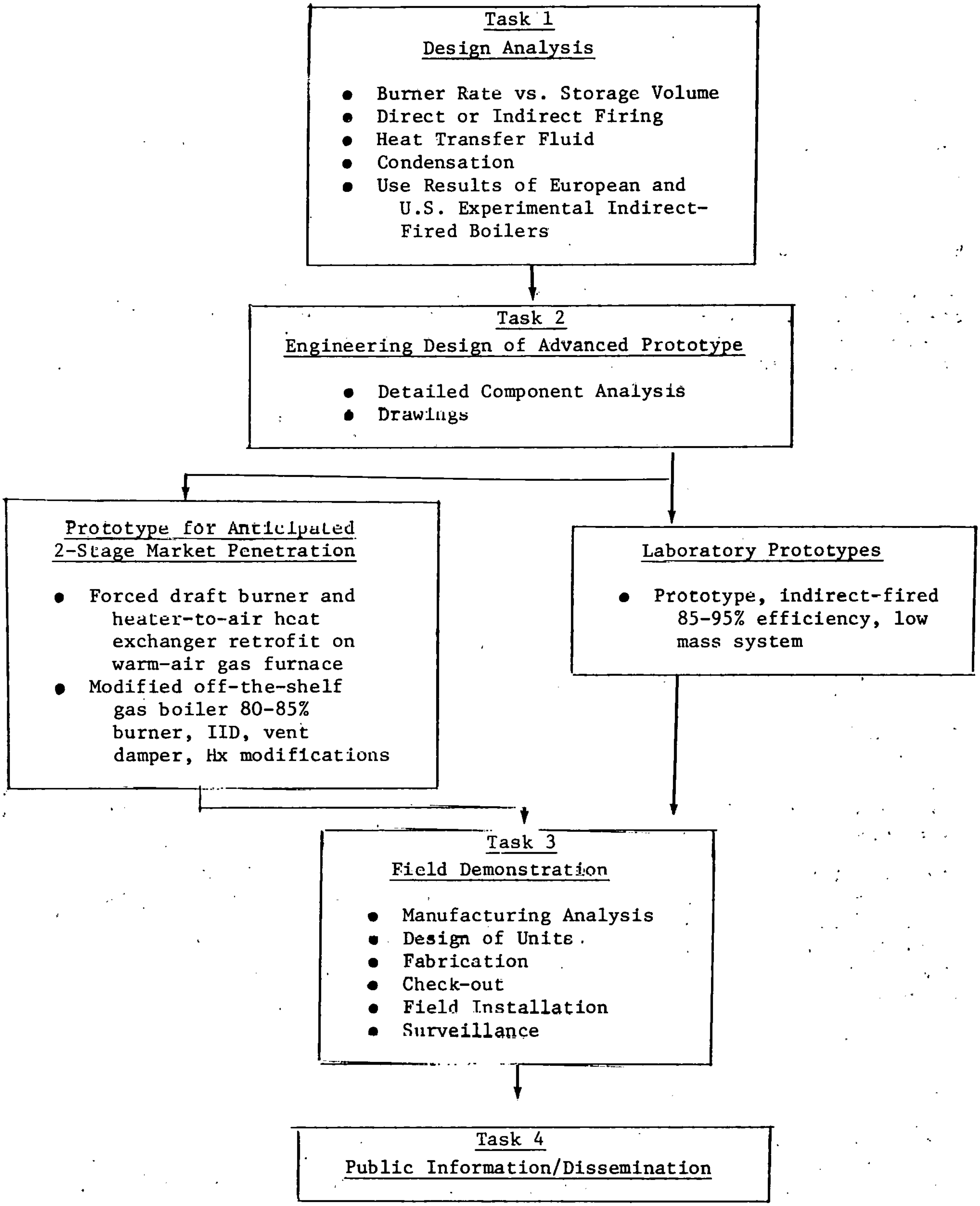


FIGURE 5.5 RECOMMENDED DEMONSTRATION PLAN

Schedule

The recommended schedule for the National Demonstration Plan is shown in Figure 5.6.

Estimated Costs for Demonstration Plan

Based on discussions with manufacturers and our own estimates of manpower requirements, we judge that the total program cost (industry and ERDA) to complete the Demonstration Program will be $500,000, with the emphasis of the program effort broken down as follows:

Task	Recommended % of Program Effort
Development	50
Demonstration	40
Public Information Dissemination	10

5.4 POTENTIAL BENEFIT OF NATIONAL DEMONSTRATION PLAN

The combined furnace/water heater discussed in the previous section is expected to save the homeowner the energy shown in Table 5.4. The national average for years to payback is 1.0 years. The cost basis for this is an added first cost of $120 for the combined furnace/water heater over separate utility functions.

The cumulative energy savings was estimated through the same approach as in Section 4.4 as given in Table 5.9. The estimate of the acceleration profile with and without an ERDA-sponsored program is a judgment based on discussions with potential manufacturers. The acceleration profile used, along with the projected total sales of furnaces used in the formula:

$$\text{OEM sales} = (.05 + \frac{.60}{\text{YTPB}}) \times (\text{furnace sales}) \times (\text{acceleration})$$

(see Section 4.4) is shown in Table 5.10

The maximum effect of an ERDA-sponsored program is on the order to 1135 trillion Btu's accumulating from 1980 to 1990. The minimum effect of an ERDA program would be around 400 trillion Btu's. In the latter case, the effect of the ERDA program is to accelerate the production of the product by about three years, while in the more optimistic scenarios, the development and demonstration programs will have a material effect on the consumer acceptance and added first cost of the product.

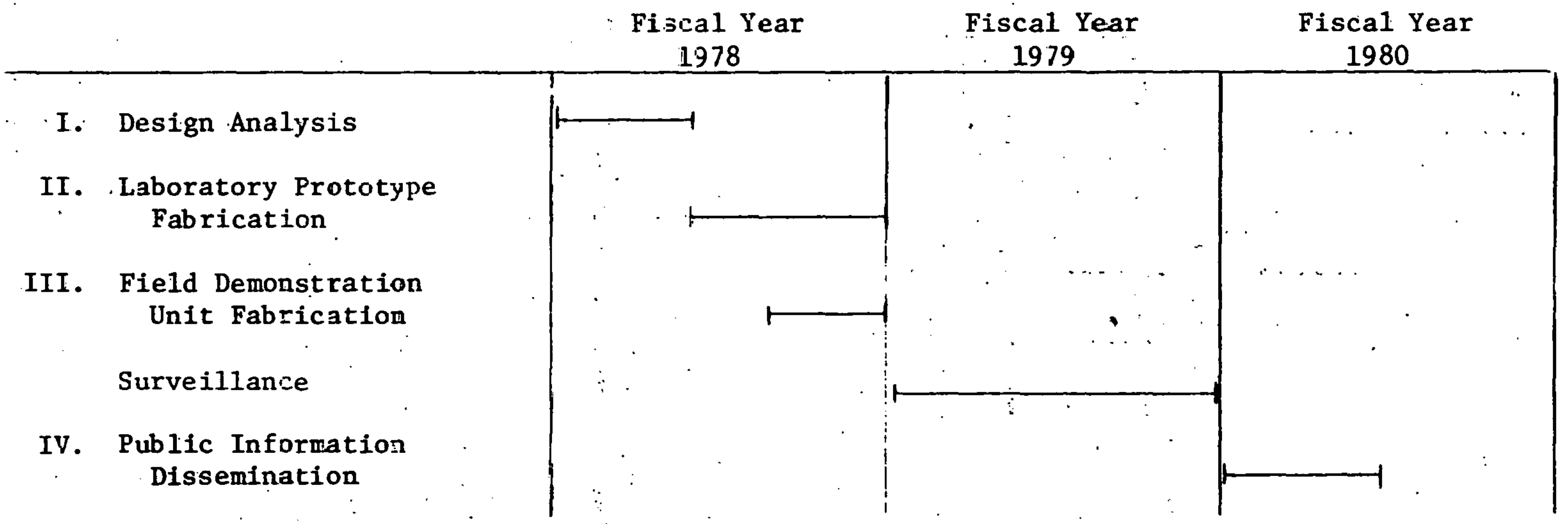


FIGURE 5.6 SCHEDULE OF NATIONAL DEMONSTRATION PLAN

TABLE 5.9

ACCELERATION PROFILE FOR FURNACE/WATER HEATER

Year	Furnace and Boiler Sales	Acceleration Without ERDA	Acceleration With ERDA
1977	2,500,000	0.000	0.000
1978	2,200,000	0.000	0.000
1979	2,300,000	0.010	0.000
1980	2,500,000	0.050	0.000
1981	2,500,000	0.070	0.000
1982	2,500,000	0.100	0.010
1983	2,600,000	0.300	0.050
1984	2,700,000	0.500	0.070
1985	2,800,000	0.900*	0.100
1986	3,000,000	0.950	0.300
1987	3,200,000	1.000	0.500
1988	3,400,000	1.000	0.900*
1989	3,500,000	1.000	0.900
1990	3,600,000	1.000	1.000

*3-year acceleration

TABLE 5.10

ESTIMATED EFFECT OF
ERDA-SPONSORED DEMONSTRATION PROGRAM

Possible ERDA Effect	Added First Cost ($)	Average Annual Primary Energy Savings (10^6 Btu per unit)	Average Nationwide Years to Payback	Max. Percent Annual OEM Sales Captured	Max. Percent of In-Place Facilities Retrofitted*	Average Sales Rate 10^3 (1985) OEM	Retrofit	Percent of 1990 In-Place Furnace With Water Heater	Cumulative Energy Savings 1980-1990 in 10^{12} Btu Primary	Effect of ERDA
Maximum										
With ERDA	120	34.3	1.0	65	0	1604	0	29	1344	1135
W/O ERDA	200	28.0	2.0	37.5	0	97	0	11	209	
Likely										
With ERDA	120	34.3	1.0	65	0	1604	0	29	344	884
W/O ERDA	120	34.3	1.0	65	0	178	0	19	460	
Minimum										
With ERDA	200	28.0	2.0	37.5	0	870	0	18	611	402
W/O ERDA	200	28.0	2.0	37.5	0	97	0	11	209	

*This grows at a linear rate from 1/10 of the value shown in 1980 to equal the value shown in 1990.

We believe that a cumulative energy savings of around 880 trillion Btu's accumulated from 1980 to 1990 and increased sales shown in Figure 5.7 would result from an industry-ERDA development and demonstration program of the furnace/water heater.

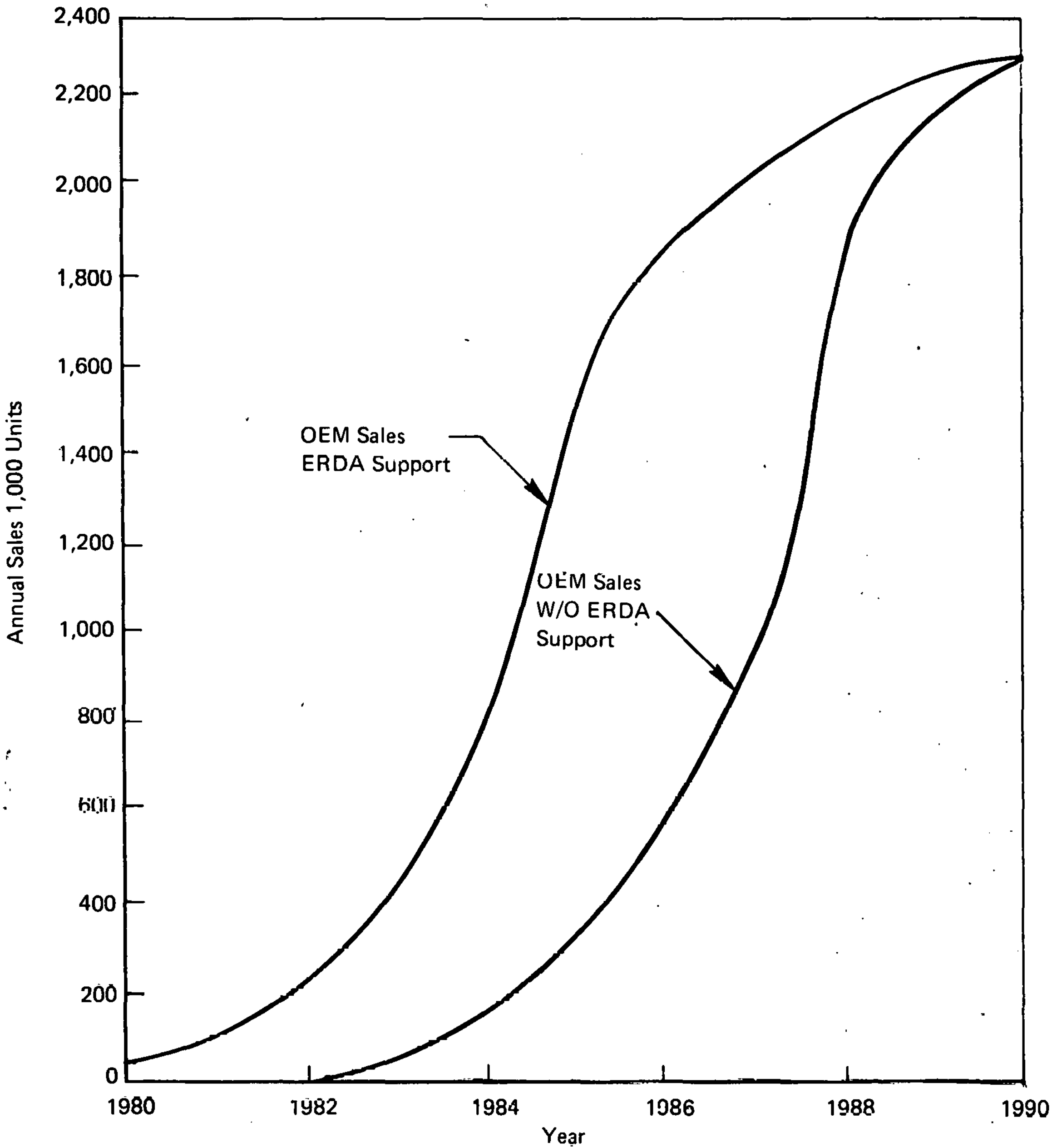


FIGURE 5.7 PROJECTED SALES OF THE INTEGRATED WATER & SPACE HEATING SYSTEM

6.0 COMMERCIAL RANGE HOOD HEAT RECOVERY SYSTEM

6.1 DESCRIPTION OF CONCEPT

6.1.1 Overview

The integrated appliance described in this section is based on the recovery of heat normally exhausted through the hood used over cooking equipment in commercial kitchens. Depending on the system configuration, this energy can be used either for space heating or water heating.

6.1.2 Precedents

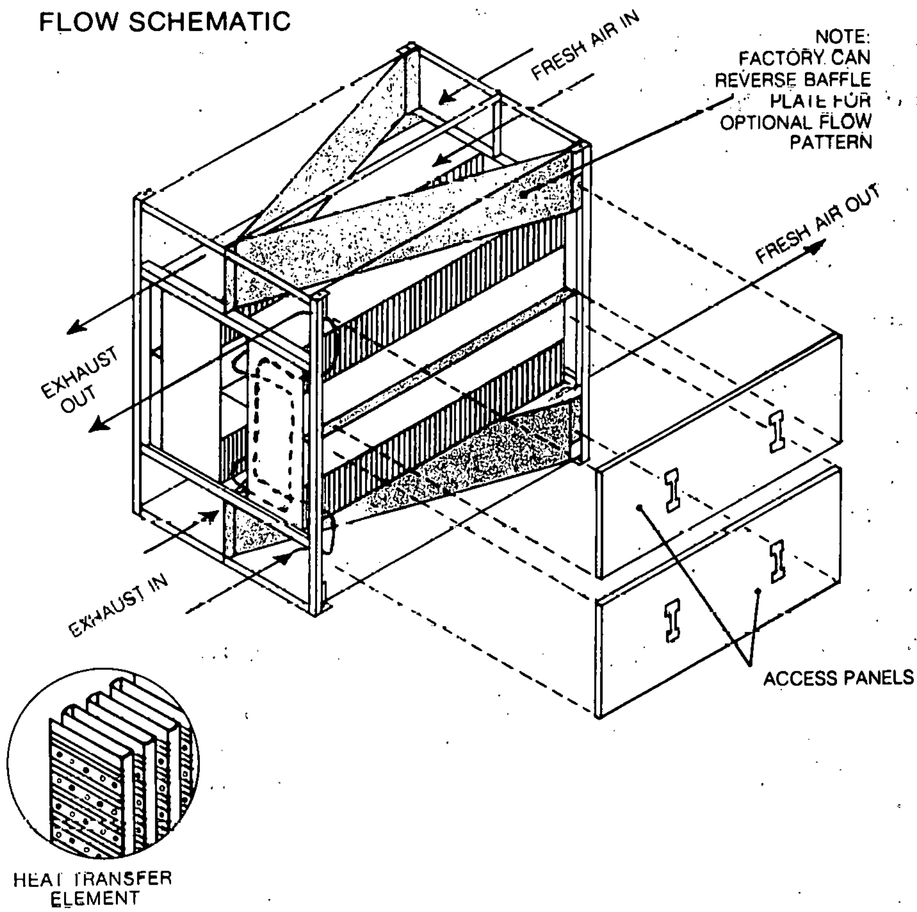
At this time several manufacturers have begun to develop and market devices designed to recover some of the energy exhausted through kitchen ventilation systems. The different types of systems are discussed below.

Air-to-Air Recovery Unit for Space Heating

Most units currently used in restaurants recover energy from the exhaust using an air-to-air type exchanger to heat the makeup air. There are two major manufacturers of this system--Gaylord Industries and Des Champs Laboratories Incorporated (DLI).

Gaylord is a hood manufacturer and sells their unit as an extension of their product line. The heat exchanger employed is a Q-Dot heat pipe based heat recovery unit. Their system is equipped with an automatic wash system which is tied in with the wash system used for cleaning the hood. Also included is the complete air handling system for both the exhaust and makeup air. Test results with an experimental unit installed on a White Tower Restaurant open 24 hrs/day in Toledo which were monitored by Toledo Edison have shown that during the winter months of November 1973 through March 1974 an average of approximately 80×10^6 Btu/month were recovered.

The DLI system employs their patented Z-Duct type heat exchanger consisting of folded thin aluminum sheets as shown in Figure 6.1; DLI sells a packaged unit with air handling equipment and a built-in wash system direct to the end user and also markets OEM to a number of companies including Air Systems and Air Distribution Associates who also sell kitchen hoods. Other major OEM purchasers of DLI units are Weather Rite, Inc., Proctor and Stuart, Applied Air Systems and Jackson and Church, all of whom sell makeup air heating equipment. According to DLI, makeup air heating equipment suppliers are concerned that their market will deteriorate due to the restrictions on the use of gas in new installations and are expanding their product lines to include heat recovery equipment.



*DLI – Des Champs Laboratories Incorporated.

FIGURE 6.1 DLI* Z-DUCT™ HEAT RECOVERY UNIT

These units have been in service in restaurant applications for approximately 2 years and sales are accelerating. According to DLI, there are no problems associated with the corrosion of the aluminun exchanger surfaces as long as the wash detergent is slightly alkaline.

Air-to-Water Recovery Unit for Space Heating

Gaylord Industries has installed two units on an experimental basis which recover heat from the exhaust using air-to-water exchangers. One of these units is installed in a Burger King in Anchorage, Alaska (see Figures 6.2 and 6.3). The unit consists of a heat recovery coil mounted on the hood. The coil is connected into the existing hydronic heating system and water is circulated by a pump through the heat exchanger and into the boiler when heat is required. This installation was able to meet a major portion of the building's heating load; however, the installation is somewhat unusual as Burger King uses high-energy input chain broilers* with inputs of 250,000 to 400,000 Btu/hr.

Since a large access area is not required for the conveyor broiler, insulated panels and removable skirts (see Figure 6.2) can be used to minimize the airflow volume and still maintain sufficient air capture velocities to carry away fumes. The reduced hood airflow volume for this type of unit means increased exhaust air temperature (in excess of 200°F) which results in a higher fraction of recovered energy with the same size recovery unit than in the case of an open broiler with higher volumes of exhaust air.

Gaylord has another installation employing a similar air-to-water exchanger which preheats water contained in a large storage tank which is part of a solar energy system. This system was designed as an energy conservation experiment for Burger King.

Another experimental system for recovering heat for preheating water is under test by Elsters in Colonie, New York. These tests are sponsored in part by ERDA and involve a number of different energy recovery systems. One system recovers heat from the hood using a heat pump and air-to-refrigerant recovery coil. This system has only recently come on line and operating data is limited.

6.1.3 System Designs

From the preceding description of the various systems being used to recover energy from the hood exhausts, it is apparent that a variety of kitchen waste heat recovery systems are possible. The choices are given in Figure 6.4.

The primary heat exchanger located in the exhaust system may be either an air-to-air or an air-to-water exchanger. If an air-to-air exchange system is used, the recovered energy heats the makeup air. This is a direct exchange and the energy must be used at once. If no space heating is required either because the outdoor ambient is warmer than the indoor temperature or because

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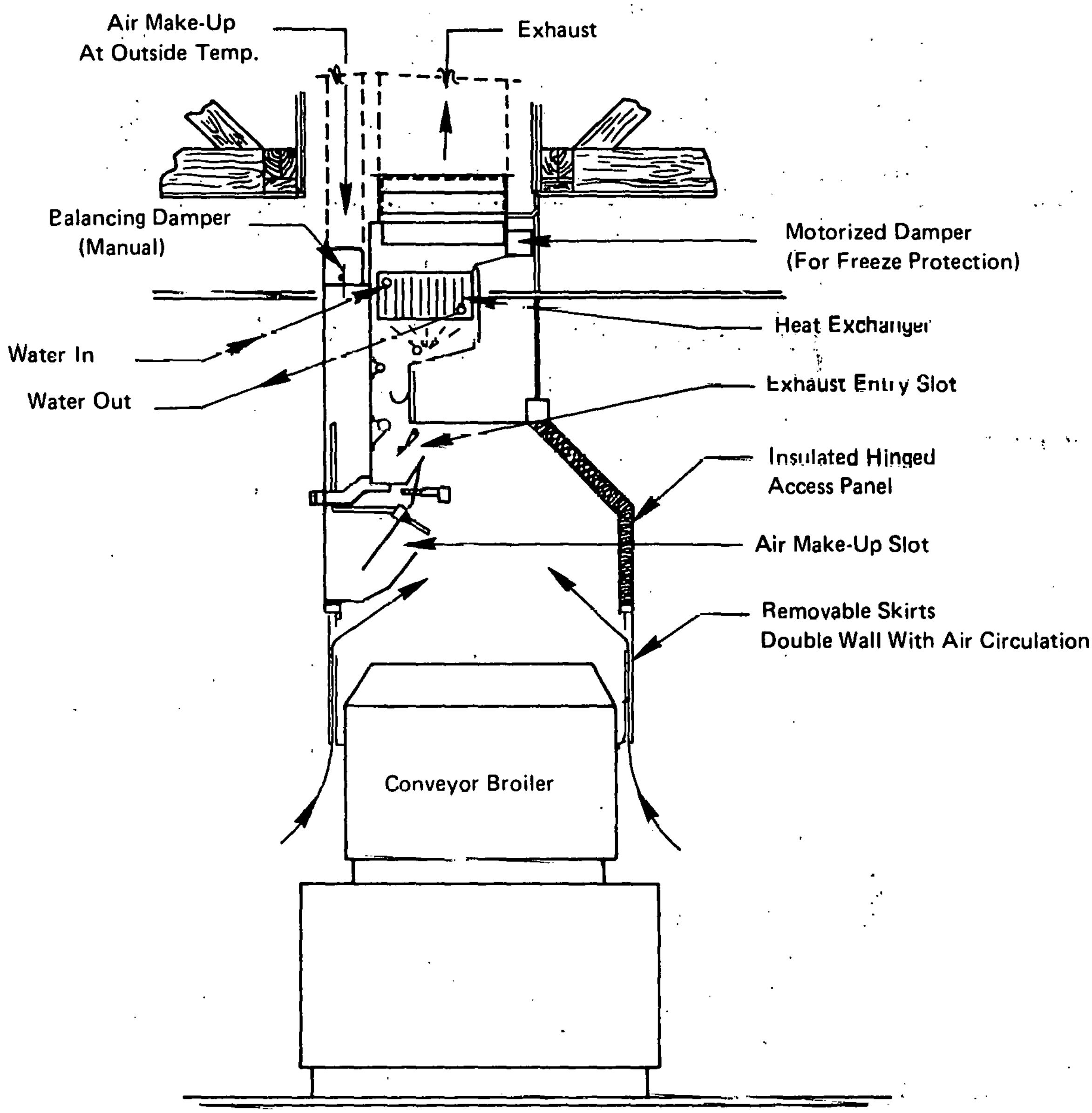
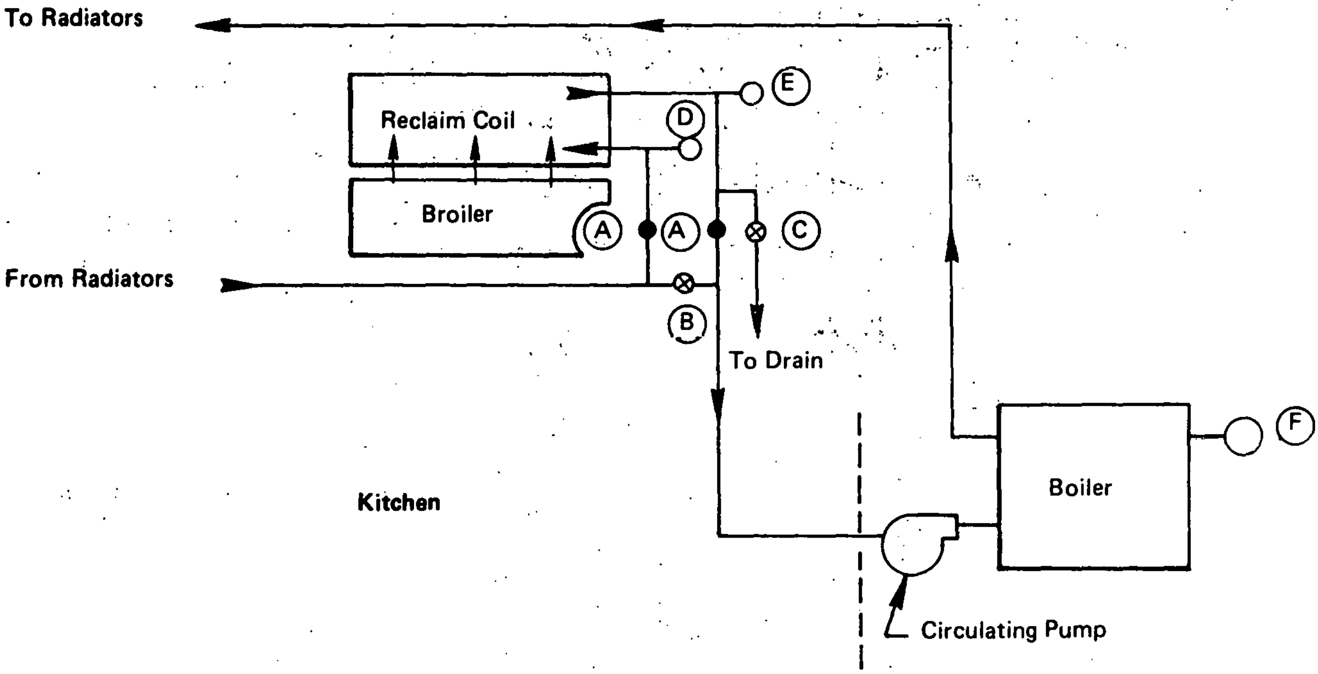
* Continuous-fired, conveyor-type broilers



Typical Section

Source: Gaylord Product Literature.

FIGURE 6.2 GAYLORD HEAT RECLAIM VENTILATOR FOR CONVEYOR BROILER



(A) – Valve Cocks To Coil
(B) – By-Pass Valve
(C) – Coil Drain Valve
(D) – Thermometer – Coil Inlet Temp.
(E) – Thermometer – Coil Outlet Temp.
(F) – Thermometer on Boiler – Water Temp.

Source: Gaylord Product Literature.

FIGURE 6.3 GAYLORD HEAT RECLAIM SYSTEM USED IN BURGER KING, ANCHORAGE, ALASKA

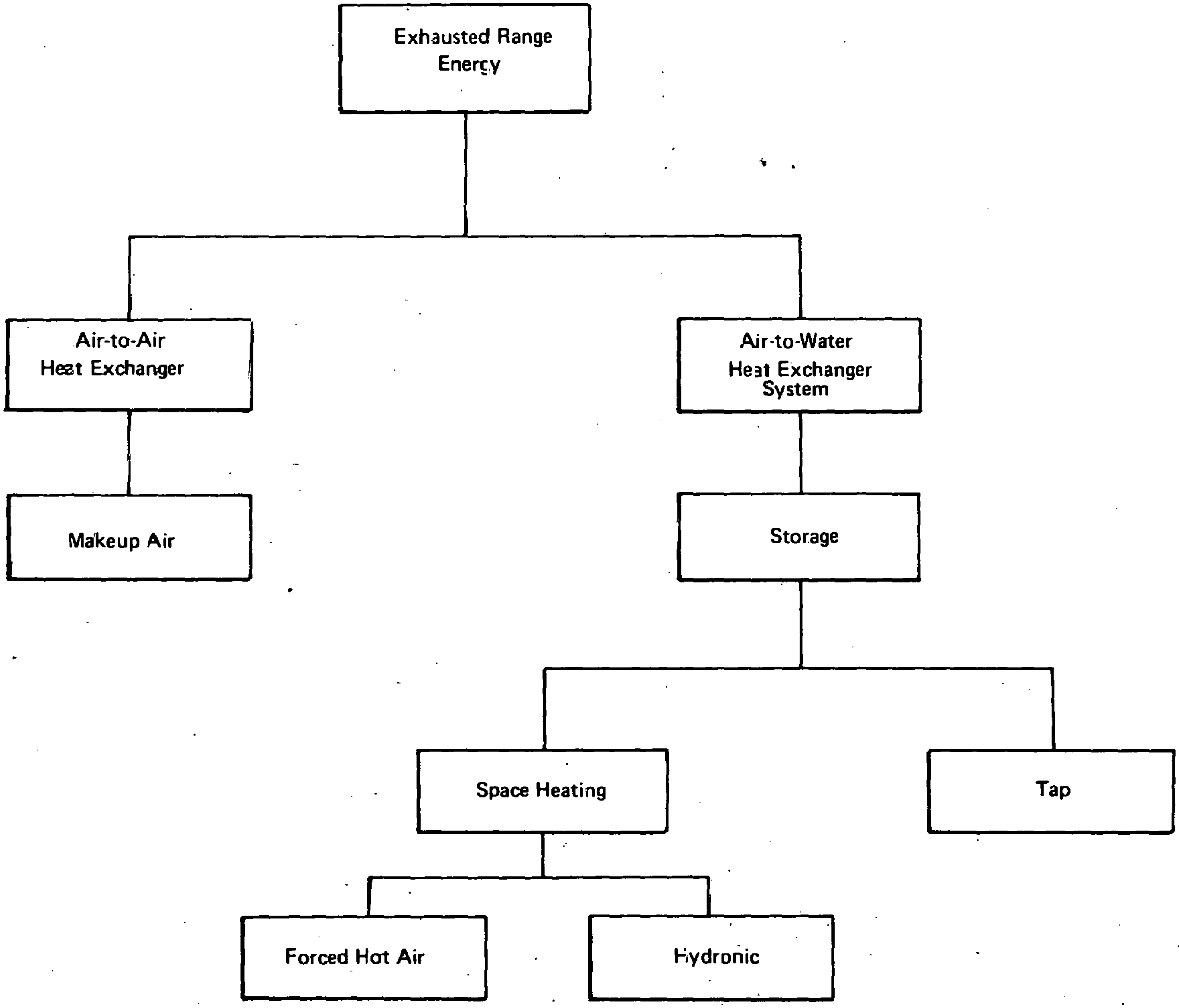


FIGURE 6.4 USES FOR RECOVERED ENERGY

heat sources within the kitchen offset losses, no recovered energy can be used. As noted previously, the only commercially available recovery units are of this type.

When an air-to-water recovery system is used, water is circulated through the primary exchanger and into a storage tank. This water may be withdrawn directly for use as tap water or it may be passed through a secondary heat exchanger for space heating. The space heating may either be forced hot air with a water-to-air exchanger positioned in the inlet air duct or a hydronic system with convectors positioned throughout the heated space. The Gaylord hydronic heating system falls into this category. The Elsters system is a variant of this system in which a heat pump replaces the water circulation loop for transferring recovered energy to storage.

In addition to these basic considerations pertaining to system configurations, other modifications are possible. For example, the heat exchanger can be positioned (1) in the hood, (2) between the hood and the exhaust blower, or (3) after the exhaust blower. The heat exchanger can be sold as (1) part of the hood, (2) part of the air handling system, or (3) as an independent system. The optimum selection depends on a variety of marketing questions which will be considered in later sections.

6.1.4 Major System Components

The components used will depend on the particular configuration chosen. The major elements of a hood recovery system are:

- hood,
- air handling system,
- heat exchanger,
- wash system, and
- storage (air-to-water system only).

Each of these is discussed in detail in the following sections.

Hood

A hood is basically a sheet metal fabrication, usually stainless steel, which is shaped to cover the cooking surfaces and to provide three primary functions. First, it collects cooking vapors. It does this by drawing a blanket of air continuously over the cooking surface. Second, it removes grease from the exhaust to prevent accumulations in the duct and the resulting fire hazard. Third, it serves as a fire control system. The hoods are equipped with fire dampers which close off the duct should a fire start on the cooking surface. In addition, they contain dry chemical extinguishing systems, which are heat activated. These systems are designed to extinguish fires on the cooking surfaces and in the hoods.

Hoods do not contain any air moving equipment. In an installation, one or more hoods are connected by a duct system to a blower located remotely in an equipment room or on the roof.

There are two types of hoods:

- filter hoods,
- ventilators or grease extractors.

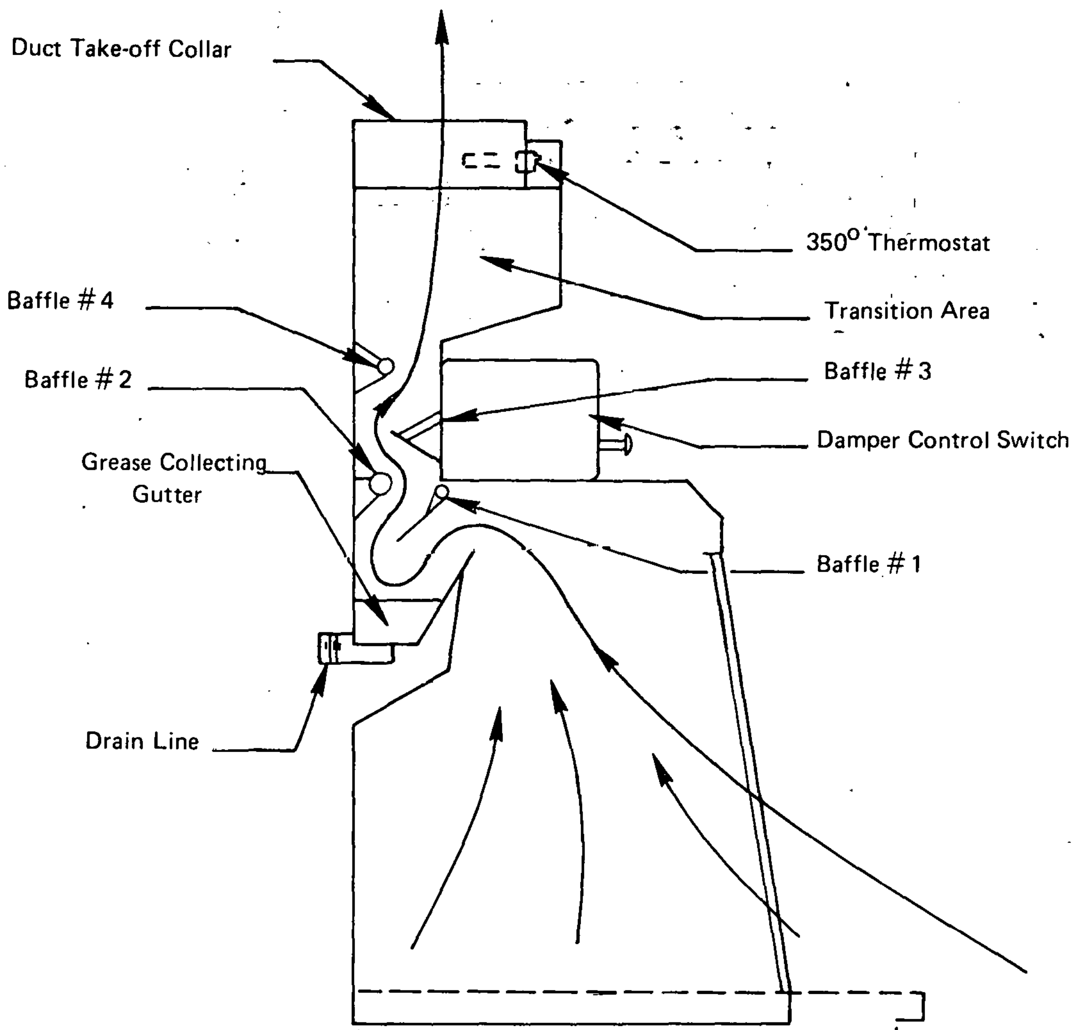
The filter hood removes grease by trapping it in metal mesh filters placed across the air inlet. These filters must be cleaned manually several times a week or they become clogged, thereby reducing the ventilation rate and increasing the fire hazard. The grease removal efficiency of these hoods is related to the airflow through the filters, but normally 50 to 60 percent of the grease in the air is removed. By NFPA code, airflow through a filter hood must be at least 100 SCFM for each square foot of canopy. The canopy must extend 6 inches beyond the cooking surface on all sides.

The ventilator or grease extractor removes grease by carrying the grease-laden air through several sharp turns where the grease is whipped against the hood walls by centrifugal action (see Figure 6.5). Once a day, these surfaces are automatically cleaned by a built-in washing system, which sprays them with hot water and detergent for 3-5 minutes. This system extracts more than 95% of the grease from the air. The code states that airflow should be "according to the manufacturer's specifications." The required flow is determined by the length of a 3-inch high air inlet slot which runs the length of the hood in the back. Generally, around 300 SCFM per linear foot is used.

The basic cost of a ventilator is two to three times as much as an equivalent filter hood. However, in many cases this difference is offset by a requirement for less air handling equipment due to the fact that the volume of airflow required by a ventilator is around 40% less than that required by the filter hood. (The pressure drop is approximately double.) This affects blower size plus the size of the heater and air conditioner required for the makeup air. In addition, because the grease extraction is more complete, the fire extinguishing system does not have to be as complex for the ventilator-based system. A rule of thumb for cost estimating is that a ventilator is roughly \$350/foot while a filter hood is \$120/foot. Both figures exclude the fire extinguishing system but the automatic wash equipment is included in the ventilator price.

Air Handling System

The role of the hood in the overall restaurant ventilation system is shown schematically in Figure 6.6. Typically, a roof-mounted exhaust fan pulls air from the kitchen and dining area across the cooking surface and out of the building. A second roof-top blower adds makeup air to the kitchen which is heated or cooled as required. The airflow



Hot, grease and odor laden air from the cooking surface merges with the cool blanket of air, then passes through the grease extracting chamber where the grease is removed by centrifugal force.

Cool air from the kitchen moves from the floor, blankets the front and top of the cooking unit and into the air inlet of the ventilator.

FIGURE 6.5 CENTRIFUGAL GREASE EXTRACTING PRINCIPLE

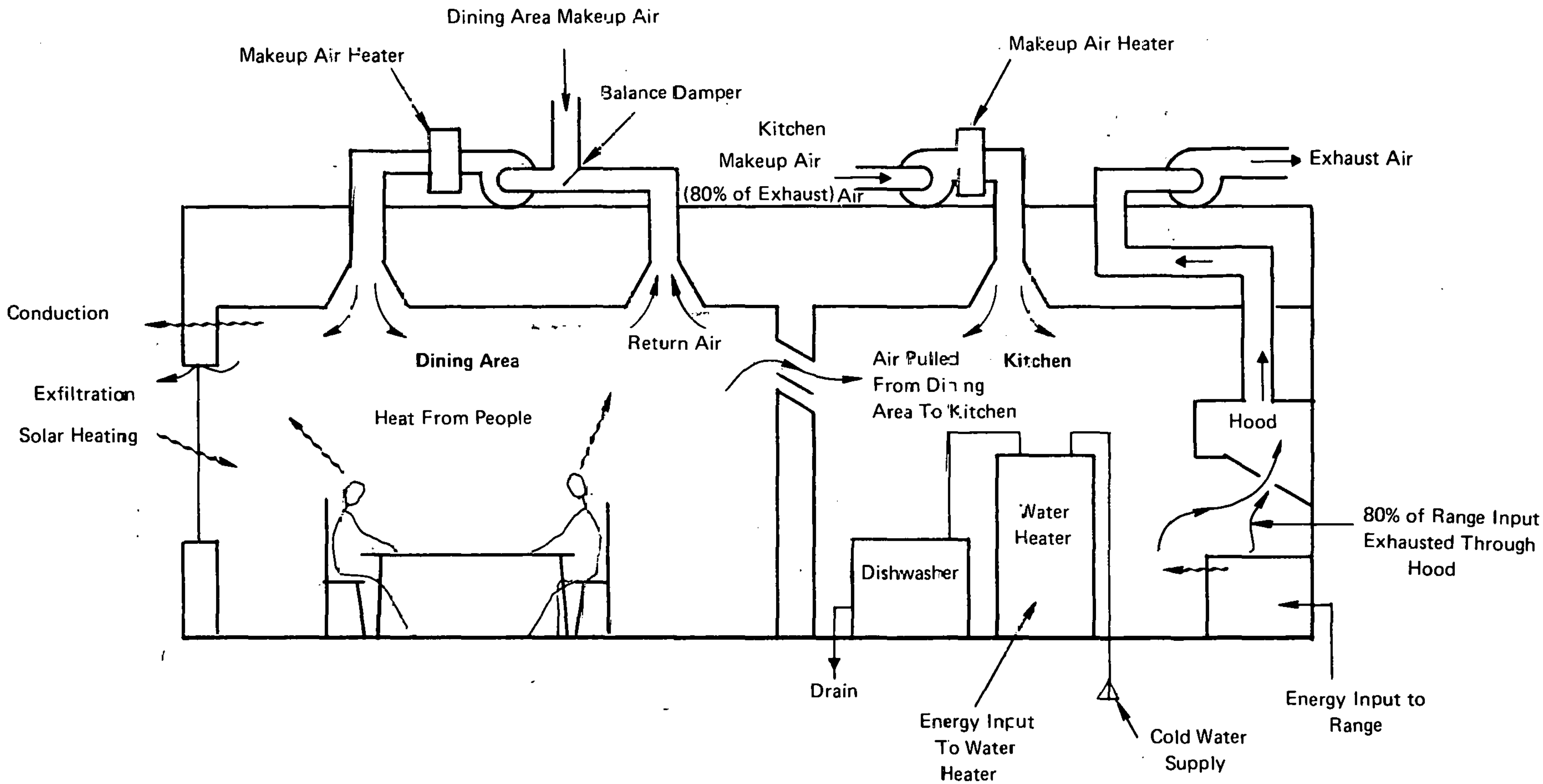


FIGURE 6.6 SCHEMATIC OF RESTAURANT ENERGY BALANCE

through this system is normally about 80% of that passing out through the hood. The additional 20% is drawn from the dining area. Air supply systems are designed in this way so that kitchen odors will not pass into the dining area.

Usually the dining area will have one or more independent systems supplying conditioned air. These systems recirculate some air but also add makeup air. The amount of makeup air is fixed by the maximum number of people the restaurant can seat. Normally the makeup air is greater than the air drawn from the dining area to the kitchen by the kitchen exhaust.

In calculating the heating and cooling requirement of a restaurant, a number of heat losses and gains must be considered. The major loss occurs from exhausted air. This air is heated not only by the space heating system, but also by the cooking equipment. Studies have shown that on the average, 80% of the actual input to the cooking equipment is lost through the hood[6.1]. In addition to this loss, normal losses occur due to conduction through the walls and the roof plus exfiltration.

Gains in the kitchen come primarily from the cooking equipment. This input is approximately 20% of the actual input to the equipment. In the dining area, the primary gains are due to the people being served and solar flux through the windows.

In a restaurant, another energy consumer is the hot water heater. In full menu operations, hot water in significant volumes is required for dishwashing. In addition, if the cooking facility is part of a larger building such as a hotel or hospital, hot water will be required for other uses as well.

Heat Exchanger

For air-to-air recovery systems, any heat exchanger made from acceptable materials can be used, provided that fin spacing is large enough to permit grease removal by an automatic wash system. As mentioned above, the Q-Dot heat pipe and the Z-Duct heat exchanger have been used, and other plate-type exchangers (such as Temp-X-Changer or Harrison) could be used just as well if the recovery efficiency and cost are acceptable.

The air-to-water exchanger must also have wide fin spacing (6-8 fins per inch) and like the air-to-air units must be made from corrosion-resistant materials. Based on the Z-Duct experience, copper tubes with aluminum fins should be acceptable.

A representative unit manufactured by Trane is shown in Figure 6.7. This exchanger is designed for heating air using hot water. A unit sized by Trane for a kitchen with an exhaust airflow rate of 9,000 SCFM, an air inlet temperature of 105°F, a water inlet temperature of 60°F, and water flow rate of 20 gpm would recover 283,000 Btu/hr using four rows of coils and a face velocity of 300 feet/minute.

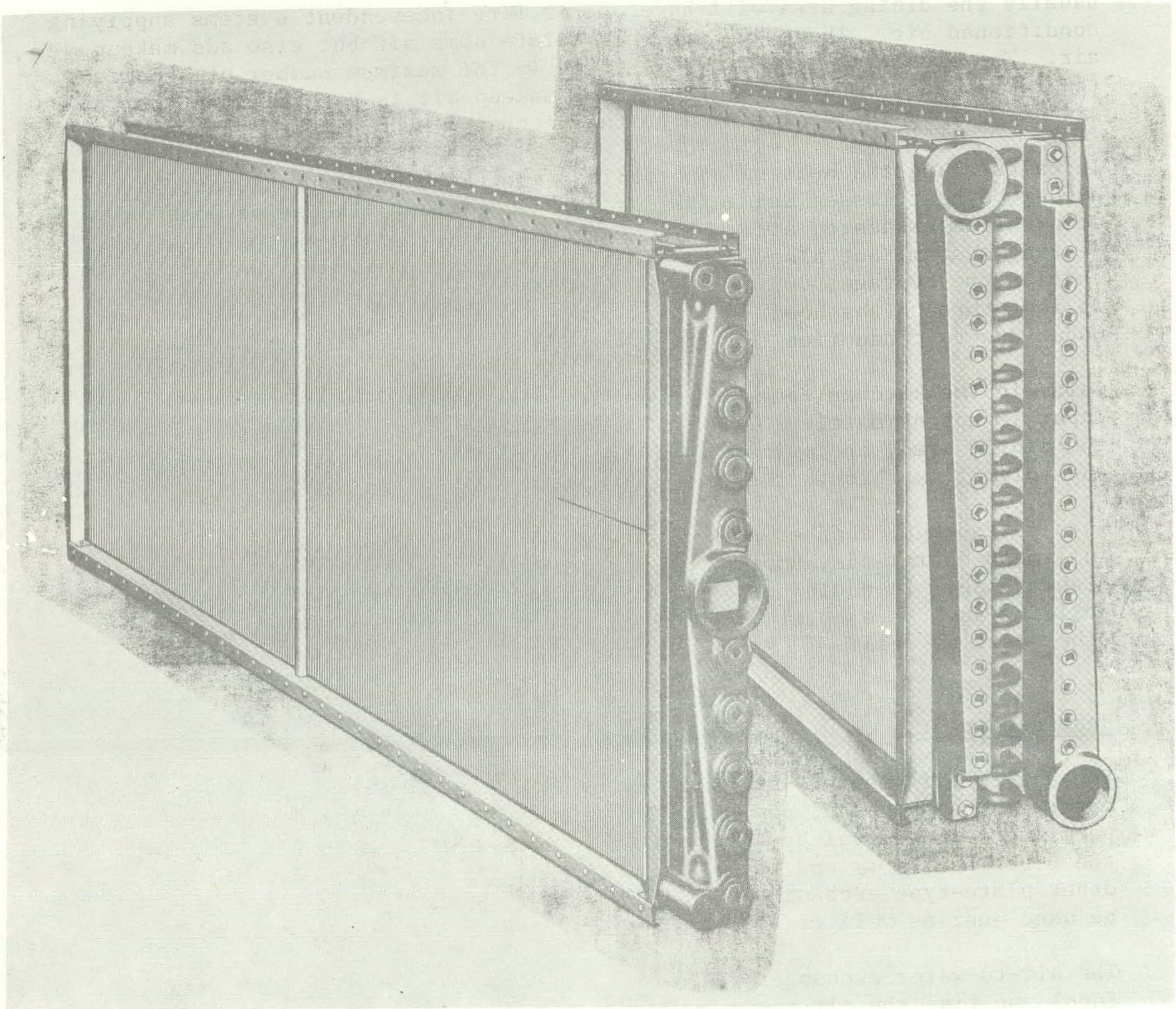


FIGURE 6.7 AIR-TO-WATER HEAT EXCHANGERS

Wash System

Two types of wash systems are currently used. In one system a bank of nozzles sprays the area to be cleaned for a fixed period of time (usually 4-8 minutes). A hot water and detergent-solution is used. For larger applications, this system may not work due to the pressure drop resulting from the high flow rates. An alternative system employs a single row of nozzles which is driven across the exchanger by a lead screw arrangement. This has the obvious disadvantages of requiring moving parts in a relatively severe environment.

In both cases, the operation of the wash system is keyed to the exhaust fan. Whenever the fan is turned off, the wash cycle is initiated under control of an automatic timer. Detergent is metered into the hot water supply line. Flows are controlled by solenoid valves and pressure regulators.

Storage

As was shown in Figure 6.4, storage is required when heat is being recovered for purposes other than immediate space heating.

Although other mediums might be considered, water is the most reasonable choice for this application. First, in this temperature range no other liquid has as high a specific heat. Second, storage technology is well developed. Third, water is often the ultimate media to be heated and the use of any other fluid would require a secondary exchanger with its additional cost and loss of recovery potential. Therefore, all system designs employ conventional glass-lined water heater tanks without heaters and controls as the energy storage device.

6.2 ANALYSIS OF DIFFERENT CONFIGURATIONS

6.2.1 Analytic Approach

Background Data

In order to estimate the energy-savings potential of this appliance, it is necessary to derive specifications for prototypical cooking facilities, to estimate the numbers of different types of establishments, and to project market penetration rates based on payback periods derived from cost estimates for different units.

Because the energy use patterns of different types of eating facilities vary greatly, it is necessary to consider several different categories for this analysis. Those used are:

- Fast foods,
- Restaurants,
- Health care facilities,

- Schools,
- Cafeterias,
- Hospitals, hotels, and motels; and
- Institutions (business cafeterias).

Data on the yearly operating characteristics for the different categories are given in Table 6.1. To obtain energy use estimates, first, specifications for a typical restaurant were derived[6.2] using data from an MRI report[6.3] and the Bureau of Census[6.4]. Next, specifications for other types of facilities were developed using the typical restaurant as the base.

The typical restaurant is described in Table 6.2. Atlanta was selected as the location for calculating space heating loads because it conservatively* represents the amount of recoverable waste energy usable for space heating.

Many restaurants of this size are actually located in rows of stores and have only a single window in the front. This configuration would have a smaller solar gain than the building chosen; however, the conduction losses would also be smaller due to the presence of buildings on each side. It is assumed that these factors are offsetting.

This kitchen size will also be assumed to be representative of a health care facility. However, no dining area will be used when calculating space heating loads. The average fast-food establishment will be taken to be 2/3 this size. Schools and cafeterias are double this size.

Larger hotels, motels, hospitals, and institutional feeders are three times this size based on industry-supplied data on average cooking lines. In this case, however, the cooking and eating facility will be assumed to be contained within a larger building so that kitchen ventilation requirements will be the only space heating load.

Inventory values for these categories are presented in Table 6.3. These values are based on Bureau of Census Data[6.5] and ADL estimates[6.6] plus contacts with industry sources, the U.S. Department of Agriculture, and the National Restaurant Association.

The energy-savings potential of this recovery unit depends on the average hood exhaust air temperature. This can be estimated by the following equation[6.7] for gas cooking equipment:

$$T_{exhaust} = T_{room} + \frac{.24 \text{ (nameplate rating (Btu/hr))}}{\text{exhaust rate (CFM)}}$$

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*The population-weighted mean number of degree days heating in the U.S. is 4361 degree days, and in Atlanta the number of degree days is 2983. The space heating energy requirements, and hence savings, are underestimated by this amount.

TABLE 6.1

OPERATING CHARACTERISTICS

Cooking Facility	Hours of Ventilator Operation Per Day	Days/Year Requiring Space Heating	Gallons of Hot Water Used Per Day	Days/Year Hot Water Required
Fast Foods	12	207	None	None
Restaurants	12	207	790[1]	312
Health Care Facilities	12	207[2]	790[1]	365
Schools	5	148	1,710[3]	180
Cafeterias	12	207[4]	2,110[1]	312
Hospitals, Hotels, Motels	12	207	15,000[3]	365
Institutions	5	148[4]	2,370[1]	260

[1] Water for kitchen only.

[2] No dining area.

[3] Includes usage for water in other areas.

[4] Heat required for makeup air only.

TABLE 6.2

TYPICAL RESTAURANT

Physical Plant

Free-standing cement block building, 13 feet high
Location: Atlanta, Georgia
Kitchen Area: 1,000 ft^2--windowless
Dining Area: 2,000 ft^2--35% of walls are 1/4 inch thick glass
Cooking Line: 10 ft long with nameplate rating of 370,000 Btu/hr
Seating Capacity: 65 persons

HVAC

Dining area ventilation: 65 seats x 15 CFM/seat = 975 CFM
Exhaust Hood: 10 ft long at 300 CFM/ft = 3,000 CFM
Separate, roof-mounted furnaces for kitchen and dining areas

Loads

150 customers/day
- 390 Btu/hr heat input/person*
- 1 gallon hot water for dishwater/person**

2,983 degree days per year
50°F mean temperature
207 day heating season
Hot water usage varies depending on type

Sources:

*ASHRAE, Handbook of Fundamentals
**Derived from sizing information in Hobart catalog #11.25/Ho 1976

TABLE 6.3

PRESENT AND PREDICTED INVENTORY OF EATING PLACES

Class of Cooking Facility	Number of Establishments 1970	1990[1]
Fast Food	69,000 ± 2,000	126,000 ± 15,000
Restaurants	116,000 ± 10,000	125,000 ± 10,000
Health Care	24,000	40,000
Schools	27,000 ± 5,000	120,000 ± 10,000
Cafeterias	12,000 ± 500	25,000 ± 5,000
Hospitals, Hotels, Motels	11,000 ± 1,000	14,000 ± 1,500
Institutions	5,600 ± 500	12,000 ± 2,000
Total	264,000	462,000

[1] Arthur D. Little, Inc. Estimates

For the typical restaurant with an exhaust rate of 3,000 CFM and an input rating of 370,000 Btu/hr, this equation predicts an average temperature rise of 30°F or an exhaust air temperature of 105°F for a kitchen air temperature of 75°F. At peak periods during the day, the temperature can be expected to be higher and at other times lower than this average value.

Configurations Considered

As described in Section 6.1.3, the energy recovered from the kitchen exhaust can be utilized in three different ways. First, it can be used with an air-to-air recovery system for heating makeup air directly while the facility is in operation. Second, it can be recovered using an air-to-water heat exchanger, put into storage for use as required for space heating. Third, it can be recovered and put into storage as with the second option but the water can be withdrawn for use as tap water when required.

A recovery system can be based on any one or a combination of these configurations as long as sufficient energy is available. In most cases, the energy which is exhausted through the hood can meet both the water heating and space heating loads. Thus, most cooking facilities could use an air-to-air recovery unit for space heating and an air-to-water recovery unit for water heating. In this case, the recovery system for water heating would be placed in series with the recovery system for space heating. The air temperature entering the space heating unit would be reduced an amount depending on the water heating load, but in most cases sufficient energy would still be present to heat the makeup air. The energy savings resulting from both systems combined is approximately equal to the sum of the individual systems for the average Atlanta facility where there is sufficient recoverable energy to meet both the space and water heating requirements. In more northern climates, the same relation holds true since energy recovered for water heating is extracted from the exhaust above 70°F, and the space heating recovery draws the remainder of the waste heat down to close to the outside air temperature.

Space heating from water storage has a prohibitively long payback period (greater than 17 years for most applications) due to the comparatively low exhaust temperatures (100-110°F) and the requirement for a secondary heat exchanger with its reduction in overall recovery efficiency.

A single heat exchanger, as opposed to two in series, might be designed for the combined space heating and water heating application. A plausible design consisted of a heat exchanger made from "Roll Bond*" heat exchangers to serve as both an air-to-air and an air-to-water unit. On close examination, it was found that the cost of the system would be greater than two

*A heat exchanger made by Olin Brass, consisting of two sheets of metal bonded and rolled together with liquid passageways maintained between the flat plates.

independent units. The major reasons for this were that a fairly complex assembly procedure would be required to insure that no cross contamination could occur, resulting in additional material use and labor cost. Because of the high relative cost per square foot of surface area of the "Roll Bond" versus a conventional finned heat exchanger, the overall system cost was found to be excessive.

The configuration used for the analysis of the air-to-air recovery system is designed around the Z-Duct heat exchanger described above. The costs are based on retrofitting the recovery system to the existing makeup air unit.

The air-to-water heat recovery coil could be positioned in the hood, in the duct between the hood and the blower, or on the exhaust side of the blower. In order to make the system retrofittable, the last position has been selected. In order to install a coil in the hood itself, each hood design would have to be treated separately. Major modifications to the hood would be required including cutting through the end, reworking the wash system if one were in place or adding it if not, attaching mounts for the coil and putting it in, etc. In addition, if these modifications were made in the field, conflicts with local building codes are likely because building codes require NFPA and/or Underwriters Laboratory approval on hoods and extensive modifications after installation could void the certifications. Also, the installation time would add significantly to the cost of the unit and would in many cases interfere with the operation of the cooking facilities. Similar problems would be involved in developing a universal design for installation between the hood and the exhaust blower.

The air-to-water system design developed for this analysis is based on retrofitting a heat recovery system at the exit of the existing exhaust blower. This system avoids most of the difficulties associated with the other configurations. In addition, it has the obvious advantage that its sales potential is not restricted to new installations.

The general layout of this system is shown in Figure 6.8. The recovery unit is attached to the exhaust fan. Water is circulated through a heat exchanger coil mounted in the recovery unit by a pump and into a storage tank. When hot water is drawn for dishwashing, the water preheated by the recovery unit is drawn into the existing water heater where it is heated to the desired level.

A more detailed drawing of the system is shown in Figure 6.9. The system actually consists of two basic parts: the recovery unit mounted on the roof and the storage system with associated plumbing positioned near the existing water heater. The recovery unit contains a coil, a bank of nozzles for cleaning the coil and a drain system to carry wash water and condensation to the existing drain system in the building. The coil has copper tubes with aluminum fins spaced 6 1/2 to the inch for easy cleaning. It has 6 rows and a face area of 30.6 ft^2 for an exhaust flow of 9,000 SCFM. The unit is a standard Trane product for water-air heating systems. The nozzle bank is similar to that employed by Gaylord in the HRU space heating system. It is attached to an existing plumbing enclosure which handles the automatic wash of the hood surfaces. The wash cycle is executed each time the exhaust system is shut off.

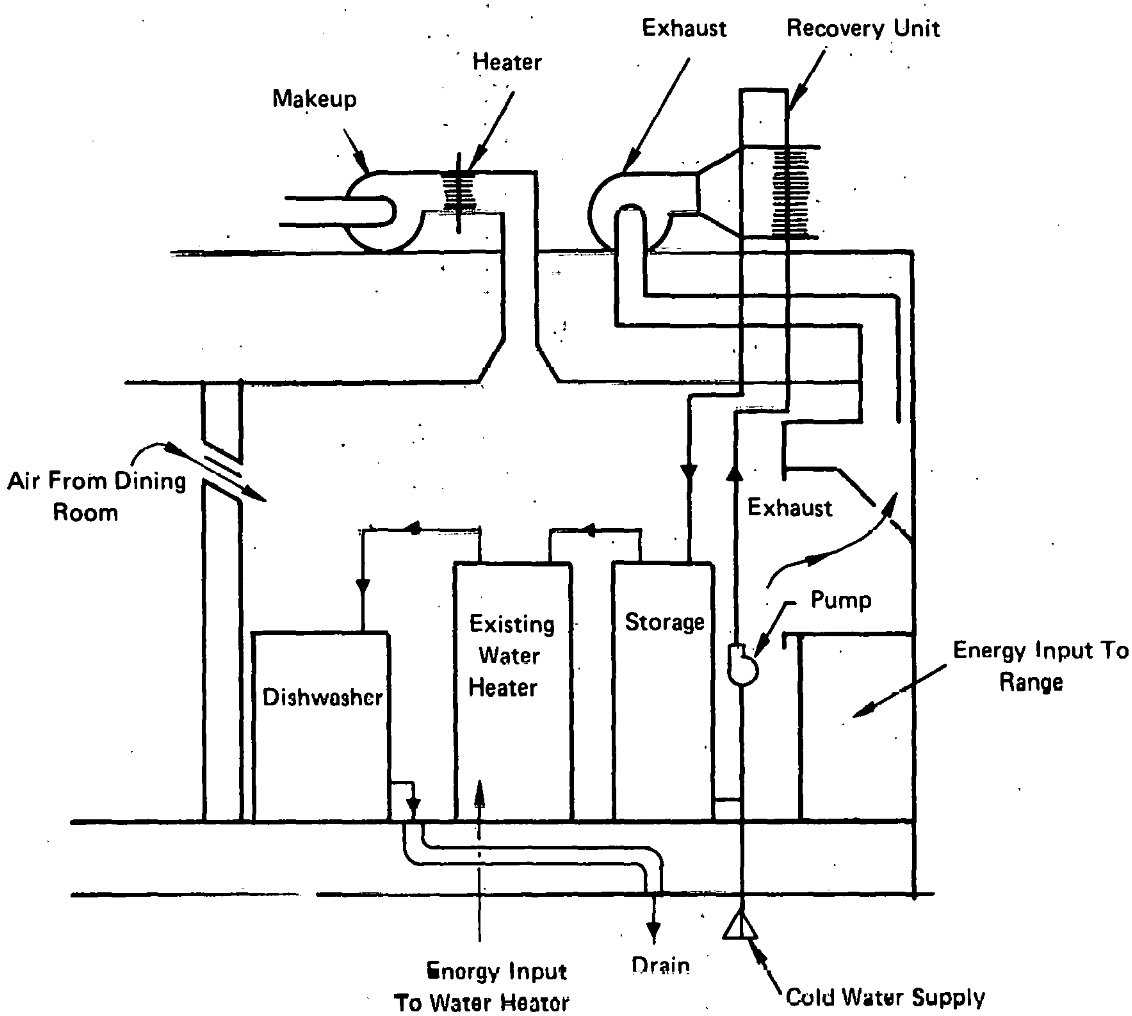


FIGURE 6.8 RANGE HOOD RECOVERY UNIT INSTALLATION

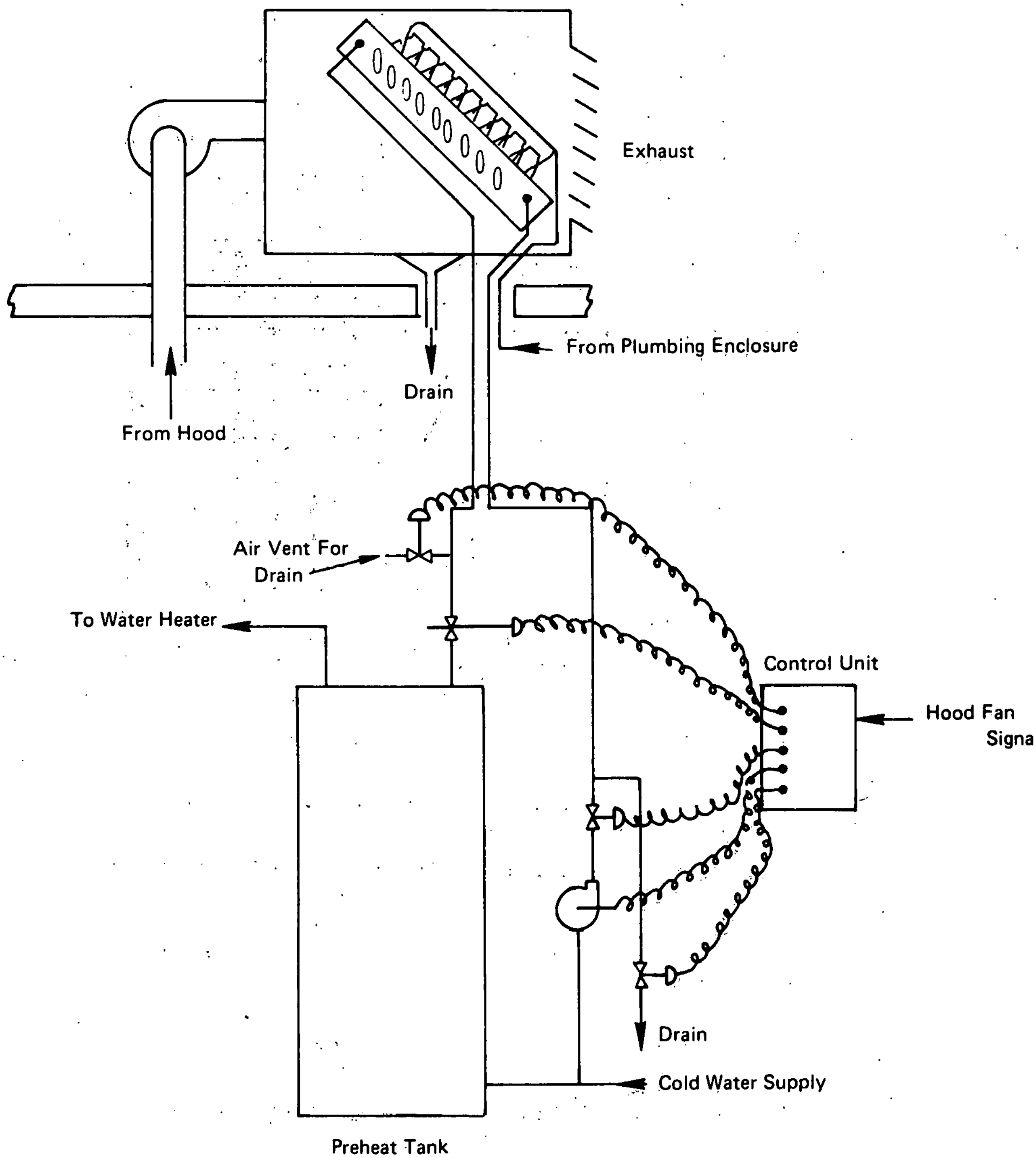


FIGURE 6.9 RECOVERY AND STORAGE SYSTEM

The storage system contains a storage tank, a circulating pump, a freeze protection system, and associated plumbing. These are all mounted and wired in a self-contained unit. At installation, the unit must be plumbed in as shown and wires from the plumbing enclosure attached to the appropriate wires in a small enclosure mounted on the tank. These wires will bring power to the unit whenever the exhaust fan is turned on. When power is applied, the circulating pump will start and the solenoid valves will be energized opening the pump-coil storage tank loop. When power is turned off, the de-energized solenoid valves close this loop and open a path between the coil and the drain which automatically empties the coil to prevent freeze-ups from occurring.

The circulation rate and the storage tank are sized by considering the expected daily water use load. The first item to be sized is the pump circulation rate. This is selected so that on the average, slightly less water will be circulated by the pump than would be used in a day. This assures that all the water preheated during the day is actually used and minimizes the storage required and the standby losses. Thus, for example; if an average total hot water use of 1,200 gallons per day over a 12-hour day were anticipated, the pump flow rate should be slightly less than 100 gallons/hr or about 1.5 gallons/minute.

The storage tank is sized so that during a normal day, assuming a well stratified tank, the warm-cold interface never reaches the point at which water is drawn by the pump. This is done to insure a supply of low temperature water to the coil in order to maximize the energy recovery. In addition, the circulating pump is attached to the cold water inlet so that, as long as hot water is drawn at a rate greater than the circulating rate, cold water will be drawn by the pump. The cold water inlet at the bottom of the tank is fitted with a diffuser tube to help maintain stratification in the tank. The actual tank size selection must be made based on the type of installation as different types of eating facilities have widely different hot water uses depending on the type of menu and other activities in the building which may require hot water. The typical storage value lies between 1.5 and 3 times the average hourly use.

6.2.2 Parametric Analysis of Systems

Air-to-Air Recovery for Space Heating

In this case, an air-to-air heat exchanger used to heat makeup air with no storage is analyzed. Several factors should be considered when examining the results of the analysis on this system. First, although the maximum energy-savings potential is high, it is based on the assumption that everyone will have a system in use in 1990. Actually, as will be shown later, the relatively poor payback period will cause the actual penetration to be significantly below this value. Second, the poor paybacks are based on national average weather conditions. Thus, there are many places in the country where there will be much better payback periods and sales potentials. Estimates of the maximum energy savings potential in different commercial kitchens using a hood heat recovery system are presented in Table 6.4. Also shown in Table 6.4 are specific costs, built up from component costs shown in Table 6.5 and discussed below. In some cases, heated makeup air is provided to both the kitchen plus a dining area; in others, only the kitchen is heated.

The general configuration on which this analysis is based contains a heat exchanger, a wash system, controls, ducting, and a fan to overcome the added resistance due to the pressure drop through the exchanger. It is likely that the existing exhaust fan will have to be removed and an additional fan added to the makeup air circuit in series with the present blower. The heat recovery package configured in this way will be suitable either for new or retrofit sales.

For the typical restaurant with a makeup airflow rate of 2400 CFM (80% of 3,000 CFM exhaust air) for the kitchen and 975 CFM for the dining area, plus conduction losses and gains from various sources throughout the day, the peak demand is 73,600 Btu/hr when the outside air temperature is 50°F. The basis for estimating the costs of these components for different systems is discussed below.

An aluminum-type exchanger can be used for this application. A series 7500 Model M-4 Z-Duct made by Des Champs Laboratories Incorporated, East Hanover, New Jersey would be suitable. With an automatic wash system, it sells for $3,100. An additional $750 should be allowed for duct work, by-pass dampers, installation, and transportation, and $1,500 for added fans. This unit would be retrofitted to the existing roof top ventilation system containing a makeup heater for backup purposes.

For schools, the exhaust airflow rate is 6,000 CFM, and the peak load during the day is 162,000 Btu/hr. The additional cost of this added capacity is estimated to be $1,000 for the heat exchanger plus $1,500 for the fans.

For cafeterias, the requirements are simply double those of the typical restaurant. The required unit is estimated to cost $2,000 more than the restaurant for the added heat exchanger and wash capacity plus the same fan size as the schools.

Units for hospitals, hotels, and motels will typically handle three times the energy of the restaurant. The added heat recovery capacity will cost $3,000 more than the restaurant unit, plus $4,500 for the fan.

Institutions (companies), although they have cooking equipment inputs roughly three times those of the typical restaurant, have different load characteristics since only one meal/day is served. The peak demand for these facilities is 222,000 Btu/hr. To meet this requirement with an exhaust flow rate of 9,000 CFM requires a heat exchanger which is the same size as the unit required for the hospital, hotel, motel case.

TABLE 6.4

YEARS TO PAYBACK AND NATIONWIDE ENERGY SAVING FOR SPACE HEATING

	Yearly Energy[1] Savings/Unit (10^5 Btu's)		Installed Cost of Recovery Unit ($)	Years to Payback		Total Primary Energy Savings Potential in 1990 (10^{14} Btu/yr)[6]
	Gas[1]	Electric[2]		Gas Heat[4]	Electric Heat[5]	
Fast Foods	159	95	5,350	9.5	5.0	.24
Restaurants	237	142	5,350	6.4	3.3	.36
Health Care Facilities	145	87	5,350	10.6	5.6	.07
Schools	95	57	7,850	23.6	12.5	.14
Cafeterias	473	284	8,850	5.3	2.9	.14
Hospitals, Hotels, Motels	711	426	11,350	4.6	2.3	.12
Institutions	116	70	11,350	2.8	14.6	.01
					TOTAL	1.08

[1] Assumes replace gas makeup air heater with recovery unit plus resistance heater. Seasonal efficiency of gas unit was 0.6.

[2] Replace electric heat with recovery system plus electric, seasonal efficiency of old system was 1.0.

[3] Although exhaust requirements slightly smaller, no appreciable savings possible.

[4] $3.50/mm Btu

[5] $0.038/kwh ($11.13/mm Btu) for projected commercial rate for electric power in 1985.

[6] Assumes 75% gas, 25% electric

TABLE 6.5

SPACE HEATING (AIR-TO-AIR) COMPONENT COSTS
(in Dollars)

	Heat Exchanger and Wash System	Duct Work, Installation, Transportation	Fans	Total
Fast Food	3,100[1]	750	1,500	5,350
Restaurant[2]	3,100	750	1,500	5,350
Health Care Facilities[2]	3,100	750	1,500	5,350
Schools	4,100	750	3,000	7,850
Cafeterias	5,100	750	3,000	8,850
Hospitals, Hotels, Motels	6,100	750	4,500	11,350
Institutions	6,100	750	4,500	11,350

[1] DLI Quote, other numbers Arthur D. Little, Inc., estimates

[2] Same ventilation requirements as fast food

Air-to-Water Recovery for Water Heating

Basis for Energy Savings

The operating characteristics and the energy saved by recovering energy for water heating using this system are summarized in Table 6.6 for the different categories of eating places.

Cost of Systems

The major cost elements for this system are:

- Heat Exchanger,
- Storage Tank,
- Automatic Wash System,
- Installation Costs, including materials, controls, and pump.

Each element is discussed below, and the costs are summarized in Table 6.7.

1) Heat Exchanger

Several types of heat exchangers are possible for this application. The primary constraint is that the configuration be such that it can be easily cleaned. If used with a centrifugal type grease extractor, the amount of grease which might collect on the exchanger during a 24 hour period is small. In order to simplify the wash system, it is desirable to design a system employing a compact heat exchanger and to configure the system so that the existing wash system control can be used.

A unit meeting these requirements was selected for larger installations (hotels, motels, hospitals) using a Trane computer program to provide performance data. Several different sizes with fin spacing of 6.5 fins/inch were considered, and four possible exchangers identified. To select the best unit, the cost of each type of exchanger was added to the cost of the other system elements (assumed independent of heat exchanger configuration) and divided by the effectiveness to obtain a number proportional to dollars per Btu recovered. The results are shown in Figure 6.10, giving an optimum effectiveness around 0.73. This compares favorably with the predicted optimum of 0.77 based on the general heat exchanger optimization guide given in Appendix C.

The general heat exchanger optimization technique given in Appendix C was used to size the heat exchanger for the other applications (restaurant, etc.).

TABLE 6.6

ENERGY RECOVERY FOR WATER PREHEATING

	Heat Exchanger Area (ft^2)	Water Circulation Rate (GPM)	Storage Required (Gal)	Water Preheated Per Day (Gal)	Operating Days Per Year	Energy Recovered Per Day (Btu)	Energy Recovered Per Year (10^6 Btu)
Restaurants	112	1.5	100	790	310	190,000	59
Health Care Facilities	112	1.5	100	790	365	190,000	70
Schools	448	6.0	240	1,710	180	397,000	71
Cafeterias	224	3.0	240	2,110	310	508,000	158
Hospitals, Hotels, Motels	1,460	20.0	500	14,400	365	2,750,000	1,000
Institutions	635	8.5	500	2,370	260	508,000	132

TABLE 6.7

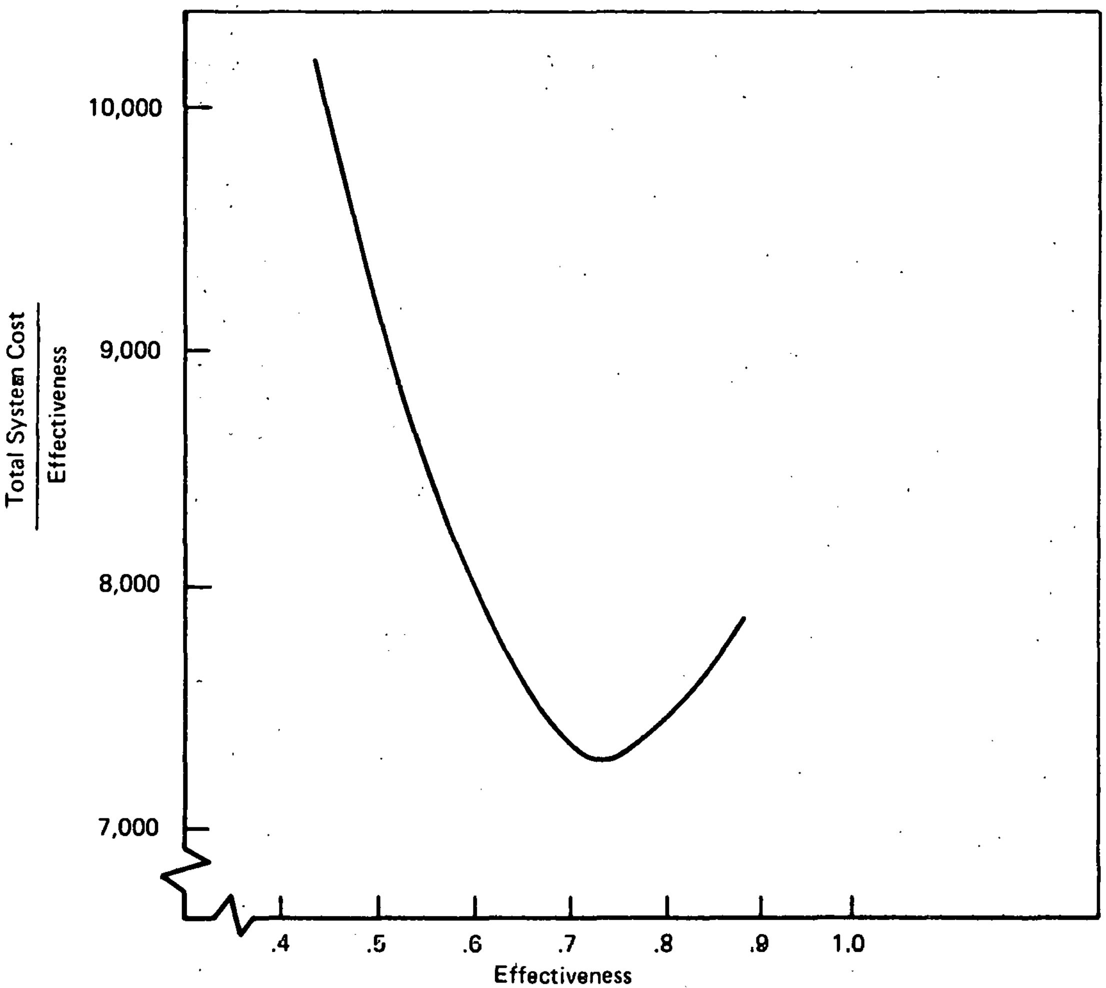
COST BASIS HEAT RECOVERY
FOR RETROFIT WATER HEATING SYSTEM

	Coil Area (ft^2)	Storage (gal)	COST ($) Heat Exchanger*	Storage	Wash	Inst.	Total
Restuarants and Health Care	112	100	210	200	300	450	1,160
Schools	448	240	550	480	450	850	2,330
Cafeteria	224	240	320	480	350	550	1,700
Hospitals, Hotels, Motels	1,460	500	1,550	1,000	980	1,800	5,330
Institutions	635	500	740	1,000	560	870	3,170

*The costs of the heat exchangers were calculated using the relationship:

$$\text{Cost (\$)} = \$100 + [\text{Surface Area } (ft^2)] \times \$1/ft^2$$

where the surface area is the area of the side of the heat exchanger with the smaller fluid capacity rate ($C_p\dot{m}$). The price of the optimum heat exchangers for hospitals, hotels and motels quoted by Trane of $1,500 fits this expression well. Other data points have been used to verify it for smaller sizes.



Note: Total System Cost is Fixed Cost Plus Cost of Different Heat Exchanger Sizes.

FIGURE 6.10 SELECTION OF OPTIMUM HEAT EXCHANGER SIZE

2) Storage Tanks

The price of insulated, glass-lined pressurized storage tanks is taken to be $2 per gallon for the range of sizes used in this system. Unfortunately, the requirement for good stratification makes it necessary to use a separate tank for this purpose.

3) Automatic Wash System

The automatic wash system consists of nozzles, solenoids, a timer, a pump to meter out detergent, and various controls. If a system is in use in the hood, some of these items do not need to be duplicated. The cost for a complete wash system for a Z-Duct exchanger is $1,000. An add-on system is estimated to cost $300. It is estimated that each additional 100 square feet of exchanger surface will add $50 to the cost of the wash system.

4) Installation and Miscellaneous Items

Materials in this category include ducting tubing, wiring, solenoids and controls for freeze protection, circulation pump, etc. In addition, an allowance must be made for installation labor and transportation. A materials cost of $150 plus an installation cost of $200 and an average transportation cost of $100 give a total cost of $450 for this item. These values will change slightly for larger systems, but for the purposes of this analysis, the increase, except for transportation which will be scaled by unit capacity, will be neglected as a second-order effect.

Energy Savings

Table 6.8 presents energy savings and payback periods for the different applications with water heating only using those cost figures.

Conclusions

As discussed in Section 6.2.1 earlier, the recovery system for space heating and the recovery system for water heating are not competitive systems for both can be used in a single facility for increased energy savings. Indeed, the system may be considered complementary since the increased acceptance of one will enhance the credibility and acceptance of the other. However, the space heating recovery system is a developed product and is already available in the marketplace, while the retrofittable water heating unit has not been developed. Both systems offer about an equal (additive) national energy-savings potential if fully implemented into all possible facilities. Since the water heating heat recovery system is still a number of years away from production, its

TABLE 6.8

YEARS TO PAYBACK AND NATIONWIDE ENERGY-SAVING POTENTIAL FOR RETROFIT WATER HEATING SYSTEM

	Yearly Energy Savings Unit (10^6 Btu's)		Installed Cost of Recovery Unit ($)	Years to Payback		Total Primary Energy Savings Potential in 1990[5] (10^{14} Btu/yr)
	Gas[1]	Electric[2]		Gas Water Heating[3]	Elec.Water Heating[4]	
Restaurants	74	59	1,160	4.5	1.8	.12
Health Care Facilities	87	70	1,160	3.8	1.5	.05
Schools	89	71	2,330	7.5	2.9	.14
Cafeterias	198	158	1,700	2.5	1.0	.07
Hospitals, Hotels, Motels	1,250	1,000	5,330	1.2	0.5	.24
Institutions	165	132	3,170	5.5	2.1	.03
					TOTAL	.65

[1] Recovery efficiency is 0.8

[2] Recovery efficiency is 1.0

[3] $3.50/mm Btu fuel cost

[4] $.038/kwh power cost

[5] 75% of gas, 25% electric water heaters

potential 1990 energy savings will not be realized without some federal assistance, such as the National Demonstration Plan outlined in the following section.

6.3 NATIONAL DEMONSTRATION PLAN

6.3.1 Introduction

One of the major barriers to the commercialization of the range heat recovery system for water heating is skepticism among potential customers as to its real value. In commercial establishments, the person making the decision to purchase a heat recovery system has typically been contacted by several salespersons offering a variety of "energy-saving" appliances. Competitors may make unfounded claims for their equipment and unjustified criticisms of the equipment of others. Because of lack of data from unbiased sources, the buyer is unable to assess the validity of these claims and in many cases elects to follow a safe course and does nothing. The difficulty lies in the fact that the actual energy used to perform different functions and the various duty cycles must be carefully considered for each individual application before a judgment can be made as to the efficiency of a particular energy-recovery system. For example, schools which have a reasonably high hot water usage and larger cooking facilities would seem like obvious candidates for this appliance. However, in many cases, because the cooking is done at a high rate over a short time, the cost of the heat exchanger and storage tank is so large that given the comparitively small number of days of use per school year, the payback period for schools using gas will be too long. Commercialization of this appliance will be accelerated through the development of guidelines for system selection.

Two types of firms are logical suppliers of this product. They are hood manufacturers and manufacturers of HVAC equipment. Both types of companies are currently marketing similar equipment to potential customers for the range hood-water heater. Several manufacturers of hoods have recently started selling heat recovery systems to preheat the makeup air required for kitchen ventilation. This represents a departure from established marketing methods for these companies. Traditionally, hoods are selected by kitchen designers rather than by the engineers who specify the HVAC equipment. The marketing of the proposed appliance requires a bridging of the gap between these two groups. Recently, suppliers of centrifugal type hoods have been working with HVAC engineers to develop an appreciation of their product as an energy-saving device due to its requirement for less airflow. In addition, they have been marketing heat recovery units for space heating to the same customers. This shift in marketing emphasis presents a very real barrier to commercialization which can be overcome in part by an ERDA-sponsored demonstration.

Manufacturers of HVAC equipment such as Trane and Carrier have obviously developed marketing contacts with these customers over the years. In addition, they have long standing reputations among buyers of this type of equipment and have the engineering background to support their marketing organizations.

In the long term, either type of organization could manufacture this appliance. For the short term, the hood manufacturers are the better choice for the lead role because they have developed the technology for automatically cleaning surfaces in this environment. They can design and build systems employing heat exchange coils supplied by HVAC equipment manufacturers. These coils would be similar to those presently used in water-to-air heating systems. Since both types of companies are already manufacturing various parts of the system, minimal investment in equipment would be required to begin producing the complete appliance. Likewise, distribution methods are currently established for similar products by these companies. Therefore, no real barrier to commercialization exists due to marketing, manufacturing, or distribution limitations.

6.3.2 Recommended National Demonstration Plan

Work Plan

The objective of this program is to demonstrate the energy-savings potential of the range hood water heater integrated appliance and thereby significantly accelerate its commercialization. This should be accomplished by:

- Building demonstration units;
- Field testing these units in restaurants and hospitals; and
- Publishing results in technical papers and displaying units at appropriate trade shows.

These three major steps are described in the following sections.

Designing and Building of Demonstration Units

There are three primary areas to be considered during the design phase of this program. First, the sites for the demonstration must be selected, their operating requirements determined and system specifications including storage requirements and water circulation rates for each location must be derived. Second, the heat exchanger coil must be configured for each location. Third, the rooftop recovery unit containing the coil and wash system must be designed. In general, much of the hardware required should be assembled from off-the-shelf items. The only major exception to this will be in the fabrication of the housing for the heat recovery coil and wash system.

Field Tests

Cooking facilities in hospitals, hotels, motels, and restaurants offer the highest potential energy savings and the best life-cycle cost savings, because of the high levels of energy usage in the kitchen. The specific sites selected for field testing should meet certain requirements. First, they should have a reasonably high demand for hot water and cooking loads which match these demands. The ventilation system must be suitable for installation of the recovery system. Also the sites selected should have good information on past energy use for various functions. Tests should be made on kitchens using gas as well as kitchens using electricity for water heating. The performance of the system with electric cooking equipment should also be demonstrated.

The primary parameters to be measured during these tests is the reduction in energy use due to recovery of heat from the hood for preheating water. This can be determined by integrating the temperature difference across the circulation loop with respect to time if the pump flow rate is constant and known. In addition to this measurement, it is desirable to know the amount of energy in the form of preheated water withdrawn from the storage tank. This can be obtained by multiplying an output signal proportional to flow by the temperature difference between the tank inlet and outlet and integrating the result with time. Standard instrumentation packages are available from several manufacturers to do this.

Other parameters which should be recorded are:

- inlet water temperature;
- exhausted air temperature;
- water usage;
- gas or electric usage to both cooking and water heating equipment.

Some of these will be recorded automatically on a continuous basis using thermocouples and chart recorders. Others will be recorded automatically when the system is shut off.

Presentation of Results

At the conclusion of this program, after the results of the tests have been analyzed, presentations will be made at appropriate trade shows and professional meeting and papers published in trade journals to encourage the rapid commercialization of this product. Because this product is used in the commercial sector, it is anticipated that presenting impartial test results to engineers, kitchen consultants, and managers of eating facilities will be the most effective mechanism of accomplishing the program objective.

Schedule

The recommended program schedule is shown in Figure 6.11.

Estimated Costs for Demonstration Plan

Based on discussions with manufacturers and our own estimates of manpower requirements, we judge that the total program cost (industry and ERDA) to complete the Demonstration Plan will be $170,000, with the emphasis of the program effort broken down as follows:

Task	Recommended % of Program Effort
Development	40
Demonstration	50
Public Information Dissemination	10

6.4 POTENTIAL BENEFIT OF NATIONAL DEMONSTATION PLAN

The benefits of the recommended Demonstration Plan are:

- The engineering risk is low, while the potential energy savings is significant.
- The time required to have demonstration systems in operation is short.
- Manufacturer interest is high.
- A demonstration of water heating potential will develop awareness of the benefits of hood heat recovery systems in general.

As outlined in Section 4.4, an acceleration profile is used to characterize the fraction of full manufacturing implementation in any year. An acceleration profile with and without ERDA support is given in Table 6.9. This estimate of the impact of the ERDA-sponsored program is a judgment based on discussions with potential manufacturers.

As in Section 4.4, an analysis of the cumulative nationwide energy savings with and without ERDA support was performed using the foregoing acceleration profile and the energy savings and consumer acceptance value shown in Table 6.10. Also given in Table 6.10 is the cumulative energy savings with and without ERDA support. Figure 6.12 shows the sales profile corresponding to Table 6.10.

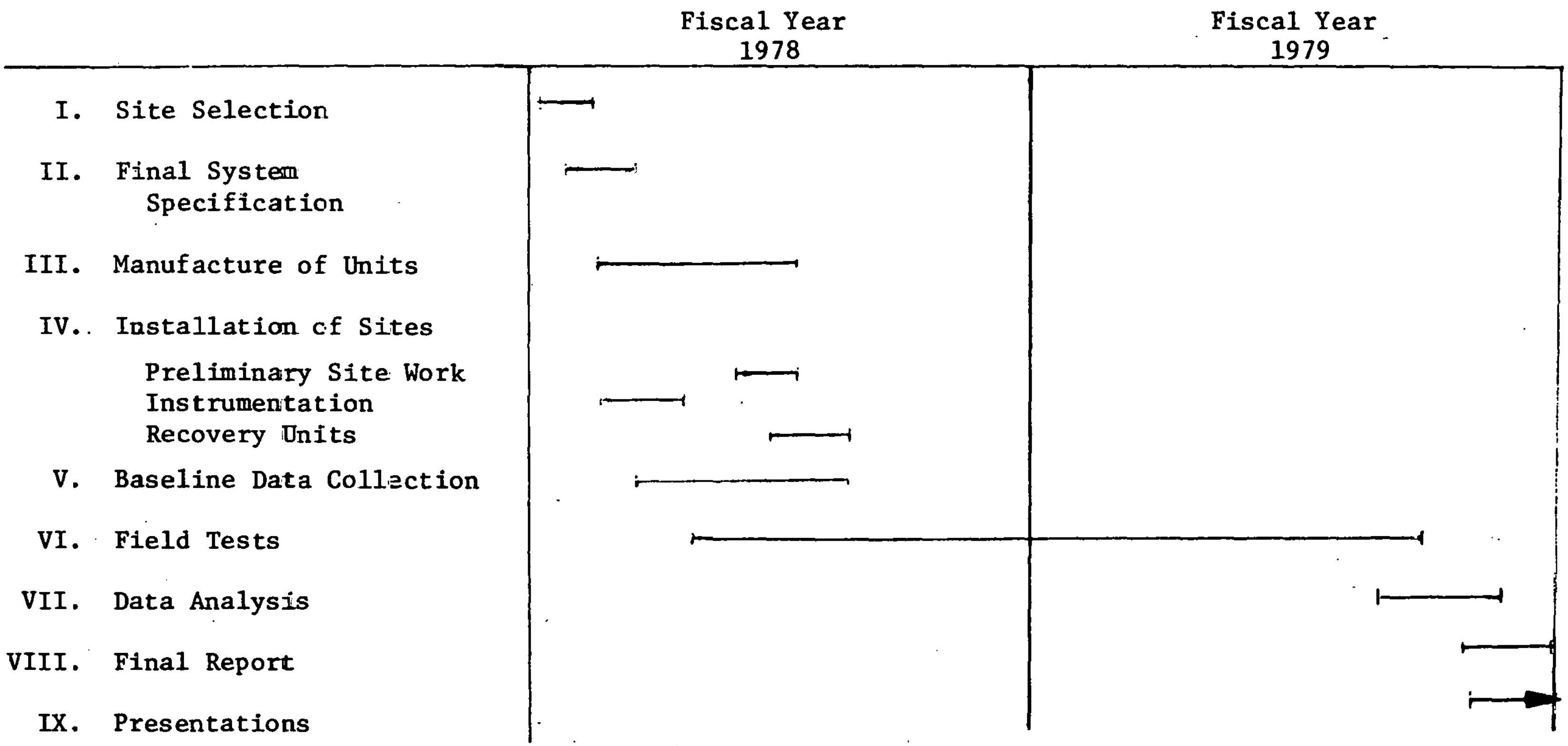


FIGURE 6.11 SCHEDULE OF NATIONAL DEMONSTRATION PLAN

TABLE 6.9

ACCELERATION PROFILE FOR RANGE HOOD HEAT RECOVERY SYSTEM

Year	Acceleration Without ERDA	Acceleration With ERDA
1977	0.000	0.000
1978	0.000	0.000
1979	0.000	0.500*
1980	0.000	1.000
1981	0.000	1.000
1982	0.250*	1.000
1983	0.500	1.000
1984	0.750	1.000
1985	1.000	1.000
1986	1.000	1.000
1987	1.000	1.000
1988	1.000	1.000
1989	1.000	1.000
1990	1.000	1.000

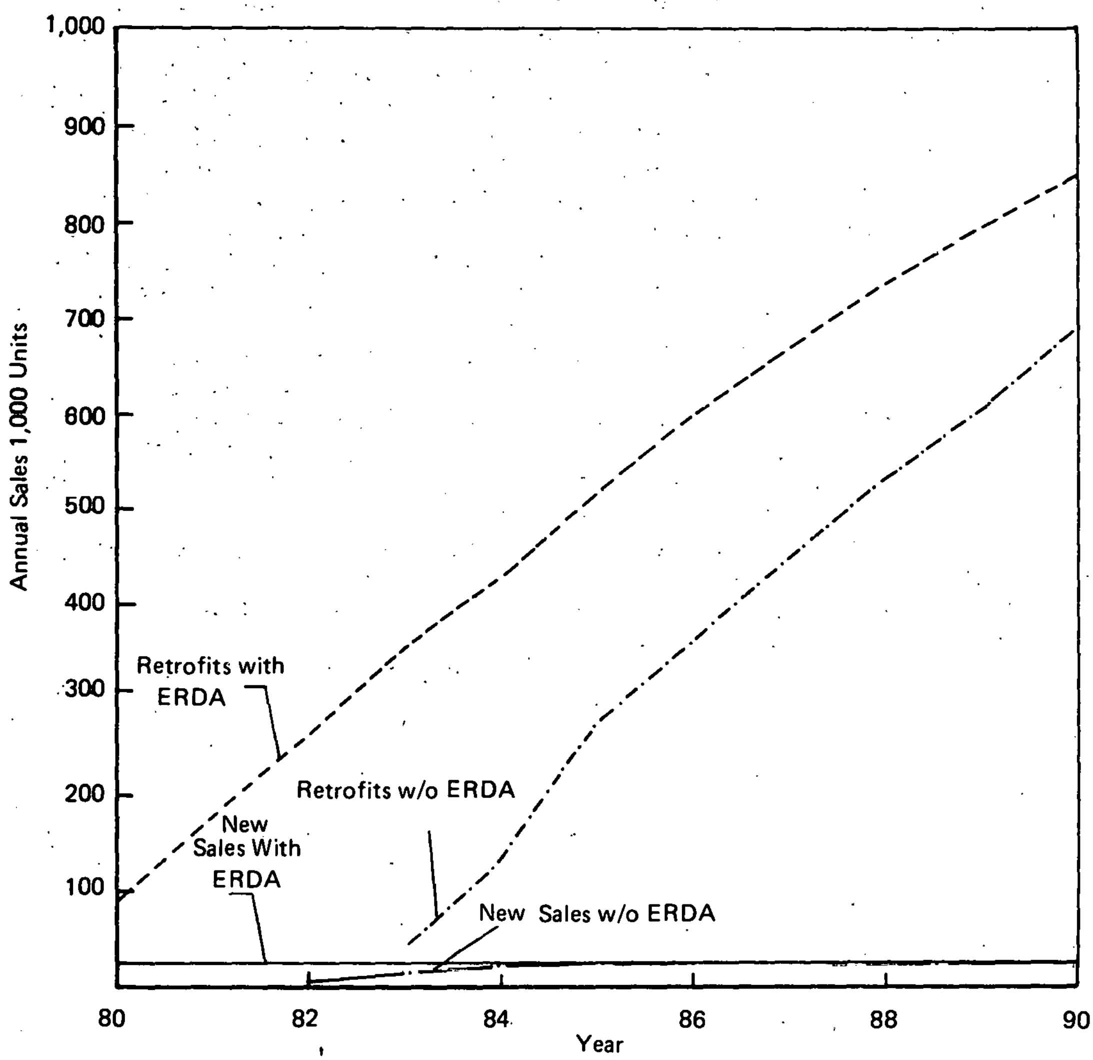
*3-year acceleration

TABLE 6.10

ESTIMATED EFFECT
ERDA-SPONSORED DEMONSTRATION PROGRAM

	Weighted Annual Primary Energy Savings (10^6 Btu per unit)	Weighted Years to Payback	Max. Percent Annual New Sales Captured	Maximum Percent of In-Place Facilities Retrofitted*	Average Sales Rate (1985) Units/Year		Percent of Facilities with HRS by 1990	Cumulative Energy Savings 1980-1990 in 10^{12} Btu Primary		
					OEM	Retrofit		Without ERDA	With ERDA	Effect of ERDA
Gas										
Hospital, Hotel, Motel	1,250	1.2	-	36	0	940	94	29	67	38
Others	92	5.4	13	4	273	4,970	24	11	27	16
Electric										
Hospital, Hotel, Motel	3,000	0.6	40	50	140	223	96	29	63	34
Others	224	2.5	45	24	1,260	4,910	81	28	66	38
TOTALS								97	223	126

*This grows at a linear rate from 1/10 of the value shown in 1980 to equal the value shown in 1990.



For: Gas Supplied; Restaurants, Cafeterias, Schools and Institutions

Sales to Electric Facilities, and Larger Gas Supplied Facilities are Not Shown.

FIGURE 6.12 PROJECTED SALES OF THE RANGE – WATER HEATER RECOVERY SYSTEM

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A. APPLIANCE INVENTORY

A.1 RESIDENTIAL APPLIANCE INVENTORY

This section presents the appliance inventory and supporting documentation which has been developed for the residential sector. A description of the approach, assumptions, calculations, and data sources is presented, followed by the inventory data in the form of tables. The backup data for the energy usage table is also given for reference and documentation purposes.

The appliance inventory was developed solely for the purpose of screening the hundreds of possible combinations of appliances in an orderly fashion. After the screening process, detailed analysis of the most promising candidates (top six) was undertaken, and in many instances new appliance inventory figures were developed for the specific appliances examined.

Energy usage and inventory figures are presented in Table 1. For certain appliances the Table A.1 figures differ from the figures in Chapters 3, 4, 5, and 6 where the detailed analysis of the top six candidates is given. Wherever significant differences arise, the discrepancy is noted in the chapter specific to the appliance. A quick reference summary of the single family home energy usage is given in Tables A.2 and A.3.

The nationwide energy consumption estimates given in Table A.1 are subject to a level of uncertainty as discussed below.

The nationwide energy consumption is the product of several terms; these items are shown below with our judgment of the uncertainty of each:

	Uncertainty*
Nationwide energy consumption =	
total number of housing units	± 10%
x percent of housing units with specific appliance	± 10%
x percent of appliances of fuel type (gas, electric, or oil)	± 10%
x energy consumption per unit per hour	± 15%
x average hours per year of operation	± 20%

*Uncertainty is based on our judgment of about 4 to 1 odds of being correct.

TABLE A.1

1970 Appliance Inventory
Residential Sector

	(1)DETACHED & DUPLEX	(2)MOBILE HOME	(3)LOW RISE
INVENTORY	46300000.0	2100000.0	727000.0
NEW/YR	1100000.0	400000.0	33300.0
PCT NEW	2.5	20.0	6.0
PCT HEATING SEASON	60.0	60.0	60.0

APPLIANCE

	(1)DETACHED & DUPLEX	(2)MOBILE HOME	(3)LOW RISE
(1)HOT WATER-BATH			
EL BTU/APPL-YR	1.270E 07	9.741E 06	1.769E 08
EL PCT OUTSIDE	81.0	81.0	81.0
EL BTU/YR(US)	1.633E 14	1.164E 13	2.122E 13
EL NO. APPL	12860459.0	1194479.0	119954.9
GS BTU/APPL-YR	2.160E 07	1.656E 07	3.010E 08
GS PCT OUTSIDE	80.0	80.0	80.0
GS BTU/YR(US)	6.703E 14	1.319E 13	1.827E 14
GS NO. APPL	31031904.0	796319.3	607044.8
NO./BLDG	0.948	0.948	1.000
BTU/YR (US)	8.336E 14	2.482E 13	2.039E 14
(2)RANGE/OVEN			
EL BTU/APPL-YR	4.000E 06	3.083E 06	2.550E 06
EL PCT OUTSIDE	20.0	20.0	20.0
EL BTU/YR(US)	8.797E 13	9.259E 11	8.371E 12
EL NO. APPL	21992480.0	300299.8	3282546.0
GS BTU/APPL-YR	1.400E 07	1.064E 07	8.797E 06
GS PCT OUTSIDE	20.0	20.0	20.0
GS BTU/YR(US)	3.403E 14	1.914E 13	7.729E 13
GS NO. APPL	24307488.0	1799699.0	8785639.0
NO./BLDG	1.000	1.000	16.600
BTU/YR (US)	4.283E 14	2.007E 13	8.566E 13
(3)REFRIG./FREEZER			
EL BTU/APPL-YR	5.606E 06	4.300E 06	3.550E 06
EL PCT OUTSIDE	20.0	20.0	20.0
EL BTU/YR(US)	3.426E 14	1.183E 13	5.678E 13
EL NO. APPL	61115952.0	2750995.0	15993996.0
NO./BLDG	1.320	1.310	22.000
BTU/YR (US)	3.426E 14	1.183E 13	5.678E 13

	Detached & Duplex	Mobile Home	Low Rise
(4)CLOTHES DRYER*			
EL BTU/APPL-YR	4.201E 05	7.019E 06	1.320E 07
EL PCT OUTSIDE	90.0	90.0	90.0
EL BTU/YR(US)	7.121E 12	2.022E 12	1.511E 13
EL NO. APPL	16950416.0	288119.8	1145024.0
GS BTU/APPL-YR	8.147E 05	1.000E 07	2.550E 07
GS PCT OUTSIDE	88.0	88.0	88.0
GS BTU/YR(US)	5.918E 12	1.235E 12	1.251E 13
GS NO. APPL	7264466.0	123479.9	490724.8
NO./BLDG	0.523	0.196	2.250
BTU/YR (US)	1.304E 13	3.257E 12	2.763E 13
(5)WH/CLOTHES WASHER*			
EL BTU/APPL-YR	4.600E 06	3.530E 06	1.100E 07
EL PCT OUTSIDE	83.0	83.0	83.0
EL BTU/YR(US)	4.437E 13	2.117E 12	5.278E 12
EL NO. APPL	9645345.0	599759.7	479819.9
GS BTU/APPL-YR	7.800E 06	5.980E 06	1.865E 07
GS PCT OUTSIDE	82.0	82.0	82.0
GS BTU/YR(US)	1.815E 14	2.391E 12	4.529E 13
GS NO. APPL	23273920.0	399839.8	2428179.0
NO./BLDG	0.711	0.476	4.000
BTU/YR (US)	2.259E 14	4.508E 12	5.056E 13
(6)WH/DISHWASHER*			
EL BTU/APPL-YR	4.600E 06	3.530E 06	2.910E 07
EL PCT OUTSIDE	79.0	79.0	79.0
EL BTU/YR(US)	6.240E 13	4.448E 12	7.959E 13
EL NO. APPL	13565890.0	1259999.0	2734970.0
GS BTU/APPL-YR	7.800E 06	5.980E 06	4.940E 07
GS PCT OUTSIDE	79.0	79.0	79.0
GS BTU/YR(US)	2.553E 14	5.023E 12	6.550E 14
GS NO. APPL	32734080.0	839999.8	13259020.0
NO./BLDG	1.000	1.000	22.000
BTU/YR (US)	3.177E 14	9.471E 12	7.346E 14
(7)TELEVISION			
EL BTU/APPL-YR	1.500E 06	1.281E 06	1.114E 06
EL PCT OUTSIDE	20.0	20.0	20.0
EL BTU/YR(US)	8.994E 13	2.494E 12	1.196E 13
EL NO. APPL	59958416.0	1946698.0	10730514.0
NO./BLDG	1.295	0.927	14.760
BTU/YR (US)	8.994E 13	2.494E 12	1.196E 13

*Electric motor energy consumption <u>not</u> included.

	Detached & Duplex	Mobile Home	Low Rise
(8)LIGHTING			
EL BTU/APPL-YR	2.500E 06	1.154E 06	1.442E 06
EL PCT OUTSIDE	0.0	0.0	0.0
EL BTU/YR(US)	1.157E 14	2.423E 12	2.307E 13
EL NO. APPL	46299984.0	2099999.0	15993996.0
NO./BLDG	1.000	1.000	22.000
BTU/YR (US)	1.157E 14	2.423E 12	2.307E 13
(9)ROOM AIR COND.			
EL BTU/APPL-YR	4.368E 06	2.022E 06	2.71E 06
EL PCT OUTSIDE	330.0	330.0	330.0
EL BTU/YR(US)	7.321E 13	1.495E 12	1.0E 13
EL NO. APPL	16760592.0	739199.6	3720055.0
NO./BLDG	0.362	0.352	5.117
BTU/YR (US)	7.321E 13	1.495E 12	1.0E 13
(10)CENTRAL AIR COND.			
EL BTU/APPL-YR	1.390E 07	6.416E 06	1.900E 08
EL PCT OUTSIDE	330.0	330.0	330.0
EL BTU/YR(US)	7.081E 13	1.630E 12	9.697E 13
EL NO. APPL	5092993.0	254099.8	510353.8
NO./BLDG	0.110	0.121	0.702
BTU/YR (US)	7.081E 13	1.630E 12	9.697E 13
(11)HEATING			
EL BTU/APPL-YR	4.571E 07	2.719E 07	1.165E 08
EL PCT OUTSIDE	30.0	30.0	30.0
EL BTU/YR(US)	1.439E 14	5.711E 12	1.083E 13
EL NO. APPL	3148397.0	209999.9	93782.9
GS BTU/APPL-YR	1.200E 08	8.430E 07	3.610E 09
GS PCT OUTSIDE	30.0	30.0	30.0
GS BTU/YR(US)	5.178E 15	1.593E 14	2.286E 15
GS NO. APPL	43151552.0	1889999.0	633216.6
NO./BLDG	1.000	1.000	1.000
BTU/YR (US)	5.322E 15	1.650E 14	2.297E 15

TABLE A.2

PRIMARY WASTE ENERGY
IN SINGLE FAMILY HOMES

1970 Inventory 10^6 Units	Appliance	INPUT ENERGY Point of Use 10^6 Btu/yr Per Unit	INPUT ENERGY Primary 10^6 Btu/yr Per Unit	INPUT ENERGY Primary 10^{14} Btu/yr Aggregate	WASTE ENERGY 10^{14} Btu/yr (Primary) Drain	Flue or Vent	Jacket*	Total
	Hot Water							
31	gas	37.2	37.2	11.5	5.2	4.3	.7	10.2
12.8	electric	21.9	73.0	9.3	7.7	0	.8	8.5
	Range/Oven							
24.3	gas	13.8	13.8	3.4	0	0	1.7	1.7
22	electric	4.0	13.3	2.9	0	0	1.4	1.4
61.1	Refrigerator	5.6	18.6	11.4	0	0	5.5	5.5
	Clothes Dryer							
7.2	gas	8.2	8.2	.6	0	.5	.05	.5
16.9	electric	4.2	14.0	2.4	0	2.0	.2	2.2
60	Television	1.5	5.0	3.0	0	0	1.5	1.5
46.3	Lights	2.5	8.3	3.8	0	0	1.9	1.9
16.7	Room A/C	4.4	14.6	2.4	0	2.4	0	2.4
	Subtotal							
	gas			15.5	5.2	4.8	2.4	12.4
	electric			35.2	7.7	4.4	11.3	23.4
5.1	Central A/C	13.9	46.3	2.3	0	0	0	2.3
	Space Heat							
43.1	gas	119.2	119.2	51.4	0	21.5	0	21.5
3.2	electric	46.0	153.0	4.9	0	0	0	0
	TOTAL RESIDENTIAL			109.3	12.9	33.0	13.7	59.6

*Assumes 1/2 of annual jacket waste heat goes into useful space heating.

TABLE A.3

POINT OF USE WASTE ENERGY IN SINGLE FAMILY HOMES

1970 Inventory 10^6 Units	Appliance	Point of Use 10^{14} Btu/yr Aggregate	Percent Point of Use	Point of Use Waste: 10^{14} Btu/yr Aggregate	Point of Use Waste: Value in $\$10^6$	Point of Use 10^6 Btu per unit Single Family House
	Hot Water					
31	gas	11.5	88	10.1	1,717	37.2
12.8	electric	2.8	91	2.5	2,352	21.8
	Range					
24.3	gas	3.4	52	1.77	300	13.8
22	electric	.9	52	.46	432	4.0
61.1	Refrigerator	3.4	48	1.6	1,505	5.6
	Clothes Dryer					
7.2	gas	.6	91	.54	92	8.2
16.9	electric	.7	91	.64	602	4.1
60	Television	.9	50	.45	423	1.5
46.3	Lights	1.2	50	.6	565	2.3
16.7	Room A/C	.7	330	2.4	2,164	4.2
5.1	Central A/C	.7	330	2.3	2,164	13.7
	Space Heat					
43.1	gas	51.4	42	21.58	3,668	119.2
3.2	electric	1.5	0	0	0	46.8
	TOTAL	79.7		44.84	15,984	

Holman[A.1] shows that the overall uncertainty for the product of several independent uncertainties (all of equal probability) is the square root of the sum of the squares of the uncertainty. For the component uncertainties listed above, we judge the nationwide energy figures reported in this report to have a ± 30% uncertainty.

A.1.1 Approach

The inventory consists of eleven appliance categories. Several of the groups are subsets and/or combinations of other appliances. For example, energy usage for water heaters is broken down into clothes washing, dishwashing, and bathing. The appliance categories considered are:

- water heaters - bathing
- water heaters/clothes washers
- water heaters/dishwashers
- range/ovens
- refrigerators/freezers
- clothes dryers
- televisions
- lighting
- room air conditioners
- central air conditioners
- heating systems

The energy usage for each appliance is subdivided into gas and electricity (except in the cases of televisions, lighting, and air conditioners, for which the energy source is only electricity). Gas includes both propane (bottled) and natural gas, oil, and other miscellaneous fuels which supply energy at the user's location (i.e., wood).

Four different types of residences are included in the inventory: single family homes, duplexes, mobile homes and low rise multifamily complexes. Single family units and duplexes are grouped together. The data are from the <u>1970 Census of Housing</u>.[A.2] This source also supplied data on the number of the various appliances by energy type (gas or electric) found in each building group.

The values for energy usage per appliance per year were established after careful review of various sources. These sources include manufacturer's reports, reports from the Federal Energy Administration/National Bureau of Standards[A.3], Arthur D. Little, Inc.[A.4], Booz-Allen[A.5], and American Gas Association[A.6]. Other articles in the literature were also reviewed. A comprehensive list of sources used and values considered for the various appliances in a single-family home can be found at the end of this section under Supporting Data.

The number of appliances per building was obtained by dividing the number of appliances by the number of buildings in the sectors and is used in the appliance matching program (Appendix B) to recognize the difference between many appliances in one building from many buildings with the appliance.

The annual energy usage for appliances in mobile homes and low rise buildings were adjusted according to occupancy rate (i.e., occupants of dwelling/occupants of single-family homes) or type of use.

A.1.2 Supporting Data

This section summarizes the data and models used to evaluate appliance energy usage and the percentage of the energy that is lost and that contributes to reducing the heating load. This data was used to construct Table A.1 presented above. The data is references and the complete reference citation is given at the end of this report.

Appliance Energy Usage - Residential Sector

Gas Water Heaters

For gas water heaters, the following values of annual energy usage have been reported:

37×10^6 Btu/year	Arthur D. Little, Inc.
30×10^6 Btu/year	Boston Gas[A.7] and FEA/NBS
28×10^6 Btu/year	Appliance Manufacturer[A.8]
31.6×10^6 Btu/year	AGA, 1973[A.9]
28.8×10^6 Btu/year	AGA, 1971
35×10^6 Btu/year	Consumer Reports[A.10]

The selected value was 37.2×10^6 Btu/year. The distribution (Arthur D. Little) is as follows:

For showers and baths	290 gallon/week
For dishwashing	105 gallon/week
For clothes washing	105 gallon/week

Thus, the energy was apportioned accordingly:

Baths	21.6×10^6 Btu/year
Dishwashing	7.8×10^6 Btu/year
Clothes washing	7.8×10^6 Btu/year

It is recognized that efficiency improvements have been made to gas water heaters (insulation, reduced input, etc.) between 1972 and 1976 which yield about a 20% savings (currently available units use under 30×10^6 Btu/year).

Electric Water Heaters

Total energy usage figures are:

22×10^6 Btu/year	Arthur D. Little
15×10^6 Btu/year	Long[A.11]
13.5×10^6 Btu/year	Livermore[A.12]
14.4×10^6 Btu/year	EEA[A.13]
16.4×10^6 Btu/year (quick recovery)	EEA

The value selected was 21.6×10^6 Btu/year based on the Arthur D. Little report. Using the same distribution of water usage, the energy usage becomes:

Baths	12.7×10^6 Btu/year
Dishwashing	4.6×10^6 Btu/year
Clothes washing	4.6×10^6 Btu/year

Electric Ovens/Ranges

Reported values are:

3.4×10^6 Btu/year	Long
4.0×10^6 Btu/year	EEA
4.0×10^6 Btu/year	EEA

The value selected was 4.0×10^6 Btu/year.

Gas Ranges/Ovens

Reported values are:

10.5×10^6 Btu/year	AGA, 1973
13.8×10^6 Btu/year	AGA, 1971
9×10^6 Btu/year	FEA/NBS
10.5×10^6 Btu/year	FEA/NBS

A value of 13.6×10^6 Btu/year was chosen, based on the following analysis:

[(2 burners) (1.2×10^4 Btu/hr-burner) (2 meals/day)
(0.5 hr/meal) (365 days/year)]

+ (4.8×10^6 Btu/year) [for pilot]

= 13.6×10^6 Btu/year.

Refrigerators

A value of 5.6×10^6 Btu/year was adopted based on Arthur D. Little, Inc., and FEA/NBS figures of 4.5 kwh/day (15358.5 Btu/day).

Gas-Fired Clothes Dryers

Data on clothes dryers are as follows:

4.6×10^6 Btu/year	AGA, 1971
4.8×10^6 Btu/year	Booz-Allen
4.0×10^6 Btu/year	Appliance Manufacturer

A value of 8.15×10^6 Btu/year was selected, based on the following reasoning:

($2.20^* \times 10^4$ Btu/hr) (35 min./load) (1 hr/60 min.) (410 loads/year)

= 5.3×10^6 Btu/year

+ 2.85×10^6 Btu/year for pilot

= 8.15×10^6 Btu/year

Electric Clothes Dryers

The following equation was used to calculate the annual energy consumption of this appliance:

(3.0^* kwh.load) (410 loads/year (3.41×10^3 Btu/kwh)

= 4.2×10^6 Btu/year

This compares with a value of 3.4×10^6 Btu/year in EEA.

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*Includes approximately 1,000 Btu/hr of indoor warmed air thermal loss in the exhaust during the winter months.

Television

Data on televisions include the following:

1.2×10^6 Btu/year (black-and-white)	FEA/NBS
1.7×10^6 Btu/year (color)	FEA/NBS
2.7×10^6 Btu/year	Appliance Manufacturer
1.4×10^6 Btu/year	Sylvania [A.14]

A value of 1.5×10^6 Btu/year was chosen using FEA values and the average sales of color and black-and-white televisions.

Lighting

The AHAM value of 2.5×10^6 Btu/year was selected. This is consistent with 18 hrs/day of operation of a 100-watt bulb. EEA shows a value of 4.0×10^6 Btu/year.

Room Air Conditioners

A value of 4.4×10^6 Btu/year was selected based on the following data:

4.5×10^6 Btu/year	AHAM [A.15]
4.9×10^6 Btu/year	Booz-Allen

Central Air Conditioners

Appliance Manufacturer data supports the U.S. average of 13.9×10^6 Btu/year.

Gas-Fired Space Heating

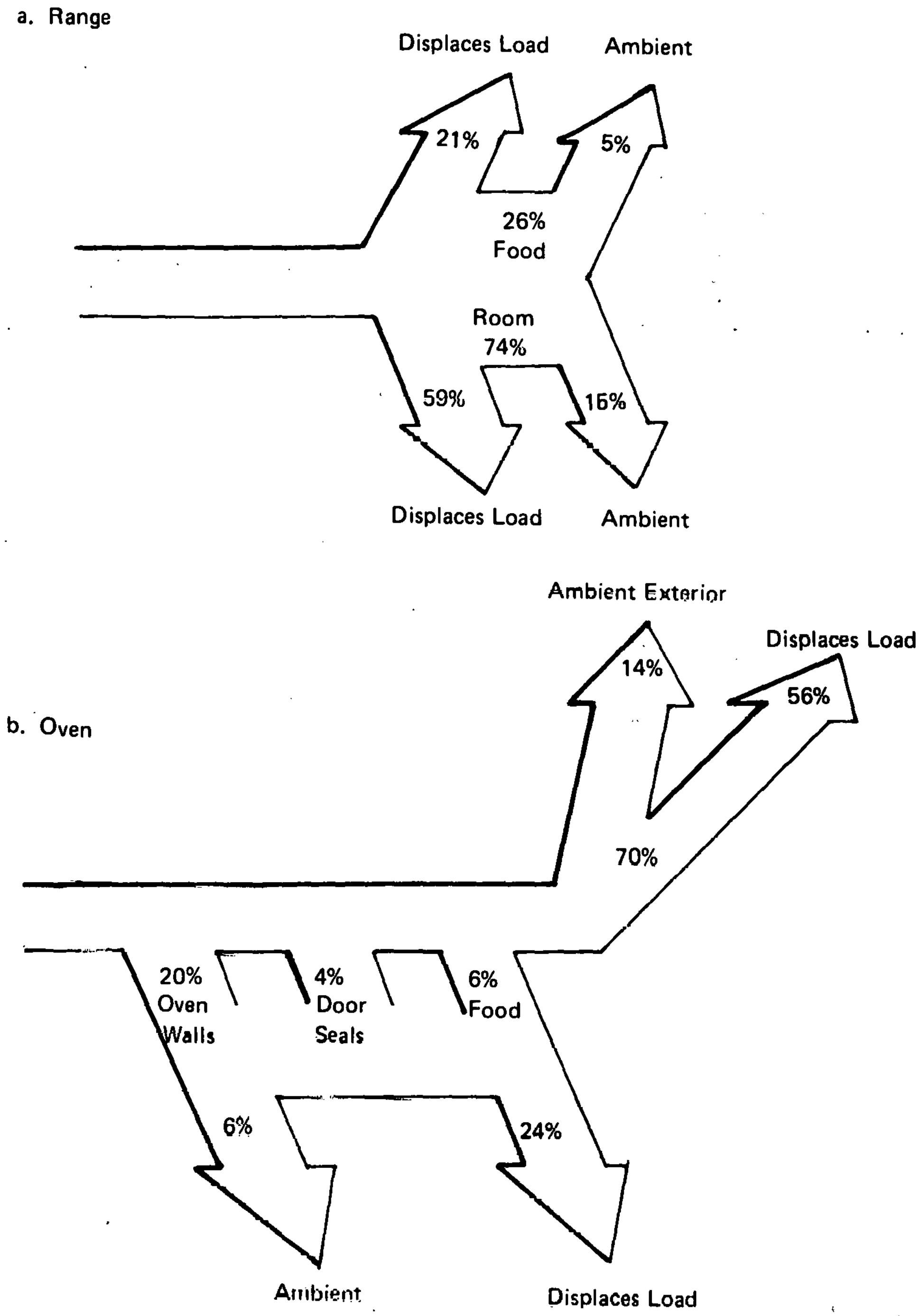
AGA data (1971) supports the U.S. average of 119.2×10^6 Btu/year over and above the heating contribution of appliances.

Electric Space Heating

A value of 46×10^6 Btu/year was used. This was based on ORNL [A.16] data which indicate that the energy used for electric space heating was equal to approximately 40% of the gas-fired space heating energy consumption.

Energy Partitioning

The following figures contain the energy partitioning diagrams for the appliance. References, values, and reasoning are presented at the bottom of each page.

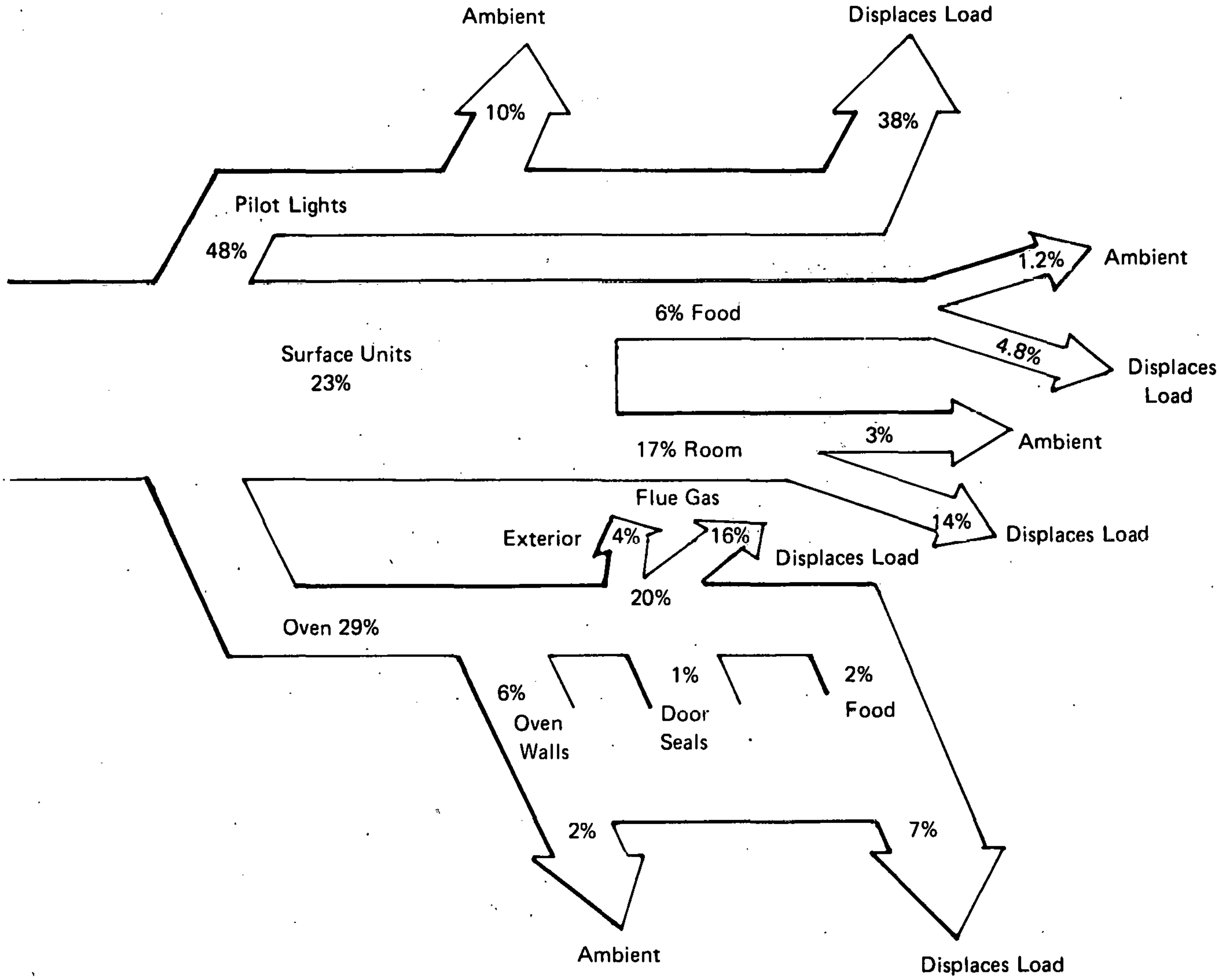


Pilot Lights	48%
Surface Units	23%
Oven	29%

Source: FEA/NBS

FIGURE A.1 GAS RANGE/OVEN ENERGY PARTITIONING

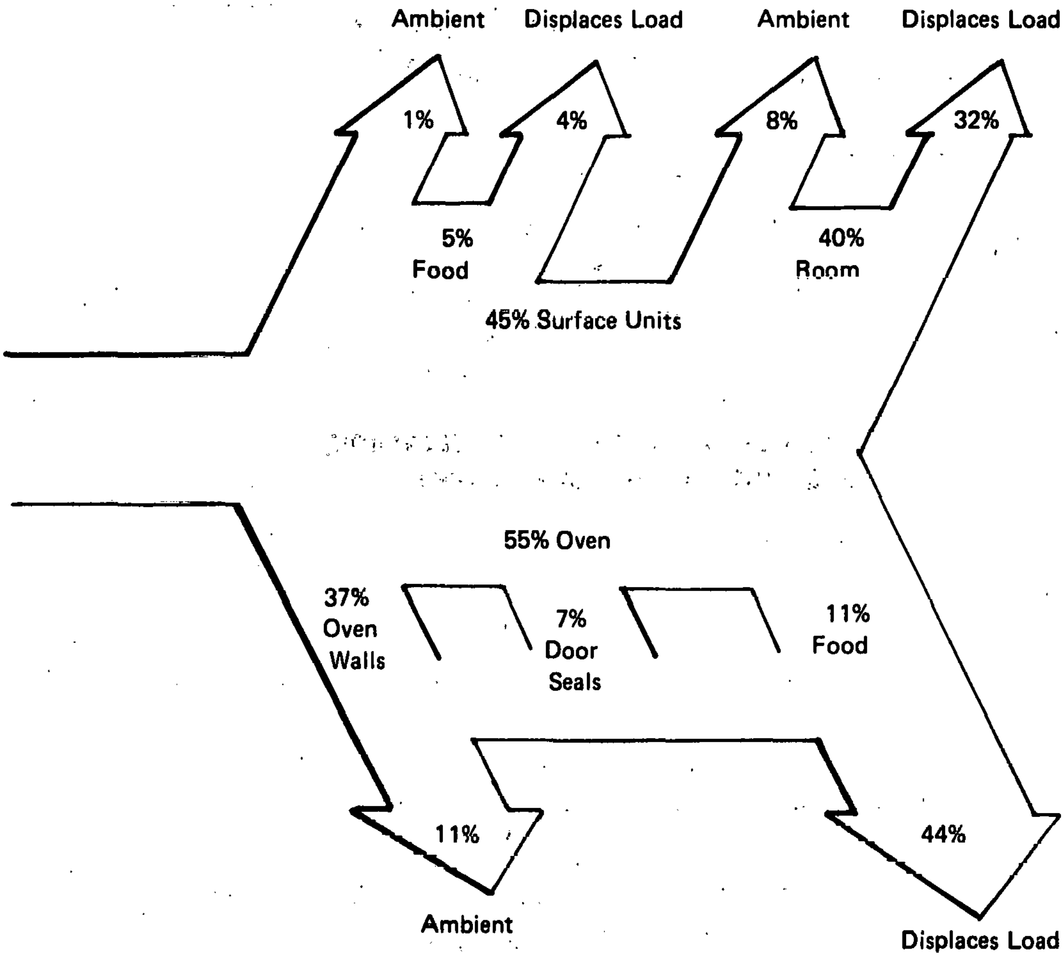
c. Range and Oven



For Oven:		
	Exhaust	70%
	Oven Walls	20%
	Door Seals	4%
	Food	6%

Source: Booz-Allen

FIGURE A.1 (Continued)



Surface Units	45%
Oven	55%

Source: Booz-Allen.

FIGURE A.2 ELECTRIC RANGE/OVEN ENERGY PARTITIONING

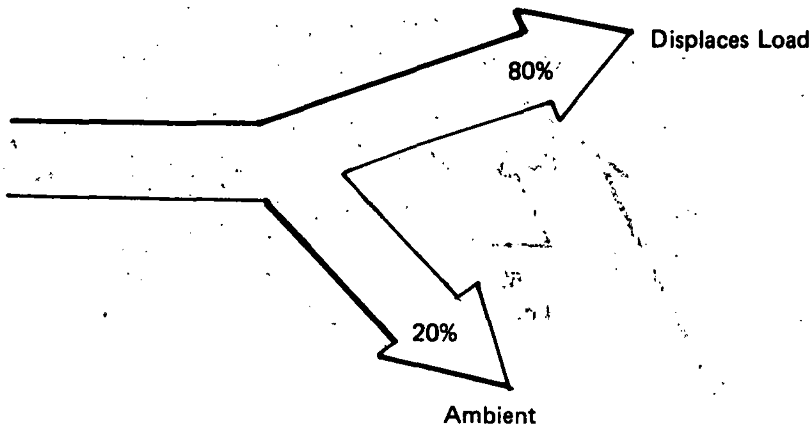
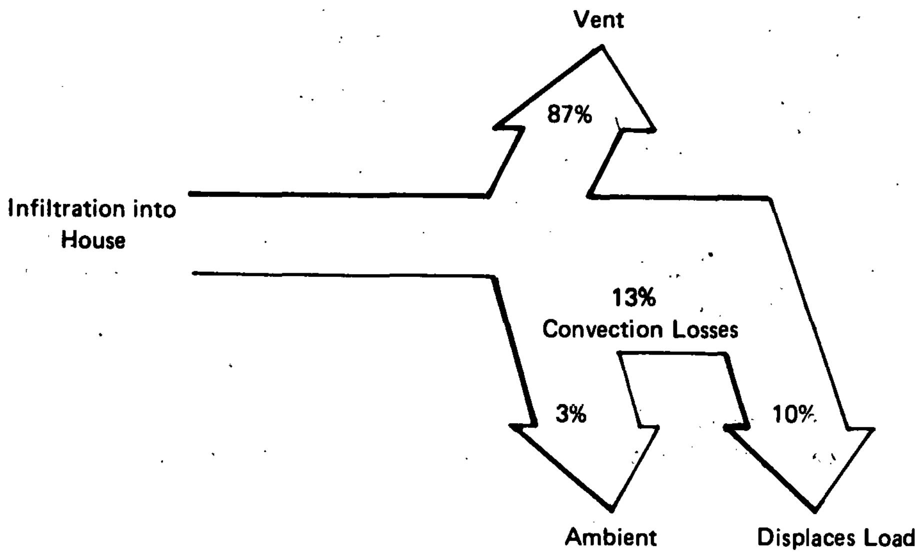


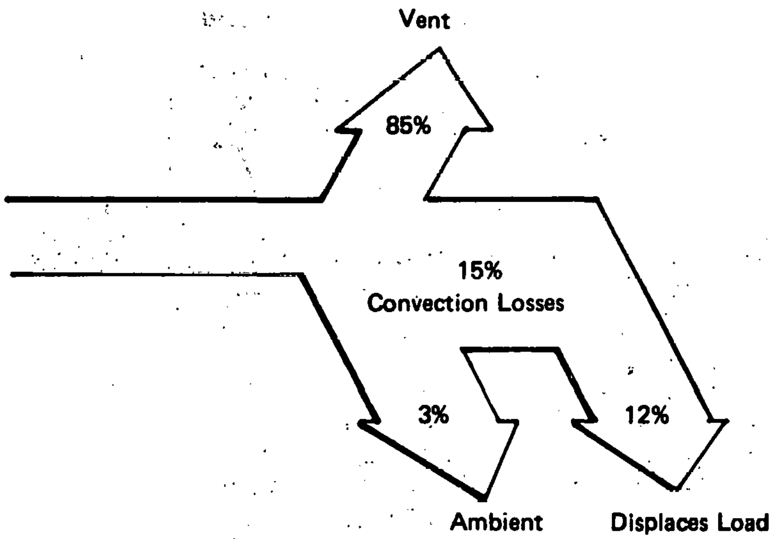
FIGURE A.3 REFRIGERATORS/FREEZERS, TELEVISION, LIGHTING ENERGY PARTITIONING



3.0 kwh Heats The Air
13% of This Energy is Lost to The Room Due to Convection (20% of This Goes to Ambient and 80% Displaces Load).

Source: Arthur D. Little, Inc., estimates.

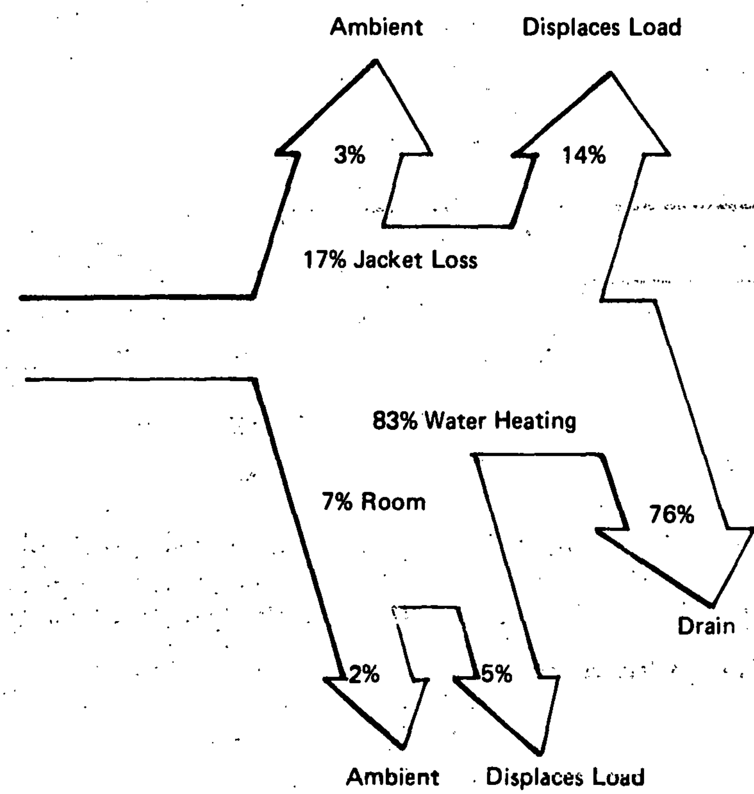
FIGURE A.4 ELECTRIC CLOTHES DRYERS ENERGY PARTITIONING



15% of the energy enters the room due to convection and radiation; motor energy is neglected and an electric pilot is assumed (FEA/NBS target for 1980).

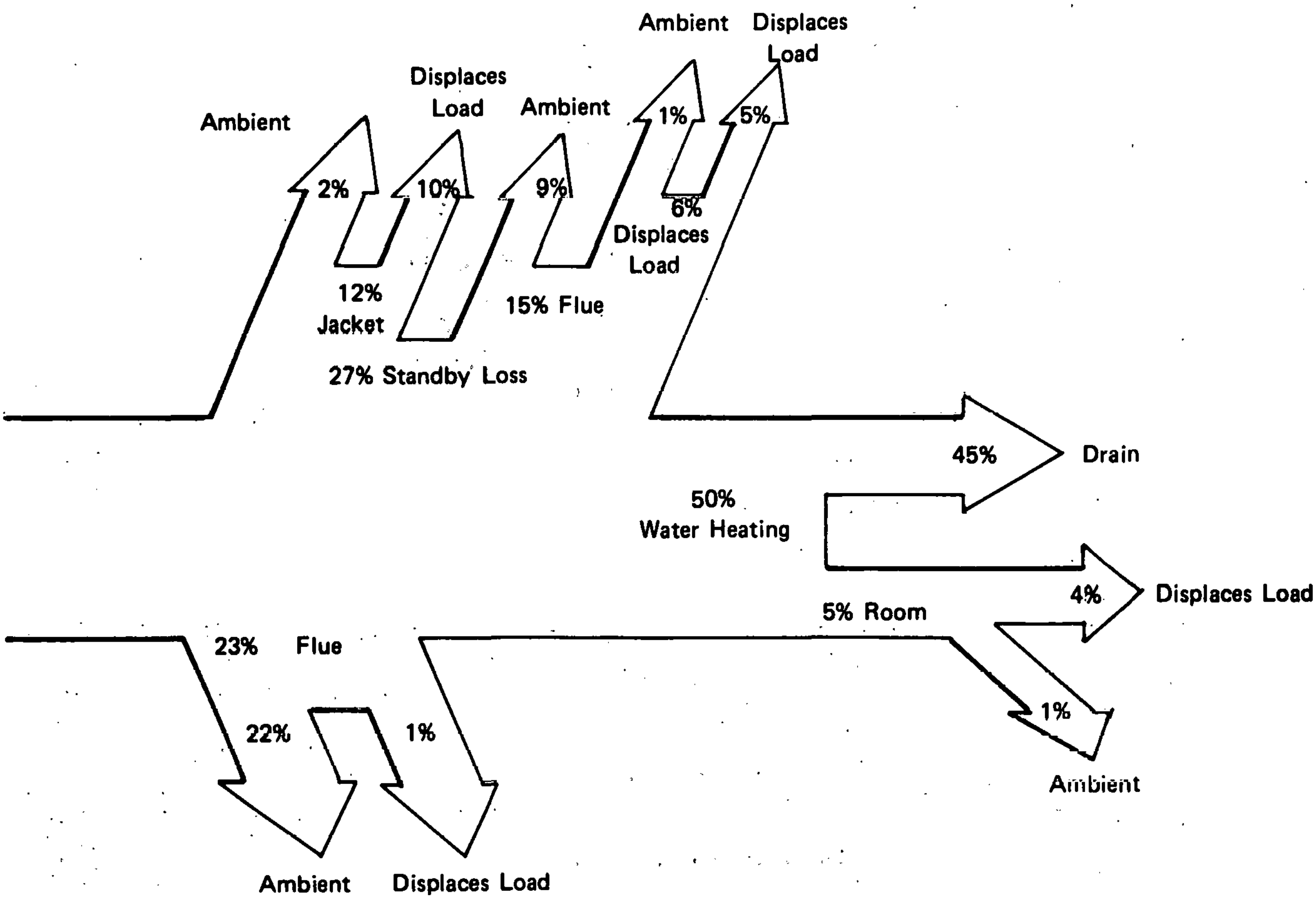
Source: Arthur D. Little, Inc., estimates.

FIGURE A.5 GAS CLOTHES DRYERS ENERGY PARTITIONING



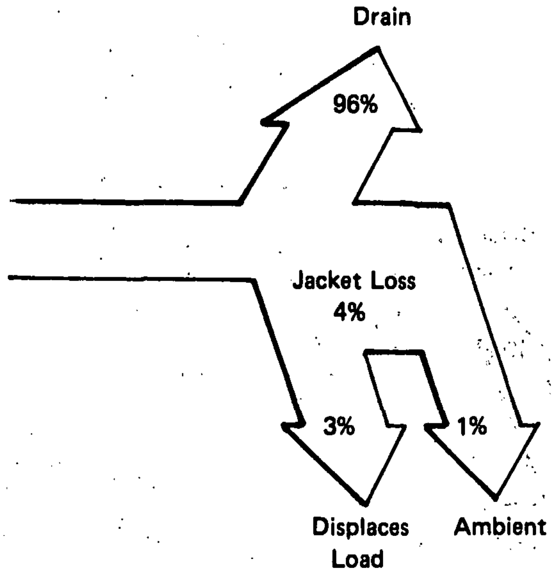
17% is Jacket Loss
83% Goes to Heating The Water
Approximately 10% of This is Recovered and Goes to The Room to be Split
80%/20% Inside/Outside

FIGURE A.6 **ELECTRIC WATER HEATERS/BATH ENERGY PARTITIONING**



The gas water heater is 50% efficient and 27% of the energy used is due to standby losses (source: FEA/NBS, Arthur D. Little, Inc). Standby losses are divided between the flue and jacket. Of the 50% that heats the water, 10% (i.e., 5% of total) is recovered. Energy is split according to the model between ambient and displacement of load.

FIGURE A.7 GAS WATER HEATER/BATH ENERGY PARTITIONING



Average Wash: 48.5 Gallons

Average Temperature: 95°F

Thus:

(1 Btu/lb °F) (48.5 gal/load) (8.34 lb/gal) (95°–60°F)
= 14,200 Btu/load

To Heat Up Washer:

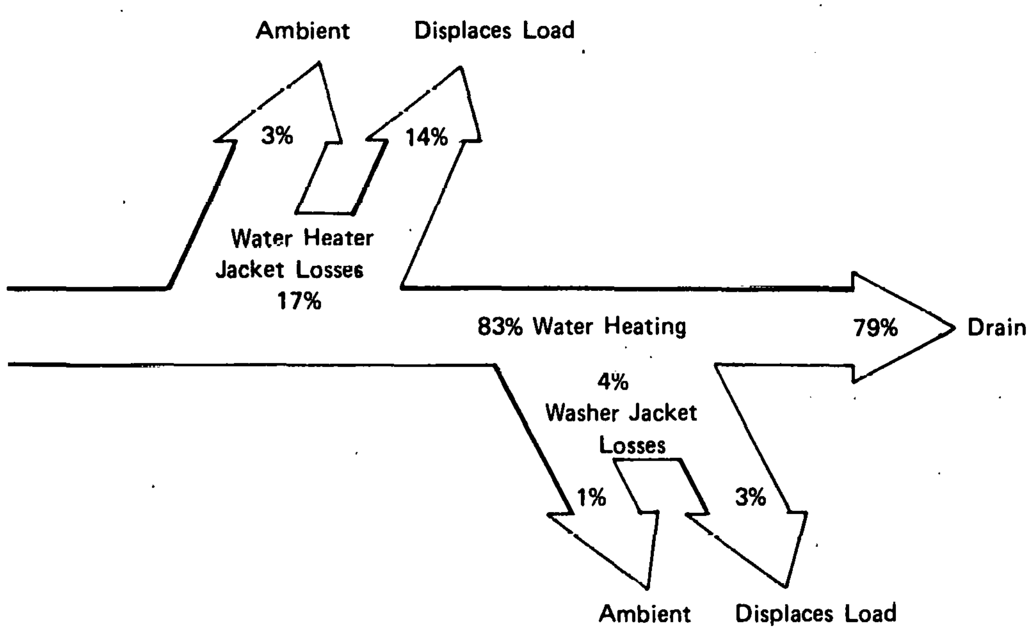
(.11 Btu/lb °F) (150 lb) (95°–70°F)
= 410 Btu/load

Convection Loss During Cycle: 200 Btu/load

Neglect Energy of Motor.

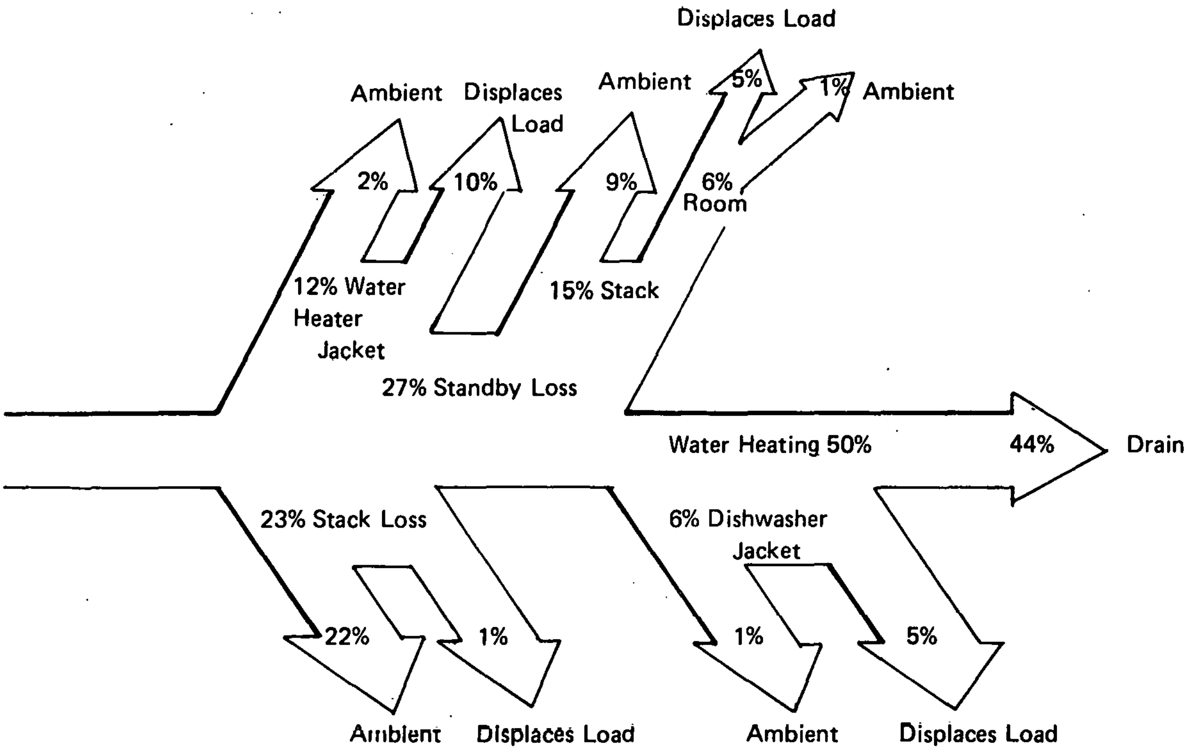
Source: Booz-Allen.

FIGURE A.8 CLOTHES WASHER ENERGY PARTITIONING



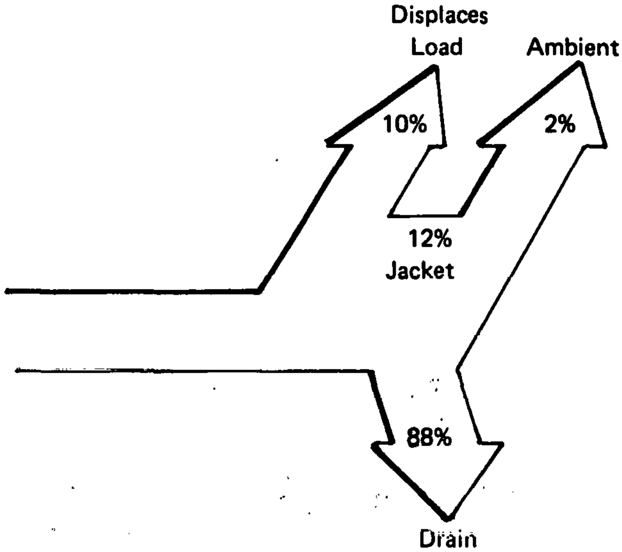
(See clothes washer and electric water heater for analysis.)

FIGURE A.9 CLOTHES WASHER/ELECTRIC WATER HEATER ENERGY PARTITIONING



(See clothes washer and gas water heater for analysis.)

FIGURE A.10 CLOTHES WASHER/GAS WATER HEATER ENERGY PARTITIONING



Average Wash: 15 Gallons

Average Temperature: $145^{\circ}F$

Thus:

$$(1\ \text{Btu/lb}\ ^{\circ}F)\ (15\ \text{gal})\ (8.34\ \text{lb/gal})\ (145^{\circ}-60^{\circ}F)$$
$$= 10{,}600\ \text{Btu/load}$$

To Heat Up Dishwasher:

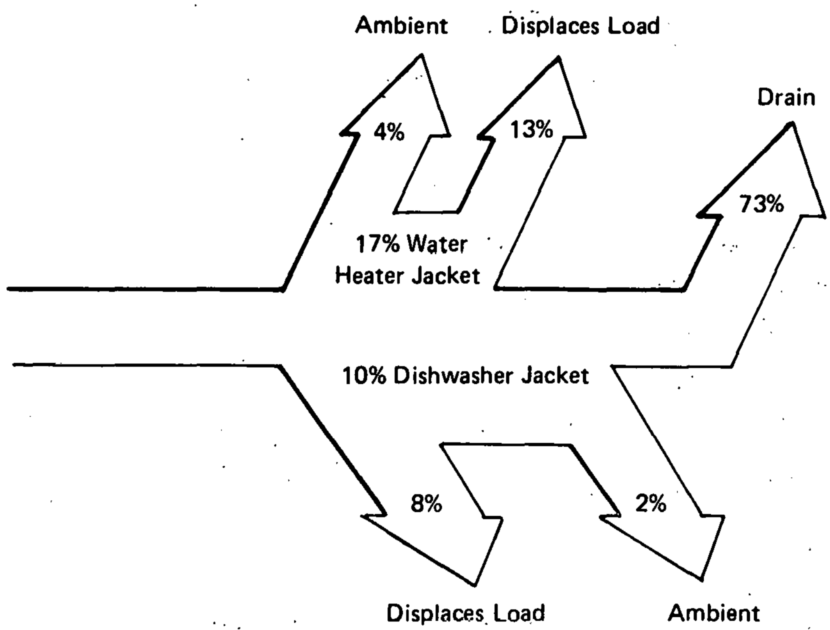
$$(150\ \text{lb})\ (.11\ \text{Btu/lb}\ ^{\circ}F)\ (130^{\circ}-70^{\circ}F)$$
$$= 1{,}000\ \text{Btu/load}$$

Convection Losses: 300 Btu/load

Neglect Motor

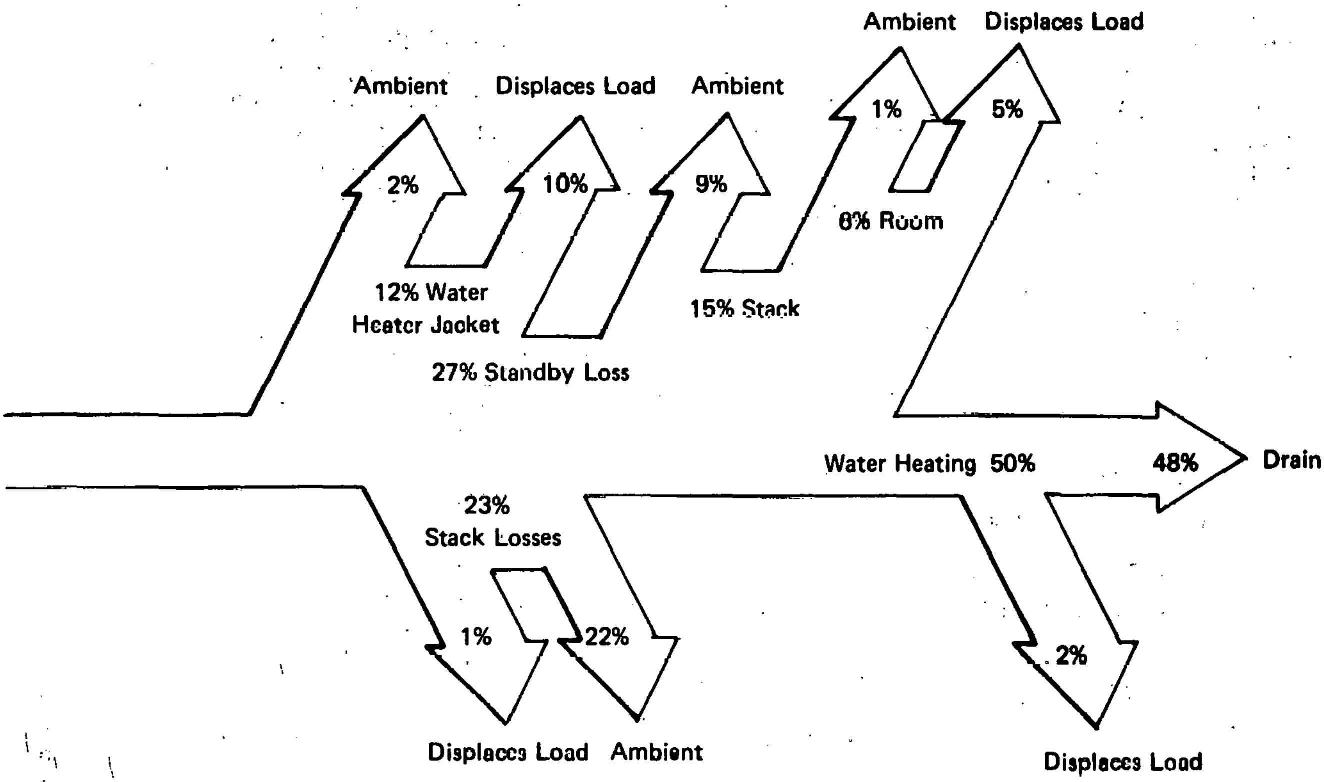
Source: FEA/NBS, Arthur D. Little, Inc.

FIGURE A.11 DISHWASHER (AUTOMATIC) ENERGY PARTITIONING



(See dishwasher and electric water heater for analysis.)

FIGURE A.12 DISHWASHER/ELECTRIC WATER HEATER ENERGY PARTITIONING



(See dishwasher and gas water heater for analysis.)

FIGURE A.13 DISHWASHER/GAS WATER HEATER ENERGY PARTITIONING

Table A.2 gives a summary of the waste energy in single family homes calculated at the point of use. Table A.3 gives the same elements calculated as primary energy, where waste energy from electric appliances is valued at the power plant at about three times the value of gas or oil. The reader will note that the sum values are different but the percentage of waste based on point of use (56.4%) and primary (59.7%) are quite close. This is due in a large measure to the compensating effect of the central and room A/C on these two methods of tracing energy, for the value of point of use waste energy is 1 + COP = three times the input energy, while the value of the primary energy is also about three times the input energy. Also given is the dollar equivalent of the waste energy at the point of use.

A.2 COMMERCIAL INVENTORY AND APPLIANCE ENERGY USAGE

This section summarizes the most significant energy usage in the commercial sector. The sectors of concern are:

- hotels and motels,
- supermarkets,
- restaurants,
- laundries,
- schools, and
- hospitals;

and the appliances of concern are:

- water heaters (for baths, dishwashing, and clothes washing),
- ranges/ovens,
- refrigerator-freezers,
- clothes dryers,
- air conditioning, and
- heating.

Table A.4 summarizes the data by sector and appliance. The values are in Btu/building/year; blank spaces indicate that the combination is possible but because of their low priority, no information was found to estimate the usage. Spaces with crosses indicate "impossible" combinations, i.e., ranges/ovens in a laundry.

The methods and rationale used to arrive at the various figures are summarized below. Due to the lack of information, most of the values and calculations are Arthur D. Little estimates which were arrived at through review of limited data available, discussions with people in the industry, and information supplied by equipment vendors. The summary is presented by appliance, rather than by sector.

These figures are meant only to represent broad sector-wide estimates. Refinement of the water and range usage figures was undertaken in Chapter 6 in the analysis of the range-water heater.

TABLE A.4

SUMMARY OF COMMERCIAL INVENTORY AND ENERGY USE DATA

Sector	Hot Water			Ranges/ Ovens	Refrigerator/ Freezers	Clothes Dryers	Air Conditioning	Heating
	Baths, Sinks	Clothes Washing	Dish Washing					
Hotel/Motel 4,100	2.2×10^9		2.1×10^8	1.6×10^9	4.5×10^7			
Supermarkets 43,000					7.5×10^8		2.0×10^9	3.9×10^9
Restaurants 130,000	1.3×10^7		1.3×10^8	7.9×10^8	1.2×10^7			3.6×10^8
Laundries 44,145		7.0×10^8				1.2×10^9		
Schools 75,000	2.4×10^7	5.7×10^6	4.8×10^7	4.7×10^8	7.8×10^6		2.9×10^8	3.4×10^9
Hospitals 7,000	4.4×10^9	5.2×10^8	3.7×10^3	2.5×10^9	3.6×10^7	1.6×10^9	4.1×10^8	4.4×10^9

Values are in Btu/building/year

A.2.1 Inventory

The inventory figures were obtained from Selected Services[A.17] and other publications and revised to reflect the number of establishments most likely to be impacted by integrated appliances.

- Hotels and motels -
  limited to those over 250 rooms; the breakdown is:

hotels	2,100
motels	1,000
motor hotels	1,000
	4,100

- Supermarkets -
  those supermarkets doing over $1,000,000 worth of business each year;

- Restaurants -
  excludes 'ma and pa' establishments and fast-food establishments;

- Coin-operated laundries -
  excludes dry cleaners;

- Schools -
  includes all primary and secondary schools with kitchen facilities (USDA[A.18]);

- Hospitals -
  hospitals only, excludes health care facilities, i.e., clinics.

A.2.2 Water Heaters

Water heating includes baths, showers, hand washing, dishwashing, and clothes washing. A ΔT of 85°F was assumed for all uses except dishwashing, where the ΔT was assumed to be 120°F.

Baths, showers, etc. (Hand washing)

- Hotel/Motel

  Hot water used: 30 gal/person/day (ASHRAE, 1966[A.19])
  Rooms: 250 (weighted average)
  Occupancy rate: 76% (190 rooms) at 1.5 persons/room (285 people/day)

(365 days/year) (30 gal/person/day) (285 people)
(1 Btu/lb °F) (85°F) (8.3 lb/gal)
$= 2.2 \times 10^9$ Btu/year

- Restaurants

150 customers/day
1/2 wash hands (75 people/day)
hot water needed: 0.8 gal/handwashing
operate 300 days/year

(75 people/day) (0.8 gal/person) (300 days/year)
(85 °F) (1 Btu/lb °F) (8.3 lb/gal)
$= 1.27 \times 10^7$ Btu/year

- Schools

450 students/school
180 school days/year

(810 gal/day) (180 days/year) (8.38 lb/gal)
(1 Btu/lb °F) (85°F)
$= 2.4 \times 10^8$ Btu/year

- Hospitals

85 gal/bed/day (includes water for therapeutic baths, etc.) (ASHRAE, 1966)
200 beds, 100% occupancy

(200 beds) (85 gal/bed/day) (365 days/year)
(8.38 lb/gal) (1 Btu/lb °F) (85°F)
$= 4.4 \times 10^9$ Btu/year

<u>Clothes Washing</u>

- Laundry

18 washers/laundry
8 loads/washer/day
13.4×10^3 Btu/load (36 gal, 45°F rise)

(8 loads/day) (365 days/year) (13.4×10^3 Btu/load)
$= 39.1 \times 10^6$ Btu/washer/year

(39.1 Btu/washer/year) (18 washers)
$= 7.0 \times 10^8$ Btu/year

- Hospitals

  10 gal/patient/day
  200 patients

  (10 gal/patient/day) (200 patients) (365 days/year)
  (8.38 lb/gal) (1 Btu/lb °F) (85°F)
  = 5.2×10^8 Btu/year

- Schools

  810 gal/day at 15 gal/shower is an average of 54 showers
  54 towels require about 45 gal of hot water

  (45 gal/day) (180 days/year) (8.3 lb/gal) (85°F)
  = 5.71×10^6 Btu/year

Dishwashing

The calculations below use:

2 meals/rack
2 gal/rack (Hobart)

- Hotel/Motel

  2 meals/person/day (570 meals/day)

  (1 rack/2 meals) (570 meals/day) (365 days/year)
  (2 gal/rack) (8.38 lb/gal) (1 Btu/lb °F) (120°F)
  = 2.1×10^8 Btu/year

- Restaurant*

  150 meals/day

  (150 meals/day) (300 days/year) (1 rack/2 meals)
  (2 gal/rack) (8.38 lb/gal) (1 Btu/lb °F) (120°F)
  = 4.5×10^7 Btu/year

*In the detailed analysis in Chapter 3, Table 3.2, the following additional volumes of hot water were added (including items on the following page):

40 gals initial dishwasher fill
180 gallon food preparation

resulting in a total consumption of 650 gallons recoverable hot water for the restaurant. In Chapter 6, an additional 140 gallons of water for steam kettles was added to the water consumption, yielding 790 gallons of hot water demand.

In addition:

240 gal/day for pot washing

(300 days/year) (240 gal/day) (8.38 lb/gal)
(1 Btu/lb °F) (120°F)
= 7.2×10^7 Btu/year

40 gal/day for slop sinks

(300 days/year) (40 gal/day) (8.38 lb/gal)
(1 Btu/lb °F) (120°F)
= 1.2×10^7 Btu/year

- Schools*

265 meals/day

(265 meals/day) (180 days/year) (1 rack/2 meals)
(2 gal/rack) (8.38 lb/gal) (1 Btu/lb °F) (120°F)
= 4.8×10^7 Btu/year

In addition:

180 gal/day for pot washing

(180 gal/day) (180 days/year) (8.38 lb/gal)
(1 Btu/lb °F) (120°F)

= 3.3×10^7 Btu/year

40 gal/day for slop sink

(40 gal/day) (180 days/year) (8.38 lb/gal)
(1 Btu/lb °F) (120°F)
= 7.2×10^6 Btu/year

*Although number of students/number of schools = 450; USDA information for October, 1976 shows:

$$\frac{24{,}131{,}000 \text{ meals}}{90{,}907 \text{ schools}} = 265 \text{ meals/day/school}$$

indicating that only 60% of the students were served.

- Hospitals

5 gal/patient/day (ASHRAE, 1966)

(5 gal/patient/day) (200 patients) (365 days/year) (8.38 lb/gal) (1 Btu/lb °F) (120°F)

$= 3.7 \times 10^8$ Btu/year

Range/Ovens

- Hotel/Motel

The restaurant in a hotel/motel with 250 rooms has the following equipment:

	Total Energy Consumption (10^3 Btu/hr)	Hours/Day
2 ranges	250	6.3
2 broilers	200	3.5
1 open flame broiler	100	10
rotary oven	100	6
2 fryers	260	6

This results in 5.4×10^6 Btu/day. The total energy use, operating 300 days/year, is 1.6×10^9 Btu/year.

- Restaurants

A typical restaurant has the following equipment

	Total Energy Consumption (10^3 Btu/hr)
4 burner range/oven	175
griddle	50
electric griddle	140
fryer	80
broaster	40
convection oven	40

Assuming operation at 10 hours/day, 300 days/year, at a 50% duty cycle, the total energy consumption is 7.9×10^8 Btu/year.

- Schools

Schools have basically the same equipment as restaurants, but operate only 180 days/year. Thus, the energy consumption is 4.7×10^8 Btu/year.

- Hospital

A hospital kitchen is equipped as follows:

	Total Energy Consumption (10^3 Btu/hr)
3 steam kettles	120
1 baking oven	40
2 roasting ovens	80
2 ranges	280
2 even heat range tops	80
2 steamers	100
2 deck broilers	80
2 fryers	130

This equipment is operated 7.5 hours/day, 365 days/year. The total energy consumption is 2.5×10^9 Btu/year.

Refrigerator/Freezers

- Hotel/Motel

570 meals/day
1.5 lb. of food/meal - 50% refrigerated
7 days of storage
2.33 lb/ft^3 refrigerated food
2 Btu/hr/ft^3

(1.5 lb/meal) (570 meals/day) (7 days)
= 5,985 lb.

(1/2 refrigerated) (5,985 lb.) (1 ft^3/2.33 lb)
(2 Btu/hr/ft^3) (8,760 hr/year)
= 4.5×10^7 Btu/year

- Supermarkets

A typical supermarket has the following refrigeration equipment:

display case for packaged food	15 hp
display case for meat	3 hp
product @ 50°F	5 hp
dairy display	5 hp
freezer	3 hp
cooler (meat)	2 hp
dairy cooler	2 hp
produce cooler	2 hp
meat packaging	3 hp
ice cream	5 hp
	45 hp

Source: Hussman[A.20]

duty cycle - 18 hours of operation/day - 75%

$0.75 \times 45 \text{ hp} = 33.75 \text{ hp}$

$(33.75 \text{ hp})\ (2{,}547 \text{ Btu/hr})\ (8{,}760 \text{ hr/year})$
$= 7.5 \times 10^8 \text{ Btu/year}$

- Restaurants

7 days of storage - 1.5 lb/meal - 50%
1.5 lb/meal - 50% refrigerated
150 meals/day
2.33 lb/ft^3 refrigerated food
2 Btu/hr/ft^3

$(1.5 \text{ lb/meal})\ (150 \text{ meals/day})\ (7 \text{ days})$
$= 1{,}575 \text{ lb}$

$(1/2 \text{ refrigerated})\ (1{,}575 \text{ lb})\ (1 \text{ ft}^3/2.33 \text{ lb})$
$(2 \text{ Btu/hr/ft}^3\ (8{,}760 \text{ hr/year})$
$= 1.2 \times 10^7 \text{ Btu/year}$

- Schools

7 days of storage - operating 9 months/year
265 meals/day
1.5 lb/meal - 50% refrigerated
2.33 lb/ft^3 refrigerated food
2 Btu/hr/ft^3

(1.5 lb/meal) (265 meals/day) (7 days)
= 2,783 lb

(2,783 lb) (1/2 refrigerated) (1 ft^3/2.33 lb)
(2 Btu/hr/ft^3) (6,570 hr/year)
= 7.8×10^6 Btu/year

- Hospitals

900 meals/day
7 days/storage
1.5 lb/meal - 50% refrigerated
2.33 lb/ft^3 refrigerated food
2 Btu/hr/ft^3

(900 meals/day) (7 days) (1.5 lb/meal)
= 9,450 lb

(1/2 refrigerated) (9,450 lb) (1 lb/2.33 ft^3)
(2 Btu/hr/ft^3) (8,760 hr/year)
= 3.6×10^7 Btu/year

Heating

- Supermarket

Heating accounts for 8% of the total energy usage in a supermarket (115 kwh/ft^2/year, 20,000 square feet). This is 3.9×10^9 Btu/year.

- Restaurants

The heating load for a restaurant is 3.6×10^8 Btu/year. This is based on a building with an area of 6,000 square feet, a height of 13 feet, and 35% glass. The mean outside temperature is 50.6°F for 207 days/year while the inside is maintained at 70°F. There are 2,983 degree days/year.

- Schools

Heating for schools is 86.15×10^3 Btu/ft^2/year (Arthur D. Little, Inc.). For a school with 40,000 square feet, heating requires 3.4×10^9 Btu/year.

- Hospitals

Hospital heating requires 4.4×10^9 Btu/year, based on 73.7×10^3 Btu/ft^2/year and 60,000 square feet (Arthur D. Little, Inc.).

Clothes Dryers

- Laundry

7 dryers, 12 loads/dryer/day
30 min/load
Burner rate: 90,000 Btu/hr
6 days/week

(7 dryers) (12 loads/dryer/day) (0.5 hr/load)
(90,000 Btu/hr) (312 days/year)
= 1.2×10^9 Btu/year

- Hospital

2 dryers, 48 loads/dryer/day
30 min/load
Burner rate: 90,000 Btu/hr

(2 dryers) (48 loads/dryer/day) (0.5 hr/load)
(90,000 Btu/hr) (365 days/year)
= 1.6×10^9 Btu/year

Air Conditioning

- Supermarket

An average supermarket requires 115 kwh/ft^2/year, based on 20,000 square feet of sales area. Air conditioning represents 4% of this total, or 2.0×10^9 Btu/year.

- Schools

To air condition a school of 40,000 square feet required 7.25×10^3 Btu/ft^2/year (Arthur D. Little, Inc.). This is 2.9×10^8 Btu/year.

- Hospitals

Air conditioning for a hospital of 60,000 square feet is 6.9×10^3 Btu/ft^2/year or 4.1×10^8 Btu/year.

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B. PARTITIONING OF WASTE HEAT INSIDE A HOME

B.1 PURPOSE

The purpose of the analysis was to develop an analytical model for evaluating the portion of appliance waste heat that displaces the winter space heating load.

B.2 GENERAL APPROACH

A 1,500 ft^2 house (38.7 feet x 38.7 feet) with 15% fenestration, insulated roof and walls, and four identical rooms was modeled as a matrix of four nodes, each node representing a room of the house. The resistance to heat transfer between each pair of rooms and between the outside and each room was calculated for a typical house.

Assuming that the appliance and thermostat are in adjacent rooms, the model predicted the portion of the appliance waste heat that was transferred to the thermostatted room. This waste heat kept the thermostat from calling for heat and thus reduced the heating load by a factor of 4 times the waste heat that entered the thermostatted room.

The premise of this analysis is that only the portion of the appliance waste heat that reached the thermostatted room actually contributed to displacing central heating demand. The remaining portion was lost through the house walls.

B.3 MODEL DESCRIPTION

Outline of the House

The outside walls were assumed to be of wooden frame construction, filled with 3-1/2 inch of fiberglass between the 2-inch by 4-inch studs. The ceiling was assumed to have 6 inches of insulation. The windows were single glazed. The walls between rooms were gypsum wallboard on 2-inch by 4-inch studs 16 inches apart. The floor between the rooms and the basement was plywood over 2-inch by 6-inch joists. Table B.1 shows the resistance of each element to heat transfer. Given the surface area of each part of the house, the heat flow per degree temperature difference was calculated in Table B.2.

With the thermostatted room set at a known temperature T_{U2}, as shown in Figure B.1, and useful heat flow into the room of Q_A from the room with the appliance, and Q_H from the central heating system, the room temperature would drop to T_T if the Q_A were discontinued and the thermostat inhibited so that Q_H is maintained. Therefore in steady state, the useful contribution of heat to the thermostatted room is:

TABLE B.1

RESISTANCE TO HEAT TRANSFER
OF VARIOUS PARTS OF THE MODEL HOUSE

Item	Resistance* $\frac{Hr\text{-}Ft^2\text{-}°F}{Btu}$
Window	0.91
Outside wall	13.5
Ceiling	25.3
Inside wall	3.28
Floor	4.55

*These numbers include inside and outside film heat transfer coefficients.

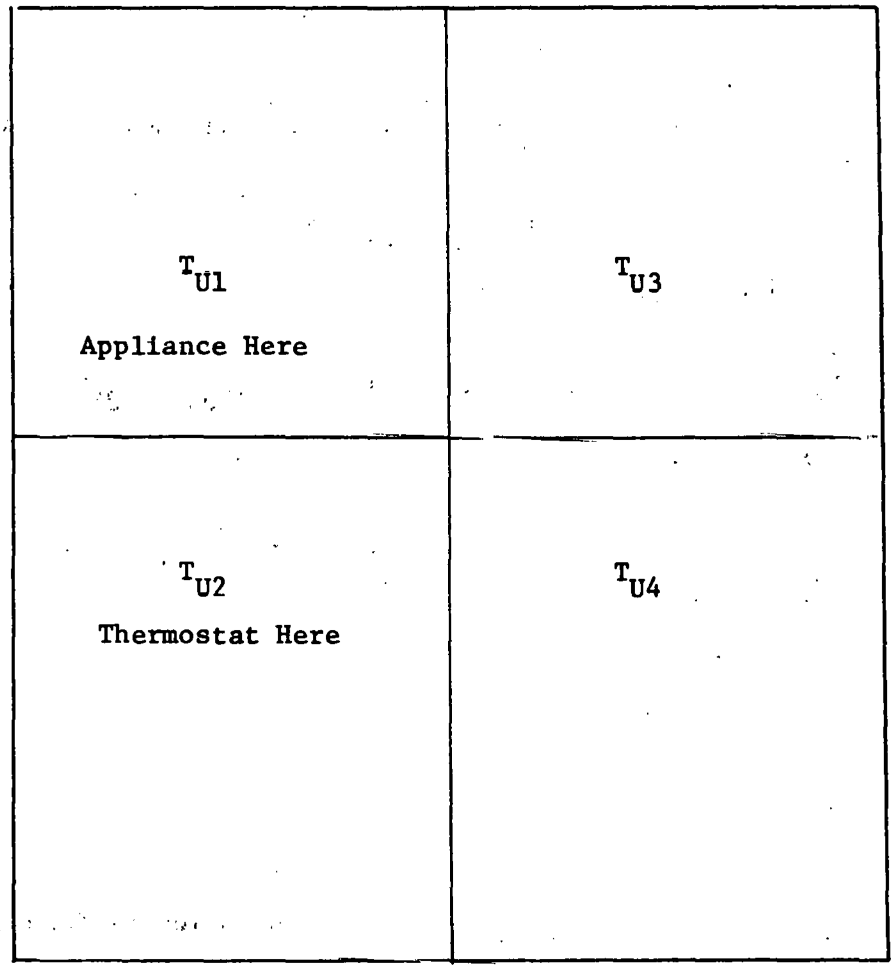
TABLE B.2

HEAT FLOW THROUGH EACH PART OF THE HOUSE

Item*	UA** (Btu/Hr-°F)
Windows, outside walls, and roof	106
Inside walls	122
Floor	82

*Each room is identical

**Does not include the effect of infiltration/ventilation. For example, one air change per hour is equivalent to an increase of UA of 67.4 Btu/Hr-°F. If a door is open between two rooms, the air changes per hour will be $3\sqrt{\Delta T}$ where ΔT is the temperature difference between the two rooms in °F.



A = Ambient
B = Basement
UA = Heat Transfer Coefficient

UA_A = UA to ambient (same for all rooms)
UA_B = UA to basement (same for all rooms)
UA_U = UA between rooms (same for all rooms)
Q_H = $Q_{\text{heating system}}$ for thermostatted room
Note: Q_H = 1/4 Boiler Output
Q_A = Q from appliance

Figure B.1 Actual Room Temperatures

Useful heat to thermostatted room = $(UA_A + UA_B)\ (T_{U2} - T_T)$

and since all four rooms are heated by the same central heat, the total heat saved is:

$$Q_{Saved} = 4(UA_A + UA_B)\ (T_{U2} - T_T) \tag{1}$$

The five other governing equations involving the six unknowns, Q_H, T_{U1}, T_{U3}, T_{U4}, Q_{Saved}, T_T are:

$$Q_H + Q_A = UA_B\ (T_{U1} - T_B) + UA_A\ (T_{U1} - T_A) + 2\ UA_U\ (T_{U1} - T_{U2}) \tag{2}$$

$$Q_H = UA_B\ (T_{U2} - T_B) + UA_A\ (T_{U2} - T_A) + UA_U\ (T_{U2} - T_{U1}) + UA_U\ (T_{U2} - T_{U4}) \tag{3}$$

$$Q_H = UA_B\ (T_{U4} - T_B) + UA_A\ (T_{U4} - T_A) + 2UA_U\ (T_{U4} - T_{U2}) \tag{4}$$

$$Q_H = UA_A\ (T_T - T_A) + UA_B\ (T_T - T_B) \tag{5}$$

By symmetry:

$$T_{U3} = T_{U2} \tag{6}$$

Through algebraic manipulation, the following solution is found:

$$Q_{Saved} = 4(UA_A + UA_B)\ \left(\frac{1 - C_2 \sum UA}{-2\ UA_U}\right)\ Q_A$$

where

$$\sum UA = UA_B + UA_A + 2\ UA_U$$

$$C_2 = \frac{\dfrac{UA_U}{\sum UA} - \dfrac{\sum UA}{2UA_U}}{2UA_U - \dfrac{\sum UA \sum UA}{2UA_U}}$$

TABLE B.3

AIR CHANGES PER HOUR UNDER VARIOUS CONDITIONS

	Air Changes per Hour
Forced Air Heating (cold day)	18
Natural Circulation Between Adjacent Indoor Rooms (1°F Temperature Difference between Rooms)	
Large interconnecting door	3
Small interconnecting door	1
Ventilation/Infiltration	
Ordinary house	1-1/2
Weather-stripped house	1

B.4 NUMERICAL ANALYSIS

If the air circulation between interior rooms is infinite, the reduction in heating required will be exactly equal to the heat output of the appliance. If there is no air circulation between rooms (i.e., all doors between rooms are shut) and one air change per hour takes place between each room and the outside air, only 65% of the heat generated by an appliance goes toward reducing the heating load. If the doors were open, about one to three air changes per hour would occur between rooms (depending on the appliance heating rate) and if the infiltration were one air change per hour, about 75% to 85% of the appliance heat would displace the heating load requirements (see Table B.3 and Figure B.2).

B.5 CONCLUSION

Although the percent of the appliance heat that may contribute to the heating load varies from about 65% to 100% in the examples given, the 75% to 85% range is probably typical of an average house. Thus, a value of 80% has been chosen for use in the analysis.

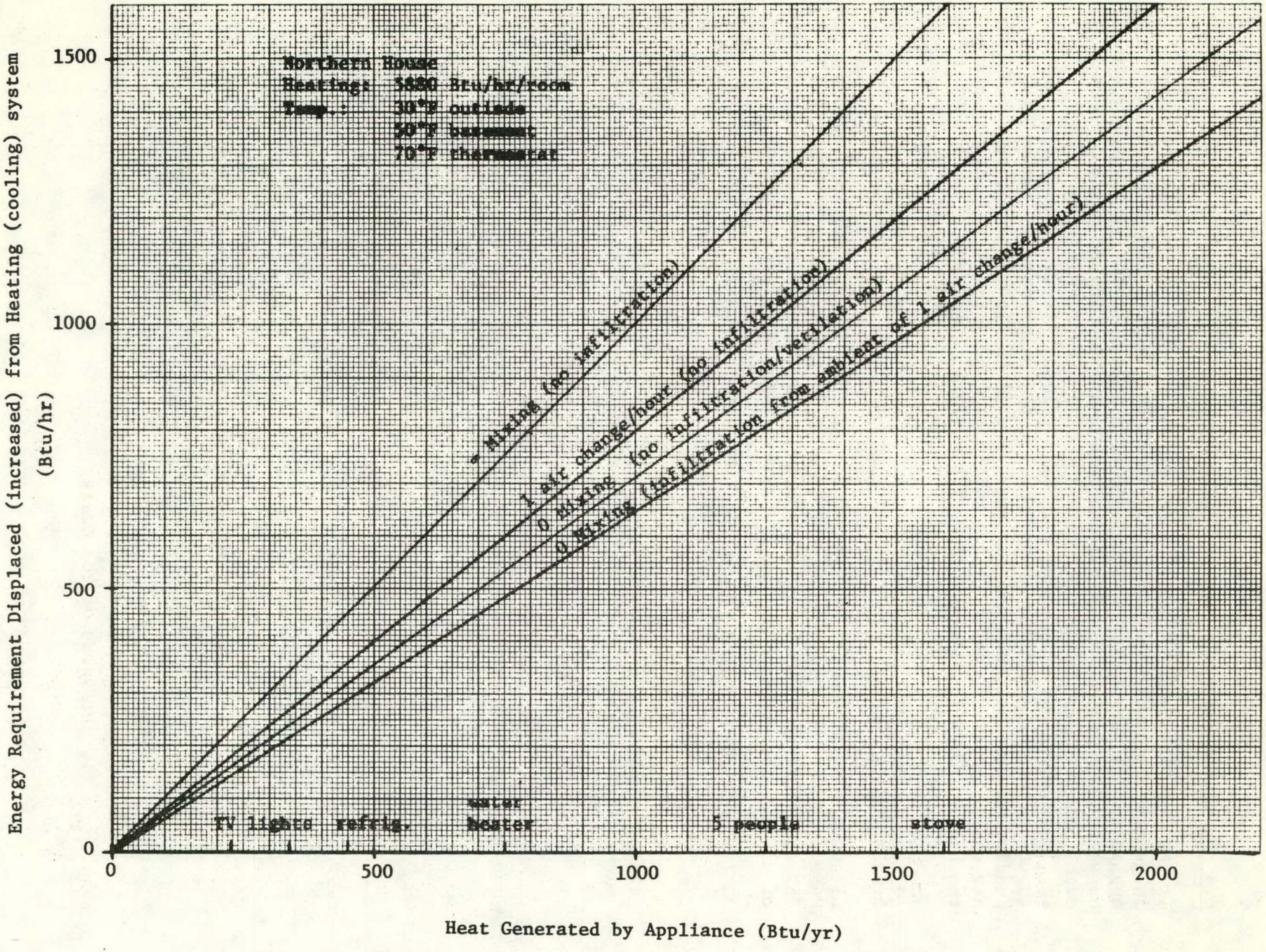


Figure B.2 Energy Partitioning of Waste Heat

C. GENERAL HEAT EXCHANGER OPTIMIZATION

Heat exchanger performance is characterized by the effectiveness - NTU curves; an example of several are shown in Figure C.1. The effectiveness is the ratio of the actual heat transfer to the maximum possible transfer between the media, and the NTU or number of transfer units is a measure of the heat exchanger size, defined as:

$$\frac{UA}{C_{min}}$$

where

UA is the surface conductance U and surface area A,

and

C_{min} is the smaller of the two values.

For initial screening in Chapter 2, a heat exchanger effectiveness was selected in the design regions shown by crosshatching in Figure C.1 following. A value of 80% for refrigerant to liquid heat exchangers and 70% for gas-to-gas or gas-to-liquid were chosen. These values were used in the initial screening. A more precise means for selecting heat exchanger sizes was needed for the detailed analysis phase of the study, as described below.

C.1 MINIMUM YEARS TO PAYBACK

As the effectiveness increases so does the heat exchanger cost, until a point is reached where further increase in size does not cause a comparable increase in heat transfer.

The effectiveness can be written as:

$$E = 1 - e^{-\alpha\ NTU}$$

where α is a constant given in Table C.1 for various heat exchanger configurations.

The value energy saved $E(t) by the heat exchanger over a time t is:

$$\$E(t) = C_{min}\ E\Delta T\ (\frac{\$}{Btu})\ t$$

where $\frac{\$}{Btu}$ is the value of the energy.

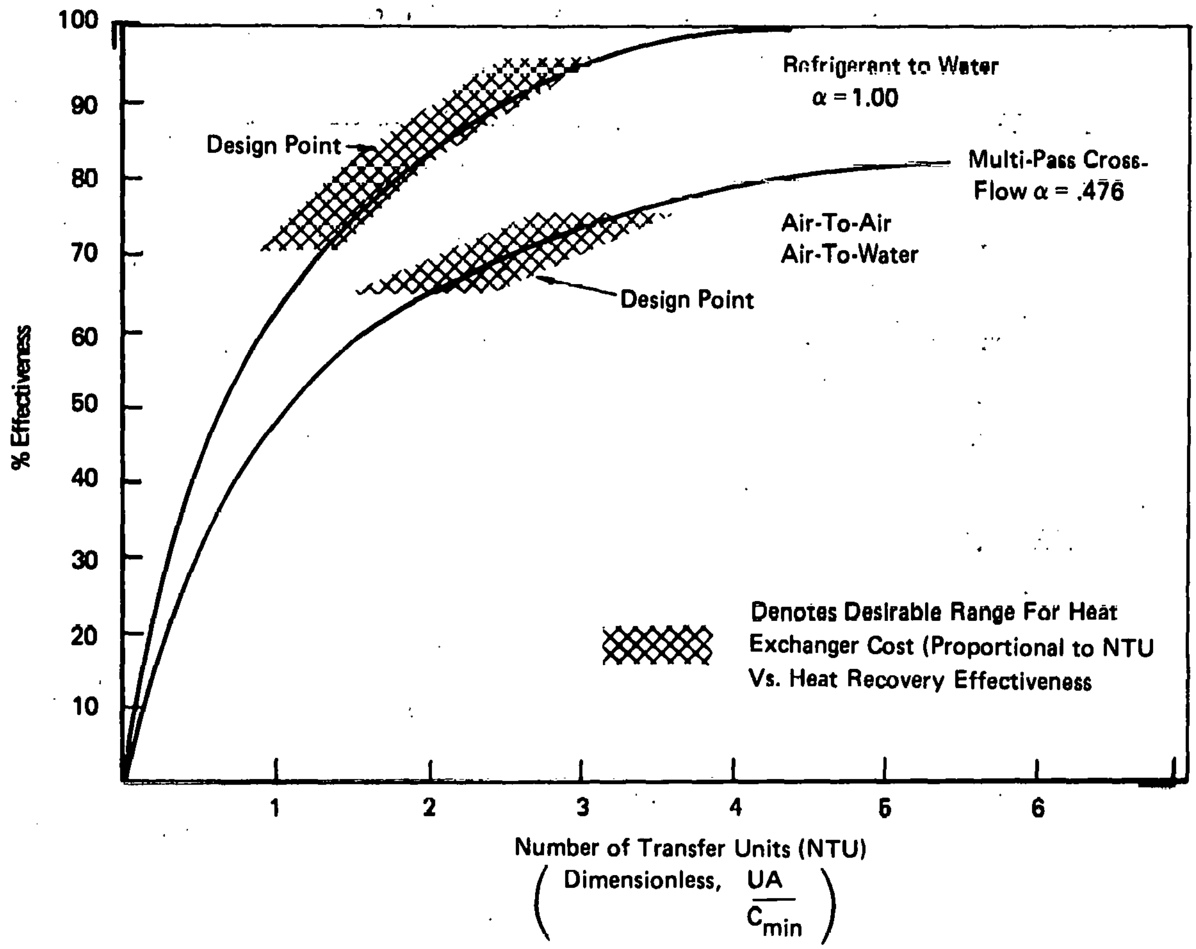


FIGURE C.1 HEAT EXCHANGER EFFECTIVENESS

TABLE C.1

VALUE OF EFFECTIVENESS CONSTANT α

Type of Heat Exchanger	α
Counterflow	
Water-to-air	.549
Phase change - refrigerant-to-water	1.000
Non-phase change- refrigerant-to-water	.549
Crossflow	
Water-to-air	.476

The cost of the heat exchanger plus the various fixed costs (F) of other system components is:

Cost of system = B (A) + F

where

B = a constant,

A = the heat exchanger surface area, and

F = the system fixed costs.

The years to payback the cost B (A) + F is

$$\text{Years to payback} = \frac{B \dfrac{NTU\ C_{min}}{U} + F}{C_{min}\ \Sigma\Delta T\ \dfrac{\$}{Btu}}$$

The minimum years to payback is found by setting the derivative with respect to NTU of this expression to zero and solving for NTU. The solution of the optimum NTU for different values of the dimensionless quantity U/C_{min} F/B is shown in Figure C.2 following. Indicated on these curves are the specifc values of the U/C_{min} F/B for the three final candidates in Chapter 4, 5, and 6. Table C.2 gives the backup data for these three systems. The reader should note that these optimum values of NTU and their corresponding values of effectiveness E are very close to the design values selected for screening.

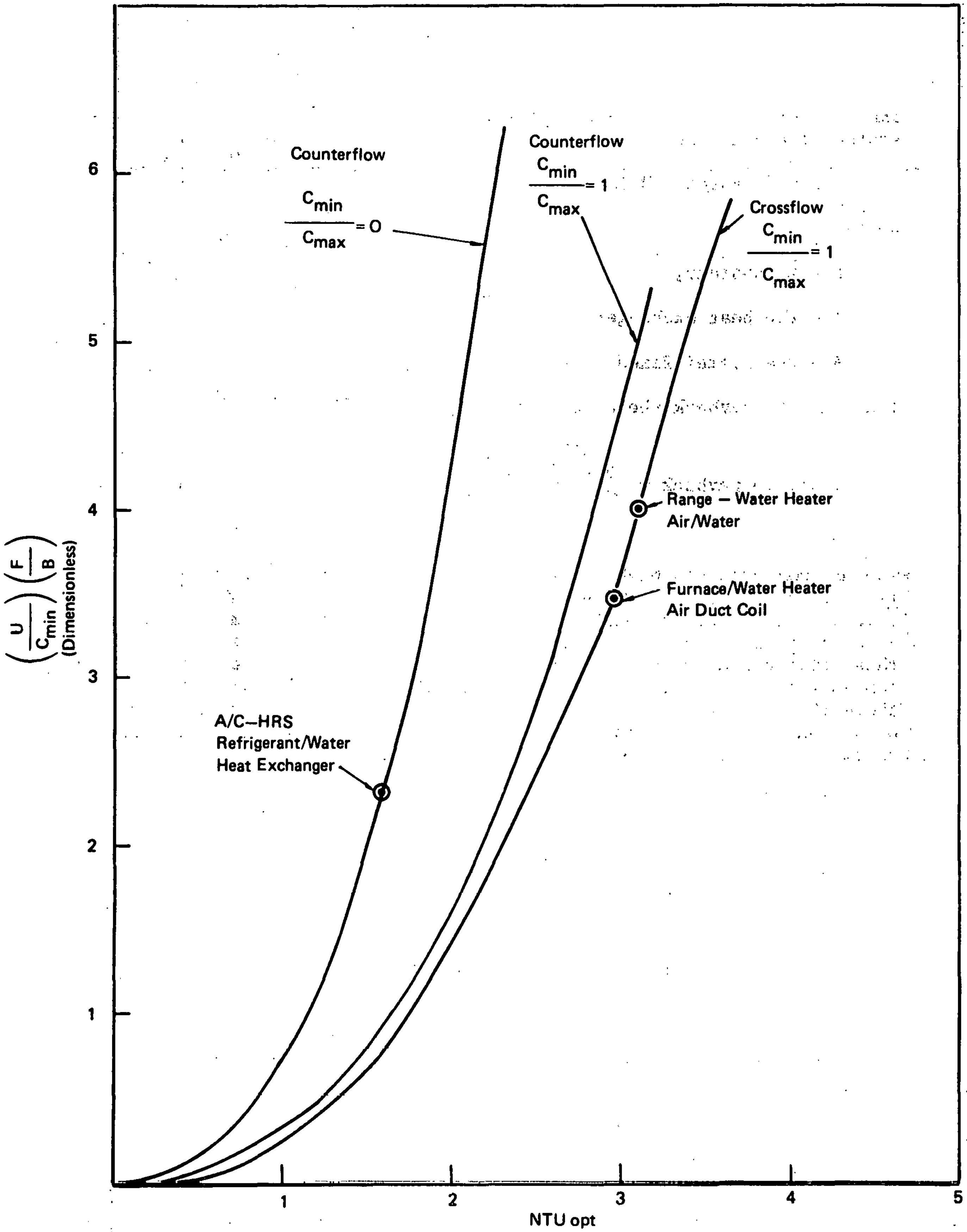


FIGURE C.2 OPTIMUM NTU

TABLE C.2

HEAT EXCHANGER PARAMETERS FOR INTEGRATED APPLIANCE CANDIDATES

Candidate	Configuration	U $\frac{Btu}{Hr-°F-Ft^2}$	C_{min} $\frac{Btu}{Hr-°F}$	F $	B $\frac{\$}{Ft^{2*}}$	$\frac{U}{C_{min}} \frac{F}{B}$	$NTU_{optimum}$	E
A/C-HRS Refrigerant to Water Heat Exchanger	Counterflow $\frac{C_{min}}{C_{max}} = 0$	300	500	233.60	59.16	2.3	1.5	.78
Furnace-Water Heater (Air Duct Heat Exchanger)	Crossflow $\frac{C_{min}}{C_{max}} = 1$	10	2770	480	.5[1]	3.48	2.9	.75
Range-Water Heater	Crossflow $\frac{C_{min}}{C_{max}} = 1$	10	9800	3880	1[2]	3.96	3.1	.77

*Area is based on surface area corresponding to the side with the minimum mass heat capacity ($\dot{m}C_p$)

[1] Uses a normal 10 fin/inch fin spacing for handling clean air

[2] Used a special 5 fin/inch fin spacing for handling grease-laden air to permit cleaning

D. COMPUTER PROGRAM USED TO MATCH AND SORT HYPOTHETICAL INTEGRATED APPLIANCES

D.1 INPUT

1) The energy use on a yearly basis (Btu/year) for each type of appliance.
2) The percent of gas and electric appliances.
3) The number of appliances per building.
4) The number of buildings.

D.2 OUTPUT

For both gas and electric appliances:

1) The Btu/year for each type of appliance of each type of building = Input (1) x Input (3)
2) The Btu/year on a nationwide basis is obtained by multiplying the energy use per year by the number of appliances per building by the number of buildings.
3) A matching of waste heat users to water heat suppliers is performed by taking the minimum of (a) the waste heat required, or (b) the waste heat available. On a per building basis, the number of appliances per building is always taken as one. On a U.S. basis, the number/building is used to calculate the energy savings on a probability basis, i.e.:

   Savings = (Btu/year) (Number/Building of A) (Number/Building of B)

   That is, if a savings of 1×10^6 Btu/year/building are possible with the appliance "A" and appliance "B" combination and if there are 0.5 "A" appliances per building and 0.6 "B" appliances per building, then only 30% (30% = 0.5 x 0.6) of all buildings have BOTH appliance "A" and appliance "B". Thus, an integrated appliance is possible for only 30% of the building (not 50% as might have been expected). So, for the entire country, the savings is the savings of:

   (one "A"-"B" pair/year) x (the number of buildings) x (30% to correct for less than full market penetration).

   In cases where the number of appliances/building is greater than 1, the program in calculating the above sets the number/building = 1.0. This may have caused problems in the low rise on a U.S. basis, since the program always assumed only one "A"-"B" pair at most is possible; and in a low rise, 20 or 30 "A"-"B" pairs are possible, thus the savings are possibly low by a factor of 20 or 30 on a U.S. basis for low rise buildings.

D.3 WINTER OUTPUT

Heat delivered to the room is not "waste" heat, and thus is not considered to be worth recovering. Thus, in winter, the only waste heat is that heat that leaks to ambient. For example, only 20% of a refrigerator's energy consumption is truly waste heat in the winter (the other 80% heats the place).

D.4 SUMMER OUTPUT

All heat in the summer is waste heat (the value of lowering the air conditioning load by removing waste heat is not considered). Thus, in the summer, 100% of the refrigerator's rejected heat is "waste" heat and all worth recovering.

D.5 AIR CONDITIONER-HEAT PUMP MODEL

The system schematic for the air conditioner-heat pump model is shown in Figure D.1. The governing equations for the heat exchangers are given below.

D.5.1 Condenser

Desuperheat Section of Condenser

The heat exchanger is a crossflow type with the air unmixed and freon mixed. The general equations are:

$$q = C_{min}\,(T_3 - T_{a2})\,\varepsilon$$

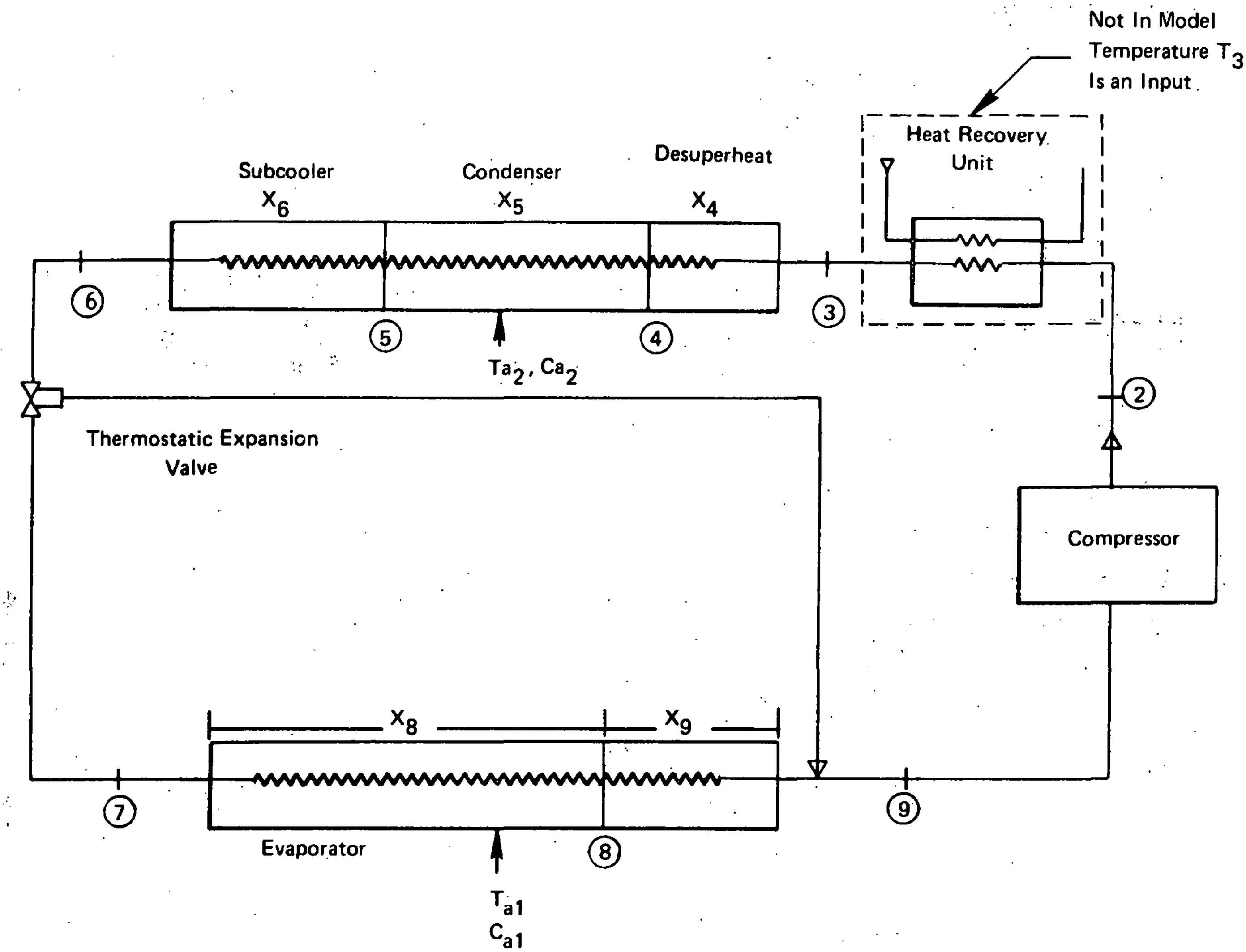
$$\varepsilon = 1 - e^{-\frac{\Gamma}{r}}$$

$$\Gamma = 1 - e^{-NTU \cdot r}$$

In the case where $C_f < C_a$, then

$$r = \frac{C_f}{C_a}$$

and



Symbols Used In Text

T_a – Approach air temperature to evaporator (1) and to condenser (2)

C_a – Air flow heat capacity ($\dot{m}\ C_p$) in $\frac{Btu}{hr\ ^oF}$

C_f – Refrigerant mass heat capacity ($\dot{m}\ C_p$) in $\frac{Btu}{hr\ ^oF}$

C_{min} – mass flow stream with smaller ($\dot{m}\ C_p$) in $\frac{Btu}{hr\ ^oF}$

FIGURE D.1 AIR CONDITIONER/HEAT PUMP MODEL

$$q = C_f\ (T_3 - T_{a2})\ [1 - e^{-\left(\frac{1 - e^{UA/C_a}}{C_f/C_a}\right)}]$$

In the case where $C_f \geq C_a$, then

$$r = \frac{C_a}{C_f}$$

and

$$\epsilon = \frac{1}{r}\ [1 - e^{-(1 - e^{-NTU})\ r}]$$

$$q = C_f\ (T_3 - T_{a2})\ [1 - e^{-\ (1 - e^{-\ UA/C_a})\ C_a/C_f}]$$

Both expressions for q are identical.

Thus,

$$q = C_f\ (T_3 - T_{a2})\ [1 - e^{-\ (1 - e^{-\ UA/C_a})\ C_a/C_f}]$$

regardless of which stream has C_{min}.

If UA and C_a are total values for whole condenser and the portion occupied by the desuperheater is X_4, then

$$\left(\frac{UA}{C_a}\right)_{DS} = \frac{X_4 \cdot UA}{X_4 \cdot C_a} = \frac{UA}{C_a} = NTU_2$$

$$\left(\frac{C_a}{C_f}\right)_{DS} = X_4 \cdot \frac{C_a}{w\ C_g}$$

$$(T_3 - T_4) = (T_3 - T_{a2})\ [1 - e^{-(1 - e^{-NTU_2})(X_4\ C_{a2})/(w\ C_{g3})}]$$

$$\frac{T_3 - T_4}{T_3 - T_{a2}} = 1 - e^{-(1 - e^{-NTU_2})(X_4\ C_{a2})/(w\ C_{g3})} \qquad (1)$$

Condensing Section of Condenser

$$C_{min} = X_5 C_{a2}$$

$$\frac{C_{min}}{C_{max}} = \simeq 0$$

$$\varepsilon = 1 - e^{-NTU_2}$$

$$w h_{fg4} = X_5 C_{a2} (T_4 - T_{a2}) (1 - e^{-NTU_2}) \tag{2}$$

Subcooling Section of Condenser

Same basic equation for HT as desuperheater.

$$T_5 = T_4$$

and

$$C_f = w C_p \text{ (liquid refrigerant)}$$

$$(T_4 - T_6) = (T_4 - T_{a2}) [1 - e^{-(1 - e^{-NTU_2}) X_6 C_{a2}/w C_{g2}}] \tag{3}$$

Total Hx

$$X_4 + X_5 + X_6 = 1 \tag{4}$$

D.5.2 Evaporator

Evaporating Section of Evaporator

$$C_{min} = X_8 C_{a1}$$

$$\frac{C_{min}}{C_{max}} = \simeq 0$$

$$\varepsilon = 1 - e^{-NTU_1}$$

$$w\ y_7\ h_{fg7} = X_8\ C_{a1}\ (T_{a1} - T_7)\ (1 - e^{-NTU_1}) \tag{5}$$

Superheating Section of Evaporator

$$(T_9 - T_7) = (T_{a1} - T_7)\ [1 - e^{-(1 - e^{-NTU_1})\ X_9\ C_{a1}/w\ C_{g9}}]$$

$$\frac{T_9 - T_7}{T_{a1} - T_7} = 1 - e^{-(1 - e^{-NTU_1})\ X_9\ C_{a1}/w\ C_{g9}} \tag{6}$$

Total Hx

$$X_8 + X_9 = 1 \tag{7}$$

D.5.3 Superheat Control

$$T_9 - T_7 = \Delta T_{sh} = \text{Constant} \tag{8}$$

D.5.4 Compressor

A typical compressor curve characteristic is shown in Figure D.2 where the capacity in Btu/hr and input watts (e) are given as a function of the condensing temperature T_4 and evaporating temperature T_7. The capacity can be written in terms of the compressor mass flow rate w, a variable needed for the solution, as given below:

$$w\ \frac{\text{lbs}}{\text{hr}} = \frac{\text{capacity}}{h_g\ (P_{evap},\ T = 95°F) - h_1\ (P_{cond},\ T = T_c - 15°F)}$$

This is evaluated at each T_7 and T_4; and the w and e can be written as:

$$e = \alpha_1\ T_4 + B_1\ T_4\ T_7 + \gamma_1$$

$$w - \alpha_2\ T_4 + B_2\ T_7 + \gamma_2$$

where for the specific compressor shown in the figure:

$$\alpha_1 = .57$$

$$\alpha_2 = -1.4$$

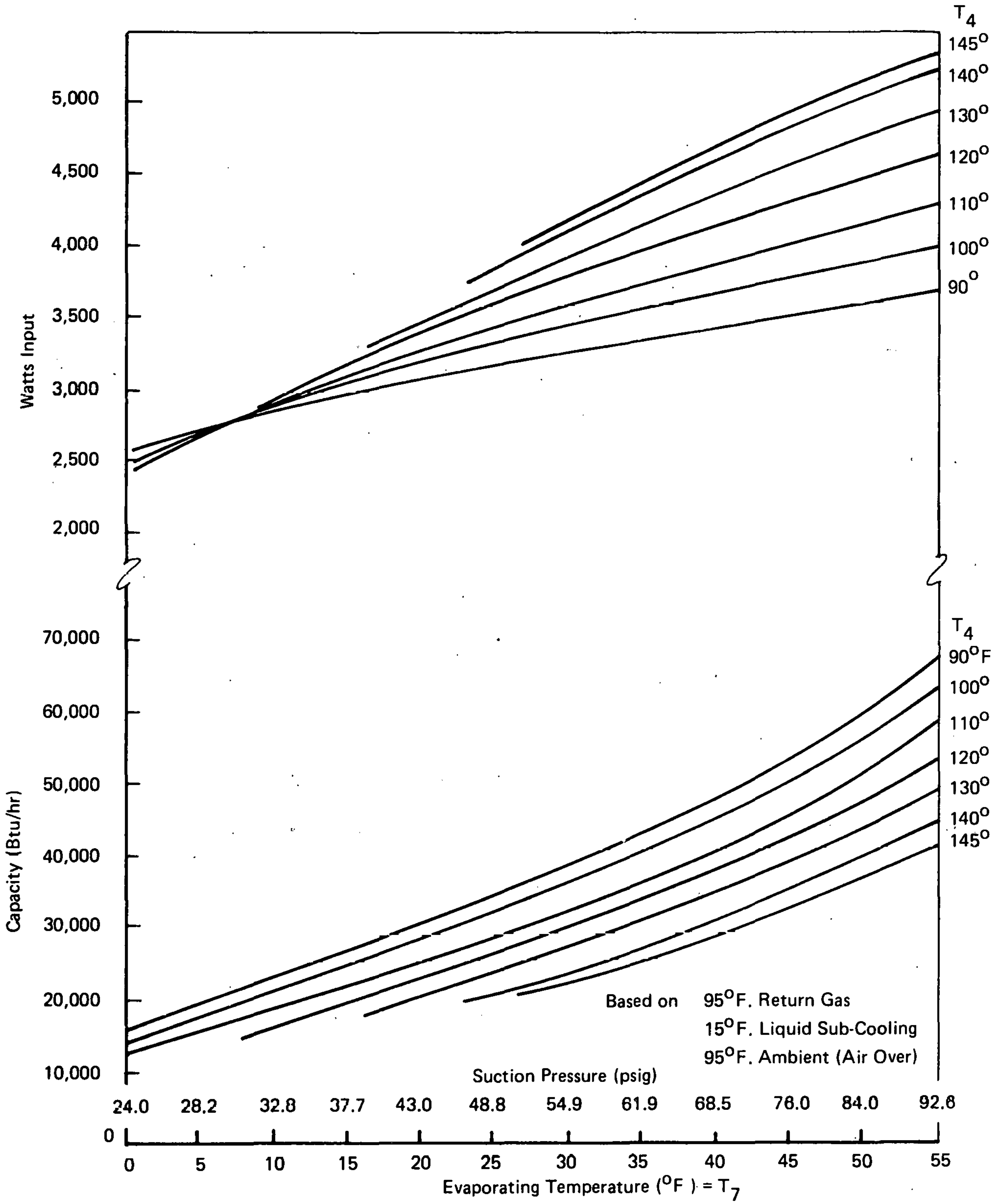


FIGURE D.2 TYPICAL COMPRESSOR CURVE CHARACTERISTIC

$B_1 = -34.5$

$B_2 = 11.7$

$\gamma_1 = 3{,}293$

$\gamma_2 = 211$

In summary, we have nine equations and ten unknowns, as shown in Table D.1. The equation for the throttle valve is missing.

D.5.5 Solution

Without the equation for the throttle, the solution is found over the range of permissible values of T_4 that satisfy the nine equations. The solution is started with a T_4 and a T_7. The physics of the problem indicate that T_7 be less than room temperature and T_4 must be greater than the ambient temperature. The equations are solved for w, T_9, X_8, X_9, and T_6. If T_6 is greater than T_4 (which is physically impossible), the initial estimate of T_7 was too high, and conversely if T_6 is less than either T_7 or the outdoor ambient air temperature (both situations are impossible), the initial estimate of T_7 was too low. In either case, another value of T_7 is selected and the procedure restarted for the beginning. If T_6 falls within acceptable bounds, X_6, X_5, X_4, and a new T_4 are calculated. If the new calculated T_4 is greater than the initial T_4 estimate, then the T_4 estimate was too low, and conversely if the calculated T_4 was less than the estimated T_4, the estimated T_4 was too high. A new value of T_4 is selected and the iteration is restarted at the beginning until the estimated T_4 is equal to the calculated T_4 (within acceptable limits).

A printout of a series of values of T_9 and T_4 falling into the acceptable range is shown in Figure 4.7 in Chapter 4.

TABLE D.1

SOLUTION MATRIX

Equation	w	T_4	T_6	X_4	X_5	X_6	T_7	T_9	X_8	X_9	T_3
1	x	x		x							An input, not an unknown
2	x	x			x						
3	x	x	x			x					
4				x	x	x					
5	x		x						x		
6	x						x	x		x	
7									x	x	
8							x	x			
9	x						x				

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E. STATEMENT OF WORK

The contractor shall accomplish Phase I of the project, consisting of the following tasks:

Task I

Submit, within 15 days of execution of a contract, a detailed program plan for ERDA review and approval which indicates more completely than the proposal does, final allocation of financial and personnel resources, timing of principal events that are to occur during the execution of the project, decision points and major milestones, program management plan, technical approach, and other items of direct relevance to timely and successful accomplishment of the program objectives. The contractor shall not proceed with Task II or beyond, until this plan is approved by the ERDA Program Manager. No changes to the plan shall be made without approval of the Program Manager.

Assemble a Review Committee that represents manufacturers, builders, and major user groups.

Task II

Assemble background information and data on conventional appliances, including patterns of energy usage, technical attributes, functional and energy characteristics, interaction between various appliances and climate conditioning systems, manufacturing and other cost data, and current market information. This information shall be presented in the Final Report (see Task VI).

Task III

Develop criteria for identifying and evaluating the potential of various integrated appliance candidates, including all promising applications. Such criteria shall include, but not be limited to, the potential for national energy savings, the time schedule on which such savings might realistically be achieved, the cost-effectiveness of a given approach, institutional or other factors that would have a strong effect bearing on consumer acceptance and commercialization, manufacturer capital requirements, the applicability of present manufacturing/installation practice to the application, capital and labor requirements, and the importance of variable geographical, climatic, and demographic factors. These criteria shall be presented and justified in a Task Report to be submitted in accordance with Exhibit II instructions.

Task IV

Identify and evaluate the most promising combinations and designs of integrated appliances that give life-cycle cost-effective energy savings, according to the criteria developed in Task III. It is anticipated that this task may require employing some techniques of computer simulation modeling and/or limited laboratory and experimental work. Results shall be presented in a Task Report.

Task V

Develop a detailed proposed approach for the Phase II demonstration phase of this program, and a plan for production of prototype units and a larger number of demonstration units for actual application, test, and evaluation in practical field use. Arrange for the participation of a major user group and a manufacturer in the demonstration of enough units to obtain adequate information to determine actual energy efficiency, reliability, quality of performance, safety, cost, etc. Results of this task shall be presented in a Task Report.

Task VI

The results of Tasks II, III, IV, and V shall be presented in a detailed and comprehensive draft Final Report, to be submitted within nine and one half months of execution of a contract. The ERDA Program Manager will review the draft report within 15 days of receipt and make criticisms and suggestions where necessary. The contractor shall then incorporate all such comments and criticisms into a Final Report, and shall deliver one camera-ready original and 200 copies within 30 days of receiving the Program Manager's comments.

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