MASTER

RESIDUAL ENERGY APPLICATIONS PROGRAM

SYSTEMS ANALYSIS REPORT

October, 1980

SOUTH CAROLINA

Energy Research Institute

WORK PERFORMED UNDER CONTRACT NO. DE-AC09-77ET12866

WITH THE UNITED STATES DEPARTMENT OF ENERGY

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

DISCLAIMER

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Department of Energy, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, mark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

RESIDUAL ENERGY APPLICATIONS PROGRAM
SYSTEMS ANALYSIS REPORT

P. W. Yngve, Principal Investigator
F. J. McCrosson, Project Manager

South Carolina
ENERGY RESEARCH INSTITUTE
Suite 670
First National Bank Building
1401 Main Street
Columbia, South Carolina 29201
(803) - 256-6400

Prepared for the
United States Department of Energy
Office of Coal Utilization
Assistant Secretary for Fossil Energy
Under Contract No. DE-AC09-77ET12866

ABSTRACT

Current DOE plans call for building an Energy Applied Systems Test (EAST) Facility at the Savannah River Plant in close proximity to the 140°F-150°F waste heat from one of several operating nuclear reactors. The waste water flow from each reactor, approximately 165,000 gpm, provides a unique opportunity to test the performance and operating characteristics of large-scale waste heat power generation and heat pump system concepts. This report provides a preliminary description of the potential end-use market, parametric data on heat pump and the power generation system technology, a preliminary listing of EAST Facility requirements, and an example of an integrated industrial park utilizing the technology to maximize economic pay back.

The parametric heat pump analysis concluded that dual-fluid Rankine cycle heat pumps with capacities as high as 400×10^6 Btu/hr, can utilize large sources of low temperature residual heat to provide 300°F saturated steam for an industrial park. The before tax return on investment for this concept is 36.2%. The analysis also concluded that smaller modular heat pumps could fulfill the same objective while sacrificing only a moderate rate of return.

The parametric power generation analysis concluded that multi-pressure Rankine cycle systems not only are superior to single pressure systems, but can also be developed for large systems (≈ 17 MW_e). This same technology is applicable to smaller systems at the sacrifice of higher investment per unit output.

ACKNOWLEDGEMENT

This report is submitted to the Department of Energy by the South Carolina Energy Research Institute to fulfill part of the obligations under contract number DE-AC09-77ET12866. As the first report in a series discussing the technology of heat recovery systems, its objectives were to provide parametric information on the current technology and to provide an example of the technology for a specific location.

This report includes information from the Advanced Energy Conversion Systems Group of the Energy Research Laboratory at United Technology Research Center. Particular individuals deserve recognition for their contribution:

F. R. Biancardi
R. W. Bass
H. E. Kalifa
A. I. Chalfant

SYSTEMS ANALYSIS REPORT

TABLE OF CONTENTS

	<u>PAGE</u>
I. Introduction	1
A. Background	1
B. Report Objectives	2
II. Residual Heat Resources	3
A. Introduction	3
B. Federal Facilities	5
1. Paducah Gaseous Diffusion Plant	5
2. Oak Ridge Gaseous Diffusion Plant	11
3. Portsmouth Gaseous Diffusion Plant	15
C. Base-Load Utilities	20
1. Utilities with Cooling Towers	22
2. Utilities with Once Through Cooling	25
3. Factors in Utility Implementation	26
4. Industrial Plants	29
D. References	35
III. Heat Pump	37
A. Introduction	37
1. Objectives	37
2. Approach	39
3. Conclusions	41
B. Application Characteristics	45
1. Industrial Park End Uses	46
2. In-Plant End Uses	50

	<u>PAGE</u>
C. Heat Pump Cycles and Terminology	52
1. Selection of Rankine Cycle	52
2. Typical Rankine Thermodynamic Cycles	53
3. Terminology and Definitions	56
D. Configuration	59
1. Working Fluid Selection	60
Closed Cycles	60
Rejected Fluids	64
Semi-Open and Open-Cycles	65
Dual-Fluid Cycles	65
2. Working Fluid Comparison	66
Performance	66
Component Sizing	71
Major Component Cost	72
E. Heat Pump Drivers	82
1. Gas Turbine	82
2. Diesel	86
3. Steam Turbine	88
4. Electric Motor	88
5. Residual-Heat-Driven Rankine Cycle	90
6. Driver Selection	91
F. System Selection for Portsmouth Gaseous Diffusion Plant	92
G. Alternative Application Evaluations	101
1. Utility Application	101
2. Industrial Applications	103
H. Prototype System Characteristics	107
I. Test Facility Specifications	110
1. Heat Pump System	110

	<u>PAGE</u>
J. References	114
IV. Power Recovery Systems	115
A. Introduction	115
1. Objectives	116
2. Conclusions	117
B. Residual Heat Characteristics	119
C. Configuration Design and Performance Evaluation	120
1. Thermodynamic Considerations	120
2. Working Fluid Evaluation	123
Performance	126
Component Sizing	129
Effect of Isolation Heat Exchangers	133
D. System Cost Analysis	137
E. Sensitivity and Trade-Off Studies	139
1. Pipeline Optimization	139
2. Multi-Pressure Systems	144
3. Effect of Residual Stream Temperature	148
4. Effect of Condenser Temperature	152
5. Effect of Pinch Temperature Difference	152
6. Effect of IHE Temperature Difference	155
7. Effect of Size	155
8. Alternate Isolation Approaches	155
F. Cooling Water Requirements	160
G. Assessment of Economic Merit	162
H. Power Recovery at Portsmouth GDP	167
Operating Conditions	168
I. Prototype System Characteristics	170

	<u>PAGE</u>
J. Test Facility Specifications	173
K. References	178
Appendix IV-A - Thermodynamic Efficiencies	179
Appendix IV-B - Rankine Cycle Working Fluids	182
Appendix IV-C - Heat Transfer Coefficients	183
References	190
V. Central Energy Management System	191
A. Introduction	191
B. Conclusions	193
C. CEMS Site Description and Requirements	194
1. Portsmouth GDP Residual Heat Source	194
2. Industrial Park Energy Requirements	194
D. Selection of CEMS Heat Pump and Power Plant Systems	196
E. Description of Heat Pump and Power Generation Systems	200
1. Heat Pump System	200
2. Power Systems	205
F. Energy Savings with CEMS	213

LIST OF FIGURES

<u>FIGURE NUMBER</u>		<u>PAGE</u>
II-1	OPERATING POWER LEVEL FOR THE PADUCAH GASEOUS DIFFUSION PLANT	6
II-2	TYPICAL POWER PLANT LOSSES	21
III-1	APPROACH TO HEAT PUMP ANALYSIS	40
III-2	INDUSTRIAL PROCESS HEAT REQUIREMENTS BELOW 400°F	47
III-3	TYPICAL INDIVIDUAL INDUSTRIAL PLANT PROCESS HEAT REQUIREMENTS	49
III-4	RANKINE CYCLE HEAT PUMP	54
III-5	MULTI-STAGE RANKINE CYCLE HEAT PUMP	55
III-6	DUAL-FLUID RANKINE CYCLE HEAT PUMP	67
III-7	HEAT PUMP PERFORMANCE	80
III-8	GAS TURBINE PERFORMANCE	84
III-9	PRIME MOVER COSTS	85
III-10	HEAT PUMP SYSTEM ECONOMICS FOR PORTSMOUTH APPLICATION	100
III-11	THERMALLY DRIVEN HEAT PUMP SYSTEM ECONOMICS	106
IV-1	RANKINE CYCLE SCHEMATIC	122
IV-2	EFFICIENCY OF LOW-TEMPERATURE SINGLE-PRESSURE RANKINE CYCLE SYSTEMS	127
IV-3	SPECIFIC NET OUTPUT AND HEAT REJECTION RATE OF SINGLE-PRESSURE R717 SYSTEMS	135
IV-4	SPECIFIC COOLING WATER REQUIREMENTS OF SINGLE PRESSURE R717 SYSTEMS	136
IV-5	INSTALLED COST OF POWER PLANT	138
IV-6	INSTALLED COST OF POWER PLANT INCLUDING A 3-MILE PIPELINE	141
IV-7	EFFECT OF PIPELINE ON THE INSTALLED COST OF OPTIMIZED SINGLE-PRESSURE SYSTEMS	142

LIST OF FIGURES

<u>FIGURE NUMBER</u>		<u>PAGE</u>
IV-8	DUAL-PRESSURE RANKINE-CYCLE SYSTEM	145
IV-9	OPTIMIZED UTILIZATION EFFICIENCY OF R717 MULTI-PRESSURE SYSTEMS	146
IV-10	INSTALLED COST OF OPTIMIZED DUAL-PRESSURE SYSTEMS	147
IV-11	EFFECT OF PIPELINE ON INSTALLED COST OF OPTIMIZED DUAL-PRESSURE SYSTEMS	149
IV-12	EFFECT OF WASTE-STREAM TEMPERATURE ON THE GROSS UTILIZATION EFFICIENCY OF R717 SYSTEMS	150
IV-13	EFFECT OF WASTE STREAM TEMPERATURE ON THE INSTALLED COST OF SINGLE-PRESSURE SYSTEMS	151
IV-14	EFFECT OF CONDENSER TEMPERATURE ON THE INSTALLED COST OF SINGLE-PRESSURE SYSTEMS	153
IV-15	EFFECT OF HEAT EXCHANGER PINCH TEMPERATURE DIFFERENCE ON THE OUTPUT AND INSTALLED COST OF SINGLE-PRESSURE SYSTEMS	154
IV-16	EFFECT OF THE TEMPERATURE DIFFERENCE ON THE INSTALLED COST AND POWER OUTPUT OF R717 SINGLE- PRESSURE SYSTEMS	156
IV-17	EFFECT OF SIZE ON THE INSTALLED COST OF SINGLE- PRESSURE SYSTEMS	157
IV-18	COOLING WATER REQUIREMENTS AND NET OUTPUT OF DUAL-PRESSURE R717 SYSTEMS	161
IV-19	ECONOMIC FIGURE OF MERIT	165
IV-20	COST AND NET OUTPUT OF A DUAL-PRESSURE R717 SYSTEM AT PORTSMOUTH GDP	169
IV-21	SCHEMATIC FLOW DIAGRAM OF THE CEMS POWER GENERATION SYSTEM	174
V-1	EFFECT OF FLOW SPLIT ON THE ECONOMICS OF A CENTRAL ENERGY MANAGEMENT SYSTEM AT THE PGDP	199
V-2	ELEVATION VIEW OF HEAT PUMP SYSTEM	201
V-3	END VIEW OF HEAT PUMP SYSTEM	202

LIST OF FIGURES

<u>FIGURE NUMBER</u>		<u>PAGE</u>
V-4	SCHEMATIC FLOW DIAGRAM OF THE CEMS POWER GENERATION SYSTEM	208
V-5	ELEVATION VIEW OF POWER GENERATION SYSTEM	210
V-6	PLAN VIEW OF POWER GENERATION SYSTEM	211

LIST OF TABLES

<u>TABLE NUMBER</u>		<u>PAGE</u>
II-1	WASTE HEAT CHARACTERISTICS PADUCAH GASEOUS DIFFUSION PLANT	8
II-2	TEMPERATURE CHARACTERISTICS OF THE OHIO RIVER IN THE VICINITY OF THE PADUCAH GASEOUS DIFFUSION PLANT	9
II-3	WASTE HEAT CHARACTERISTICS OAK RIDGE GASEOUS DIFFUSION PLANT	12
II-4	TEMPERATURE CHARACTERISTICS OF THE CLINCH RIVER IN THE VICINITY OF THE OAK RIDGE GASEOUS DIFFUSION PLANT	14
II-5	WASTE HEAT CHARACTERISTICS AT THE PORTSMOUTH GASEOUS DIFFUSION PLANT	16
II-6	TEMPERATURE CHARACTERISTICS OF THE SCIOTO RIVER IN THE VICINITY OF THE PORTSMOUTH GASEOUS DIFFUSION PLANT	18
II-7	PROJECTED COOLING TOWER INLET TEMPERATURE WATTS BAR NUCLEAR POWER PLANT	24
II-8	DUKE POWER PLANT ALLEN AVERAGE INLET AND OUTLET TEMPERATURES	27
II-9	INDUSTRIES WITH POTENTIAL RESIDUAL HEAT RECOVERY HEAT PUMP APPLICATIONS	31
II-10	INDUSTRIES WITH POTENTIAL WASTE HEAT RECOVERY POWER GENERATION APPLICATIONS	34
III-1	CANDIDATE HEAT PUMP APPLICATIONS	42
III-2	COMPARISON OF HEAT PUMP OPTIONS FOR FEDERAL FACILITY APPLICATIONS	70
III-3	COSTS OF HEAT PUMP WORKING FLUIDS	74
III-4	HEAT PUMP COST COMPARISON	75
III-5	HEAT EXCHANGER OVERALL HEAT TRANSFER COEFFICIENTS	76
III-6	HEAT PUMP COST PERFORMANCE COMPARISON	79
III-7	TYPICAL GAS TURBINE DRIVES	83
III-8	TYPICAL DIESEL MECHANICAL DRIVES	87
III-9	STEAM TURBINE PERFORMANCE FOR HEAT PUMP DRIVER	89

LIST OF TABLES

<u>TABLE NUMBER</u>		<u>PAGE</u>
III-10	COMPARISON OF HEAT PUMP SYSTEM INSTALLED COSTS FOR PORTSMOUTH APPLICATION	93
III-11	COMPARISON OF HEAT PUMP SYSTEMS WITH BOILERS FOR PORTSMOUTH APPLICATION	95
III-12	ASSUMPTIONS USED IN HEAT PUMP ECONOMIC ANALYSIS	96
III-13	COMPARISON OF HEAT PUMP DRIVERS FOR PORTSMOUTH APPLICATION	98
III-14	COMPARISON OF HEAT PUMP SYSTEM INSTALLED COSTS FOR UTILITY APPLICATION	102
III-15	COMPARISON OF HEAT PUMP SYSTEMS WITH BOILERS FOR UTILITY APPLICATION	104
III-16	HEAT PUMP REQUIREMENTS - EAST FACILITY	112
III-17	HEAT PUMP TEST MEASUREMENT REQUIREMENTS	113
IV-1	WASTE ENERGY SOURCE CHARACTERISTICS	121
IV-2	WORKING FLUID CHARACTERISTICS	125
IV-3	POWER RECOVERY SYSTEM ASSUMPTIONS	128
IV-4	SUMMARY OF PERFORMANCE AND DESIGN DATA	130
IV-5	SUMMARY OF PERFORMANCE AND DESIGN DATA AT PEAK UTILIZATION EFFICIENCY	134
IV-6	COMPARISON BETWEEN AMMONIA ISOLATION APPROACHES	159
IV-7	POWER GENERATION REQUIREMENTS - EAST FACILITY	176
IV-8	POWER CONVERSION TEST MEASUREMENT REQUIREMENTS	177
V-1	ECONOMIC ASSUMPTIONS	197
V-2	HEAT PUMP SYSTEM MAJOR COMPONENT SPECIFICATIONS AND AVAILABILITY	203
V-3	POWER GENERATION SYSTEM SUMMARY	212
V-4	FUEL SAVINGS ACHIEVABLE USING PORTSMOUTH CEMS	214

SECTION I

INTRODUCTION

A. BACKGROUND

This East Facility Systems Analysis report is submitted by South Carolina Energy Research Institute to cover work accomplished for DOE under the Residual Energy Applications Program (REAP - Contract No. DE-AC09-77ET 12866). Much of the technical information was developed by United Technologies Research Center (UTRC) as a subcontractor to ERI.

The EAST Facility at the Savannah River Plant will provide unique facilities for:

- Developmental testing of large-capacity residual energy recovery systems prior to service at the gaseous diffusion plants; i.e., Portsmouth, Ohio; Paducah, Kentucky; Oak Ridge, Tennessee; or other Federal installations.
- Application and confidence testing of heat recovery components and systems to produce data needed for successful equipment marketing in the private sector.

The initial project activity has been to define the equipment to be tested, the test program and the related test equipment. In order to manage this initial activity, ERI elected to divide the project into (4) tasks: Systems Analysis, Test and Operations, Environmental Analysis, and Support and Integration. Each task is submitting a summary report under separate cover. Together these complementing reports define the preliminary scope of the EAST Facility.

B. REPORT OBJECTIVES

This report deals with the Systems Analysis. It provides:

- An overview of potential resources for residual heat recovery (Section II).
- A description (including parametric data) of typical equipment to be tested at EAST Facility. Heat pumps are discussed in section III and power generation systems in section IV. The equipment descriptions form the basis for specifying the EAST Facility requirements.
- An integrated concept for an industrial park located at the Portsmouth G.D.P. (Section V). This concept illustrates, at a particular site, the optimization potential of combining and managing the total performance of heat pumps, power generation systems, and direct uses.

This report represents only one portion of the potentially available equipment sizes and types which can be tested at EAST. Subsequent reports will add information from other contributing manufacturers which will result in an expansion of technology type and equipment capability. For example, the report does not deal with absorption coolers or with power recovery systems other than the Rankine cycle. Later reports may add information on these systems.

Furthermore, the power recovery and heat pump technology are discussed from the viewpoint of the Federal facility application, which is toward the larger size and lower temperature end of the application scale. While the resulting equipment will fit a large part of the available residual heat, there are certainly other source areas available.

SECTION II

RESIDUAL HEAT RESOURCES

A. INTRODUCTION

Several factors determine the potential market for residual heat recovery systems. The primary factors require identifying potential sources and applications. Sources need to be described in terms of types, quantity available, temperature and location. Applications need to describe the end form, i.e., heat or power. Other important factors are the technology availability and the resulting economics.

This report presents only the information on the residual heat sources at three Federal facilities (Paducah, Oak Ridge and Portsmouth), base-load utilities, and industrial processes. The Savannah River Plant has not been included as it is the site selected for EAST Facility.

Information on potential applications will follow in future reports as deemed necessary. This division of task was done for the following reasons:

- Actual studies on potential applications must be site specific because of the interaction with the local surroundings. Examples of application studies at the Federal facility sites can be found in references II-1 - II-4. These DOE-funded studies by necessity include an analysis of the technology impact on the surrounding area. Similar studies can be found in literature on specific utilities. Such site-specific studies are much too detailed for the scope of this report.

- The information on residual heat sources should provide some overall picture of the potential market for different recovery systems. The information in this report represents only one part of the market picture. Future reports will provide additional source information.

B. FEDERAL FACILITIES

This section summarizes the residual heat source characteristics, cooling water availability, on-site energy use, and industrial park energy use for the three federally-owned nuclear facilities. In addition, the design requirements for residual heat recovery systems which can produce power and useful process heat at each of the three sites are defined.

1. Paducah Gaseous Diffusion Plant

The Paducah gaseous diffusion plant is located on the Ohio River in McCracken County in western Kentucky. The facility produces enriched uranium by isotope separation from uranium in the form of uranium hexafluoride (UF_6). As a result of a current plant upgrading and improvement program, the full power electrical load will eventually be 3040 MWe. Even though the plant operates continuously, projected fluctuations in the product stream will result in a projected power requirement as shown in Figure II-1. The power level proposed for use in subsequent phases of this study is 75 percent of the full load value, or 2300 MWe. This level is consistent with the level of plant operation planned for 1982 and beyond.

FIGURE II-1

a. Residual Heat Source Characteristics

More than 90 percent of the plant electrical load is dissipated by means of cooling water during the enrichment process. The plant complex consists of a series of diffusion stages arranged in four separate buildings. The buildings are each served by a pair of pipes which direct the cooling water from the diffusion operation to cooling towers. A second set of pipes return the water at a reduced temperature. The mass flow and temperature for each building at full, 75 percent (2300 MWe) and minimum projected power are given in Table II-1. Thus, the minimum total plant water flow at the 75 percent power condition occurs during the winter and is 2.44×10^5 gpm (1.22×10^8 lb/hr) at a temperature at or slightly in excess of 150°F.

b. Cooling Water Availability

Cooling water is required for heat rejection from residual heat recovery power generation systems. The boundary of the plant site is approximately three miles from the Ohio River. The minimum ten-year flow at this point in the river is 2.16×10^7 gpm (1.08×10^{10} lb/hr); the yearly average temperature is 61.3°F. Monthly average temperatures are given on Table II-2.

c. On-Site Energy Use

The plant will consume 2300 MWe at the 75 percent load level anticipated to be the normal operating condition after the planned upgrading is completed in 1981. While there is no requirement for on-site process steam, seasonal building heat is required. However, a current study is underway to use the plant cooling water directly for space heating.

TABLE II-1

WASTE HEAT CHARACTERISTICS
PADUCAH GASEOUS DIFFUSION PLANT
(Reference II-1)

Building	C-331	C-333	C-335	C-337
Pipe Diameter, in.	48	60	42	60
Summer Return Temperature, °F	95	95	95	95
Winter Return Temperature, °F	90	90	90	90
Summer Full Power Flow Rate, 10^3 gpm	64.2	123.0	63.2	123.4
Winter Full Power Flow Rate, 10^3 gpm	57.78	110.7	56.88	100.26
Summer 75 percent Power Flow Rate, 10^3 gpm	48.15	92.25	47.4	92.55
Winter 75 percent Power Flow Rate, 10^3 gpm	43.34	83.03	42.66	75.20
Full and 75 percent Power Temp., °F	150	150	151	152
Summer Minimum Power Flow Rate, 10^3 gpm	9.19	76.0	12.02	106.93
Winter Minimum Power Flow Rate, 10^3 gpm	8.27	65.73	10.74	90.73
Minimum Power Temp., °F	140	127	137	123

TABLE II-2

TEMPERATURE CHARACTERISTICS OF THE OHIO RIVER
IN THE VICINITY OF THE PADUCAH GASEOUS DIFFUSION PLANT
(Reference II-1)

<u>Month</u>	<u>Temperature, °F</u>
January	37.1
February	38.2
March	47.8
April	57.9
May	68.3
June	80.2
July	86.7
August	82.7
September	79.1
October	63.3
November	55.0
December	39.8
Annual Average	61.3
Three Hot Month Average	83.2
Three Cold Month Average	38.4

d. Design Requirements for Residual Heat Recovery Systems

For the design of residual heat recovery systems, the Paducah plant should be characterized by a cooling water rate of 1.22×10^8 lb/hr at a temperature of 150°F . This is the total flow from all four buildings. However, low power operation has historically occurred (Figure II-1) where cooling water temperatures are greatly reduced by using more flow than required (see Table II-1). It will be assumed that this occurrence can be avoided in the future.

Any interface between the cooling water piping system and all offsite use is presently restricted by DOE. The restriction not only precludes the removal of water from the site; it also prohibits heat exchange with the cooling water by any fluid other than water. These restrictions will require a secondary heat transfer loop with its inherent temperature drop in any power recovery or heat pump system which does not use water as the working fluid.

For power recovery, Ohio River water can be used for cooling. The temperature data for annual cooling water availability is given in Table II-2. The minimum river flow rate is more than adequate for such a use.

For thermal recovery by means of a heat pump, Ref. II-1 study indicated that industries compatible with the raw materials and markets of the Paducah area could only be attracted to a co-located industrial park if process heat at 300°F could be generated by residual heat recovery. While a higher temperature capability would have more applications, 300°F steam must be considered the minimum acceptable.

2. Oak Ridge Gaseous Diffusion Plant

The Oak Ridge gaseous diffusion plant is located in Roane County approximately 36 miles west of Knoxville, Tennessee. The facility produces enriched uranium by the isotope separation technique. As a result of a current upgrading and improvement program, the full power electrical load will be 2080 MW. However, operation at a power level of 1525 MWe is projected for 1982.

a. Residual Heat Source Characteristics

In excess of 90 percent of the electrical load is dissipated by means of cooling water during the enrichment process. The Oak Ridge complex consists of a series of diffusion stages arranged in two separate buildings. Building K-31 has one cooling line, while building K-33 has three lines. The cooling water flow and temperature for each building are given in Table II-3 for the nominal 1982 operating power level (1525 MWe) as well as for the minimum allowable operating level (600 MWe). The minimum power level data is given both with and without an internal heat use option.

b. Cooling Water Availability

The Clinch River flows through the Department of Energy (DOE) Reservation which contains the Oak Ridge plant. The diffusion buildings are located within one mile of the river at Campbell Bend. At this point, the average river water is 2207×10^3 gpm (1.1×10^9 lb/hr). The flow exceeds 678×10^3 gpm (3.4×10^8 lb/hr) 76 percent of the time. The Tennessee Valley Authority (TVA) has formulated regulations on use of the river based on the three-day, 20-year, low-flow condition of 4.8×10^3 gpm (2.4×10^7 lb/hr).

TABLE II-3

WASTE HEAT CHARACTERISTICS
 OAK RIDGE GASEOUS DIFFUSION PLANT
 Based on fouled condenser surfaces
 Reference II-5

	K-31 Building	K-33 Building
<u>1525 MW Power Level</u>		
Summer Mass Flow, 10^3 gpm	34.7	118.9
Winter Mass Flow, 10^3 gpm	32.4	109.6
Temperature, °F	165	154
<u>600 MW Power Level</u>		
Summer Mass Flow, 10^3 gpm	40.1	78.7
Winter Mass Flow, 10^3 gpm	34.5	64.1
Temperature, °F	126	117
<u>600 MW Power Level With Heat Option⁽¹⁾</u>		
Mass Flow, 10^3 gpm	13	21.8
Temperature, °F	176	159

(1) Building heat satisfied by using 6800 gpm of this flow, winter only.

Regulations governing heat rejection to the river require that the maximum temperature change not exceed 3°C (5.4°F) nor the absolute temperature exceed 30.5°C (86.9°F). Monthly temperature data (Ref. II-9) are presented in Table II-4.

c. On-Site Energy Use

The plant will consume a minimum of 1525 MWe as described above. There is no requirement for process steam. A current study is underway to provide space heating by tapping the cooling water stream.

d. Design Requirements for Residual Heat Recovery Systems

For designing residual heat recovery systems, the Oak Ridge plant should be characterized by a winter cooling water flow rate of 3.24×10^4 gpm (1.62×10^7 lb/hr) from K-31 building and 1.1×10^5 gpm (5.5×10^7 lb/hr) from K-33 building. These flows would have temperatures of 165°F and 154°F , respectively. However, if the operating power level is significantly lower than 1525 MW, reduced temperatures are likely (see Table II-3). It will be assumed that reduced power operation does not occur. A heat recovery interface similar to the Paducah plant is required.

For power recovery, Clinch River water is available for cooling purposes, but cooling towers may be required. For thermal recovery by means of a heat pump, the Ref. II-2 study indicated that 267°F process heat was needed to satisfy the needs for the textile finishing, coal-derived methanol and salt industries. It can be assumed that a 300°F level of process steam would add industries to this list, as shown in the Ref. II-1 study.

TABLE II-4

TEMPERATURE CHARACTERISTICS OF THE CLINCH RIVER
IN THE VICINITY OF THE OAK RIDGE GASEOUS DIFFUSION PLANT
(Reference II-2)

<u>Month</u>	<u>Temperature, °F</u>
January	48.5
February	48.5
March	51.0
April	54.0
May	60.5
June	66.8
July	68.5
August	69.8
September	69.8
October	65.5
November	57.5
December	49.5
Annual Average	59.2
Three Hot Month Average	69.4
Three Cold Month Average	48.8

3. Portsmouth Gaseous Diffusion Plant

The Portsmouth gaseous diffusion plant is located on 4,000 acres in Pike County about 22 miles north of Portsmouth, Ohio. The facility produces enriched uranium in a manner similar to the Oak Ridge and Paducah plants. As a result of a current upgrading and improvement program, the full power electrical load will be 2,260 MWe. The plant operates continuously. However, fluctuation in the product stream will result in a power requirement expected to average about 1,940 MWe by 1982.

a. Residual Heat Source Characteristics

In excess of 90 percent of the electrical load is dissipated by means of cooling water during the enrichment process. The Portsmouth complex consists of a series of diffusion stages arranged in three separate buildings. Process buildings X-333, X-330, and X-336 have four, two, and one cooling tower, respectively. Each cooling tower is supplied by two sets of incoming and return pipes and each building is provided with its own pump house. The mass flow and temperature for the three buildings are given in Table II-5. The temperature variations between systems indicate that the output from the south pipe of building X-333 would be the most attractive residual heat source.

b. Cooling Water Availability

The boundary of the plant is approximately one mile east of the Scioto River, a tributary of the Ohio River. The Scioto River has an annual mean flow of 2.08×10^6 gpm (1.0×10^9 lb/hr) and monthly temperatures given in Table II-6 (Ref. II-10). However, water flow as low as 350×10^3 gpm (1.8×10^8 lb/hr) is common in the fall, and the seven-day, ten-year projected low flow is 128 x

TABLE II-5

WASTE HEAT CHARACTERISTICS AT THE PORTSMOUTH GASEOUS DIFFUSION PLANT

Based on 1980 Condenser Surface Conditions
Reference II-7

Building	X-326	X-326	X-330	X-330	X-333	X-333
Supply Line	West	East	North	South	North	South
Winter Flow Rate at 1940 MW, 10^3 gpm	5.30	4.23	34.90	23.59	53.14	60.38
Winter Temp. at 1940 MW, °F	144	90-95	131 ⁽¹⁾	126	152 ⁽²⁾	156
Summer Flow Rate at 1940 MW, 10^3 gpm	5.84	4.23	42.15	44.62	64.87	78.16
Summer Temp. at 1940 MW, °F	144	90-95	141	139	156	156
Winter Flow Rate at 660 MW, 10^3 gpm	5.30	4.23	18.92	8.25	24.12	0 ⁽³⁾
Winter Temp. at 660 MW, °F	144	90-95	114 ⁽¹⁾	127	120 ⁽²⁾	---
Summer Flow Rate at 660 MW, 10^3 gpm	5.84	4.23	16.89	14.52	23.94	20.78
Summer Temp. at 660 MW, °F	144	90-95	143	134	133	135

(1) Reflects use for in-plant space heating.

(2) Reflects use for gas centrifuge plant space heating.

(3) Supply line not used at this power level.

10^3 gpm (6.41×10^7 lb/hr). The use of this river as a heat rejection medium for a residual heat recovery Rankine-cycle power generating system requires additional study due to the periodic low flow.

c. On-Site Energy Use

The plant is expected to consume a minimum of 1940 MWe during operation after 1982. There is no requirement for process steam. A current study is underway to use cooling water directly for space heating. This use is reflected in the data in Table II-5, as indicated.

d. Design Requirements for Residual Heat Recovery Systems

For the design of residual heat recovery systems, the Portsmouth plant should be characterized by a cooling water flow rate of 1.14×10^5 gpm (5.7×10^7 lb/hr). This is the flow from the two pipes of building X-333 during winter operation. It would have a temperature of 154°F . However, if the operating power level is significantly different, reduced temperatures are likely (see Table II-5). It will be assumed that reduced power operation does not occur. A heat recovery interface similar to the Paducah plant is required.

For power recovery, Scioto River water is available for cooling; but it may have flow limitation at low water that restricts its use. The temperature data for annual cooling water availability is given in Table II-6.

For thermal recovery by means of a heat pump, the Ref. II-3 study failed to identify industries that can utilize process steam for the reason previously given in

TABLE II-6.

TEMPERATURE CHARACTERISTICS OF THE SCIOTO RIVER
IN THE VICINITY OF THE PORTSMOUTH GASEOUS DIFFUSION PLANT
Reference II-7

<u>Month</u>	<u>Temperature, °F</u>
January	37.4
February	38.3
March	49.1
April	51.8
May	59.9
June	73.4
July	76.1
August	75.2
September	70.7
October	57.2
November	46.4
December	40.1
Annual Average	56.3
Three Hot Month Average	74.9
Three Cold Month Average	38.6

the discussion of the Paducah Diffusion Plant. However, the Ref. II-1 and II-2 studies can be used to set an acceptable level of 300°F for steam that can be of significant use to industries that are compatible with the Ohio River Valley raw materials and markets.

C. BASE-LOAD UTILITIES

The largest source of low-temperature residual heat is utility power plants. Using even the most modern energy conversion techniques, approximately 60 percent of the input energy is dissipated as low-temperature residual heat. Utility power plants can be classified in one of two categories depending on their operating schedule. Base-load units are designed to run continuously except for scheduled or emergency shutdown, while cycling units run for various periods of time each day necessary to match the load. In general, the most efficient units are used in the base-load mode. Only base-load units will be considered in this study, since only they can be relied on to provide continuous residual heat during their period of availability.

The basic components of a steam electric generating plant are shown in Figure II-2. High temperature steam is generated and fed to the turbine where thermal energy is converted to mechanical work to produce electric power. The low pressure steam exits the turbine and goes to a condenser where cooling water removes heat from the steam. The condensed steam is pumped back to the steam generator and the cooling water is rejected back to its source (once-through cooling) or pumped to a cooling tower for removing the gained heat and then recycled in a closed system.

Figure II-2 also illustrates a typical heat balance of a 1000 MWe fossil steam plant operating at full load. By comparison, a typical nuclear plant will have lower efficiencies (33% vs. 38%) with 62 percent discharged to the cooling water (vs. 48%) and 5 percent lost in the plant.

TYPICAL POWER PLANT LOSSES

FIGURE II-2

In either fossil or nuclear plants, the cooling water is typically heated from 10 to 30°F depending on plant design. The cooling water flow rate can be approximated by the relation

$$Q \text{ (ft}^3\text{/Hr)} = .547 \times 10^5 \left(\frac{P \text{ (MWe)}}{\Delta T \text{ (}^\circ\text{F)}} \right) \left[\left(1 - \frac{\eta_I}{100} \right) \frac{1}{\eta_P/100} \right] - \quad (\text{Ref. II - 21})$$

- P = plant capacity (MWe)
 ΔT = allowable temperature rise through the condenser (°F)
 η_I = Efficiency lost in-plant and in stack
(average fossil plant = .15, average nuclear = .05)
 η_P = overall plant efficiency

In general, a nuclear plant will require ~40 percent more cooling flow than a fossil-fired plant to maintain the equivalent ΔT across the condenser.

The following paragraphs discuss the two basic cooling systems (cooling towers and once-through cooling).

1. Utilities With Cooling Towers

In the cooling tower mode of heat rejection, air is ultimately used to withdraw heat from the power generating system. There are two basic ways to accomplish this task. The first is to reject heat from the power system condenser to cooling water and then to the ambient by direct contact with air. This is referred to as a wet cooling tower. The second is to reject heat directly from the condenser to ambient air without an intermediate fluid; this is referred to as a dry cooling tower. However, water may be used on the air side of the condenser to enhance heat transfer, especially during periods of high air temperature. In the following paragraphs, the two types of cooling towers will be discussed.

a. Wet Cooling Towers

There are 39 utility installations with outputs in excess of 500 MWe that use wet cooling towers (Ref. II-11). Wet cooling tower base-load utility power plants are typified by the soon-to-be activated TVA nuclear installations at Watts Bar on the Tennessee River. Projected temperatures for the cooling water at the inlet to the cooling towers are shown in Table II-7. The Watts Bar power plant consists of two independent 1270 MWe units, each of which uses 410,000 gpm (205×10^6 lb/hr) of cooling water to transport condenser heat to the cooling tower. The water returns to the condenser at a temperature that is 38°F cooler. The proposed utilization for each unit is 74 percent. Thus, it can be anticipated that at least 205×10^6 lb/hr of residual heat water will be available from at least one of the units 93 percent of the time. The performance of a temperature-augmenting heat pump system designed to match this type of heat rejection could be ascertained assuming a source temperature of 112.8°F for six months, of 120.9°F for three months, and of 105.3°F for three months (see Table II-7).

b. Dry Cooling Towers

Dry cooling towers are used in utility power plant applications only when adequate cooling water is unavailable. The rejection of the heat is accomplished by the transfer of sensible heat from the hot water to cooling air in a heat exchanger with no contact between air and water. The primary advantages are less environmental impact and virtual elimination of water consumption. The disadvantages are the higher required air flow ($\sim 3\times$ the air flow of a comparable wet tower), the higher cost of producing electricity compared to wet

TABLE II-7

PROJECTED COOLING TOWER INLET TEMPERATURE
WATTS BAR NUCLEAR POWER PLANT

	Maximum °F	Minimum °F	Average °F
January	108.6	99.2	103.9
February	109.4	100.0	104.7
March	114.1	102.2	108.1
April	118.9	105.9	112.4
May	122.0	110.0	116.0
June	124.7	114.4	119.5
July	125.8	117.5	121.7
August	125.9	117.0	121.4
September	124.3	112.3	118.3
October	120.3	106.1	113.2
November	114.7	99.8	107.2
December	110.0	98.7	107.3

Annual Average 112.8°F

Three Hot Month Average 120.9°F

Three Cold Month Average 105.3°F

towers and a greater dependence of performance on ambient conditions.

Outside the United States dry cooling systems are being installed routinely, especially in the U.S.S.R., Hungary, West Germany, Spain, and South Africa. Within the U.S., there is only one pure dry tower plant of reasonably large-scale size. That installation is a coal fired 330 MWe unit operated jointly by Black Hills Power and Light and Pacific Power and Light at Wyodak, Wyoming. The heat-rejection system works directly on the turbine exhaust flow. At full power, 2.6×10^9 lb/hr of 90 percent quality steam rejects heat in forced-air cooled finned tubes. In normal operation, the temperature of the condensing steam is 160°F. Thus, 2.3×10^9 Btu/hr is rejected to the air. The present operating utilization is 90 percent.

Though not a pure dry cooling tower system, there is a wet/dry hybrid system in operation at the San Juan III plant owned by Public Service Company of New Mexico and Tucson Gas and Electric and located at Farmington, New Mexico.

2. Utilities With Once Through Cooling

Power plant cooling by means of once through river water flow is common practice where sufficient water is available. There are 268 such units with outputs greater than 500 MWe in current operation (Ref. II-11). Operation of this type of base-load power plant is generally altered by reducing the heat rejection temperature when lower temperature cooling water is available so that the cooling water temperature rise is held to a predetermined value. Older installations (20 years or more) tend to have a 20°F temperature rise; newer, more efficient and more environmentally acceptable power plants are usually limited in

temperature rise to 15°F. For the purpose of this study, the 15°F limit will be used, since this represents future units as well as the existing installations which are likely to be in service for the longest period of time. An example of a fossil-fired electric generating plant is illustrated by the Duke Power Plant Allen (Ref. II-16).

Plant Allen has five (5) independent generating units with a total name plate capacity of 1155 MWe. Units 1 and 2 are each rated at 165 MWe with units 3, 4 and 5 each rated at 275 MWe. During winter months the designed cooling water flows are 360×10^3 gpm (180×10^6 lb/hr) and during the summer months the flows increase to 599×10^3 gpm (300×10^6 lb/hr).

Table II-8 gives monthly average outlet temperatures and max./min. outlet temperatures for the period 1968-1974.

A second example is the Widows Creek installation on the Tennessee River, with a net continuous output of 1762 MWe, and a cooling water flow of 1.15×10^6 gpm (5.76×10^8 lb/hr). A typical utilization factor based on net continuous output of such a plant is 55 percent. Since the Widows Creek plant has a total of eight units, the probability that all are not operating is low. In fact, the minimum output normally encountered is 900 MWe, requiring 5.87×10^5 gpm (2.94×10^8 lb/hr) of cooling water.

3. Factors in Utility Implementation

Although base-load utility residual heat offers a very attractive resource, significant implementation factors beyond the development of the technology must be faced. These factors, although not insurmountable, need to be considered at the start of any specific site application. The following list is representative but not intended to be all-encompassing (Ref. II-19).

TABLE II-8

MONTHLY AVERAGE INLET TEMPERATURES, ΔT AND OUTLET TEMPERATURES

1968-1974

DUKE POWER PLANT ALLEN

	<u>Avg. Inlet Temp. °F</u>	<u>Plant ΔT</u>	<u>Avg. Outlet (Min.) Temp. °F (Max.)</u>
January	46.4	21.2	(65.5) 67.6 (69.6)
February	46.1	21.5	(61.0) 67.6 (70.3)
March	52.0	19.8	(69.5) 71.8 (73.7)
April	62.4	17.1	(75.3) 79.5 (80.3)
May	69.4	16.7	(83.4) 85.1 (89.4)
June	77.0	15.4	(77.3) 92.4 (94.8)
July	82.2	15.8	(95.2) 98.0 (101.2)
August	83.3	16.4	(97.8) 99.4 (101.6)
September	79.2	16.0	(91.8) 95.2 (100.0)
October	70.4	16.1	(84.3) 86.5 (89.6)
November	59.2	18.5	(75.1) 77.7 (79.7)
December	50.0	18.4	(64.8) 68.4 (71.2)

- Legal constraints - The Federal Water Pollution Control Act and state public utility commissions are just two examples of the legal constraints which must be considered in an application.
- Utility operating requirements - Several of the obvious constraints are:
 - (1) Since power is the prime concern of the utility, any application of residual heat must not interfere with power levels.
 - (2) The amount of residual heat is not a constant.
 - (3) Retrofitting to existing utilities is costly.
 - (4) Many existing plants do not have available land due to close proximity to other industry and residential communities.
 - (5) The use of large quantities of process steam produced by heat pumps requires locating generating plants near an industrial park, which in turn would result in additional institutional problems.
- Biological considerations - particularly in those areas where the water is used directly without using intermediate heat exchangers.
- Economic feasibility

4. Industrial Plants

Industrial residual heat can be grouped into four basic types: recycled cooling water, contaminated liquid, stack exhaust and product heat. Only the first three types will be considered; the fourth represents an energy loss that cannot be easily recovered and is too site specific for a general study. Under normal industrial operating conditions, cooling water tends to be less than 120°F. Contaminated liquid occurs, however, over a temperature range from ambient to 200°F. Stack exhaust has an even wider temperature range since gasses can be exhausted to the atmosphere as low as 300°F without incurring acid condensation and in excess of 2000°F in the case of some metal processing furnaces. For this study, an exhaust temperature lower limit of 300°F is used to define the amount of recoverable stack heat.

Recovery of industrial residual heat is only feasible if a need (end-use) is present for the recovered energy. For the power generating application, the end-use is assumed to be power consumed on-site. For a heat pump application, the end-use is usually hot water or low pressure steam that would otherwise be generated in a fossil-fueled boiler. The industrial plant residual heat source and end-use data used in this study were obtained from Refs. II-5 and II-6. Industries which do not have an on-site end-use were not considered. The ranges of plant sizes were obtained from Ref. II-5 or estimated from 1972 Census of Manufacturers data. Both residual heat sources and end-uses were assumed to be collectable and distributable, respectively, at a single site in the plant; whereas, in practice multiple residual heat streams and steam boilers are common in plants. This will have to be addressed before a residual heat recovery system could be implemented.

a. Heat Pump Applications

Residual heat in the form of cooling water or low-temperature contaminated process water ($\sim 200^{\circ}\text{F}$) is quite common from industrial plants. Table II-9 summarizes those industries identified in the survey as having a process steam requirement and residual heat in the form of water in any of three temperature categories: 95° to 120°F , 130° to 160°F , and 170° to 200°F . This list is not purported to be complete since the data sources used (Refs. II-5 and II-6) give only typical plant energy flows and are limited to certain industries, albeit large energy consuming ones. However, the wide variety of industries listed indicate that residual heat at potentially recoverable temperatures is common in many industrial plants.

The amount of residual heat associated with a given type of single product plant is, in general, proportional to the output; in most industries plant outputs vary over a considerable range. For example, there are 59 plants which produce phosphoric acid. The smallest has an annual capacity of 10^4 tons while the largest has a capacity of over 10^6 tons. Plant size ranges are even larger in the food processing industry, where many plants operate with 10 or fewer employees, while some large plants have in excess of 2,500 employees. For this study, the sizes for heat pumps will be determined by the minimum process heat requirement for average plants in the majority of the industries listed in Table II-9. The survey indicated that an output of 10^4 lb/hr of $250\text{--}300^{\circ}\text{F}$ saturated steam can satisfy this criteria for nearly all industries which have a supply of 95 to 120°F residual water, the only exception being the plastics industry where small amounts of higher temperature steam are required.

TABLE II-9

INDUSTRIES WITH POTENTIAL RESIDUAL HEAT RECOVERY HEAT PUMP APPLICATIONS

SIC	Industry	Waste Heat Temperature, °F			Process Heat Temp., °F
		95-120	130-160	170-200	
2011,13	Meatpacking	X			140/180/250
2026	Fluid Milk	X			250
2033	Canned Fruits & Vegetables	X	X	X	250
2046	Wet Corn Milling	X			250
2062	Cane Sugar	X	X		185/250
2063	Beet Sugar	X	X		250
2075	Soybean Oil	X			250/350
2082	Malt Beverages	X	X		185/250
2211,21,31,41	Weaving Mills	X	X		250
2261,2	Finishing Mills	X	X		246
2611,21,31,41	Pulp & Paper Mills	X			300
2812	Alkalies & Chlorine			X	250/350
2813	Industrial Gases			X	250
2816	Inorganic Pigments	X			250/350
2819	Industrial Inorganic Chemicals	X	X		250/300
2821	Plastics	X	X		250/400
2822	Synthetic Rubber	X			250
2823	Cellulosic Fibers		X	X	200/250
2824	Noncellulosic Fibers	X	X		250/500
2865	Cyclic Crudes & Intermediates	X			320/400
2869	Industrial Organic Chemicals	X			270/350
2874	Phosphatic Fertilizers	X	X		250
2951	Paving Mixtures & Blocks			X	328
3296	Minerals Ground or Treated			X	320
3312	Steel Mills			X	320

However, there are some industries which have plants which are large enough to require over 10^5 lb/hr of steam (e.g., meatpacking, fluid milk, wet corn milling, pulp and paper mills, alkalies and chlorine, inorganic pigments, noncellulosic fibers, cyclic crudes and intermediates and phosphoric fertilizers). Thus, a unit having an output of this magnitude would find a market and alleviate the use of 10 or more of the smaller units to achieve the desired output. A few of the industries listed on Table II-9 require process heat temperatures below 250°F ; however, this need can be satisfied using commercially available equipment.

Residual heat in the 130° to 160°F temperature range is less common than the lower temperature range, but still occurs in a wide range of industrial plants (Table II-9). Most needs of these industries could be satisfied with single or multiple heat pump units which deliver 30,000 lb/hr of 250° to 300°F saturated steam. Residual heat in the 170° to 200°F temperature range is the least common of the liquid streams; however, where it does exist, the temperature differential between it and useful process heat is so reduced compared to the other residual heat sources that a heat pump application appears quite attractive. A unit which provides 10^5 lb/hr of 250° to 350°F saturated steam could satisfy moderate or large plants with single or multiple units, respectively.

b. Power Generation Applications

In order to use industrial plant residual heat to generate power, a suitable heat rejection medium must be present. While river or pond water is available at the Federal facilities, in general no such heat sink can be assumed at a typical industrial plant. Industrial

residual heat is, however, commonly dissipated using a wet cooling tower which receives water at about 95°F and returns it at about 75°F depending on the local weather conditions. Using this type of heat sink, the only appropriate liquid residual heat source for a power recovery system would be 170° to 200°F water. To produce at least 1 MWe of power, a plant would have to have in excess of 10^6 lb/hr of residual water for use in boiling the working fluid. Industries which possess this quantity of residual heat in some of these plants are indicated in Table II-10. However, the total amount of residual heat in the 170-200°F temperature range that is rejected by these plants is only one quarter of that identified in the entire industrial sector (Ref. 10). Therefore, the number of power recovery applications from liquid residual heat is much greater than indicated by the data on Table II-10.

Stack exhaust represents a much larger source of residual heat. Since it is at a higher temperature, the efficiency of a Rankine cycle power system would be somewhat higher for a stack exhaust system than for one using a 170-200°F water heat source. Two levels of stack exhaust temperature are used in this study to estimate the potential applications for such a system: 300 to 600°F, and >600°F. A 300°F lower limit in ultimate exhaust temperature is used in all cases to eliminate acid condensation. Table II-10 shows those industries which have been identified as having substantial amounts of residual heat available (sufficient to provide 2 MWe of power).

TABLE II-10

INDUSTRIES WITH POTENTIAL WASTE HEAT RECOVERY POWER GENERATION APPLICATIONS

SIC	Industry	Waste Streams		
		170-200°F Water	300-600°F Stack	600°F Stack
2033	Canned Fruits & Vegetables	X	X	
2046	Wet Corn Milling		X	
2062	Cane Sugar		X	
2075	Soybean Oil		X	
2082	Malt Beverages		X	X
2611,21,31,41	Pulp & Paper Mills		X	
2812	Alkalies & Chlorine	X	X	
2813	Industrial Gases	X		
2816	Inorganic Pigments			X
2819	Industrial Inorganic Chemicals		X	X
2822	Synthetic Rubber		X	
2823	Cellulosic Fibers	X	X	
2824	Noncellulosic Fibers		X	
2865	Cyclic Crudes & Intermediates		X	X
2869	Industrial Organic Chemicals		X	
2873	Nitrogenous Fertilizers			X
2874	Phosphatic Fertilizers			X
2911	Petroleum Refining		X	X
2951	Paving Materials & Blocks	X		
3011	Tires & Tubes		X	
3211	Flat Glass			X
3221	Glass Containers			X
3241	Hydraulic Cement			X
3274	Lime			X
3296	Mineral Wood	X		
3312	Steel Mills	X		X
3334	Aluminum			X

D. REFERENCES

- II-1 U.S. Department of Energy Waste Heat Utilization Project for the Paducah, Kentucky, Uranium Enrichment Gaseous Diffusion Plant, Final Report prepared by J. E. Sirrine Co., Vols. I and II, September 1978.
- II-2 Oak Ridge Gaseous Diffusion Plant Reject Heat Study, Final Report prepared for DOE by Exxon Nuclear Company, Inc., Vols. I through IV, September 1978.
- II-3 Use of Reject Heat from Uranium Enrichment Gaseous Diffusion Plant Portsmouth, Ohio, Final Report prepared for Department of Energy by Bechtel National, Inc., Vols. I and II, October 1978.
- II-4 The Savannah River Plant Low Level Waste Heat Utilization Project, Report prepared for DOE by South Carolina Energy Research Institute, August 1978.
- II-5 Cogeneration Technology Alternatives Study, Final Report prepared for NASA/DOE by Power Systems Division of United Technologies Corporation, June 1979.
- II-6 Energy Analysis of 108 Industrial Processes of Industrial Applications Study, prepared by Drexel University for DOE, June 1979.
- II-7 Private communication with Mike Boyd, DOE Operations Office at Oak Ridge, August 1979.
- II-8 Private Communication from Charles A. Keller, DOE Operations Office at Oak Ridge to Managers of Gas Diffusion Plants, November 22, 1977.
- II-9 Private Communication with Claude H. Smith, Chief Data Services Branch, TVA, August 22, 1979.
- II-10 Final Environmental Impact Statement - Portsmouth Gaseous Diffusion Plant Site, Vols, 1 and 2, ERDA-1555, May 1977.
- II-11 Steam-Electric Plant Construction Cost and Annual Production Expenses 1977 - Department of Energy Report No. DOE/EIA-0033/3 (77), December 1978.
- II-12 Private communication with Randy Snipes, TVA, August 1979.
- II-13 Private communication with Jim Morgan, Pacific Power & Light, August 1979.
- II-14 Industrial Applications Study, prepared for DOE by Drexel University, Report No. HCP/T2826-01, March 1978.

- II-15 Paper by R. M. Jameson & G. G. Adkins, "Waste Heat Disposed in Power Plants", presented at 1970 AIChE conference in Houston.
- II-16 Paper by D. W. Anderson & A. Gnilka, "The Biological Impact of a Thermal Discharge Exceeding 95°F - A Case Study of Allen Steam Station, North Carolina", presented at 1977 conference in Miami.
- II-17 Paper by D. E. Miller, et al, "Modifications to Once Through Cooling Water, Discharge Structure to Achieve Entrainment Mixing and Lateral Transport of Thermal Plume", presented at 1977 conference in Miami.
- II-18 Paper by R. C. Smith and H. J. Teas, "Biological Effects of Thermal Effluent from the Cutler Power Plant in Biscayne Bay, Florida", presented by 1977 conference in Miami.
- II-19 Paper by V. E. de Pass, et al, "Social Aspects of Regulating Waste Heat", presented at 1977 conference in Miami.
- II-20 Paper by E. R. Burres, et al, "Waste Heat in a Controlled Environment Greenhouse", presented at 1977 conference in Miami.
- II-21 Paper by P. P. Pailey, et al, "Cooling Water Resources of the Upper Mississippi River for Power Generation", presented at 1976 conference in Miami, Florida.

SECTION III

HEAT PUMP

A. INTRODUCTION

1. Objectives

The objectives of this section were to:

- Determine the potential cost and performance characteristics of heat pumps designed for selected large capacity residual heat applications.
- Evaluate the various cycle, component and working fluid design options.
- Select systems that could be offered commercially based on return-on-investment criteria for the various applications. A conceptual design is presented of a heat pump system using the selected configuration which utilizes 150°F residual heat from the Portsmouth gaseous diffusion plant to generate 300°F saturated process steam for an industrial park.
- Utilize the preferred system to specify the preliminary requirements for EAST facility.

The results of the heat pump analysis are presented in seven parts.

- Heat Pump Cycles and Terminology - Part C describes the relative merits of Rankine, Sterling and Brayton cycles for the identified heat pump applications. The various

types of Rankine cycles are also described and the terminology used in this report is defined.

- Selected Configuration - Part D describes the four types of Rankine cycles and evaluates the working fluids in terms of performance and cost.
- Heat Pump Drivers - Part E describes five types of prime movers for use as heat pump drivers. Each driver is evaluated for its impact upon the heat pump system performance, size and cost characteristics.
- System Selection for Portsmouth GDP - Part F describes a system which provides the most attractive return-on-investment for the Portsmouth DP. The characteristics are described for this complete heat pump system.
- Considerations of Alternate Applications - Part G evaluates the Portsmouth system for five other potential residual heat applications.
- Prototype System Specification - Part H defines the prototype system which will be proposed to test in EAST Facility.
- Test Facility Specification - Part I provides the preliminary facility specification.

The material presented was developed for ERI by R.W. Bass at the United Technologies Research Center (UTRC). Subsequent reports will expand this data to include the inputs of additional manufacturers.

2. APPROACH

Three elements are required to successfully accomplish the application of heat pumps to low-level residual streams: (1) identification must be made of available sources; (2) potential end-uses must be defined; and (3) economically attractive technology must be available. The approach used in this study was to survey the published literature and manufacturers' literature to define the available source characteristics and potential end-uses. Section II of this report provides a discussion of the available sources. Section III-B provides some discussion of application characteristics applied to these sources.

To define the economic viability of heat pump systems, a cascading analysis was undertaken (Figure III-1). These steps included:

- a. A review was done of the thermodynamic heat pump cycles, which resulted in selecting the Rankine cycle as the only practical near-term approach for recovering residual heat.
- b. An examination was done of Rankine cycle heat pump configurations. Single fluid closed cycle, steam semi-open and open-cycle and dual fluid cycles were identified for further study.
- c. An analysis was done of working fluids for flammability, toxicity, cost, critical temperature, thermal stability and minimum cycle pressure. Ten fluids were selected for study in the closed cycle systems.
- d. An examination was made of five prime movers: gas turbines, diesels, steam turbines, electric motors and residual-heat-driven Rankine power systems. Gas turbines

Approach to Heat Pump Analysis

Figure III-1

and diesels were selected for further analysis because their residual heat can be recovered in the form of process heat at the same temperature levels required for industrial applications. Their overall performance is therefore relatively high compared to steam turbine and electric motor drives.

- e. An evaluation was made of return-on-investment calculations using the Portsmouth GDP residual heat source and the thermal energy requirements of a hypothetical nearby industrial park. This evaluation considers various combinations of the selected drivers and heat pump configurations. A system was selected which showed the highest rate-of-return and met other practical cost criteria.
- f. An analysis was made of the five (5) alternate applications (see Table III-1) using the configuration developed for Portsmouth. The dual-fluid, closed cycle thermally driven heat pump system was shown to be applicable to these alternatives.

3. Conclusions

- Large capacity, heat-engine driven, dual-fluid Rankine cycle heat pumps can effectively provide industrial thermal process heat (300°F steam) at attractive return-on-investment rates from low-temperature residual sources located at Federal nuclear facilities and numerous base-load utilities.
- a. Before tax, return-on-investments ranging from 29 to 36 percent are feasible with Rankine cycle heat pumps capable of providing 100 to 400 x 10⁶ Btu/hr of process steam at 300°F using 150°F hot water from

TABLE III-1
CANDIDATE HEAT PUMP APPLICATIONS

Application	Waste Heat Source			End-Use Application		
	Location	Flow Rate 10 ⁶ lb/hr	Temp. °F	Location	Heat Requirements 10 ⁶ Btu/hr	Temp. °F
#1	Fed. Facilities	20 ⁽¹⁾	140-165	Ind. Park	400-1000	300
#2	Utility Cooling Tower	20 ⁽²⁾	105-121	Ind. Park	400-1000	300
#3	Industry	0.2	95-120	In-Plant	10	300
#4	Industry	2.0	95-120	In-Plant	100	300
#5	Industry	0.3	130-160	In-Plant	30	300
#6	Industry	0.5	170-200	In-Plant	100	300

(1) Cooresponds to approximately the smallest single pipe flow at Paducah, Oak Ridge and Portsmouth.

(2) Equivalent to approximately 125 MWe of power production.

the four Federal facilities for on-site applications.

- b. If the heat pumps are located at nearby industrial parks, such as one considered in conjunction with the Portsmouth, Ohio, gaseous diffusion plant, the cost of the mile-and-a-half (1 - 1/2) pipelines in each direction will lower the before-tax return-on-investment by about 6 percentage points for the large (100 to 400 x 10⁶ Btu/hr) capacity units.
- c. Since the residual heat from base-load utilities is generally available at some 40 to 50°F lower temperature than at the Federal facilities, the return-on-investment rates are reduced by about 10 percentage points; therefore any industrial parks or individual industries utilizing 300°F steam must be located on-site. If lower output temperatures, i.e., 250°F, are suitable for the industrial applications, the large capacity heat pumps can provide before-tax rates-of-return above 30 percent.
- Residual heat normally available at 150 to 180°F from individual industrial plants can also be economically converted to 300°F thermal process steam by heat-engine driven heat pumps if the plant requires more than 50 x 10⁶ Btu/hr. As the available residual heat temperature decreases below 150°F, the output capacity of the heat pump must increase to 100 x 10⁶ Btu/hr to maintain a 20 percent or higher before-tax return-on-investment.
- Large gas turbines and diesel engines (> 5000 HP) are the most effective heat engines for these applications, for they are highly efficient and their residual heat can be efficiently used to supplement the heat pump output to

provide industrial thermal process needs at 300°F and above. Smaller gas turbines and diesels are less attractive because they are less efficient and have higher specific costs. Electric motors are less acceptable because they do not permit recovery of the power generating plant residual heat.

- All the components selected in the preferred heat pump configuration are classified as state-of-the-art. For this reason, the development program envisioned for this particular equipment for the EAST Facility is system, as opposed to component, oriented. Future considerations and other manufacturers may include component oriented programs.
- By using a modular approach ($\sim 100 \times 10^6$ Btu/hr/Module) to meet the large residual heat flows, the EAST Facility will be sized to test actual full scale modules which could then be combined with other modules and be installed at a residual heat source.

B. APPLICATION CHARACTERISTICS

Use of a heat pump requires both a source of low-temperature (residual) heat and an end-use for the higher-temperature output heat. A discussion of potential sources is contained in Section II of this report. This discussion groups the sources into (3) categories; Federal Facilities, Utilities and Industrial.

The identified sources are predominantly cooling water, although some process water and non-returned condensate were also found. Each source is defined in terms of its flow rate and temperature. In cases where the residual source originates in an industrial plant, the potential in-plant end-use requirement was identified in terms of heat required (Btu/hr) and its temperature ($^{\circ}\text{F}$). In most of these plants, the end-use was process steam, for it represents the most common heat transfer medium and has a wide range of uses from indirect (i.e., jacketed heaters) to direct (i.e., distillation).

In those cases where the residual heat source is cooling water from Federally-owned gaseous diffusion plants or utilities, there generally is no large co-located process heat need. Therefore, for these cases, all industrial process heat requirements were surveyed to identify steam heat loads and temperatures that are common to a wide variety of industries suitable for an industrial park located near the residual heat source.

The resultant data was used to define six different potential applications, summarized in Table III-1. Two of these applications are for the very large sources of residual heat from Federal facilities and large central power stations that could be used to provide process heat to a nearby industrial park. The remaining four applications are for residual heat sources and process heat requirements that exist within a given industrial

plant. The following paragraphs describe each of the six applications.

1. Industrial Park End-Uses

A unique, large relatively high-temperature source of cooling water exists at the three Federal facilities (i.e., the gaseous diffusion uranium enrichment plants at Paducah, Kentucky; Oak Ridge, Tennessee; and Portsmouth, Ohio). At these locations, sources of cooling water at minimum flow rates of 20×10^6 lb/hr (40000 gpm) are available at temperatures ranging from 140 to 165°F. The heat in this water is presently being dissipated by means of forced-air wet cooling towers. The gaseous diffusion plant cooling water is in a closed loop and is returned to the plant at about 95°F.

There are no large process steam requirements located near the large residual heat sources at the Federal facilities. Therefore, the end-use requirements were obtained from an assessment of general industrial plants needs (Refs. III-1 to III-4). These studies identified a wide range of industrial plants that are compatible with the raw materials and markets in the vicinity of each Federal facility. The process temperature needs of the identified industries are generally distributed in a manner similar to that for the entire industrial sector as shown in Fig. III-2. This plot shows the cumulative percentage of required process heat below 400°F as a function of process heat temperature levels (approximately 95% of total industrial process heat requirements are below 400°F). The temperatures required in various typical industries are also in the figure. The data shows that 50 to 75% of the process heat is in the 275°-300°F range. Therefore, a large number of industries could potentially locate in an industrial park if a heat pump could provide 300°F saturated steam economically using the 150°F residual heat as input.

FIGURE III-2

INDUSTRIAL PROCESS HEAT REQUIREMENTS BELOW 400°F

APPROXIMATELY 95 PERCENT OF TOTAL INDUSTRIAL STEAM NEED IS AT 400°F OR LESS

Industrial plants which produce the same product often vary in size by as much as 100 times. The data presented in Fig. III-3 shows the cumulative percentage of total process steam need as a function of individual plant requirement. Also shown along the abscissa of Fig. III-3 are typical steam demands (10^6 Btu/hr) of various individual large, modern plants in several industries. For example, it is not uncommon for a single food processing plant to require in excess of 100×10^6 Btu/hr. The amount of process heat required at an industrial park depends on the type and size of the industries that may be attracted to the site. The size of the heat pumps will be dependent on the economics of the system and will be a major part of the discussion in this section.

The utility application (#2) shown in Table III-1 is similar in flow rate characteristics to the Federal facility application but identifies a much more common large source of residual heat. Utility base-load nuclear or coal-fired steam power generation plants often reject condenser heat by means of cooling towers due to the absence of river water or some environmental restrictions. Generally, the water leaves the power system condenser at temperatures ranging from 105° to 121°F depending on system design, ambient temperature and humidity. The cooling water flow rate is directly proportional to the power generated and inversely proportional to the temperature drop achieved in the cooling tower. With typical cooling tower temperature drops of 35° - 40°F , a flow rate of 20×10^6 lb/hr is associated with a generating capacity of approximately 125 MWe.

As with the Federal facilities, there is no need at most utility sites for process heat. Following the same argument as that used for the Federal facility application, the end-use for recovered heat at this type of utility power plant is to provide useful heat in the form of 300°F steam to an industrial park located nearby. In such utility applications where land is not as readily available as the Federal facilities, the factors

FIGURE III-3

TYPICAL INDIVIDUAL INDUSTRIAL PLANT PROCESS HEAT REQUIREMENTS

mentioned in section II become as critical as the technology itself.

2. In-Plant End Uses

The survey identified low-temperature residual heat in industrial plants in the form of cooling water (95-120°F), process water (130-160°F) and nonreturned condensate (170-200°F). Each of these residual heat sources can be found in a large number of industries. Within a specific industry quantities will exhibit wide variations dictated by the plant size as well as the manufacturing processes. In order to define a finite number of specific design requirements, it is necessary to select sizes from the spectrum of possible sizes that will have wide applicability. In this manner, very small plants might use one heat pump at part load, a larger plant one at full load, and a still larger plant multiple units.

Cooling water is an ubiquitous form of low temperature industrial residual heat. It occurs in such a wide variation of quantities that selecting only one size in order to assess its economic viability would not be prudent. Therefore, two sizes (in terms of flow rate), an order of magnitude apart, were selected to typify potential equipment. The smaller size (application #3) would use 2×10^5 lb/hr of water and a larger size (application #4) would use 2×10^6 lb/hr of water.

For application #3, a heat pump would have to produce approximately 10×10^6 Btu/hr of steam to satisfy the needs of a typical plant of this size. Moreover, the steam would have to be 300°F (see Fig. III-2) to have wide applicability. Application #4 is typical of a plant 10 times as large as application #3 and is commonly encountered in industries such as food processing. This is by no means the upper limit of plant sizes but is a convenient design condition to divide the plants with 95°-120°F residual heat sources. This reduces the number of heat pumps

sized for application #3. Some large industrial plants (notably in the paper and pulp, chemical and petroleum industries) may require as much as 1000×10^6 Btu/hr of process steam. This requirement could be met by using several heat pumps sized for application #4 in parallel, each unit designed in the range of 100 to 500×10^6 Btu/hr.

Residual heat at 130° - 160° F from process streams is not as common as the lower temperature cooling water. However, many plants in the food and chemical industries reject heat at that temperature level. A typical plant has 3×10^5 lb/hr (600 gpm) of 130° - 160° F residual heat available and requires 30×10^6 Btu/hr of 300° F process steam (application #5). Multiple units each with this output capability could be used to satisfy the needs of larger plants, although justification for two sizes (as in the cooling water applications) may be found if a more detailed market study were made.

Nonreturned condensate (application #6) is well suited for a heat pump source since its temperature is normally high (170° - 200° F). Nonreturned condensate, however, is not as common as some of the other sources since only a few industrial plants, typically those which wash products or cook, reject heat at this temperature. A typical plant has 5×10^5 lb/hr (1000 gpm) of residual water and requires 100×10^6 Btu/hr of 300° F steam.

C. HEAT PUMP CYCLES AND TERMINOLOGY

A heat pump uses power to raise the temperature of heat from the source level to a level at which it can be utilized. This section presents a brief discussion of the selection of residual heat recovery heat pumps which operate on the Rankine cycle. Following this, a detailed discussion is provided on the Rankine cycle options considered and terminology used in the remainder of the report.

I. Selection of Rankine Cycle

Although other thermodynamic cycles (i.e., Stirling and Brayton) have been considered and sometimes used in heat pumps, essentially all of the commercially available industrial heat pumps are based on the use of the Rankine cycle. This cycle has a number of performance and cost advantages relative to the other cycles and therefore was selected as the basis for all heat pump systems in this study. These advantages include:

- a high COP relative to the Carnot limit due to the large component of constant temperature heat input and output.
- use of a working fluid that has a large heat capacity in the phase change process, permitting reduced cycle flow rate.
- a large and familiar line of commercially available equipment and technology.
- use of a working fluid that changes phases during the heat transfer process, thereby having high heat transfer coefficients and requiring only small heat exchangers.

A general discussion of the three thermodynamic cycles (Stirling, Brayton and Rankine) is presented in Reference III-1.

2. Typical Rankine Thermodynamic Cycles

Figure III-4 shows the working fluid pressure-enthalpy diagram and a component schematic for a typical Rankine cycle heat pump in a residual heat application. Residual heat is used to vaporize the working fluid at low pressure to state 1. Saturated vapor at low temperature (and pressure) is then compressed from state 1 to a higher pressure at state 2. The actual work required is higher than the isentropic work (state 2') due to inefficiency in the compression process. The high pressure vapor leaving the compressor is cooled and condensed to a saturated liquid (state 3), releasing its heat at a temperature slightly above that required in the industrial process. The liquid is then throttled to low pressure (state 4) and a portion of the remaining liquid is vaporized by absorbing heat from a low-temperature residual heat source to return the fluid to state 1.

The performance of the heat pump cycle can be improved by multistaging the working fluid compression and expansion process as shown in Fig. III-5. In this multi-pressure cycle, the compressed fluid at state 2 is desuperheated by adding saturated liquid at the same pressure. Vapor generated in the throttling process occurring between states 5 and 6 is then added to the stream at state 3. The desuperheating process reduces the amount of work required to compress the working fluid. The mixing of vapor reduces the amount of mass flow requiring compression from state 1 to state 2 but does not affect the evaporator heat transfer process. In systems where working fluid compression requires two or more compression stages, desuperheating and vapor transfer between each stage will provide a performance advantage as well. This design feature is widely used in commercial centrifugal water chillers.

RANKINE CYCLE HEAT PUMP

a. PRESSURE—ENTHALPY DIAGRAM

b. SYSTEM SCHEMATIC

Figure III-4

a PRESSURE—ENTHALPY DIAGRAM

3. Terminology and Definitions

- Coefficient of performance: As indicated above, the function of a heat pump is to utilize some form of work input to take heat from a lower temperature (source) and raise it to some level at which it can be utilized. The heat pump performance can be measured by the coefficient of performance (COP) defined as

$$\text{COP} = \frac{\text{HEAT OUT}}{\text{WORK IN}} = \frac{Q_{\text{out}} (\text{BTU/lb})}{W_{\text{in}} (\text{BTU/lb})} \quad (3-1)$$

Referring to Figure III-4, (3-1) can be expressed as

$$\text{COP} = \frac{h_2 - h_3}{h_2 - h_1} \quad (3-2)$$

where h_1 = enthalpy entering the compressor, h_2 = enthalpy leaving the compressor and h_3 = enthalpy leaving the condenser. This performance is limited by the 2nd law of thermodynamics and the temperatures of the residual heat source and the output heat. These limitations will be discussed further in this report as they effect equipment and working fluid selection.

This measure of performance can, however, be misleading when comparing two or more methods of providing the compressor drive work. A more accurate parameter for defining heat pump performance is the Coefficient of Performance based on primary fuel input (COPF). This COP is defined as the ratio of the heat output from the heat pump to the heat content of the fuel used (by whatever means) to provide the compressor work. For example, a heat pump with a COP of 3.0 would have a COPF of 1.0 if

the compressor were electrically driven and the utility generation transmission process was 33 percent efficient. However, a COPF of 1.2 would be possible if the compressor was driven on-site by a high-performance, low-speed diesel which generates electrical or mechanical power at 40 percent efficiency.

- Application lift: The temperature difference between the process heat requirement (saturation temperature in the case of steam) and the inlet residual heat temperature.
- Internal lift: The temperature difference between the saturation temperature of the working fluid in the condenser and the saturation temperature at the inlet to the first stage of compression.
- Pinch Temperature difference: The minimum temperature difference occurring in each heat exchanger to facilitate heat transfer. The amount that internal lift exceeds application lift is due to the pinch temperature difference of each heat exchanger and the temperature drop of a sensible residual heat source. For this preliminary study, the commonly used values of 10°F for heat exchanger pinch temperature differences and 10°F for residual heat temperature drop have been adopted. Detailed design not covered by this study will require an economic tradeoff to determine exact values.

For the heat pump needed to satisfy the demands of the Federal facilities, the application lift is 150°F (based on residual heat input stream temperature of 150°F and steam output requirement of 300°F). However, using a heat pump where the working fluid is neither the residual stream or the output stream (i.e., a closed-cycle requiring heat exchangers), the internal lift can be 180°F. The additional 30°F comes from 10°F in the condenser and 20°F in the evaporator (10°F due to residual heat

stream temperature loss and 10°F to facilitate heat transfer between it and the working fluid).

D. CONFIGURATION

Selection of the most economically attractive Rankine cycle heat pump for a residual heat recovery application requires performance and cost comparisons of the various configuration alternatives. These configurations can be categorized as follows:

- The most common heat pump configuration is a closed cycle. The working fluid is confined to a closed loop and only interfaces with the residual heat stream and with the output stream across heat transfer surfaces. A wide variety of fluids can be employed in a closed cycle configuration.
- For the applications being considered in this study, the output stream is steam. Thus steam can be used as the working fluid without benefit of a condenser. This is defined as a semi-open cycle.
- Again for the applications being considered, the residual stream is water that can be flashed to produced steam. This allows application of the open-cycle, where steam is the working fluid as well as the output fluid.
- A number of single-fluid heat pumps with different working fluids can be placed in series to pump the heat from the residual heat source to the desired output temperature. Multi-fluid heat pump configurations of this type can be used so the selected working fluids are matched to their optimum temperature range to achieve high overall performance and low equipment cost.

In the following paragraphs, the various working fluids are examined and the most attractive in terms of performance and equipment cost are identified for evaluation in other parts of

this report. First a brief discussion of the criteria used to select candidate working fluids is presented.

1. Working Fluid Selection

Closed Cycles

The most appropriate fluids are those that come closest to meeting the following several criteria. Each factor affects system performance and cost. Without knowledge of fluid characteristics, performance and component size calculations are impossible.

(a) A latent phase change at the heat input temperature

Although it is possible to conceive of rejecting the heat at supercritical conditions, it is inefficient to pump heat to super critical conditions (i.e., heat transferred per pound of working fluid), especially if the required end-use is isothermal (i.e., generation of saturated steam). A heat pump which provides 300°F output condenser temperature requires a 310°F internal condenser temperature. At least a 25° to 50°F difference between the output temperature and the critical temperature (i.e., 335° to 360°F) would be desirable to ensure adequate latent heat for constant temperature heat transfer to the output stream.

(b) The heat rejection temperature below the fluid critical temperature

(c) Adherence to safety regulations

Fluid safety factors can influence both the system design and the number of permissible application sites. While use of a highly flammable and toxic substance

might be tolerated in a chemical plant where adequate safety precautions should be available, other industries might refuse to install a heat pump using such a fluid. OSHA has instituted maximum exposure limits for various commonly encountered chemical fluids. The term used to define the acceptable exposure limit is TLV (Threshold Limit Value) and is specified in parts per million (ppm). The sophistication in design and the cost of all seals and connections in the heat pump will be a function of the TLV to ensure that the established value is not exceeded.

(d) Lowest reasonable cost

Costs of working fluids vary widely. If a substance is a commonly used chemical like ammonia, it can be purchased cheaply from a variety of sources. If a substance is not normally used as an industrial feedstock or a working fluid in a commercial product, it can be quite expensive. For example, hexafluorobenzene working fluid, which has several desirable properties, is available only in small quantities and is quite expensive. The amount of fluid used in a heat pump is the quantity needed to ensure flooded tube boiling during heat pump operation and is, therefore, set by the heat exchanger volume. For typical commercially available chillers which use flooded-tube boilers, fluid cost rarely exceeds 10 percent of the equipment cost.

(e) Thermal stability

Thermal decomposition of the working fluid must be avoided to limit the accumulation of noncondensable gases, the depletion of the working fluid and the generation of corrosive and toxic elements such as chlorine from common refrigerants. Therefore, the

stability temperature limit of the fluid must exceed the highest operating temperature.

(f) Operating pressure above ambient

Noncondensable gas contamination of a working fluid is more likely to occur if the operating pressure in the heat pump is below ambient. The contamination occurs as a result of air and moisture leaking into the system. Since the lowest pressure occurs in the evaporator, air leakage can be eliminated by requiring that the working fluid saturated pressure at the evaporator temperature exceeds one atmosphere. For the Federal facility application, this would occur at 130°F.

Thermodynamic physical property data (P,T,h,v,s) based on experimental measurements are available in tabular form in great detail for steam and for most commonly used refrigerants. Although equations of state have been devised which characterize large categories of fluids, in many cases these characteristics have not been experimentally verified. Additional data needed to size components (i.e., viscosity and thermal conductivity) are often poorly defined or totally lacking in the literature. In order to insure that calculated performance and component sizes accurately reflect the selected fluid, we decided to limit consideration to fluids for which published data, based on actual measurements, are available and not to rely on equations of state approximations.

The following paragraphs discuss some of the properties of the fluids evaluated in this study. In addition, reasons for eliminating some commonly used fluids are discussed.

Steam (H₂O)

Steam is a commonly used Rankine power cycle working fluid. It is nonflammable, nontoxic, and although requiring chemical treatment, inexpensive. Water has good temperature properties for a heat pump application. Its drawbacks include low vapor pressure (less than 1 atmosphere for the condenser temperatures required for all of the identified applications) and relatively high isentropic work. Although these factors will lead to large compressors and higher speed operation, this is not sufficient to eliminate steam from further consideration.

R113 (C₂Cl₃F₃)

R113 (trichlorotrifluoroethane) is the only commonly used refrigerant that has a critical temperature (417.4°F) above the 350°F desired for the identified applications and is chemically stable (at 300° to 350°F) under the proper conditions. It is nonflammable and nontoxic. It does, however, have subatmospheric vapor pressure at temperatures below 117°F and could ingest noncondensable gas for the utility application (#2), the industrial cooling tower applications (#3 and #4), and during shutdown periods. This is, however, not sufficient to eliminate it from consideration.

Methanol (CH₃OH)

Methanol is an inexpensive fluid that has the desired temperature properties for use in a 300°F output heat pump. It was given additional consideration even though it is flammable, has a subatmospheric vapor pressure below 150°F, and has an OSHA TLV limit of 200 ppm (compared to a R113 value of 1000 ppm).

Propanol (C_3H_7OH)

Propanol has the same advantages and disadvantages as methanol. Due to its increased molecular weight (60 compared to 32 for methanol) it could be used in turbomachinery to achieve smaller units and was evaluated in detail.

Toluene (C_7H_8)

Toluene is flammable and toxic, having the same OSHA TLV value as methanol (i.e., 200 ppm). It is chemically stable to temperatures in excess of $600^{\circ}F$ and has a high critical temperature ($609^{\circ}F$). Even though it has a subatmospheric vapor pressure below $230^{\circ}F$, its use as a working fluid for Rankine power cycles justified further consideration.

Hexafluorobenzene (C_6F_6)

Hexafluorobenzene is a nontoxic, nonflammable, highly stable working fluid that has been suggested for a number of Rankine cycle applications. It was considered even though it has a subatmospheric vapor pressure below $180^{\circ}F$ and costs about \$100 per pound.

Fluorinol-85 ($C_2H_2F_3OH + H_2O$)

Fluorinol-85 is a mixture of trifluoroethanol and water and is considered nontoxic and nonflammable. Even though it is costly (\$7.90 per pound) and has a subatmospheric vapor pressure below $170^{\circ}F$, it was evaluated.

Rejected Fluids

A number of fluids for which experimentally verified thermodynamic properties are available were eliminated for various reasons. Common fluorocarbon refrigerants with the

exception of R113 have critical temperatures that are too low or chemical stability limits below the maximum temperature of the heat pump cycle required by the identified application. Ammonia has a critical temperature of 271.4°F and therefore would have to operate supercritically. Benzene and pyridine have OSHA TLV limits of 10 and 5, respectively, which are so low that more expensive nonconventional compressor shafts and seals would be required. Furthermore, they have properties very similar to toluene, which has a higher OSHA limit.

Semi-Open and Open-Cycles

As previously discussed, when steam is the desired output, it can also be used as the heat pump working fluid. This eliminates the need for a condenser and the associated temperature differential for heat transfer. Since steam can be generated by flashing the liquid (water) residual streams in each of the identified applications, a fully open-cycle can be used where no heat exchangers are needed. Thus, open and semi-open configurations have useful features and were evaluated in addition to the closed-cycle heat pump for performance and cost comparisons.

Dual-Fluid Cycles

Design of a heat pump to recover 150°F residual and deliver 300°F useful heat requires a working fluid that possesses not only a high critical temperature but a low atmospheric boiling temperature and low vapor specific volume. High critical temperature is needed for good performance; the remaining characteristics for prevention of noncondensable gas contamination and to permit use of small components. Since these requirements are somewhat contradictory, the best that can be expected with a single fluid is a compromise. However, the use of a two-fluid heat pump cycle permits selection of one fluid to use between the source and some intermediate temperature and a

second fluid for heat pumping up to higher output temperatures. The first fluid must have a low atmospheric boiling temperature and a low vapor specific volume. The second fluid must have a high critical temperature. However, this two-fluid configuration does, in general, require one more heat exchanger, which penalizes the performance by the temperature difference across this heat exchanger. This penalty can be eliminated by using steam as the high temperature fluid (see Fig. III-6). This is a logical use of steam since it eliminates the subatmospheric evaporator, an objection to steam in a single-fluid heat pump.

Three dual-fluid cycles were considered in this study, each using steam as the high temperature fluid and R11 (CCl_3F), R133a ($\text{C}_2\text{H}_2\text{ClF}_3$) and R216 ($\text{C}_3\text{Cl}_2\text{F}_6$), respectively, as the low temperature fluids. Each of these fluorocarbon refrigerants is nontoxic, nonflammable and stable to temperatures of 275° to 325°F or higher under selected conditions.

2. Working Fluid Comparison

Although final selection of the most appropriate working fluid for any of the defined applications cannot be made until a complete system (i.e., including compressor driver) is defined, the candidate working fluids can be compared in terms of performance and cost, and some eliminated if no advantages are found.

Performance

In order to assess the relative merits of the candidate working fluids selected for the various cycles, system performance was calculated for the Federal facility application (i.e., a 150°F residual water stream and a 300°F saturated steam product end-use). A 10°F drop in residual heat temperature (equivalent to 200×10^6 Btu/hr) and 10°F heat exchanger pinch temperature differences were assumed for the closed-cycle case,

FIGURE III-6
DUAL-FLUID RANKINE CYCLE HEAT PUMP
 TEMPERATURES IN °F
 PRESSURES IN PSIA

yielding evaporator and condenser temperatures of 130°F and 310°F, respectively. Cycle calculations were made assuming an 80 percent compressor efficiency, an isenthalpic expansion and no internal pressure losses. The number of required compressor stages was based on a maximum tip Mach number of 1.0. Furthermore, the flow out of each compressor stage was desuperheated and added to the portion of expanding fluid that was vaporized at that pressure (see Fig. III-5 for a schematic of this interstage mixing).

The schematic shown in Fig. III-6 illustrates how inter-stage desuperheating and vapor transfer is designed into the two-stage R11 heat pump. The R11 evaporates at 130°F and is compressed to a pressure (220 psia) so that it condenses at 268°F. Return water condensate at 300°F saturated pressure is expanded to a pressure corresponding to a saturation temperature of 258°F (i.e., a 10°F pinch temperature difference with the condensing R11) and the liquid component is vaporized by the heat released in the R11 condensation process. The saturated steam is subsequently compressed to 67 psia (saturated pressure at 300°F) and made available for process use for the industrial park near the Federal facility.

The COP, the required number of compressor stages, and the fluid inventory estimates of compressor stages, are given in Table III-2 for all heat pump configurations. The fluid inventory estimates are based on the use of a flooded, tube-and-shell evaporator.

For the closed-cycle heat pump, the COP for the single fluid varies from a low of 2.80 for R113 to a high of 3.30 for steam. The low value for R113 is due primarily to the low reduced latent heat at the condenser resulting from the low critical temperature. The low COP for hexafluorobenzene is due to the shape of the saturation curve of the fluid which yields a large vapor fraction during expansion. Since this vapor must be

recompressed without the benefit of absorbing any heat, it acts as a detriment to system performance. The remaining fluids in the single-fluid, closed-cycle heat pump have comparable COP's indicating there is no outstanding working fluid from a performance viewpoint and each fluid must be judged on its overall merits (i.e., cost, flammability, toxicity, etc.).

The semi-open cycle configuration has a slightly higher COP. Since steam is both the working fluid and the required process fluid, no condenser is needed and the machine internal lift is 10°F less than in the closed-cycle. However, this assumes that all the condensate returns from the process use and is saturated at 300°F .

The open-cycle configuration involves two additional considerations. First, heat is recovered by flashing the residual heat stream and recovering the vapor component as saturated steam at 140°F (the heat recovered is equivalent to the same 10°F drop in the residual stream as in the closed-cycle configuration). Second, the return condensate, assumed to be saturated liquid at 300°F , is expanded in steps to permit vapor transfer and desuperheating as before but the portion remaining as liquid at 140°F must be discarded since it is not valuable as a residual heat stream. All other assumptions are the same. The performance (Table III-2) shows that the COP of the open-cycle is 4.4 percent higher than the semi-open cycle and 6.7 percent higher than the closed-cycle configuration. However, this concept involves using some of the residual heat stream fluid as the output and if the residual stream is contaminated with noncondensable gases this may be impractical.

The performance for the three dual-fluid heat pumps given on Table III-2 vary almost as much as the single fluid heat pumps. While the R113a/steam and the R216/steam heat pumps have poorer performance than most of the single fluid heat pumps, the

TABLE III-2

COMPARISON OF HEAT PUMP OPTIONS FOR FEDERAL FACILITY APPLICATIONS

Waste Heat Temperature = 150°F

Output = 300°F Steam

Heat Input = 200×10^6 Btu/hr

Configuration & Fluid Single Fluid Closed Systems	COP	Fluid Inventory 10^3 lb	No. of Compression Stages	No. & Arrangement of Compressor Stages	Total No. of Compressors
Steam	3.30	176	4	3-2-1-1	7
R113	2.80	290	3	2-1-1	4
Methanol	3.16	138	4	2-1-1-1	5
Propanol	3.16	138	4	4-2-1-1	8
Toluene	3.06	160	4	8-5-3-1	17
Hexafluorobenzene	2.88	292	3	4-3-2	9
Fluorinol-85	3.06	249	4	2-2-1-1	6
<u>Semi-Open System</u>					
Steam	3.37	174	4	3-2-1-1	7
<u>Open System</u>					
Steam	3.52	171	4	3-2-1-1	7
<u>Dual Fluid Systems</u>					
R11/Steam	3.14	171**	3*	1-1-1	3
R133a/Steam	2.93	214**	3*	1-1-1	3
R216/Steam	2.80	258**	3*	1-1-1	3

*2 organic and 1 steam stage

** organic only

R11/steam working fluid configuration has comparable performance with all but the pure steam single-fluid configurations.

Component Sizing

Since the Rankine cycle requires evaporation of the working fluid, the vapor phase thermodynamic properties have a profound effect on the size of the compressors and heat exchangers. This is particularly true when operation is at a low temperature relative to the critical temperature. For example, the specific volume for saturated steam increases by almost 28 percent as the temperature is reduced from 140°F to 130°F. For large heat pumps with a high specific volume working fluid, the use of a single component (e.g., one compressor) may exceed the economic limits for a commercial application. For the Federal Facility application where 200×10^6 Btu/hr of residual heat is recovered for a 10°F drop in temperature, multiple compressor units and heat exchangers are required. For example, the volumetric flow into the lowest pressure compressor for a four-stage, closed-cycle steam heat pump configuration is 534,000 ft³/min. To do this work with a single compressor would require a unit which is approximately 7 feet in diameter, whereas units approximately 5 ft in diameter are the largest commonly produced. To stay within this 5 ft diameter limit, the flow was divided into several parallel streams, each with its own compressor for the first compression stage. The resulting total number of compressors and the unit arrangement are shown on Table III-2 for each fluid. Thus, for the closed cycle steam heat pump, three units are required for the first stage. The total exit steam is then desuperheated and increased by the vapor from the intermediate expansion process and then divided in half for the second stage of the compression to keep this stage within the diameter limitation. Desuperheating and vapor augmenting the flow after its second stage of compression results in a volumetric flow which can be handled by a single compressor for each of the remaining two stages. Thus, seven compressors would

be required for the closed-cycle steam heat pump for the Federal facility application.

Although the semi-open and open cycle steam configurations have a slightly lower work requirements, and in the open cycle configuration a slightly greater inlet density, than does the closed-cycle configuration, the stage loading is not substantially different and the number of stages and unit arrangement are the same. Methanol also requires four stages of compression. However, the first-stage inlet volumetric flow is lower than in the case of steam so that only two units are required in the first stage and one is sufficient for the remaining stages. A similar situation exists for Fluorinol-85, though two compressor units are required for each of the first two stages. However, propanol and toluene have greater volumetric flows than steam and require more compressor units. Furthermore, R113 and hexafluorobenzene require less enthalpy change to achieve the needed temperature lift and require only three compressor stages. The volumetric flow of R113 is low so that only a total of four compressors are needed, whereas the volumetric flow of hexafluorobenzene is high and as many as nine compressors are needed.

Three compressor stages are required (two organic and one steam) for each of the dual-fluid heat pumps based on a stage Mach number limit of 1.0. Moreover, due to the low specific volumes of the low temperature fluids, only one low pressure compressor unit is required to handle the entire vapor flow. Thus, the use of two fluids for the Federal facility application gives a substantial degree of compressor simplification not found in single-fluid closed-cycle configurations.

Major Component Cost

A preliminary comparison of the various heat pump configurations based on the cost of the major components was

defined. The following subsections discuss the correlations used to define budgetary costs for the evaporator, compressors and condensers and fluid inventory for each configuration. These correlations were used to obtain equipment costs for the heat pump configurations using the various working fluids.

a. Working Fluid Inventory

Costs for the working fluids on a per pound basis are given in Table III-3. The value for steam reflects the cost in preparing demineralized water from tap water. Methanol, propanol and toluene are generally available in any quantity at reasonable cost. R113, R11, R216 and Fluorinol-85 are more expensive (in part due to their fluorine content) but relatively easy to obtain. Hexafluorobenzene is currently restricted to small lot production and quite expensive. The costs for the fluid inventory needed for a heat pump which recovers 200×10^6 Btu/hr are given in Table III-4. This does not include leakage replacement or maintenance purge requirements.

b. Heat Exchangers

The evaporator for each heat pump, except for open-cycle steam configuration, is assumed to be a flooded, tube-and-shell design with low fin tubes. Heat transfer coefficients were estimated for each working fluid and an uncontaminated water residual stream (Table III-5). The required surface areas were then calculated. The costs shown on Table III-4 are calculated using $\$5/\text{ft}^2$ for the heat exchanger. This value is typical of exchanger costs in the 200×10^6 Btu/hr size range. The evaporator for the open-cycle configuration is merely a flash tank.

TABLE III-3

COSTS OF HEAT PUMP WORKING FLUIDS

<u>Fluid</u>	<u>\$ Per Pound</u>
Steam	0.01
R113	1.25
Methanol	0.07
Propanol	0.10
Toluene	0.04
Hexafluorobenzene	100.00
Fluorinol-85	7.90
R11	0.70
R133a	3.00
R216	3.00

TABLE III-4
HEAT PUMP COST COMPARISON
Thousands of Dollars

<u>Configuration & Fluid</u> <u>Single Fluid Close Systems</u>	<u>Fluid</u> <u>Inventory</u>	<u>Evaporator</u>	<u>Compressor</u>	<u>Condenser</u>	<u>Total</u>	<u>\$/10⁶Btu/hr</u>
Steam	2	375	4200	675	5252	18.5
R113	363	625	3000	1300	5288	16.8
Methanol	10	500	2000	940	3450	11.6
Propanol	14	625	7200	1130	8939	30.2
Toluene	7	625	13600	1130	15342	51.0
Hexafluorobenzene	29200	625	6700	1300	37825	122.6
Fluorinol-85	1967	500	4970	1130	8567	28.6
<u>Semi-Open System</u>						
Steam	2	375	4200	--	4577	16.1
<u>Open System</u>						
Steam	2	45	3900	--	3947	14.2
<u>Dual Fluid System</u>						
R11/Steam	164	500	1550	895	3144*	10.8
R133a/Steam	642	530	1150	1000	3357*	10.3
R216/Steam	774	690	2770	1375	5644*	38.0

*Includes 35 for gear

TABLE III-5

HEAT EXCHANGER OVERALL HEAT TRANSFER COEFFICIENTS
 Btu/hr ft² F Based on Tube Outside Total Surface Area

<u>Working Fluid</u>	<u>Evaporator*</u>		<u>Condenser**</u>	
	Typical Range	Estimated Value	Typical Range	Estimated Value
Steam	100-250	190	150-300	210
R11	80-170	135	100-200	160
R113	70-150	115	100-150	120
R133a	80-170	130	100-200	150
R216	60-140	110	100-150	120
Methanol	90-180	140	100-200	150
Propanol	70-150	115	100-200	130
Toluene	70-150	115	100-150	120
Hexafluorobenzene	70-150	115	100-150	120
Fluorinol-85	90-180	140	100-160	130

* Clean waste water on hot side

** Clean boiling water on cold side

c. Compressors

The cost for the compressors shown on Table III-4 reflect the sum of the costs for the units identified on Table III-2. The cost for each stage was obtained from the correlation:

$$\text{COST} = 1550 \times \text{DIAMETER}^{1.54}$$

Cost is in dollars and diameter is in inches. The correlation was derived from manufacturer's prices for several individual single-stage general purpose compressors and is given as a continuous function. However, not all sizes are readily available on a commercial basis.

Compressor costs are a strong function of the standard to which they are designed. For example, redundancy in the lube oil system required in API approved designs can add as much as 50 percent to the cost of a compressor. At this stage of design, it is not known whether API design philosophy is required so the cost correlation chosen reflects typical non-API industrial applications. It can be seen that the dominant factor in the compressor costs shown in Table III-4 is the number of units required, making such low density fluids as toluene very unattractive.

d. Gear

Compressors with a common fluid can be mechanically linked to run at the same speed without much loss in efficiency over a wide range of sizes by adjusting the hub/tip ratio of successive stages. However, when compressors which use two or more fluids are joined, this is not the case. Thus, the cost data for the dual fluid heat pumps on Table III-4 contains the cost of a gearbox needed to provide a single-shaft drive.

e. Total Cost

The costs of the major components for a heat input of 200×10^6 Btu/hr are totaled on Table III-4 and are also expressed in terms of cost per 10^6 Btu/hr of output heat at 300°F .

f. Comparison of Configurations

An examination of the performance and cost factors for the heat pumps was used to eliminate some of the working fluids from further consideration.

The data for the heat pumps with the twelve previously defined fluids or fluid pairs are tabulated on Table III-6 in terms of COP and major component cost per unit output. In addition, fuel savings associated with each heat pump configuration per 10^6 Btu/hr of output compared to an 85 percent efficient boiler and the fuel value at $\$5/10^6$ Btu/hr are given. Finally, the major equipment cost is divided by the value of fuel saved to obtain a cost multiplier. This term is proportional to the equipment cost required to save a dollar of fuel cost. It is clear from this data that those single-fluid heat pumps using hexafluorobenzene, toluene, R113, Fluorinol-85 and propanol and the fluid heat pumps using R113a/steam and 216/steam are not desirable since they require more cost to save a dollar of fuel. These configurations can be eliminated from further study. Furthermore, the closed-cycle steam configuration offers no advantage over the semi-open and open-cycle configurations so it too can be eliminated. None of the remaining heat pump configurations (i.e., two-fluid R11/steam and single-fluid methanol, semi-open steam and open-steam) can be dismissed without a detailed economic evaluation of cost and performance made on an installed unit which includes the driver.

In Figure III-7, the performance in terms of COP is given for a range of output temperatures for selected residual heat

TABLE III-6

HEAT PUMP COST PERFORMANCE COMPARISON

Waste Stream Inlet Temperature = 150°F
 Output = 67 psia Steam (300°F saturated)

<u>Configuration</u>	<u>COP</u>	<u>Cost</u> <u>\$/10⁶ Btu/hr</u>	<u>Fuel Savings</u> <u>Btu/hr per</u> <u>10⁶ Btu/hr</u> <u>Capacity (*)</u>	<u>Cost Savings</u> <u>\$ per</u> <u>10⁶ Btu/hr</u>	<u>Cost Multiplier</u>
Open-cycle Steam	3.52	14.2	.38	1.90	7.47
R11/Steam	3.14	10.8	.23	1.16	9.33
Methanol	3.16	11.6	.24	1.20	9.70
Semi-open Steam	3.37	16.1	.32	1.61	10.0
Closed-cycle Steam	3.30	18.5	.29	1.47	12.6
R133a/Steam	2.93	10.3	.15	0.75	13.8
Propanol	3.16	30.2	.24	1.20	25.2
Fluorinol-85	3.06	28.6	.20	1.0	28.6
R113	2.8	10.8	.10	0.49	34.3
Toluene	3.06	51.0	.24	1.20	42.5
R216/H ₂ O	2.80	38.0	.10	0.49	77.6
Hexafluorobenzene	2.88	122.6	.13	0.65	189.5

(*) Compared to 85% efficient boiler

FIGURE III-7

HEAT PUMP PERFORMANCE

WASTE HEAT STREAM TEMPERATURE DROP = 10°F

METHANOL

R11/STEAM

SEMI-OPEN STEAM

source temperatures of 100°, 150°, and 200°F for the methanol, the R11/steam, and the semi-open steam configurations. The output temperature used is the temperature at which the output fluid will condense. When steam is a working fluid in these configurations, the output is always at a higher temperature due to superheat but the saturation pressure corresponds to the prescribed temperature. Since the compressor inlet temperature is 20°F below the source temperature, performance for the open-cycle steam configuration with a 10°F flash can be determined by assuming a source temperature 10°F higher than the actual temperature. Similarly, performance for a closed-cycle steam system can be determined if desired by assuming an output temperature 10°F higher than the actual temperature.

E. HEAT PUMP DRIVERS

The shaft power required to drive the heat pump compressor can be supplied by either a gas turbine, diesel, steam turbine, electric motor or a residual heat recovery Rankine power system. The evaluation of these prime movers for such use is discussed in the following paragraphs.

1. Gas Turbine

Gas turbines are commonly used as compressor drives for gas pipeline and other power applications. Their use as heat pump drives can be particularly attractive since a portion of the heat content of the high temperature exhaust gas can be recovered to augment the heat pump output. Two approaches can be taken to recover this exhaust heat. First, the exhaust heat can be used in parallel with the heat pump to generate more of the same quality process heat (i.e., 300°F saturated steam as in the Federal facility application). Second, the exhaust can be used in series with the heat pump output to produce high pressure saturated or superheated steam by boiling high pressure water that has been preheated by the heat pump. In either case, the COPF of the turbine system can be significantly higher than one with an electric drive. A list of representative gas turbines that includes output is given in Table III-7. Also shown is the power conversion efficiency, the useful heat (exhaust above 300°F), conversion efficiency and the COPF that would result from driving a heat pump with a COP of 3.0. The values for COPF vary from 1.133 to 1.481. The predominant reason for this variation is the increase in power conversion efficiency achieved in the larger gas turbine sizes. This trend cannot be represented by a single smooth curve, since some of these gas turbines, designed a decade ago, do not reflect the latest technology. The single-curve variation of power recovery efficiency with size shown in Fig. III-8 is representative of better engines and was utilized in the parametric analyses conducted in this study. Also shown is the heat recovery efficiency. Figure III-9 gives

TABLE III-7
TYPICAL GAS TURBINE DRIVES

<u>BUILDER</u>	<u>MODEL</u>	<u>HORSE POWER</u>	<u>FRACTION OF INPUT HEAT AS POWER</u>	<u>RECOVERABLE AS HEAT</u>	<u>COPF*</u>
Detroit Diesel Allison	250-KB	370	0.213	0.541	1.180
PWA Canada	ST6-L77	654	0.224	0.520	1.192
Solar	Saturn	1160	0.223	0.489	1.158
PWA Canada	ST6-T76	1440	0.213	0.494	1.133
AVCO	Super TF25	2500	0.225	0.543	1.218
AVCO	Super TF35	3500	0.261	0.450	1.273
Solar	Centaur	3830	0.265	0.442	1.237
Detroit Diesel Allison	S01-KB	4380	0.283	0.471	1.320
Garrett	IE990	5800	0.293	0.472	1.351
Detroit Diesel Allison	571-KC	7300	0.320	0.467	1.427
Solar	MARS	10600	0.328	0.426	1.41
Westinghouse Canada	CW182MD	18490	0.293	0.467	1.346
G. E.	LM2500	27500	0.358	0.407	1.481
Curtiss Wright	Mod Pod 25	33000	0.296	0.474	1.362
U.T.C.	FT4C-3f	39000	0.319	0.436	1.393

*Based on heat pump COP of 3.0.

FIGURE III-8

GAS TURBINE PERFORMANCE

EXHAUST REJECT TEMPERATURE = 300°F

FIGURE III-9

PRIME MOVER COSTS

1981 DOLLARS

NOTE: STEAM TURBINE INCLUDES BOILER
AND FEEDWATER HEATER

the specific equipment cost for gas turbines in the size of interest for the Federal facility and utility applications.

2. Diesel

Diesel engines are widely used in industrial applications to provide shaft power and electricity. Like gas turbines, diesels are particularly attractive as heat pump drivers since a portion of their reject heat can be used to supplement the heat pump output. However, the diesel reject heat is not all in the form of high temperature gas. A substantial amount of the heat is rejected as moderate temperature water ($\sim 250^{\circ}\text{F}$) in the process of cooling the cylinder walls and the lubricating oil. Depending on the type and size of diesel, this low temperature reject heat can account for as much as 40 percent of the fuel input heat. Table III-8 lists the characteristics typical of commercially available diesels ranging in size from 500 hp to 44,000 hp. Each model shown has a wide output range which is achievable by varying the number of cylinders. For example, the Fairbanks Morse 38D8-1/8 model which operates at 514 rpm can be designed to have a nominal output of 500, 750, 1000 or 1,500 hp by using 4, 6, 8 or 12 cylinders, respectively.

Each of the diesel engines listed in Table III-8 is characterized by a power recovery fraction and a heat recovery fraction comparable to those identified for the gas turbines. Similarly, a value for COPF is defined for systems with a heat pump cycle COP of 3.0. The values of COPF vary from 1.2 to 1.44 and generally increase with size. Specific equipment cost for low speed diesels is given in Fig. III-9.

TABLE III-8

TYPICAL DIESEL MECHANICAL DRIVES

Fraction of Input Heat Recoverable

<u>Builder</u>	<u>Model</u>	<u>Horsepower Range</u>	<u>Rpm</u>	<u>Max. Number of Cylinders</u>	<u>As Power</u>	<u>As Heat</u>	<u>COPF[*]</u>
Fairbanks Morse	38D8-1/8	500 - 1500	514	12	.346	.157	1.20
		560 - 1680	600	12	.346	.164	1.20
		640 - 1920	720	12	.346	.172	1.21
		800 - 2400	900	12	.321	.173	1.14
		1440 - 2880	720	12	.371	.230	1.34
		1800 - 3600	900	12	.371	.230	1.34
	PC-2	4000 - 9000	514	18	.366	.242	1.34
M.A.N.	KSZ70/125B	8260 - 20650	145	10	.418	.158	1.41
	KSZ78/155B	10660 - 23990	122	9	.422	.158	1.42
	KSZ78/160B	18350 - 44040	122	12	.427	.158	1.44

*

Based on heat pump COP of 3.0

3. Steam Turbine

In general, steam turbines in the under 2000 hp range are not very efficient (i.e., 40-70 percent). Many current steam turbines were designed before high cost energy, when low capital cost was emphasized. Even though steam turbines in the under 2000 hp range can achieve higher efficiencies with development, 85 percent levels have not been demonstrated. Table III-9 shows typical inlet temperature and pressure as well as performance characteristics for three typical back-pressure steam turbines for the 500-to-5000, 5000-to-20,000 and the 20,000-to-50,000 hp ranges, assuming an 85 percent expansion efficiency and an 85 percent boiler efficiency. The COPF for the 300°F saturated steam heat pump cycle is between 1.10 and 1.19; in all cases the coefficient is below that achievable using either a gas turbine or a diesel. The major cause is the loss of power which results from back-pressuring the turbine to provide useful process heat (i.e., 67 psia in the example). If the steam turbine data given in Table III-9, typical of the 20,000-to-50,000 hp size, were adjusted for operation at an exhaust pressure of 2 inches of Hg, the power fraction would increase to 0.28. Even so, this fraction corresponds to a COPF of only 0.849, since none of the exhaust heat could be utilized. This operating characteristic limits the usefulness of steam turbine heat pump drives.

Specific cost data for steam turbine systems including boiler and feed water heater are given in Fig. III-9.

4. Electric Motor

Electric motors drive most commercially available heat pumps. Such motors come in a wide variety of sizes, from less than one to over 100,000 hp, and in the larger sizes can convert electricity to shaft work at efficiencies approaching 98 percent. Efficiencies in the 95 percent range are typical of units in the 500-to-1000 hp range. However, with an electric motor drive for the heat pump the heat rejected at the power generation site is

TABLE III-9

STEAM TURBINE PERFORMANCE FOR HEAT PUMP DRIVER

67 psia Back Pressure
Boiler Efficiency = 85 percent
Expansion Efficiency = 85 percent

<u>Nominal Size, HP</u>	<u>Pressure (psia)</u>	<u>Temp (°F)</u>	<u>As Power</u>	<u>As Heat</u>	<u>COPF*</u>
500-5000	465	700	.126	.725	1.10
5000-20000	665	800	.150	.700	1.15
20000-50000	865	900	.169	.681	1.19

*Based on heat pump COP of 3.0

not recovered. Therefore the overall system is not fuel efficient when the heat pump COP is low. Thus, the best that an electric motor-driven heat pump with a COP of 3.0 can do in terms of COPF is about 1.0 if the electricity is generated at 33 percent efficiency (including transmission loss). Normally, 29-31 percent is the quoted electric efficiency and transmission effect. An electric motor is an effective driver if the heat pump COP is 4 or more, since under these conditions a COPF above 1.0 can be obtained. These conditions occur when the required temperature lift is 120°F or less. Therefore, the electric motor could only find possible use for application #6.

5. Residual-Heat-Driven Rankine Cycle

In some cases the source of residual heat used by the heat pump at the evaporator can also be used to provide heat to generate power by means of a Rankine cycle. An example of this is discussed in Section IV of this report. Even though the capital cost of a residual-heat-driven heat pump will be high, the essentially zero fuel costs can make this sytem configuration quite attractive. Although it is not practical to evaluate this type of prive mover compared to the others, since it must be designed to be site specific, a few observations can be made about its use.

In terms of the heat pump applications defined in this report, a residual-heat-driven Rankine cycle may be practical for the Federal facility application. Such cycles are not practical for utility applications, due to their low residual heat temperatures. For industrial applications, where the temperature is below about 140°F, a residual-heat-drive appears impractical. When the residual heat is above 140°F, the quantity of heat to satisfy the heat pump evaporator as well as the power system requirement must be evaluated. For example, if a residual heat source of 300,000 lb/hr from 150°F water is available, a 10°F drop in temperature yields 3×10^6 Btu/hr. Using a heat pump with a COP of 3.14, 4.4×10^6 Btu/hr of heat can be delivered at

300°F. To do so requires 1.4×10^6 Btu/hr or 410 KW of compressor power. Only about 50 KW can be obtained using a 5 percent efficient Rankine cycle to recover the heat in the residual source between 140° and 130°F. Clearly, this energy source is insufficient to power the heat pump. In fact, under these assumptions (i.e., 5 percent efficient power system and a combined heat removal down to 130°F), this residual heat stream can supply only 850,000 Btu/hr of heat output (or 19.3 percent of the requirement) at 300°F using a residual heat recovery Rankine cycle for all of its power.

6. Driver Selection

In the foregoing discussion of heat pump driver concepts, gas turbines, diesels, steam turbines, electric motors and residual-heat-powered Rankine cycle were examined. Of these, the site specific problems associated with the residual-heat-powered Rankine cycles preclude it from consideration except at the Federal facilities. Since residual heat recovery for power cycles is the subject of Section IV, it will not be further discussed here. The electric motor can also be eliminated since all of the applications require such high temperature lifts that the heat pump COP will be so low that the resulting COPF will not be competitive with other drive concepts. Of the remaining drivers, each can be used for a heat pump application with a reasonable COPF (e.g., gas turbines and diesels at COPF = 1.4 and steam turbines at COPF = 1.2). Each of these drivers will be considered in the next section of this report to select the most economically attractive heat pump system for application at the Portsmouth gaseous diffusion plant.

F. SYSTEM SELECTION FOR PORTSMOUTH GASEOUS DIFFUSION PLANT

The available residual heat resource at Portsmouth has been defined (Ref. 3-5) as 143,000 gpm in summer (71.5×10^6 lb/hr) and 113,000 gpm in winter (56.5×10^6 lb/hr) at an average temperature of 150°F (actually 156°F in summer and 154°F in winter; however, the assumption of 150°F accounts for transmission loss). A typical industrial park process heat requirement for a hypothetical industrial park located just north of the Portsmouth GDP is 400×10^6 Btu/hr of 300°F saturated steam. The heat pump equipment site which is assumed to be located at the industrial park would require a 3-mile pipeline (1-1/2 miles each way) to transport residual water to and from the GDP.

Selection of the most attractive heat pump system for a particular application can be made only by determining the return-on-investment (ROI) obtainable using the combinations of heat pumps working fluid configuration (R11/steam, methanol, semi-open steam and open steam) and drivers (gas turbines, steam turbines and diesels) previously identified. Unless otherwise stated, ROI is a before-tax value and is valid for government ownership of the heat pump system. After-tax ROI's for privately owned equipment is generally 75 percent of the before-tax ROI. The matrix of configuration and drive options were analyzed by first comparing the ROI of the four configurations with a gas turbine driver. After the R11/steam dual-fluid configuration was identified as best, systems with the R11/steam heat pump but with different drivers were evaluated.

For a gas-turbine driven system, the costs for the driver, the major components (compressors, evaporator, condenser, gas turbine residual heat boiler, fluid inventory and, if needed, gear), auxiliaries (estimated at 25 percent of the major component cost) and installation (estimated to be equal to the total of the capital cost items) are summarized in Table III-10. The cost for installation does not include any residual heat

TABLE III-10

COMPARISON OF HEAT PUMP SYSTEM INSTALLED COSTS FOR PORTSMOUTH APPLICATION

Gas Turbine Drive

Waste Heat Temperature = 150 °F

Output = 400×10^6 Btu/hr of 300 °F Steam

Millions of 1980 Dollars

<u>Configuration</u>	<u>Driver</u>	<u>Major Components*</u>	<u>Auxiliaries</u>	<u>Installation</u>	<u>Total**</u>
Rll/Steam	4.3	3.7	0.9	8.9	20.5
Methanol	4.3	4.0	1.0	9.3	21.4
Semi-Open Steam	4.1	5.3	1.3	10.7	24.6
Open Steam	4.0	4.7	1.1	9.8	22.5

* Includes Gas Turbine Exhaust Waste Heat Boiler

** Includes 15% Contingency

collection or steam distribution piping. The total system costs, including 15 percent contingency, are shown in Table III-11 along with the annual fuel consumption for each configuration. Also shown is data for a conventional boiler (at 85 percent efficiency) with sufficient capacity to provide the same process heat. Each of the systems is compared to the boiler in terms of return-on-investment (ROI) based on the criteria listed in Table III-12. The same cost fuel, gas at about \$3.00/10⁶ Btu was used as fuel for all the systems. Other applications only require a different value for the gas, oil and solid fuels used in the various systems. It can be seen that the semi-open steam system has a lower ROI than the other three configurations. Furthermore, there is no cost advantage in using methanol, and the disadvantage posed by its flammability and subatmospheric operating pressure may be sufficient to eliminate it from consideration. The open-cycle has the disadvantage of operating with flashed steam from the residual water which may contain noncondensable gases that could result in much lower performance than that calculated in this study. Since the results also show no ROI advantage over the R11/steam system, the open steam system can also be eliminated. The selected system (R11/steam) has all the advantages of high ROI, minimum number of compressor units, sufficient operating pressures to reduce the possibility of noncondensable gas contamination, nontoxicity, nonflammability, widespread working fluid acceptability and extreme flexibility for use in other applications.

TABLE III-11

COMPARISON OF HEAT PUMP SYSTEMS WITH BOILERS FOR PORTSMOUTH APPLICATION

Gas Turbine Drive

Waste Heat Temperature = 150°F

Output = 400×10^6 Btu/hr of 300°F Steam

<u>Configuration</u>	<u>COPF</u>	<u>Annual Fuel Consumption, 10^{12} Btu</u>	<u>Total System Installed Cost, \$$10^6$ (1980)</u>	<u>Before Tax Return on Investment, percent</u>
Boiler	0.85	3.76	3.2	Base
R11/Steam	1.41	2.27	20.5	36.2
OS Methanol	1.42	2.26	21.4	34.8
Semi-Open Steam	1.48	2.16	24.6	32.0
Open Steam	1.52	2.10	22.5	35.7

TABLE III-12

ASSUMPTIONS USED IN HEAT PUMP ECONOMIC ANALYSIS

System Life	20 yrs.
Depreciation Period*	16 yrs.
Investment Tax Credit*	10%
Misc. Annual Charges	2%
O&M as percent of Capital Cost	2.25%
Capital Cost Year	1980
Cost Year of Fuel, Electricity, etc.	1980
Start-up Year	1982
General Inflation Rate	6%
Escalation Rate of Fuel**	4%
Escalation Rate of Electricity**	2%
Cost of Fuel	\$3/10 ⁶ Btu
Cost of Electricity	3.3¢/kWhr.
Depreciation Method*	Straight Line
Tax Rate*	48%
Discount Rate	10%
Operating Time Per Year	8000 Hrs.

* These Items Are Omitted For Before Tax Calculation

** Above General Inflation Rate

In addition to the gas turbine, the diesel and the steam turbine are potential heat pump drivers for this application. For the total system output the size of the heat pump needed using a diesel driver is larger than that needed using a gas-turbine driver since significantly less heat can be recovered from the diesel exhaust gases. This larger heat pump plus the higher capital cost for a diesel (Fig. III-9) yield a total system installed cost of $\$36.3 \times 10^6$ compared to only $\$20.5 \times 10^6$ for the gas turbine driven system. However, the higher efficiency of the diesel reduces the annual fuel requirement from 2.27×10^{12} Btu for the gas turbine to 2.14×10^{12} Btu. Even so, it can be seen from Table III-13 that the ROI for such a system is below that of the gas turbine driver and therefore eliminates it from further consideration for this application.

Using a steam turbine for the driver of the RII/steam heat pump to produce 400×10^6 Btu/hr provides a system that is quite different than the one which uses the diesel. This is due to a large fraction of the steam turbine output which is recoverable for heat (steam) and only a small fraction of its output as shaft power. For the Portsmouth application, 400×10^6 Btu/hr of output, the exhaust heat recovered from the prime mover is 124×10^6 Btu/hr, 42×10^6 Btu/hr, and 224×10^6 Btu/hr for the gas turbine, diesel and steam turbine, respectively. Thus, the capacity of the heat pump proper is much less for the steam turbine driver case and the cost for the total system is less. However, the resulting small use of Portsmouth residual heat yields a low COPE and hence a large fuel consumption. The ROI for such a system (Table III-13) is lower than the diesel-driven system or the gas turbine-driven systems. Thus, neither the diesel nor the steam turbine-driven heat pump show a greater ROI than the gas turbine-driven system. Furthermore, the disparity in driver economics is so great that the selection of the RII/steam heat pump over the other fluid configurations for this application would not be altered by use of a diesel or a steam turbine.

TABLE III-13

COMPARISON OF HEAT PUMP DRIVERS FOR PORTSMOUTH APPLICATION

R11/Steam Heat Pump Configuration

Waste Heat Temperature = 150 °F

Output = 400×10^6 Btu/hr of 300 °F Steam

<u>Driver</u>	<u>COPF</u>	<u>Annual Fuel Consumption, 10^{12} Btu</u>	<u>Total System Installed Cost, $\\$10^6$ (1980)</u>	<u>Before Tax Return on Investment, percent</u>
Boiler	0.85	3.76	3.2	Base
Gas Turbine	1.41	2.27	20.5	36.2
Diesel	1.49	2.14	36.3	21.9
Steam Turbine	1.21	3.30	13.6	19.9

The Portsmouth gaseous diffusion plant industrial park requirement of 400×10^6 Btu/hr can, and for high reliability should, be provided by a multiple number of smaller units. Using the ROI procedure previously discussed, values of this economic parameter have been calculated for units with smaller outputs both with and without the required 3-mile waste water pipeline. These results are presented in Fig. III-10 and show that larger units are more economical. However, the use of four 100×10^6 Btu units, three 133×10^6 Btu/hr units or two 200×10^6 Btu/hr units would increase overall system reliability while still providing relatively high values of ROI. The use of multiple units each below 100×10^6 Btu/hr appears much less acceptable.

FIGURE III-10

HEAT PUMP SYSTEM ECONOMICS FOR PORTSMOUTH APPLICATION

GAS TURBINE DRIVE

WASTE HEAT STREAM INLET TEMPERATURE = 150°F

OUTPUT = 67 psia STEAM (300°F SATURATED)

G. ALTERNATIVE APPLICATION EVALUATIONS

The selection of the gas-turbine-driven Rll/steam heat pump as the most appropriate system for the Federal facility application does not imply that it is the most economically attractive system for other heat pump applications. Each unique application poses different constraints on the choice of both the heat pump working fluid configuration, driver and fuel availability, and cost. In this section, these constraints are examined to determine their influence on the choice of heat pump system.

1. Utility Application

For the utility application, 20×10^6 lb/hr of residual hot water is available at an average temperature of 110°F (variations from 105°F - 121°F are expected, see Table III-1) and the desired output is saturated steam at 300°F . Thus, this application differs from the Federal facility application only in the temperature lift required.

Since increasing the temperature lift of the heat pump puts more importance on optimum working fluid selection, the concept proposed for the Federal facility application should be re-examined. The same assumptions were used to develop cost estimates for the gas turbine, major components, auxiliaries and installation for the Rll/steam, methanol, semi-open steam and open steam heat pump systems. These are listed on Table III-14. There are two reasons why these costs are higher than those for the Federal facility application (Table III-11). First, the increased lift yields a lower heat pump COP (2.58 compared to 3.14 for the Rll/steam heat pumps) which requires a larger, more expensive prime mover to produce the same heat output. Secondly, the lower residual heat temperature requires both an additional compressor stage and larger compressors to accommodate the increase in vapor specific volume. This latter factor is quite pronounced in the two steam heat pump configurations where the

TABLE III-14

COMPARISON OF HEAT PUMP SYSTEM INSTALLED COSTS FOR UTILITY APPLICATION

Gas Turbine Drive

Waste Heat Temperature = 110°F

Output = 400×10^6 Btu/hr of 300°F Steam

Millions of 1980 Dollars

<u>Configuration</u>	<u>Driver</u>	<u>Major Components</u> *	<u>Auxiliaries</u>	<u>Installation</u>	<u>Total</u> **
Rll/Steam	4.8	4.5	1.1	10.4	23.9
Methanol	4.8	5.5	1.4	11.7	26.9
Semi-Open Steam	4.6	9.0	2.3	15.9	36.6
Open Steam	4.5	7.9	2.0	14.4	33.1

* Includes Gas Turbine Exhaust Waste Heat Boiler

** Includes 15% Contingency

mass flow had to be split into three streams to permit reasonably sized compressors even for the 150°F Federal facility application.

The return-on-investment values obtained for the four systems (Table III-15) show that for this application the Rll/steam heat pump is clearly superior to the two steam systems and more than 10 percent better than the methanol system.

Earlier comments on the driver selection for the Rll/steam heat pump are unaffected by designing to the utility application. Using a diesel will increase the capital cost and not permit any substantial fuel saving. Using a steam turbine would result in a COPF of only 1.12, which is too low to justify the expense of the heat pump system. However, the cost implications for other fuels should be re-examined.

2. Industrial Applications

The basic differences between the Federal facility and utility applications and the industrial applications (see Table III-1) is the reduced amount of residual heat available and the required output. The reduction in heat pump capacity would not influence the choice of heat pump working fluid. The prime effect of the fluid is to increase the output unit cost of the compressor more rapidly than any other component; the Rll/steam heat pump has the lowest compressor cost. However, reduction in output could effect the selection of the heat pump driver. While gas turbines and steam turbines exhibit a fairly continuous variation of cost with size, diesels do not. Large diesels (>10,000 hp) are normally designed for base-load operation and generally operate at low speed. They are quite bulky and heavy and are designed for long life with a minimum downtime and maintenance problems. They are usually built to order from standard designs and are field-erected. Industrial diesels in sizes below 10,000 hp have different characteristics. Although there is no sharp dividing line, these diesel engines (commonly

TABLE III-15

COMPARISON OF HEAT PUMP SYSTEMS WITH BOILERS FOR UTILITY APPLICATION

Gas Turbine Drive

Waste Heat Temperature = 100°F

Output = 400×10^6 Btu/hr of 300°F Steam

<u>Configuration</u>	<u>COPF</u>	<u>Annual Fuel Consumption, 10^{12} Btu</u>	<u>Total System Installed Cost, $\\$10^6$ (1980)</u>	<u>Before Tax Return on Investment, percent</u>
Boiler	0.85	3.76	3.2	Base
Rll/Steam	1.26	2.54	23.9	26.0
Methanol	1.27	2.52	26.9	23.3
Semi-Open Steam	1.33	2.41	36.6	18.2
Open Steam	1.36	2.35	33.1	21.2

referred to as high-speed diesels) operate at higher speeds and tend to weigh only one-third that of the larger units on a per unit output basis. They also are produced in greater quantity. This type diesel would have a lower unit cost than a low-speed model of the same output.

Typical equipment costs for a 4,000-10,000 hp diesel is \$125/hp, roughly comparable to gas turbines. Using these driver costs and scaling the heat pump costs to reflect smaller sizes, ROI values were calculated for the two different systems as functions of output for residual heat temperatures of 110° and 150°F (Fig. III-11). These data show that for the size range where the relatively inexpensive high-speed diesel can be used ROI is insensitive to driver selection for 150°F residual heat. However, the gas turbine has a clear advantage when the residual heat is 110°F.

The use of heat-engine-driven heat pumps for the four industrial applications can be evaluated by examination of Fig. III-11. For application #3 where 110°F residual heat is available and only 10×10^6 Btu/hr of output is needed, this system is not economically attractive with currently available equipment. This is due to the combination of reduced driver efficiency and higher specific cost in the small equipment sizes. For application #4, however, where 110°F residual heat is used to provide 100×10^6 Btu/hr of 300°F steam, the before-tax return-on-investment for the gas-turbine-driven heat pump approaches 20 percent and therefore can be an economic venture. The size defined by application #5, 30×10^6 Btu/hr from 150°F residual heat has a return-on-investment of 22 percent, which makes it as attractive an investment as the larger, higher-lift application #4 system. Application #6 with 170°-200°F residual heat in a 100×10^6 Btu/hr size, though not shown on Fig. III-11 was found to be economically attractive due to its relatively low lift.

THERMALLY DRIVEN HEAT PUMP SYSTEM ECONOMICS

————— GAS TURBINE
- - - - - HIGH-SPEED DIESEL

H. PROTOTYPE SYSTEM CHARACTERISTICS

A prime-mover-driven, dual-fluid Rankine cycle system has been identified as the most economical heat pump for utilizing large liquid residual heat sources from Federal facilities, utilities and industries. Due to economies of scale and other factors, larger heat pump systems are more attractive than smaller systems; hence process heat requirements for large single industries as well as industrial parks would best be met by relatively large systems (i.e., $>100 \times 10^6$ Btu/hr). The analysis presented in this section showed that for systems above an output 100×10^6 Btu/hr, the increase in return-on-investment becomes relatively small. Thus, the prototype system size selected for testing at EAST is 100×10^6 Btu/hr, and is justified on the following basis:

- 1) the prototype will use the same type of components that the larger systems would use.
- 2) the prototype would be directly applicable to the Portsmouth GDP Central Energy Management Systems or to other industrial park sites.

System component criteria are discussed in the following paragraphs.

The major components of the heat pump system are the driver, evaporator, condenser/boiler and compressor. For the output size range from $100 \times 400 \times 10^6$ Btu/hr, the total heat pump drive power varies from about 8,000 to 35,000 hp. Within this range of power requirements, there are a number of commercially available units, gas turbines and diesel engines (see Part E). Although a specific size heat pump may not have a driver requirement that matches an exact prime mover rating, use of a derated engine operating at a part load condition (10-15 percent below aerodynamic design) will not impose a significant performance penalty and may enhance the hot section material life. Thus

there appears to be no restriction on system size imposed by the gas turbine driver. The residual heat boiler for the gas turbine exhaust can be made to order in any size, so such boilers cause no restriction. For the same heat pump output variation, the evaporator and condenser/boiler have heat transfer load ranges of 45 to 189×10^6 Btu/hr and 61 to 257×10^6 Btu/hr, respectively. Within these ranges, heat exchangers of the flooded type (identified in Part D-2) can be purchased on a built-to-order basis from various vendors. The actual design (i.e., shell diameter, number of tubes and their length and diameter and internal baffling) will undoubtedly vary with size but will not impose any performance or operating restriction.

The essential compressor design features to be demonstrated and proven by the prototype heat pump system include maintaining working fluid compatibility with the construction materials and bearing lubricating oil, and achieving the specified aerodynamic and mechanical performance parameters. Maintaining the oil for lubrication separate from the working fluid will require selection and demonstration of seals that are used in special industrial applications. Although the compressor operating speed is at a higher-than-normal limit when the compressor design is scaled to a larger capacity, this should not cause a problem, since performance levels are generally higher when radial or axial compressors are scaled to larger sizes due to reduced Reynolds' Number effects and smaller leakage flows. Thus, if the performance levels can be demonstrated in the 100 to 150×10^6 Btu/hr size system, they could be achieved in higher capacity systems as well. Since compressors are generally offered in discrete frame sizes by manufacturers, demonstration of peak efficiency may necessitate some compromise in compressor selection. Once a final design is made and suitable equipment manufacturers are contacted, the exact features of the prototype system can be resolved. Variations in compressor rotor speed with size are not anticipated to introduce any problem since a wide variety of gear boxes in the hp range of interest are available.

The second criteria for selecting prototype system size for the EAST facility is the applicability to future industrial applications; i.e., at the Portsmouth industrial park or at other sites. Although the industrial park characterized in Section V requires 400×10^6 Btu/hr, this value is only representative. If the size proves to be correct in a mature industrial park at that site, heat pump units of 100 to 200×10^6 Btu/hr capacity for the initial installation would be acceptable. As the industrial park expands, additional units of this size or larger could be added. Furthermore, the use of multiple units will permit removal of one unit from service for routine maintenance without disrupting process heat production or relying on a back-up boiler system.

Therefore, testing a 100×10^6 Btu/hr prototype heat pump system permits the demonstration of a system with applications in both Federal facilities, utilities, and large industrial plants. Furthermore, a prototype of this size will demonstrate all of the essential features required of a larger machine, thereby permitting scale-up to a 400×10^6 Btu/hr size or larger with only minimal design change. In addition, the physical size requirements of the system components would represent a reasonable compromise of the floor area and volume required for the EAST facility. Table III-16 summarizes the residual heat flow and temperature conditions required to test a 100×10^6 Btu/hr heat pump.

I. TEST FACILITY SPECIFICATIONS

1. Heat Pump System

The EAST facility will have access to large quantities of water at approximately 150°F. The prototype 100×10^6 Btu/hr heat pump will obtain its input heat from approximately 10,000 gpm of this water as it is cooled approximately 10°F. After this heat removal, the water will be returned to the normal SRP cooling channels. To overcome the pressure loss in the heat pump evaporator and ensure that the residual stream water flow is continuous, a pump capable of pressurizing the 10,000 gpm to 10 psig is required. Furthermore, the desire to simulate lower temperature residual heat sources requires a dilution system either using cooling water or a recirculated residual stream sufficient to lower the water stream temperature to a level of approximately 110°F.

During the test program, the output of the heat pump in the form of 300°F saturated steam must be dissipated and pressurized water returned to the system. This heat dissipation could be accomplished by utilizing 10,000 gpm of 70°F cooling water in a condenser. The temperature of this cooling water would rise 15° if the gas turbine residual heat boiler was in use.

As a result of the preliminary design, a 100×10^6 Btu/hr heat pump using a gas turbine drive and an exhaust residual heat boiler will require an equipment floor area of 36 ft by 56 ft and a ceiling height of 20 ft. This would require a total test cell area of approximately 2400 sq ft to allow for aisle and work space. In addition, it is estimated that an auxiliary power requirement of up to 500 kW may be needed for pump drives and up to 125 tons of building cooling to dissipate heat generated by the equipment. This cooling load may be reduced considerably by using large air circulating fans.

A variety of controls are required to permit control room operation of the unit. Among these are the driver controls and the various residual stream pump controls. Finally, there is a requirement for an exhaust stack capable of handling 80,000 cfm of 850°F exhaust if no residual heat boiler is used, or >50,000 cfm of 300°F exhaust if the residual heat boiler is used. A complete list of test facility requirements is given in Table III-16.

Minor modifications to the EAST facility would permit testing a heat pump with an output of 400×10^6 Btu/hr. This would be done by utilizing the test cell area being designed for the power generation equipment and recovering the heat by reducing the temperature of the 22,000 gpm residual stream available to that part of the facility by approximately 17°F.

As part of the prototype testing program, the temperatures, pressures, flow rates and rotor speeds listed in Table III-17 will be measured to determine whether the components and the entire system is functioning according to design. There is no need for quick response time measurements at present; however, this type of instrumentation may be desired on subsequent tests.

TABLE III-16

HEAT PUMP REQUIREMENTS - EAST FACILITY

HEATED WATER

Flow rate	10,000 GPM
Temperature	150°F
Source	Reactor Coolant
Required Pressure	20 psig

HEAT PUMP SYSTEMTEST CELL SPECIFICATIONS

Length	36 ft.
Width	24 ft.
Height	20 ft.
Pit Depth	N/A
Crane Capacity	5 tons

OUTPUT100 x 10⁶ Btu/hrADDITIONAL FLUID HANDLING & STORAGE

R-11

No. 2 Fuel Oil

Lubricating Oil

ELECTRICAL POWER REQUIRED

500 kW

AUXILIARY REQUIREMENTS

Gas Turbine Control
 Hot Water Control
 300°F Steam Dissipator
 G/T Exhaust Stack
 (80,000 CFM @ 300°F)
 125 Tons Space Cooling
 Cooling Tower

TABLE III-17

HEAT PUMP TEST MEASUREMENT REQUIREMENTS

<u>Location</u>	<u>Temperature</u>	<u>Pressure</u>	<u>Flow Rate</u>	<u>Speed</u>
Waste Heat Inlet	X		X	
Waste Heat Outlet	X			
R11 Evaporator	X	X		
R11 Compressor Inlet	X	X	X	X
R11 Compressor Outlet	X	X		
Interfluid Heat Exchanger Hot Side	X	X		
Interfluid Heat Exchanger Cold Side	X	X		
Steam Compressor Inlet	X	X	X	X
Steam Compressor Outlet	X	X		
Condensate Return	X		X	
Prime Mover Exhaust Exit	X	X	X	
W/H Boiler Exhaust Exit	X	X		
W/H Boiler Feed Water	X	X	X	
W/H Boiler Steam Output	X	X		

Note: Standard prime mover control parameters such as fuel flow and shaft speed will also be measured.

J. REFERENCES

- III-1 The Savannah River Plant Low Level Waste Heat Utilization Project, Report prepared for DOE by South Carolina Energy Research Institute, August 1978.
- III-2 Use of Reject Heat from Uranium Enrichment Gaseous Diffusion Plant Portsmouth, Ohio. Final Report prepared for Department of Energy by Bechtel National, Inc., Vols. I and II, October 1978.
- III-3 Oak Ridge Gaseous Diffusion Plant Reject Heat Study, Final Report prepared for DOE by Exxon Nuclear Company, Inc., Vols. I through IV, September 1978.
- III-4 U.S. Department of Energy Waste Heat Utilization Project for the Paducah, KY Uranium Enrichment Gaseous Diffusion Plant, Final Report prepared by J. E. Sirrine, Co., Vols. I and II, September 1978.
- III-5 Bass, R. W.: Residual Energy Applications Program Task I, Final Report, UTRC Report R79-954787-1, October 1979.
- III-6 Snyder, P. H. and W. C. Moreland: Development and Demonstration of an Electric Heat Pump for Waste Heat Recovery in Industry - The Selection of Working Fluid Candidates for Use in High Temperature Industrial Heat Pumps, Westinghouse Electric Corporation, December 1978.
- III-7 Cogeneration Technology Alternatives Study (CTAS), Prepared by Power Systems Division, United Technologies Corporation for DOE, Report No. DOE/NASA/0030-80/3, January 1980.
- III-8 R&D for Improved Efficiency Small Steam Turbines, Phase I, prepared by Northern Research and Engineering Corporation for DOE, Report No. NREC-1380-1, October 1979.

SECTION IV

POWER RECOVERY SYSTEMS

A. INTRODUCTION

Large quantities of warm water at temperatures ranging from 100°-200°F are discharged by steam power plants, industrial process plants and from the Federal nuclear facilities. Some of the energy contained in this water can be converted to electricity by means of advanced low-temperature power generation systems. Earlier Federal facilities studies performed by UTRC and others indicated this is indeed possible and can be achieved with current technology, although it requires hardware configurations that to-date remain untested (see Ref. IV-1 & IV-3). Moreover, the various investigative results differ widely as to the best system configuration, working fluid, component cost levels, and economic merit. The disparity in the conclusions can be attributed partially to the different ground rules and methods of analysis as well as to the variance in the external design constraints imposed in each case. These differences illustrate that universal statements cannot be made concerning the economic merit of power generation systems. Some of the earlier studies also failed to take into consideration certain design alternatives that may significantly affect the conclusions. For example, most studies evaluated only single-pressure Rankine cycles. The addition of a 2nd stage can significantly improve the efficiency without a substantial cost penalty and thereby change the initial conclusions.

1. Objectives:

The main objectives of this present study were to:

- provide a consistent parametric performance and economic evaluation of low-temperature Rankine cycle power generation systems. These systems would be suitable for the recovery of the residual energy rejected by industrial process plants and by the Federal nuclear facilities. Unlike the heat pump analysis (Section III) the utilities were not included in this study. Their much lower residual water temperature (110°F) was too low for good economics.
- define the optimum configuration and the operating and design parameters for a power recovery system that might be constructed at the Portsmouth Gaseous Diffusion Plant (GDP).
- use the results of this study to size the preliminary requirements of the EAST facility.

The results of this investigation are presented in six parts.

- Part C - Low Temperature Power Generation

This discussion concerns relevant thermodynamic aspects of low-temperature power generation. It includes a preliminary evaluation of the component sizes and performance for systems using various working fluids.

- Part D - Single-pressure system cost analysis

The cost analysis is presented on three selected working fluids R114, R500 and R717 (ammonia).

- Part E - Detailed Sensitivity and Trade-off Studies

This part addresses such aspects as pipeline optimization, benefits of multi-pressure evaporation cycles, effects of residual-stream temperature, condenser temperature and heat exchanger temperature differences. The evaluation is based on their effects on the system output and cost characteristics.

- Part F - Cooling Water Requirements

- Part G - Economic merits of the preferred ammonia system

- Part H - Site example - Portsmouth G.D.P.

This example demonstrates the utility of the developed techniques.

Much of this material was developed for ERI by UTRC (H.E. Khalifa). It describes power recovery systems sized to meet the Federal facility residual heat flow rates. Subsequent reporting will include additional input for smaller systems.

2. Conclusions

- Electric power can be generated economically with dual pressure Rankine cycle systems from the vast amounts of warm water (150°F) discharged by the Federal nuclear facilities.

Ammonia is the preferred working fluid for these low temperature applications because of its superior heat transfer, thermodynamic and turbomachinery characteristics relative to water and other organic fluids. In the 1-to-30 MWe size range, an ammonia system would be about 50 percent less costly than its nearest competitor--an R500 dual-pressure system.

Ammonia is the preferred working fluid even if an isolation heat exchanger (IHE) is required in the Federal facility application and no IHE is required when R500 is used as the working fluid. In this case the ammonia system is still about 30 percent less costly.

- Up to 40 MWe can be recovered at each of the three gaseous diffusion plants (Oak Ridge, Portsmouth and Paducah) using ammonia multi-pressure systems.
- The same type of power recovery systems can be used for low-temperature industrial effluents with typical sizes in the 1-2 MWe range.
- The cost and power expenditure of a pipeline from the residual heat source to the power plant site adversely affects the economics of power recovery. Therefore power recovery systems must be located as near the warm water source as possible.
- Multi-pressure Rankine cycle systems are thermodynamically superior to single-pressure systems. A dual-pressure system, for example, produces about 35 percent more output than a single-pressure system and is, therefore, less sensitive to the adverse economic impact of a pipeline.
- If the use of a pipeline is deemed necessary, it will be more economical to employ one or two large capacity power recovery systems than many small-capacity systems.

B. RESIDUAL HEAT CHARACTERISTICS

The terms residual heat or residual energy generally refer to energy that can no longer be used to perform economically some useful function at the site where it is produced and is subsequently dissipated into the environment. The temperature at which an energy stream loses its usefulness depends primarily on the requirements of the primary process and secondarily on the cost of providing an adequate supply of raw energy. As the prices of fuel escalate, lowering the temperature level at which energy is ultimately dissipated into the environment becomes more economically attractive. Examples of residual energy streams at temperature levels spanning the range 100-1000°F or higher can be found in the power and process industries; steam power plants typifying the low-temperature end and blast furnaces typifying the high-temperature end. Low-temperature residual energy is usually carried by liquid streams; high-temperature residual energy is invariably contained in gaseous effluents.

A careful survey of residual heat sources and their characteristics is summarized in Section I of this report. Since the present section covers only power generation, only those residual energy sources that were judged suitable for power recovery will be characterized. Table IV-1 contains a summary of the characteristics of selected residual energy sources. The analysis presented in the remainder of this section concentrates on the first two applications. Applications 3, 4, and 5 should be investigated separately since they all represent moderate-to-high temperature gaseous residual energy sources. Vastly different systems and designs result for these applications.

C. CONFIGURATION DESIGN AND PERFORMANCE EVALUATION

1. Thermodynamic Considerations

The basic components for a Rankine power cycle residual heat recovery system is shown schematically in Figure IV-1.

The performance of residual-heat-driven power generation systems cannot be realistically assessed on the basis of cycle efficiency or heat rate as is customarily done for fossil-fuel power plants. Unlike the energy derived directly from a fossil fuel, residual energy is usually available as low-temperature sensible heat and at little or no cost. Residual energy sources experience a significant temperature drop when energy is withdrawn from them and cannot be treated as thermal reservoirs in the strict thermodynamic sense. The optimum thermodynamic cycle for a given residual heat source need not be the most efficient since this cycle, while converting a large fraction of its heat input into work, may extract only a small fraction of the energy available in the residual stream. This is a situation where a system endowed with a high cycle efficiency offers a poor utilization of the available residual energy. For this reason, reliance on the cycle efficiency alone as a performance measure can be misleading.

A thermodynamically rigorous performance measure for residual heat utilization systems and other low-temperature energy conversion systems must express the degree of closeness to the thermodynamic ideal. This is discussed in some detail in Appendix A. A residual stream utilization efficiency η_u is defined by

$$\eta_u = \frac{\dot{W}}{\dot{W}^0} \quad (4-1)$$

in which $\dot{W}$ is the actual power produced by the system and $\dot{W}^0$ is the available power (Appendix A). Unlike the cycle efficiency,

TABLE IV-1

WASTE ENERGY SOURCE CHARACTERISTICS

	Flow Rate 10^6 lb/hr	Temperature °F	Heat Source	Cooling Water		Typical Single Unit Output MWe
				Source	Temp. °F	
1. Federal Facilities	20-80	140-165	Cooling Water	River/Pond Cooling Tower	56-73	30
2. Industry	1.0	170-200	Condensate	Cooling Tower	~75	2
3. Industry	0.6	300-600	Stack Gas	Cooling Tower	~75	2
4. Industry	0.3	> 600	Stack Gas	Cooling Tower	~75	2
5. Industry	3.0	> 600	Stack Gas	Cooling Tower	~75	20

RANKINE CYCLE SCHEMATIC

FIGURE IV-1

the utilization efficiency accounts for the loss of available work resulting from rejecting the residual stream at increasingly higher temperature. For each energy conversion cycle configuration, there is an optimum cycle operating temperature at which the utilization efficiency is a maximum.

The configuration achieving the highest utilization efficiency (or highest cycle efficiency for that matter) may not be the most economical. More efficient equipment may require too high a capital cost. When the cost of bringing the residual stream to the power generation system is dominant, economics will favor the configuration with the higher utilization efficiency because it generates more power from the residual stream. The same will be true if the cost of the conventional electric energy that would be displaced by the residual energy recovery system is high. Under high cost-of-pipeline and high cost-of-displaced-electrical-energy conditions, the utilization efficiency can provide a basis of comparison among alternative residual energy recovery systems.

2. Working Fluid Evaluation

Because of the large available quantities of warm water (150°F) at the Federal nuclear facilities, this temperature was selected as the design point. The performance and cost comparisons reported in this section apply to a typical 150°F water stream and a specified 80°F condenser temperature which is compatible with a cooling water temperature, in the 65°-70°F range.

Previous studies have considered a large number of organic and inorganic fluids as potential low-temperature Rankine cycle working media (Refs. 4-2,-3). The selection of the most suitable working fluid for a given application depends not only on its thermodynamic and transport properties, but also on such factors as chemical stability in the operating temperature range, toxicity, flammability, availability and cost. Only a few of the

working fluids considered appear to be viable candidates for low-temperature Rankine cycle applications. Table IV-2 lists the pertinent characteristics of six candidate fluids. Appendix B contains a more complete listing.

TABLE IV-2
WORKING FLUID CHARACTERISTICS

Fluid	R11	R113	R114	R500	R717	R718
Chemical formula	CCl_3F	$\text{C}_2\text{Cl}_3\text{F}_3$	$\text{C}_2\text{Cl}_2\text{F}_4$	--	NH_3	H_2O
Molecular weight	137.4	187.4	170.9	99.3	17.0	18.0
Critical pressure, psia	635	495	474	631	1657	3206
Critical temperature, °F	388.4	417.4	294.3	221.1	271.4	705.4
Sat. pressure @ 80°F, psia	16.3	6.9	33.0	117.2	153	0.507
Sat. pressure @ 150°F, psia	52.9	25.9	95	297.0	437.0	3.72
Sat. liquid density @ 80°F, lb/ft ³	91.7	97.5	90.6	71.9	37.5	62.2
Sat. vapor density @ 80°F, lb/ft ³	0.401	0.228	1.048	2.374	0.512	0.0016
Liquid thermal conductivity @ 100°F, Btu/hr. ft. °F	0.049	0.042	0.049	0.049	0.26	0.36
Flammability limit (1)	N	N	N	N	16-25	N
Toxicity (2)	5a	4-5	6	5a	2	N

(1) Percent by volume · N - Nonflammable

(2) Underwriters Laboratory classification · N - nontoxic

R500 - Azeotrope $\text{CCl}_2\text{F}_2/\text{CH}_3\text{-CHF}_2$ - Dichlorodifluoro methane and Difluoro ethane

R11 - Tri mono fluoro methane

R113 - Tri chloro fluoro ethane

R114 - Dichloro tetra fluoro ethane

R717 - Ammonia

R718 - Steam

Performance

The equilibrium thermodynamic properties* of the working fluid directly affect the overall performance and sizing parameters of the energy conversion systems. Figures IV-2 a and b show the dependence of the utilization and cycle efficiency on the boiling temperature for single-pressure Rankine cycles matched to the 150°F residual stream. The results shown are for the six working fluids listed in Table IV-2 using the basic system assumptions listed in Table IV-3. The utilization efficiency for all selected fluids exhibit the same dependence on the boiler temperature; peak values differ by less than 4 percent for any two fluids. This indicates that the most suitable working fluid cannot be chosen on the basis of thermodynamic performance alone.

*The thermodynamic properties of the fluorocarbons R11, R113, R114 and R500 are not as well known as those of air and steam. Also these fluids exhibit small values of turbine enthalpy drop for low temperature cycles (5 Btu/lb). It is not uncommon under these conditions to have a 5-10% performance uncertainty.

EFFICIENCY OF LOW-TEMPERATURE SINGLE PRESSURE RANKINE CYCLE SYSTEMS

FIGURE IV-2

TABLE IV-3
POWER RECOVERY SYSTEM ASSUMPTIONS

Waste stream temperature*, °F	150
Cooling water temperature, °F	70
Condenser temperature, °F	80
Condenser terminal temperature difference, °F	5
Boiler pinch temperature difference, °F	5
Turbogenerator efficiency, percent	85
Pump efficiency, percent	80
Boiler pressure loss, percent	10
Auxiliary Power consumption (percent of turbine output)	5

*Water delivered to power recovery system.

Component Sizing

The most important components in a Rankine cycle system are the turbine, the evaporator (boiler) and the condenser. Characteristic turbine size depends primarily on the equilibrium properties of the working fluid. Similarity relations for turbomachinery (Ref. IV-4) indicate that the characteristic diameter (D) of a turbine and its characteristic speed (N) are a function of the exhaust volume flow rate ($\dot{V}_e$) and the enthalpy drop. These relations are

$$D = K_D \frac{(\dot{V}_e)^{\frac{1}{2}}}{(\Delta h)^{\frac{1}{4}}} \quad (4-2)$$

$$N = K_N \frac{(h)^{\frac{3}{4}}}{(\dot{V}_e)^{\frac{1}{2}}} \quad (4-3)$$

where K_D and K_N = turbine similarity parameters.

Once the key cycle operating parameters have been established, these similarity relations can be used to estimate for comparison purposes the approximate turbine diameter and speed. Table IV-4 shows estimated turbine diameters and other characteristics for the different working fluids. The turbine characteristic diameters are normalized for the same water flow rate relative to the corresponding value for ammonia (R717) at the optimum thermodynamic conditions (peak η_u).

Due to their low thermal (cycle) efficiency, low-temperature power recovery systems have to handle large quantities of heat per unit work output. Heat transfer equipment therefore constitutes a major fraction of the cost of such systems. The transport properties of the working fluid (i.e., its viscosity and thermal conductivity) influence the convective heat transfer

TABLE IV-4

SUMMARY OF PERFORMANCE AND DESIGN DATA
AT PEAK UTILIZATION EFFICIENCY
(without an intermediate loop)

Parameter	R11	R113	R114	R500	R717 (NH ₃)	R718 (H ₂ O)
<u>(a) Thermodynamic</u>						
Boiler temperature, °F	112.5	112.5	112.5	112.5	112.5	112.5
Boiler pressure, psia	29.3	13.4	56.6	185.8	256.5	1.59
Condenser pressure, psia	16.3	6.9	33.0	117.2	153.0	0.507
Peak utilization efficiency*	0.37	0.37	0.37	0.36	0.36	0.36
Corresponding cycle efficiency*	0.044	0.043	0.042	0.040	0.043	0.045
Water flow rate, lb/kwh*	2605	2585	2620	2690	2450	2440
<u>(b) Design</u>						
<u>Heat Exchangers</u>						
U-factors, Btu/hr.ft ² °F						
Preheat	200	200	200	200	400	400
Boiler	300	300	300	300	800	800
Condenser	200	200	200	200	450	450
LMTD, °F						
Preheater	10.2	10.1	9.8	9.8	7.7	8.0
Boiler	16.1	16.1	16.1	16.1	16.1	16.1
Condenser	7.2	7.2	7.2	7.2	7.2	7.2
Heat transfer area, ft ² /kw*						
Preheater	2.5	3.1	4.2	4.2	1.0	0.4
Boiler	17.5	17.3	17.1	17.1	6.0	0.4
Condenser	61.3	62.0	63.3	63.2	24.8	24.1
<u>Turbines**</u>						
Relative size	5.1	7.1	3.3	2.0	1.0	10.0
Relative number (R.S.) ²						

*Based on gross output

**Normalized with respect to the size of an ammonia turbine at the same water flow rate.

coefficients which, in turn, determine the size and cost of the heat exchangers. The two inorganic fluids (H_2O and NH_3) have excellent heat transfer properties when compared with fluoro- and hydrocarbons (Refs. 3-5, 6). In fact, the thermal conductivity of liquid H_2O and NH_3 in the temperature range 0° - $150^\circ F$ is more than five times that of common fluorocarbons such as R11 (CCl_3F) or common hydrocarbons such as isobutane (C_4H_{10}). The overall heat transfer coefficients for forced-convection boilers and surface condensers have been estimated for the six working fluids listed in Table IV-2, assuming a total fouling resistance of $0.0004 \text{ hr.ft.}^2\text{O}^\circ\text{F/Btu}$ for the boilers and 0.001 for the condensers and also assuming a water-side film coefficient corresponding to a water velocity of 10 ft/sec . The film coefficients on the working fluid side were obtained by scaling those for ammonia using the well-known correlations for forced convection boiling and film condensation (Ref. 4-5). Appendix B discusses this procedure.

Because of the large heat exchanger surface area requirements, an allowance must be made for the power required to pump the water through the tubes. It can be shown that for a given tube inside diameter and flow velocity, the pumping power expenditure on the water side of a surface heat exchanger can be expressed approximately by

$$\dot{W}_p \approx 6 \times 10^{-6} \frac{AV^{2.75}}{\eta_p d^{0.25}} \quad (4-4)$$

in which $\dot{W}_p$ is the pumping power in KW, A is the bare-tube heat transfer area in ft^2 , V is the flow velocity in ft/sec , d is the tube inside diameter in ft and η_p is the pump efficiency. For a typical velocity of 6 ft/sec in one-inch tubes and a 70 percent pump efficiency, equation (4-4) reduces to:

$$\dot{W}_p \approx .0021 A \quad (4-5)$$

This result indicates that not only does a high heat transfer coefficient reduce the size of the heat exchange equipment in the power recovery system, but also creates less pumping loss, and hence, higher net output.

Table IV-4 also summarizes the performance and component size results for the six fluids/systems evaluated. Examination of this table reveals a number of important facts:

- (1) The utilization and cycle efficiencies for all of the fluids are about the same.
- (2) The total heat exchanger surface area requirements for NH_3 and H_2O are much smaller than those for R11, R113, R114 and R500.
- (3) Steam and the four fluorocarbons require much larger turbines than ammonia.
- (4) The pressure at all operating conditions in the steam cycle is below atmospheric and the R113 cycle has a subatmospheric condenser and a near-atmospheric boiler.

Thus we conclude that ammonia, which requires both smaller heat exchangers and smaller turbines, is by far the most attractive working fluid for power recovery from low-temperature residual streams. Although steam has the heat transfer advantages of ammonia, steam requires turbines that are either too large or too numerous to be economically feasible. Also, the steam cycle operates at subatmospheric pressures; air leakage into the system may be a problem. If leak-proof operation cannot be ensured, noncondensable gases may severely compromise the heat transfer advantage of steam. The power expended in extracting the noncondensable gases from the condenser could be excessive, further reducing the net output of the system. These disadvantages more than offset the heat transfer advantages of

steam. Consequently, steam was discarded as a practical working fluid in low temperature power recovery systems.

Effect of Isolation Heat Exchangers

If a power recovery system is built at one of the Federal nuclear facilities, the process cooling water may have to be isolated from the power loop. A water/water intermediate heat exchanger (IHE), interposed between the residual stream and the power loop working fluid offers one isolation technique. The IHE will reduce the temperature of the warm water delivered to the working fluid vapor generator, degrading system thermodynamic performance and increasing installed system cost. To study the effect of using an IHE on the overall ranking of the working fluids, a 10°F temperature difference in this heat exchanger and an overall heat transfer coefficient of $400 \text{ Btu/hr ft}^2^{\circ}\text{F}$ were assumed. Table IV-5 summarizes the results for R114, R500 and R717 (ammonia) at conditions corresponding to peak utilization efficiency. Figure IV-3 shows a comparison between the net outputs and the heat rejection rates for the ammonia systems both with and without the IHE. The specific cooling water requirements for the two systems are depicted in Fig. IV-4, illustrating that the peak net output is reduced by a 35% with the addition of the IHE. Taken together these results show that ammonia still maintains its commanding position ahead of the fluorocarbons, even when ammonia is burdened by the IHE and the fluorocarbons are not.

TABLE IV-5

SUMMARY OF PERFORMANCE AND DESIGN DATA
AT PEAK UTILIZATION EFFICIENCY
(with intermediate loop)

Parameter	R114	R500	R717 (NH ₃)
<u>(a) Thermodynamic</u>			
Boiler temperature, °F	107.5	107.5	107.5
Boiler pressure, psia	52.3	173.7	237.9
Condenser pressure, psia	33.0	117.2	153.0
Peak utilization efficiency*	0.27	0.25	0.25
Corresponding cycle efficiency*	0.036	0.035	0.037
Water flow rate, lb/kWh*	4295	4480	3880
<u>(b) Design</u>			
Heat transfer area, ft ² /kW*			
Preheater	5.3	5.3	1.2
Boiler	26.2	26.2	8.8
Condenser	87.1	87.1	32.8
IHE	32.5	32.5	27.6
Relative turbine size	3.6	2.0	1.0

*Based on gross output

FIGURE IV-3

**SPECIFIC NET OUTPUT AND HEAT REJECTION RATE
OF SINGLE-PRESSURE R717 SYSTEMS**

FIGURE IV-4

SPECIFIC COOLING WATER REQUIREMENTS OF SINGLE PRESSURE R717 SYSTEMS

D. SYSTEM COST ANALYSIS

A cost analysis was performed in which the cost and output characteristics of ammonia power recovery systems with and without an isolation heat exchanger (IHE) were compared with those of R114 and R500 systems without IHE. The design and sizing parameters for all the systems were based on a residual stream flow rate 50×10^6 lb/hr.

Heat exchanger cost (\$/ft²) and cooling tower cost were estimated from the correlations compiled by UTRC and reported in Ref. IV-2. The cost of the turbine/generator set was obtained from a correlation of manufacturers' data (Ref. IV-7). This correlation estimates the cost of the turbine as a function of the pitch diameter, the number of exhaust ends and the number of stages with modifications for the temperature and blade tip speed. The total installed cost of the system was taken as twice the cost of the major equipment: the heat exchangers, the cooling towers and the turbine/generator set. The costs estimated in this manner, although not based directly on actual prices quoted by manufacturers and/or suppliers, still reflect the basic trends and are adequate for the purposes of comparison and parametric analysis.

Figure IV-5 compares the estimated installed costs of the R717 (ammonia), R500 and R114 systems. The ammonia system is consistently cheaper than the corresponding R500 and R114 systems, even when the ammonia system is burdened by the IHE which both degrades its performance and increases its cost. The ammonia system is about 30 percent less expensive than the direct (no IHE) R500 system even though the IHE constitutes a severe penalty amounting to over 35 percent loss of output and a corresponding increase in the cost per net installed kilowatt.

The analysis confirms the selection of ammonia as the best working fluid for low-temperature Rankine cycle power recovery systems with R500 as a possible second choice.

FIGURE IV-5

INSTALLED COST OF POWER PLANT

(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr
CONDENSER TEMPERATURE = 80°F

E. SENSITIVITY AND TRADE-OFF STUDIES

So far performance and design analyses have been presented for a fixed configuration and for a specified set of design and operating conditions. The impact of design variations and the influence of deviations from the specified design and operating conditions must be assessed before a final conceptual site-specific design of a power recovery system is attempted.

1. Pipeline Optimization

In some residual energy recovery applications it may be impossible to locate the power plant in the immediate vicinity of the residual energy source; a pipeline extending over large distances may be needed. The cost of the pipeline and the pumping power expended to transmit the water may constitute a sizable fraction of the power plant cost and power output. Hence the power plant and the pipeline must be optimized together to ensure the most economical combination.

Pipeline losses include both heat and power losses. In low-temperature applications heat losses can be minimized through the use of modest insulation. Power losses, on the other hand, can only be significantly reduced by increasing the pipeline diameter and hence cost. The cost of length L of a pipeline whose diameter is D can be adequately expressed by

$$C^L = \alpha L D^\beta \quad (4-6)$$

in which α and β depend on the material and construction of the pipeline. For a steel pipeline, the constants α and β are such that

$$C^L \approx 32 L D^{1.5} \text{ (1980 dollars)} \quad (4-7)$$

with both L and D in feet. The net power ($\dot{W}_m$) produced by a power plant fed by this pipeline is given approximately by

$$\dot{W}_n = \dot{m} (\eta_u \dot{W}_i^o - 0.025f \dot{m}^2 L / \eta_p \rho D^5) \quad (4-8)$$

where $\dot{m}$ is the residual stream mass flow rate, $\dot{W}^o$ is the available work at the pipeline inlet (see Appendix A), η_u is the power plant utilization efficiency, ρ is the density of the residual stream, η_p is the pump efficiency, and f is the pipe friction factor. The optimum pipeline diameter is that which maximizes the net power output per unit power plant and pipeline cost. This corresponds to the solution of the equation

$$\frac{d}{dD} (\dot{W}_{n/c}) = 0 \quad (4-9)$$

in which $C = C_o + C_L$ is the combined cost of the power plant (C_o) and the pipeline (C_L).

The impact of adding 3 miles of pipeline to R500 and R717 systems is illustrated in Fig. IV-6 for a water flow rate of 50×10^6 lb/hr (about 100,000 gpm). Fig. IV-7 depicts the effect of pipeline length on the unit-installed cost of cost-optimized R500 and R717 systems. The results show that the pipeline constitutes a substantial economic penalty. Therefore, the power plant should be located at the residual energy source and the generated electric energy transmitted to the user rather than pumping the residual stream through several thousand feet of pipeline to a nearby load center.

Since the isolation heat exchanger drastically degrades the system performance, the impact of a pipeline is more pronounced for the ammonia system equipped with an isolation heat exchanger, leading to an amplification of the relative effect of the pipeline penalty. If the construction of a pipeline cannot be avoided, its adverse impact on system cost can be reduced only by

FIGURE IV-6

INSTALLED COST OF POWER PLANT INCLUDING A 3-MILE PIPELINE
(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr
CONDENSER TEMPERATURE = 80°F

FIGURE IV-7

**EFFECT OF PIPELINE ON THE INSTALLED COST OF
OPTIMIZED SINGLE-PRESSURE SYSTEMS**

(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr

CONDENSER TEMPERATURE = 80°F

improving the utilization efficiency of the residual stream. One of the most effective ways of achieving this is to employ multi-pressure Rankine cycles instead of single-pressure cycles.

2. Multi-Pressure Systems

Multi-pressure Rankine cycle systems were identified in Ref. (IV-3) as the most economical systems for utilizing low-temperature residual energy at the Savannah River Plant. A flow diagram of a dual-pressure system is given Fig. IV-8. Theoretically if an infinite number of evaporation stages is employed, the utilization efficiency of a finite stream can be increased to a level determined only by the turbine efficiency and by the allowed pinch temperature difference. Figure IV-9 shows the variation of the utilization efficiency with the number of stages. Nearly 35 percent more power can be extracted from the residual stream if two evaporation stages, instead of one, were employed. This represents a significant improvement, particularly when the power system is burdened by the cost and power expenditure of a long pipeline. However multi-pressure systems are more complex than single-pressure systems. The superior performance of the former must be balanced against the relative simplicity of the latter in order to choose the optimum configuration for a given application.

Extensive computer simulations of dual-pressure cycles were performed to determine the optimum boiling temperatures. The results indicate that peak utilization efficiency corresponds to boiling temperatures T_1 and T_2 chosen such that the available temperature difference is divided about equally among the stages. That is:

$$T_1 = T_c + 1/3 (T_i - \delta T - T_c) \quad (4-10)$$

$$T_2 = T_c + 2/3 (T_i - \delta T - T_c) \quad (4-11)$$

Here T_i is the initial residual stream temperature, T is the pinch temperature difference and T_c is the condenser temperature. The economic optimum, however, occurs at higher values of T_1 and T_2 . Figure IV-10 displays the results for R500 and R717 systems when no pipeline is needed.

FIGURE IV-8

DUAL-PRESSURE RANKINE-CYCLE SYSTEM

FIGURE IV-9

OPTIMIZED UTILIZATION EFFICIENCY OF R717 MULTI-PRESSURE SYSTEMS

TURBINE EFFICIENCY 80 PERCENT
PINCH TEMP. DIFF. 5°F

FIGURE IV-10

INSTALLED COST OF OPTIMIZED DUAL-PRESSURE SYSTEMS

(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr
CONDENSER TEMPERATURE = 80°F

Figure IV-11 shows the effect of pipeline cost and power loss on overall system cost and performance for R500 and R717 dual-pressure systems. Due to their superior thermodynamic performance, dual-pressure systems are not as severely penalized by the pipeline as are single-pressure systems.

3. Effect of Residual Stream Temperature

The effect of the residual stream temperature on the utilization efficiency of single stage R717 systems is depicted in Fig. IV-12. Stream temperature effects are amplified by an isolation heat exchanger. A general rule, which holds reasonably well for single as well as multi-pressure systems, indicates that the boiler temperatures corresponding to the peak utilization efficiency can be obtained from a generalization of the dual-pressure case, i.e.,

$$T_k = T_c + \frac{k}{n+1} (T_i - T_c - \Delta T - \delta T) \quad (4-12)$$

Here T_i is the initial residual stream temperature, T_k is the boiling temperature in the k 'th stage (starting from the lowest pressure), n is the number of evaporation stages, δT is the pinch temperature difference, ΔT is the temperature difference in the isolation heat exchangers, if any and T_c is the condenser temperature.

Since the optimum boiling temperature of a single-pressure power recovery system increases with the temperature of the residual stream, the cycle efficiency also increases. This results in a lower system cost as shown graphically in Fig. IV-13. The ordinates in Fig. IV-13 represents the system installed cost at the peak utilization efficiency point. Multi-pressure systems also exhibit the same trend.

FIGURE IV-11

EFFECT OF PIPELINE ON INSTALLED COST OF OPTIMIZED DUAL-PRESSURE SYSTEMS

(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr
CONDENSER TEMPERATURE = 80°F

FIGURE IV-12

EFFECT OF WASTE-STREAM TEMPERATURE ON THE GROSS UTILIZATION EFFICIENCY OF R717 SYSTEMS

T_i = WASTE STREAM TEMPERATURE

FIGURE IV-13

**EFFECT OF WASTE STREAM TEMPERATURE ON THE INSTALLED COST
OF SINGLE-PRESSURE SYSTEMS**

(NO CONTINGENCY)

CONDENSER TEMPERATURE = 80°F

MASS FLOW RATE = 50×10^6 lb/hr

4. Effect of Condenser Temperature

The condenser temperature is determined primarily by the temperature of the available cooling medium. For wet-cooling tower applications, this is the prevailing wet-bulb temperature of the ambient air. Ambient temperature fluctuations can cause large fluctuations in the system power output and must be taken into consideration when designing a low-temperature power plant. These power plants must not be designed for a fixed heat rejection temperature, which is the recognized practice for high-temperature steam power plants. Low-temperature power recovery systems must be designed to take advantage of the lower ambient temperature during the cold seasons. Figure IV-14 shows the impact of arbitrary changes in design condenser temperature on the unit installed cost of R717 and R500 systems for a fixed residual stream temperature of 150°F under conditions of peak utilization efficiency. The effect of condenser temperature is more pronounced for the R500 system because of its high heat exchanger cost, and for the R717 system with the intermediate heat exchanger because of the IHE's cost and power impacts.

5. Effect of Pinch Temperature Difference

In earlier analyses, the pinch temperature difference in the boiler was taken arbitrarily as 5°F . An increase in the pinch temperature difference reduces the boiler/preheater heat transfer area requirements but, at the same time, causes a decline in the system efficiency. The compound effect of these two factors is shown in Fig. IV-15, which depicts the dependence of the unit installed cost of R717 and R500 single-pressure systems on the pinch temperature difference. The pinch temperature difference has a very weak influence on the unit installed cost. Its influence on the net power output is more pronounced. The results depicted in Fig. IV-15 suggest that pinch temperature differences as small as $2\text{--}3^{\circ}\text{F}$ may be economically justifiable, although difficult to achieve in practice throughout the system lifetime.

FIGURE IV-14

**EFFECT OF CONDENSER TEMPERATURE ON THE INSTALLED COST
OF SINGLE-PRESSURE SYSTEMS**

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr

(No Contingency)

FIGURE IV-15

**EFFECT OF HEAT EXCHANGER PINCH TEMPERATURE DIFFERENCE ON THE OUTPUT
AND INSTALLED COST OF SINGLE-PRESSURE SYSTEMS**

(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr
CONDENSER TEMPERATURE = 80°F

6. Effect of IHE Temperature Difference

The interposition of an intermediate heat exchanger (IHE) between the residual stream and the power recovery system results in a sharp deterioration of the system output as well as a marked increase in its cost. The comparative studies discussed in the foregoing sections were all based on an arbitrary temperature difference of 10°F in the IHE. Lower values generally increase both the unit cost of the system and the power output. However, since the IHE pumping power requirements increase with the surface area (i.e., with lower temperature difference), very low temperature differences can actually lead to lower net power output as shown in Fig. IV-16. Little advantage is gained by increasing the IHE's temperature difference beyond about 10°F or lowering it below about 8°F .

7. Effect of Size

The effect of system size rating on the unit installed cost of thermodynamically-optimized single-pressure systems is presented in Fig. IV-17. Over the output range 2-30 MW, the unit cost is a very weak function of size. This is due to an almost direct relation between heat exchanger cost and size in this limited output range. Stronger dependence on size may be observed if there is a site preparation cost that depends only weakly on the size of the system.

8. Alternate Isolation Approaches

It has been shown that the interposition of an intermediate heat exchange loop between the residual stream and the power system results in a severe economic penalty. Effective isolation of the residual stream from the working fluid can be achieved with less cost by utilizing either a buffered water/water/ammonia heat exchanger, or by pumping the residual stream to a pressure that is slightly higher than that of the boiling ammonia, then recovering most of the pumping power by means of a hydraulic

FIGURE IV-16

EFFECT OF IHE TEMPERATURE DIFFERENCE ON THE INSTALLED COST AND POWER OUTPUT OF R717 SINGLE-PRESSURE SYSTEMS

FIGURE IV-17

EFFECT OF SIZE ON THE INSTALLED COST OF SINGLE-PRESSURE SYSTEMS

(NO CONTINGENCY)

WASTE STREAM TEMPERATURE = 150°F CONDENSER TEMPERATURE = 80°F

turbine. In this manner, any leakage will occur from the intermediate water side to the ammonia side, eliminating the possibility of residual water stream contamination by ammonia. Preliminary cost estimates indicate that the buffered heat exchanger approach may reduce the unit installed cost of an ammonia system by 5-10 percent. If the second approach is adopted, the residual stream must be pressurized to a level in excess of 300 psig in the case of ammonia. This corresponds to 1.15 Btu/lb of water for pumping, allowing for a 10 percent overpressure and an 85 percent pump efficiency. Table IV-6 presents a comparison between this approach and the IHE approach for a water flow of 50×10^6 lb/hr at 150°F. The hydraulic turbine efficiency is taken as 85 percent. The pumped system produces 22 percent more power at a 16 percent unit cost savings, when the system is operated near its peak efficiency point. Thus, although the IHE approach was analyzed and used in some comparisons, the NH_3 system output and cost characteristics could be enhanced with either of these alternate approaches to isolation.

TABLE IV-6

COMPARISON BETWEEN AMMONIA ISOLATION APPROACHES

	<u>Pumping Approach</u>	<u>IHE Approach</u>
Surface area of IHE, ft ²	-	366,500
Power loss in IHE, kW (pumping)	-	1540
Cost of IHE (@14\$/ft ² installed)	-	5.13 x 10 ⁶
Pumping power, kW	16,850	-
Hydraulic turbine output, kW	12,170	-
Power loss, kW	4,680	-
Cost of pump, turbine & drive (@ 150\$/kW installed)	4.36 x 10 ⁶	-
Gross system output, kW	20,310	14,360
Net system output, kW	15,630	12,820
Total system cost, \$	20.09 x 10 ⁶	19.55 x 10 ⁶
Installed cost per net kW, \$	1285	1525

F. COOLING WATER REQUIREMENTS

Low-temperature power generation systems reject large quantities of heat to their environment. Their performance depends strongly on the availability of an adequate cooling medium to maintain a low condenser temperature. If no large supply of cooling water, such as a river or a lake, is available for a once-through cooling system, a wet mechanical draft cooling tower must be employed. Dry cooling towers or wet/dry cooling towers are unacceptable because of their high parasitic power consumption and because of their inability to maintain sufficiently low condenser temperatures. Therefore, large amounts of water will be needed either for once-through cooling or for cooling tower makeup. Cooling tower makeup water, which compensates for cooling water loss by evaporation and drift, amounts to about 10 percent of the circulating cooling water. In the absence of a river or a lake from which this water can be withdrawn, underground aquifers may be tapped to provide the necessary water, as is currently done at the Portsmouth GDP.

Figure IV-18 depicts the cooling water needs of dual pressure ammonia systems with and without the isolation heat exchanger for a range of residual stream temperatures. The net power generated in each case is also shown. The cooling water flow is several times higher than the residual stream flow. This must be taken into account when selecting the ideal location of a residual-heat-driven power generation system. All the considerations pertaining to the optimization of residual stream pipelines apply with equal force to the optimization of cooling water conduits.

FIGURE IV-18

COOLING WATER REQUIREMENTS AND NET OUTPUT OF DUAL-PRESSURE R717 SYSTEMS

CONDENSER TEMPERATURE = 80°F MASS FLOW RATE = 50×10^6 lb/h
PINCH TEMPERATURE DIFFERENCE = 5°F, --- W/IHE, — W/O IHE

G. ASSESSMENT OF ECONOMIC MERIT

The installed cost per kilowatt represents only one economic parameter by which various power recovery systems may be compared. The use of this parameter alone as an economic figure of merit may be quite misleading. This is particularly true when the residual energy power recovery system must be supplemented by grid electricity to meet a specific demand. Under these circumstances, a power recovery system which is cheap to install (low \$/KW) but poorly utilizes the residual stream may prove economically inferior to another which is more expensive to install but provides a superior utilization of the residual stream. This point is further elucidated in the following paragraphs.

The maximum power, W^0 , that can be generated from a given residual stream was given previously. If this power were purchased from the grid, the annual cost to the consumer would be

$$C_e^0 = c_e E^0 \quad (4-13)$$

where E^0 is the annual electric energy consumption corresponding to a load W^0 operating 8000 hours per annum and c_e is the unit cost of electricity in this year. The present worth of these annual expenditures is the datum against which the various systems must be compared. If a power recovery system whose net utilization efficiency is η_{un} can be installed for c \$/KW, the total initial investment would be

$$C = c \eta_{un} W^0 \quad (4-14)$$

and the amount of electric energy saved annually would be $E = \eta_{un} E^0$. The electric energy deficiency, $(1 - \eta_{un}) E^0$, must be supplied by the grid. The net present worth of this arrangement is made of two parts: the initial investment C and the present

value of a time series $C_e = c_e (1 - \eta_{un}) E^0$, in which C_e changes according to general inflation and electric energy cost escalation. The present value of the savings is given by

$$S_p = P(\eta_{un} C_e E^0) - C \quad (4-15)$$

in which $P(x)$ designates the present value of a time series x . The normalized present value of the savings can be used as an economic figure of merit, viz

$$F_e = \frac{S_p}{P(C_e^0)} = \eta_{un} - \frac{C}{P(C_e^0)} \quad (4-16)$$

which is higher for more economical systems. For a discount rate d , a differential electricity cost inflation rate i_e and an economic life of N years, this becomes

$$F_e = \eta_{un} \left(1 - \frac{c}{t c_e} \left[\frac{i (1 + i)^N}{(1 + i)^N - 1} \right] \right) \quad (4-17)$$

In this equation $(1 + i) = (1 + d)/(1 + i_e)$, c_e is the unit cost of electricity at the beginning of the economic evaluation period and t is the number of operating hours per annum. For $d = 0.1$, $i_e = 0.02$, $N = 20$ years and $t = 8000$ hours per annum, F_e reduces to

$$F_e = \eta_{un} \left(1 - 0.00126 \frac{c}{c_e^0} \right) \quad (4-18)$$

in which c is in \$/KW and c_e^0 is in KWh at the beginning of the evaluation period. It can be seen that plants costing more than $795 c_e^0$ are not economically justifiable. More specifically, for $c_e^0 = 3.3¢/\text{KWh}$ (1980), power recovery systems costing in excess of \$2623/KW (1980) are not economically justifiable. It is evident that the economic figure of merit depends very strongly on the utilization efficiency and that the most economical

configuration (highest η_{un}) is not necessarily the cheapest configuration (lowest c). Figure IV-19 displays this relationship for R500 and R717 systems at the conditions indicated in the figure. The economic advantage of dual-pressure systems is clearly evident in Fig. IV-19, as is the superiority of R717 relative to R500.

The economic figure of merit defined in the previous paragraphs is valid also when the power recovery system generates excess electric energy, provided that the system is credited for this excess electricity at the same rate as for that bought from the grid, i.e., 3.3¢/KW in the above example. However, if only a partial credit is given for the excess power, it may prove more economical to generate only as much power as needed. In this case, the most economical system will be the cheapest and the cost per kilowatt will be the appropriate economic figure of merit.

The cost per kilowatt can be used as an economic figure of merit also if the goal is to realize the highest possible rate of return (ROR) on the initial investment. Since a residual heat driven power recovery system does not consume fuel, the highest ROR corresponds to the system which has the lowest unit cost, irrespective of the utilization efficiency. The ROR has more value as an economic figure of merit in private industry. In publicly financed projects, however, present worth analysis or, equivalently, life-cycle cost is more widely employed.

FIGURE IV-19

ECONOMIC FIGURE OF MERIT

(15% CONTINGENCY; GRID ELECTRICITY AT 3.3¢/kWh)

WASTE STREAM TEMPERATURE = 150°F MASS FLOW RATE = 50×10^6 lb/hr

CONDENSER TEMPERATURE = 80°F

The simple economic figures of merit discussed above are adequate for system selection and tradeoff studies. A thorough economic assessment of competing systems must take into account project construction time, operation and maintenance costs, taxes, insurance, type of financing, etc. The inclusion of these factors would increase the life-cycle costs and reduce the ROR, particularly for an expensive system like that using R500. Thus, it may be concluded that a dual-pressure ammonia power recovery system is the choice for the utilization of low-temperature residual stream such as those available from Federal nuclear facilities.

H. POWER RECOVERY AT PORTSMOUTH GDP

About 128,000 gpm of warm water at a seasonal average temperature of 156°F is available from the X-333 process building at the Portsmouth GDP (Ref. IV-4). Some or all of this water can be used to generate power for on-site use or for feeding an industrial park to be constructed one or two miles away from the gaseous diffusion plant. The computational procedures developed in the previous section and the obtained parametric relations were used to provide preliminary data on the design, cost and performance of a power recovery system at the Portsmouth GDP. The following paragraphs give a summary of the results.

Following the arguments of the previous sections, a dual-pressure ammonia system with an isolation heat exchanger was selected as the most appropriate for the PGDP application. Although a buffered heat exchanger or a pressurized water system can be employed more economically to isolate the working fluid from the process cooling water, a separate isolation heat exchanger was chosen because it is readily available as a commercial product and is a reliable alternative. This choice also results in a more conservative design from both performance and cost perspective.

Due to the absence of an adequate supply of water for once-through condenser cooling, the power recovery system must employ wet-cooling towers. Under these conditions the ideal location of the power plant is in the immediate vicinity of the GDP to eliminate the need for an expensive pipeline. The electricity generated by the power recovery system can be: (1) used on-site to displace a fraction of the enormous power needs of the GDP; (2) sold to the grid; (3) transmitted one or two miles to the proposed industrial park site.

Operating Conditions

Before calculating the optimum design parameters of the power recovery system, it was necessary to determine the condenser operating temperature, T_c . For the prevailing average wet bulb temperature (55°F), it was estimated that the cooling tower would be able to supply the condenser with cooling water at about 67°F . Assuming a temperature rise of 10°F and a terminal temperature difference of 5°F , the condenser operating temperature was taken as 82°F .

The other system optimum design conditions were estimated using the procedures and the parametric curves of the previous sections. Figure IV-20 depicts the variation of the unit installed cost and the power output with the low-pressure boiler temperature, T_1 . The high-pressure boiler temperature was chosen so that the lowest life-cycle cost is achieved (highest economic figure of merit). This corresponds to a high-pressure boiler temperature of $125\text{--}130^{\circ}\text{F}$ and a low pressure boiler temperature of $110\text{--}115^{\circ}\text{F}$. The values $T_1 = 100^{\circ}\text{F}$ and $T_2 = 125^{\circ}\text{F}$ were chosen for subsequent design analysis (see Section V).

About 24 MW_e of power (net) can be provided at the Portsmouth GDP site using only the X-333 building flow. The installed costs are likely to be $\$1000\text{--}\$1200/\text{kW}$. Thus, the cost of this residual heat generated power would be about $2.8\text{¢}/\text{kWhr}$, which is very competitive with power generated using coal at local utilities.

FIGURE IV-20

COST AND NET OUPUT OF A DUAL-PRESSURE R717 SYSTEM AT PORTSMOUTH GDP

I. PROTOTYPE SYSTEM CHARACTERISTICS

This section describes the characteristics of a prototype power generating system which could be tested at the EAST facility. In Part J, this prototype design will be used to specify the residual heat stream flows, auxiliary equipment, instrumentation and data reduction equipment needed to perform the tests. The prototype system size and description was developed from a concept of an industrial park using the Portsmouth G.D.P. as a residual heat source (see Section V) and by using the techniques developed in this section. A Rankine cycle system was selected employing two stages of evaporation and ammonia as the working fluid. This system can provide a net output of 17 MWe.

Since this prototype was to be used to define the preliminary requirements of EAST, a decision had to be made on whether to size the EAST facility for the 17 MWe output system or some fraction (say 1/4 size or 4.25 MWe). The following paragraphs discuss the major issues in this decision.

1. Component changes resulting from a scale reduction.

The major components of the prototype power conversion systems are the evaporators, the turbogenerators and the condensers. A vertical tube, natural-circulation boiler was selected for the evaporator. An alternative horizontal-tube, pool boiler was not selected because of uncertainty as to the effect of vapor blanketing on the large number of layers of cross-section heat transfer tubes. The natural circulation boilers for the proposed four 4.25 MWe net systems for the Portsmouth CEMS are, however, near the lower limit in capacity for this type of design. The active portions of the boilers have a length/diameter ratio of 7, being approximately 5 ft in diameter and 35 ft in length. The length cannot be reduced without compromising the heat transfer rates.

Thus, a smaller capacity system would logically not employ a vertical natural circulation boiler because of the high length/diameter ratio and large shell surface area required by such a design.

No change would be involved in scale-down of the turbogenerators. The diameters of the radial turbine wheels are approximately 18 in.; they rotate at 12,000 rpm. Efficient radial turbines driving generators are in service having diameters as small as 6 in. and rpm's as high as 40,000.

There would also be no change in basic condenser design involved as long as the scale-down is less than an order of magnitude. For greater scale-down, the cooling water flow circuit would probably be changed from single pass to two pass. In summary, scale changes between the prototype and PGDP systems would involve a change in the evaporator type but probably not in the turbogenerator or condenser type.

2. Effect of scale reduction on the direct applicability of the test results.

The direct applicability of test results primarily concerns data on the turbogenerators. The efficiencies of these units are predictable from extensive industrial experience. The primary value of the EAST tests would be to confirm the mechanical design of the units, e.g. bearing and seal endurance and absence of vibration problems. These characteristics are very specific to the particular design tested. Thus, the test results on a full-size turbogenerator unit would be directly applicable to a Portsmouth GDP design as described in Section V. However, the test results from scaled-down turbogenerators would not be directly applicable.

3. Resulting cost savings for fabrication and testing due to the reduced size.

The specific (unit) cost of the power conversion unit increases only moderately as size is scaled down below 17 MWe net output (see Figure IV-17). Thus, a power plant cost (total capital) saving is almost proportional to the reduction in output that would be realized by scaling down. Additional savings would be realized in reduced test facility costs. These costs are discussed in the Test and Operations Report for EAST Facility (Ref. IV-8).

4. The usefulness of the resulting facility.

From the discussions presented in section I, EAST Facility will have a greater utilization if it is sized at the 4.25 MWe capability rather than the 17 MWe capability.

From these issues, it would be the prudent decision to scale the prototype, and hence EAST, for the 4.25 MWe capability as opposed to the 17 MWe unit. The EAST Facility will have a better utilization and a resulting industrial park at Portsmouth will suffer only a modest cost increase for a multi-unit vs. a single-unit approach.

J. TEST FACILITY SPECIFICATIONS

In the prototype tests at the EAST Facility, the power conversion system will obtain its heat input from a reactor effluent and will reject heat to the cooling water system. The performance of the intermediate heat exchanger and of cooling tower should be regarded as predictable. Thus, the EAST Facility would test the portion of the power plant between the hot water inlet to the high pressure evaporator and the cooling water inlet to the condenser cooling water pumps. This is shown schematically in Figure IV-21. The reactor would take the place of water from the intermediate heat exchanger at Portsmouth while cooling water system would take the place of water returning from the cooling towers. A hot water flow of 22,500 gpm at a nominal temperature of 146°F and a cooling water flow at a nominal temperature of 67°F would be desired. The cooling water temperature would be available only during certain times of the year; however, the operating conditions of the actual power plant at Portsmouth would vary seasonably so that during most of the year a realistic power plant operating condition could be duplicated at the EAST Facility. The hot water temperature would probably average 4° or 5°F hotter than the desired average of 146°F. It could be adjusted to the desired temperature by dilution with cooling water.

Facility electric power requirements would be mainly for pump drives, assuming that operating of the pumps independently of the turbogenerators would be desirable to achieve flexibility of testing procedures. The pumping power for hot water circulation, ammonia feed supply and cooling water supply is 508 kW if the cooling water and hot water sources are at the same elevations (see Section V). Allowing a reasonable margin for miscellaneous requirements, a required power of 600 kW is estimated.

Service pumps or a throttled gravity circulation arrangement should be provided for the hot water since the power plant

SCHEMATIC FLOW DIAGRAM OF THE GEMS POWER GENERATION SYSTEM

FIGURE IV-21

circulating pumps are designed for the additional heat loss in the intermediate heat exchanger and would require more electric power than necessary for the EAST Facility arrangement. Circulating pump performance is predictable and could be verified by separate tests not in combination with tests of the power plant.

For testing the power plant without the intermediate heat exchangers, a plan area of approximately 40 ft by 80 ft would be required. The maximum vertical height above grade of the power plant is 57 ft. Allowing clearance for removing vessel covers, a clear height of about 70 ft would be necessary. The cooling water pumps require a pit about 28 ft below grade for their intakes.

The heaviest components are the condensers and evaporators. They would probably be hoisted into place by a portable crane of 90 ton capacity, available when required. A permanent traveling crane should be made available for lifting the turbines, generators, pumps and vessel heads. The 3600 rpm generator, driven by the high pressure turbine, appears to be the heaviest of these components, requiring a crane capacity of 5 tons.

The test program should employ sufficient instrumentation to measure the performance of each component as well as of the overall system, plus instrumentation which would warn in advance of component malfunctioning. Table IV-7 summarizes the requirements to test this power system at EAST. Table IV-8 lists the temperature, pressure, flow, speed, power and trouble shooting measurements required.

TABLE IV-7

POWER GENERATION REQUIREMENTS - EAST FACILITY

HEATED WATER

Flow rate	22,500 GPM
Temperature	146°F
Source	Reactor Coolant
Required pressure	15 psig

COOLING WATER

Flow rate	72,000 GPM
Temperature	70°F
Source	Par Pond
Required pressure	5 psig

TEST CELL SPECIFICATIONS

Length	80 ft.
Width	40 ft.
Height	25 ft.
Pit Depth	30 ft.
Crane capacity	5 tons

ADDITIONAL FLUID HANDLING & STORAGE

Ammonia (R717)
Lubricating Oil

ELECTRICAL POWER REQUIRED

350 kW (700 kW)

AUXILIARY REQUIREMENTS

Grid Connection
(Induction Gen.)
25 Tons Space Cooling

TABLE IV-8

POWER CONVERSION TEST MEASUREMENT REQUIREMENTS

Location	<u>Temperature</u>		<u>Pressure</u>		<u>Vapor Quality</u>		<u>Flow Rate</u>	<u>Speed</u>	<u>Power Output</u>	<u>Power Input</u>	<u>Voltage</u>	<u>Current</u>	<u>Fluid Level</u>	<u>Vibration Amplitude</u>
	<u>Inlet</u>	<u>Outlet</u>	<u>Inlet</u>	<u>Outlet</u>	<u>Inlet</u>	<u>Outlet</u>								
Condensate Pumps (2)	X		X	X			X	X		X				
Feed Pumps (2)	X		X	X				X		X				
Feed Heater (1)														
Ammonia Water	X	X	X	X									X	
Evaporators (2)	X	X	X	X										
Ammonia Water	X	X	X	X		X	X							
Turbines (2)														X
Ammonia Lube Oil	X	X	X	X	X	X		X						
Condensers (2)													X	
Ammonia Water	X	X	X	X										
Cooling Water Pumps (2)			X	X			X			X				
Alternators (2)									X		X	X		

K. REFERENCES

- IV-1 Residual Energy Applications Program, Task 1, Waste Heat Resource/ Application Analysis, UTRC Report R79-954787-1, October 1979.
- IV-2 Technology Data Base-Evaluation of Waste Recovery Systems, UTRC Report R77-952642-1, January 1977.
- IV-3 Preliminary Assessment of Waste Energy Recovery from Savannah River Plant, Technology Options Characterization - Power Generation, Heating and Cooling. NTIS #DOE/EC/1021-1 Vol. 1 - Vol. 3.
- IV-4 O. E. Balje: A Study on Design Criteria and Matching of Turbomachine: Part A - Similarity Relations and Design Criteria for Turbines, Journal of Engineering for Power, Trans. ASME, Series A, Vol. 84, No. 1, January 1962, p. 83.
- IV-5 W. M. Ronsenow and J. P. Hartnett: Handbook of Heat Transfer, McGraw Hill, New York, 1973.
- IV-6 ASHRAE Handbook of Fundamentals, 1972.
- IV-7 S. L. Mittra and J. W. Tester: Geothermal Energy as a Source of Electric Power, MIT Press, Cambridge, MA, 1976.
- IV-8 "Test and Operations Report", part of a systems test and evaluation study for the Residual Energy Applications Program, prepared under U.S. Dept. of Energy Contract DE-AC09-77ET-12866, by S.C. Energy Research Institute, DOE/ET/12866-4.

APPENDIX IV - A

THERMODYNAMIC EFFICIENCIES

The maximum power, $\dot{W}^0$, that can be generated from a fluid stream, $\dot{m}$, entering an energy conversion system at a thermodynamic state i , is given by

$$\dot{W}^0 = \dot{m} \left[(h_i - h_o) - T_o (S_i - S_o) \right]$$

in which h and s are the specific enthalpy and entropy, respectively and the subscript "o" refers to a thermodynamic dead state, the ambient temperature and pressure. The term $\dot{m}(h_i - h_o) = \dot{Q}^0$ represents the maximum amount of heat that can be extracted from the stream if it were cooled from its initial state to the ambient temperature. Therefore, an ideal cycle efficiency may be defined as

$$\eta_c^0 = 1 - T_o \frac{(S_i - S_o)}{(h_i - h_o)}$$

which differs from the Carnot efficiency

$$\eta_c^c = 1 - \frac{T_i}{T_o}$$

representing the efficiency of an infinitesimal Carnot cycle operating between T_i and T_o . The ideal cycle efficiency η_c^0 is in essence an integration of the efficiencies of infinitesimal Carnot cycles traversing the temperature range T_i to T_o . Although it is theoretically possible to construct a Carnot cycle whose efficiency is η_c^c , a system that can achieve this efficiency will extract no heat from the residual stream as the fluid will enter and leave such a system with no temperature change; hence no work will be generated by this system.

Actual energy conversion systems are burdened by irreversibilities, both internal and external, that degrade their output, sometimes substantially. The most significant sources of

irreversibility in a power generation system are those associated with heat transfer across a finite temperature difference and those associated with fluid friction. The power output of an actual system is always lower than that of an ideal system by an amount dictated by the various dissipative mechanisms in the system, that is

$$\dot{W} < \dot{W}^o$$

which is in fact an alternative statement of the second law of thermodynamics. The degree of closeness to the thermodynamic ideal can be expressed in terms of $\dot{W}$ and $\dot{W}^o$ by means of a utilization efficiency

$$\eta_u = \frac{\dot{W}}{\dot{W}^o}$$

Introducing the cycle thermal efficiency

$$\eta_c = \frac{\dot{W}}{\dot{Q}}$$

where $\dot{Q}$ is the amount of heat extracted from the stream in the actual system, and recalling the definition of η_c^o and $\dot{Q}^o$, the utilization efficiency can be expressed as

$$\eta_u = \eta_c \dot{Q} / \eta_c^o \dot{Q}^o$$

For a sensible heat stream leaving the power generation system T_r , this reduces to

$$\eta_u = \eta_c (T_i - T_r) / \eta_c^o (T_i - T_o)$$

which indicated that an improvement in the cycle efficiency, η_c , would result in an improved utilization efficiency only if it were not achieved at the expense of a higher fluid rejection temperature. Improvement in the cycle efficiency resulting from increasing the cycle heat addition temperature, all other factors remaining unchanged, results also in an increase in the fluid

rejection temperature. Therefore, unlike the monotonically increasing cycle efficiency, the utilization efficiency will exhibit a maximum at a cycle heat addition temperature between T_i and T_o .

RANKINE CYCLE WORKING FLUIDS

Fluid	Formula	Mol. Wt.	Freezing Temp., °F	Critical Temp., °F	Critical Pressure, Psia	Approximate Stability Temp., °F	Pressure At 100°F, Psia	Cost ^{oo}	Auto Ignition Temp., °F	Comments ^{ooo}
Water	H ₂ O	18	32	705	3206	-	1.0	A	-	D
Ammonia	NH ₃	17	-108	271	1657	-	212	A	-	D
n-Butane	C ₄ H ₁₀	58.1	-217	305	550	450	52	A	761	D
R 11	CCl ₃ F	137.4	-168	353	640	350*	23.5	A	-	D
R 12	CCl ₂ F ₂	120.9	-252	234	597	350*	130	A	-	D
R 21	CHCl ₂ F	103.2	-211	353	740	-	40	B	-	D
R 22	CHClF ₂	86.5	-256	205	722	375*	210	A	-	D
R 113	C ₂ Cl ₃ F ₃	187.4	-31	417	499	300*	10.5	A	-	D
R 133a	C ₂ H ₂ ClF ₃	113.5	-156	306	590	400*	45	-	-	D
R 216	C ₃ Cl ₂ F ₆	220.9	-194	356	400	-	15.5	-	-	D
Fluorinol 60	C ₂ H ₂ F ₃ OH	67.2	-38	483	1053	625	2.8	B	-	D
Fluorinol 85	C ₂ H ₂ F ₃ OH	87.7	-80	453	840	625	3.1	B	-	D
Toluene	C ₇ H ₈	92.1	-139	609	617	750	1.0	A	900	D
Benzene	C ₆ H ₆	78.1	42	552	706	800	3.2	A	928	D
Hexafluorobenzene	C ₆ F ₆	186	-67	470	474	750	3.7	C	-	D
Fluorobenzene	C ₆ H ₅ F	76.1	-44	543	656	700	2.7	C	1000	D
Pyridine	C ₅ H ₅ N	79.1	-44	657	817	700	0.7	A	900	D
Thiophene	C ₄ H ₄ S	84.1	-40	554	792	550	2.7	B	755	D
Iso-Butane	C ₄ H ₁₀	58.1	-256	275	529	450	74	A	864	E
Methanol	CH ₃ OH	32	-145	454	1154	-	4.4	A	727	E
Ethanol	C ₂ H ₅ OH	46	-170	470	926	-	2.2	A	685	E
Fluorinol 100	C ₂ H ₂ F ₃ OH	100	-47	440	715	600	3.1	B	-	E
" 90	"	91.8	-53	453	803	600	3.1	B	-	E
" 80	"	83.6	-66	462	874	625	3.1	B	-	E
" 75	"	79.5	-53	466	913	625	3.0	B	-	E
" 70	"	75.4	-50	472	953	625	2.9	B	-	E
" 65	"	71.3	-43	477	1000	625	2.9	B	-	E
" 55	"	63.1	-33	488	1090	625	2.8	B	-	E
" 50	"	59.0	-28	498	1175	625	2.8	B	-	E
" 45	"	54.9	-25	511	1279	625	2.7	B	-	E
Ethylenediamine	C ₂ H ₄ N ₂	60.1	47	604	-	-	0.5	B	700	E
Acetone	C ₃ H ₆ O	58.1	-139	465	691	-	7.4	A	1000	E
Ethylether	C ₄ H ₁₀ O	74.1	-177	382	523	-	16.5	A	336	E
Perfluoroether	C ₁₀ F ₂₂ O	570	-121	470	172	-	-	-	-	E
Biphenyl	C ₁₂ H ₁₀	154	156	929	477	-	Solid	A	1000	F
Monochlorobenzene	C ₆ H ₅ Cl	112.6	-49	678	656	630	0.4	A	1180	E
Pentafluorobenzene	C ₆ HF ₅	168	-54	447	363	750	3.7	C	-	G
Octafluorotoluene	C ₇ F ₈	236	-108	-	-	-	-	-	-	E
Octafluorocyclo-butane	C ₄ F ₈	200	-42	240	392	-	60	-	-	E

^{oo}Oil free in stainless steel

^{oo}Cost data in 1000 lb lot purchases

A < \$2.00 per lb

B \$2-10.00 per lb

C > \$10.00 per lb

^{ooo}Basis for detailed evaluations used in this study

D Mollier data available, performance estimated and included in report

E Insufficient Mollier data available

F Solid at condenser temperature selected

G Performance very similar to hexafluoro-

APPENDIX IV - C

HEAT TRANSFER COEFFICIENTS

Vapor Generator

This appendix contains background information used to substantiate the heat transfer data used in the power generation system analysis.

Data from the Ocean Thermal Energy Conversion (OTEC) Program

Tests of a pool boiler employing a porous surface on the boiling side (Ref. IV-C-1) furnish data on one potential type of evaporator. At a heat flux rate of 4600 Btu/ft^2 , a coefficient of $4800 \text{ Btu/ft}^2 \text{ hr}^\circ\text{F}$ was measured on the boiling ammonia side. A correction for heat flux rate (film temperature differential) should be applied in the analyses described herein since the heat transfer rates are approximately double those for OTEC. Small-scale ammonia boiling tests on porous surfaces (Ref. IV-C-2) indicated that the coefficients varied approximately as heat flux to the $1/4$ power. Therefore, at typical power system design conditions, a rate of $5700 \text{ Btu/ft}^2 \text{ hr}^\circ\text{F}$ was used. The tests of Ref. IV-C-1 also indicated an overall U of 785. This value is not believed to be usable directly in the current analyses since it is based on clean (nonfouling) conditions and on lower water side velocities than would be used in the vapor generators considered in this study.

Forced Convection Boiler Calculations for SRP

The Ref. IV-C-3 study employed a vertical forced-convection ammonia boiler. In predicting the performance of this design, the well-known Chen equations (Ref. IV-C-4) were applied. The predicted coefficient on the ammonia side was approximately $4000 \text{ Btu/hr ft}^2 \text{ }^\circ\text{F}$ for forced convection boiling on the outside of 0.5" diameter vertical tubes. The predicted overall coefficient of

about 900 is not directly applicable because a high water velocity (14 fps) was employed to utilize an available gravity head.

Estimation of Boiling Coefficient for Power System Conditions

The ammonia-side coefficient for forced convection was, conservatively, selected because of uncertainty as to the cost and availability of porous coated tubing in large quantities. In addition, the effect of vapor blanketing on the performance of very large pool boilers was unknown (in Ref. IV-C-1 only a 4' diameter bundle was tested). A water-side velocity of 10 fps, corresponding to a film coefficient of 3000 Btu/hr ft²°F, in 1/2 dia. tubes was assumed. In estimating the fouling factor, it was assumed that the water was treated and that an automatic cleanup system such as the AMERTAP system could be employed. Thus, a fouling factor of 0.0004 was assumed. With these assumptions, an overall U of 800 was estimated for ammonia.

To scale the coefficients for other working fluids, the Chen equation for the pool boiling fraction of the total boiling coefficient was employed. According to the Chen equation

$$h = 0.00122 \frac{k^{0.29} c_p^{0.45} \rho_f^{0.49} g^{0.25}}{\sigma^{0.5} \mu^{0.29} h_{fg}^{0.24} \rho_g^{0.24}} \Delta T_{SAT}^{0.24} \Delta P_{SAT}^{0.75} \quad (1)$$

Assuming the same temperature differential for each fluid, the film coefficients for other fluids should have the following ratio to that for ammonia according to the Chen equation

$$\frac{h}{h_{NH_3}} = \frac{\left(\frac{k}{k_{NH_3}}\right)^{0.29} \left(\frac{c_p}{c_{pNH_3}}\right)^{0.45} \left(\frac{\rho_f}{\rho_{fNH_3}}\right)^{0.49} \left(\frac{(\Delta P/\Delta T)}{(\Delta P/\Delta T)_{NH_3}}\right)^{0.75}}{\left(\frac{\sigma}{\sigma_{NH_3}}\right)^{0.5} \left(\frac{\mu}{\mu_{NH_3}}\right)^{0.29} \left(\frac{h_{fg}}{h_{fgNH_3}}\right)^{0.24} \left(\frac{\rho_g}{\rho_{gNH_3}}\right)^{0.24}} \quad (2)$$

Table C-1 shows the values of the boiling coefficient ratio for representatives of the Freon family (R11 and R500) and the overall U for the same fluids. The coefficients for all other fluids fall into a much lower range than that of ammonia, 200-300 as compared to 800. Since property data are not highly accurate and the use of the Chen pool boiling equation is only an approximation, a uniform value of 300 Btu/hr ft²°F was assumed in the parametric study for the Freons and a value of 800 Btu/hr ft²°F was assumed for ammonia.

Condenser

Data from the OTEC Program

Significant information for condensing coefficients from the OTEC program include data for ammonia condensing on wire-wrapped horizontal tubes (Ref. IV-C-5) and for ammonia and other refrigerants condensing on fluted vertical tubes (Ref. IV-C-6 and IV-C-7) was reviewed. Vertical tube condensers were not considered for the power generation systems because of the high gravity pumping head required for the cooling water. However, the comparative data for different refrigerants provides an indication of the probable variation in heat transfer coefficients. The condensing coefficient measured in Ref. IV-C-5 was 5180 Btu/hr ft²°F at a heat flux of 4300 Btu/ft² hr°F. Reference IV-C-5 also notes that the Nusselt equation would predict a coefficient of 2600 for unenhanced tubes. The Nusselt equation indicates that the heat transfer coefficient should vary inversely as the 1/3 power of the heat flux rate. Corrected to a typical system condition heat flux of 5300 Btu/hr ft²°F, the enhanced tube coefficient should be 4860 Btu/hr ft²°F and the bare tube coefficient 2440 Btu/hr ft²°F. The overall U of 620 Btu/hr ft²°F in Ref. IV-C-5 is not usable directly because it represents a clean (unfouled) condition with lower water velocities than would be used in a representative design.

TABLE C-1

COMPARATIVE EVAPORATION FILM COEFFICIENTS FOR SELECTED WORKING FLUIDS

(Based on Vertical Forced Convection Ammonia Boiler Scaled for Fluids
Other than Ammonia By the Chen Pool Boiling Equation)

Fluid	Ratio of Boiling Coefficient to That of Ammonia	Estimated Boiling Coefficient	Estimated Overall Heat Transfer Coefficient
Ammonia	1.00	4,000 $\frac{\text{Btu}}{\text{Hr Ft}^2\text{°F}}$	800 $\frac{\text{Btu}}{\text{Hr Ft}^2\text{°F}}$
RTI	0.063	252	205
R500	0.062	248	202

The data for a vertical tube having 48 flutes (tube F in Refs. IV-C-6 and -7) gave the following measured equivalent condensing coefficients for several fluids at a heat flux of 5300 Btu/hr ft²:

Ammonia	7000 Btu/hr ft ² °F
R114	360
R113	400
R11	600
R115	260

Estimation of Condensing Coefficients for REAP Conditions

Since the coolant could be untreated water from rivers, a fouling factor of 0.001 appears to be appropriate. A lower factor, 0.0004 was assumed in the Ref. IV-C-3 study because actual data on exchangers using Savannah River water indicated that this value was appropriate. The sand content of the water appears to reduce the fouling factor in the Ref. 3 study case, a situation not believed to exist in the general case.

With a fouling factor of 0.001, the reduced thermal resistance offered by enhanced tubes is not worthwhile and the coefficient for bare tubes is assumed. In the Ref. IV-C-3 study, enhancement was worthwhile because of the lower fouling factor of 0.0004. Using the Nusselt equation and assuming banks of two tubes separated by drainage gutters, a condensing coefficient of 2500 Btu/ft² hr°°F is estimated for ammonia. Assuming 1" OD tubes with a water velocity of 10 fps and a fouling factor of 0.001, an overall U of 500 is estimated for ammonia.

$$h = 0.728 \left[\frac{g \rho_f (\rho_f - \rho_g) k^3 h_{fg}}{n D \mu \Delta T} \right] \quad (3)$$

For other fluids, the Nusselt equation is used to scale the predicted heat transfer coefficient.

Assuming the same geometry and temperature differential, the ratios of the condensing coefficients for the other fluids to that of ammonia should be as follows:

$$\frac{h}{h_{\text{NH}_3}} = \frac{\left(\frac{\rho_f}{\rho_{\text{fNH}_3}}\right)^{1/2} \left(\frac{k}{k_{\text{NH}_3}}\right)^{3/4} \left(\frac{h_{fg}}{h_{fg \text{ NH}_3}}\right)^{1/4}}{\left(\frac{\nu}{\nu_{\text{NH}_3}}\right)^{1/4}} \quad (4)$$

Table C-2 shows the values of the condensing coefficient ratio for water, R11 and R500 and the overall U for the same fluids. It is apparent that the overall U for ammonia and water and those for the other fluids fall into two separate groups with values of 400-600 and 200-300.

The scaling procedure is not sufficiently accurate to justify fine distinctions between the heat transfer performance of the various fluids. Allowing for possible deterioration of the U values due to noncondensable gases, a uniform value of 450 was assumed for water and ammonia and a uniform value of 200 for the other fluids.

TABLE C-2

COMPARATIVE CONDENSER FILM COEFFICIENTS FOR VARIOUS FLUIDS

(Based on a Smooth Horizontal Tube Ammonia Condenser Scaled
for Other Fluids by the Nusselt Equation)

Fluid	Ratio of Condensing Coefficient to That of Ammonia	Estimated Condensing Coefficient	Estimated Overall Heat Transfer Coefficient U
Ammonia	1.00	2500 $\frac{\text{Btu}}{\text{Hr Ft}^2\text{°F}}$	500 $\frac{\text{Btu}}{\text{Hr Ft}^2\text{°F}}$
Water	1.41	3530	530
R11	0.172	430	248
R113	0.168	420	245
R500	0.230	575	291

REFERENCES

- IV-C-1 Thomas, Anthony, et al.: Performance Tests of 1 MWt Shell-and-Tube Heat Exchangers for OTEC, Conference on Ocean Thermal Energy and for the 80's, Washington, D.C., June 19-22, 1979, Vol. 1.
- IV-C-2 Czikk, A. M., H. D. Fricke and E. N. Canic: Enhanced Performance Heat Exchangers, Proceedings of the Fourth Annual Conference on Ocean Thermal Energy Conversion, University of New Orleans, March 22-24, 1977.
- IV-C-3 Chalfant, A. I., A. M. Landerman and J. J. Sangiovanni: Preliminary Assessment of Waste Energy Recovery from the Savannah River Plant, Task Report on Technology Options, Characterizations, Contract EC-77-C-09-1021-SUB-005, June, 1978.
- IV-C-4 Collier, John G.: Convective Boiling and Condensation, McGraw Hill, 1972.
- IV-C-5 Yung, David T. et al.: OTEC Performance Tests of the Union Carbide Enhanced-Tube Condenser, Argonne-National-Laboratory Report No. ANAL-OTEC-PS-2, May 1979.
- IV-C-6 Combs, S. K.: Experimental Data for Ammonia Condensation on Vertical and Inclined Fluted Tubes, Oak Ridge National Laboratory Report No. ORNL 5488, January 1979.
- IV-C-7 Combs, S. K., G. S. Mailen and R. W. Murphy: Condensation of Refrigerants on Vertical Fluted Tubes, Oak Ridge National Laboratory Report No. ORNL/TM-5848, August, 1979.

SECTION V

CENTRAL ENERGY MANAGEMENT SYSTEM

A. INTRODUCTION

When a group of industrial plants are located in an industrial park, it is possible to provide their diverse electric and thermal energy requirements efficiently by means of a single facility--a Central Energy Management System (CEMS). A central energy management system could possibly employ cogeneration systems within a single building instead of numerous on-site boilers, substations, and possibly small power generation units. Without a CEMS, the thermal needs of each industrial plant would be supplied by a separate on-site industrial boiler while power needs would be supplied through the grid by an electric utility central power plant.

Since the analyses described in Sections III and IV show that GDP rejected heat at 150°F could be used economically to generate industrial thermal energy (i.e., 300°F steam) and electric power, the CEMS concept was considered for a hypothetical industrial park located near a Federal facility. The study objective is to estimate the synergistic effect of utilizing residual heat for an industrial park CEMS and to illustrate its use by providing a conceptual design for a park located near the Portsmouth GDP.

This section presents the results of this analysis. It begins with a brief review of the residual heat characteristics at the Portsmouth GDP, then defines the thermal and electric needs of a hypothetical industrial park that would be compatible with the raw material, industrial, labor force and economic structure in the selected area. The remainder of the section is divided into three parts. Part D contains a brief description of the economic analyses used to select the heat pump and power generation system used in the CEMS. In Part E, the components,

operating features and preliminary conceptual design of the selected heat pump and power generation systems are presented. Part F contains a summary of the energy requirements for the CEMS and brief assessment of the energy savings that result from the use of the Portsmouth residual heat and CEMS facility.

B. CONCLUSIONS

1. A central energy management system located near a typical Federal gaseous diffusion plant could economically recover reject heat to provide the thermal and electrical requirements of a representative industrial park. For example, by utilizing the 128,000 gpm residual hot water stream from a selected building at the Portsmouth GDP and through careful system optimization, it is possible to cost-effectively provide about 400×10^6 Btu/hr of 300°F steam and 17 MWe for use in an industrial park containing industries indigenous to the area.

2. The actual thermal and electric output provided by the CEMS can be varied over a wide range and matched to the industrial park constituent thermal and electric needs. The optimum thermal and electric split would be a strong function of the prevailing fuel and electric energy costs and other site-specific factors.

3. This CEMS alone could save as much as 3×10^{12} Btu/yr of primary fuel energy relative to conventional power plants and industrial boilers. This is equivalent to a savings of 520,000 barrels of oil annually.

C. CEMS SITE DESCRIPTION AND REQUIREMENTS

The following paragraphs describe the characteristics of the Portsmouth GDP residual heat source used in the CEMS analysis and the aggregate electrical and thermal requirements of typical plants that might locate in an industrial park nearby. A more detailed description of Portsmouth can be found in Section II of this report. The area surrounding the Portsmouth GDP is a particularly attractive site for a cost-effective industrial park development using the CEMS concept. Within a few miles of the GDP, large areas of flat land suitable for industrial use exist. Abundant cooling water is available from underground aquifers and the Scioto River. Railroads, major highways and the Ohio River are nearby for distribution of finished product; a wide variety of raw materials are readily available.

1. Portsmouth GDP Residual Heat Source

Residual heat in the form of cooling water is discharged from three buildings at the Portsmouth GDP to adjacent cooling towers. The largest and highest temperature residual heat streams come from Building 333 located at the northern edge of the complex. The water flows in a closed loop and enters the cooling tower at 156°F. The flow rate varies with the ambient temperature but has an annual average of 128,000 gpm. Although the residual heat source is near the periphery of the site, a 1-1/2 mile pipeline with a returning line will be required to transmit the water to the potential industrial park site.

2. Industrial Park Energy Requirements

Although the area near the Portsmouth GDP is largely rural, several industries in the region produce paper, wood products, and a variety of chemicals. Since Ohio is a large producer of corn and the Ohio River is available for low-cost transportation, alcohol production from grain has been of interest to the state and local governments. Drawing upon the existing and potential

industrial interest in the Portsmouth region, the industrial park was determined to have a total thermal energy requirement of about 400×10^6 Btu/hr of steam at 300°F and below.

The electric power requirement for the plants to be served by the CEMS total about 29 MW. Thus, the thermal and electrical energy requirements to be met by the CEMS are 400×10^6 Btu/hr of 300°F or lower steam and 29 MW of electrical power.

D. SELECTION OF CEMS HEAT PUMP AND POWER PLANT SYSTEMS

The selection of the optimum heat pump and power plant sizes for the Portsmouth industrial park depend on the design requirements and economics of both the heat pumps and the power recovery systems. The analyses presented in Sections III and IV indicate that to meet the total demands of the industrial park (400×10^6 Btu/hr) by means of gas-turbine-driven heat pumps will require about 37,600 gpm of warm water (at 156°F) to produce 300°F steam. To meet the entire electric power demand of the industrial park by means of a dual-pressure ammonia power recovery system would require in excess of 155,000 gpm of warm water (at 156°F). This is 20 percent higher than the 128,000 gpm of warm water produced by the Portsmouth GDP. A three- or four-stage ammonia power recovery system could produce about 15 percent more power or a total of 27 MW; this is still not enough to meet the electric demand for the industrial park. The combined warm water needs of the heat pumps and the power plant could exceed the available flow by 40 to 50 percent.

Due to the limited flow of warm water produced by the GDP, the flow must be divided between the heat pump and the power plant in such a manner as to yield the highest economic benefit. Thermal and/or electrical energy deficiencies must be supplied from conventional sources, i.e., an industrial boiler for the former and grid electricity for the latter.

A detailed economic analysis to estimate the optimum split of the residual stream based on highest ROI was made. Table V-1 gives the assumptions used in this economic analysis. The cost and fuel consumption of the heat pump system and the cost and power output of the power recovery system were taken from the results in Sections III and IV. Due to the adverse effect of the pipelines on the power recovery systems, the power plant should be located in the immediate vicinity of the GDP. The heat pump, however, should be located at the industrial park to avoid expensive high-temperature thermal distribution systems. This

TABLE V-1

ECONOMIC ASSUMPTIONS

Construction start	1980
Operation start	1982
System life, years	20
Discount rate, percent	10
O&M (% of capital cost), percent	2.25
Misc. expenses (% of capital cost), percent	2.0
General inflation rate, percent	6
Cost of fuel (gas) \$/Btu (1980)	3.0
Fuel cost escalation rate, percent/yr	10
Cost of electricity, ¢/kWhr (1980)	3.3
Electricity cost escalation rate, percent	8

requires two legs of pipeline extending from the GDP to the industrial park and back to the GDP. The calculations were performed for a distance of 1.5 miles.

Figure V-1 depicts the before-tax rate of return (ROR), life-cycle cost and payback period for a range of heat pump warm water flow extending from zero (no heat pump) to the full amount needed to meet the total thermal demand of the industrial park (37,600 gpm). It can be seen that the variations in all three economic parameters is relatively small. The most economical split corresponds to a heat pump system delivering 100 percent of the 400×10^6 Btu/hr demanded by the industrial park (i.e., using 37,600 gpm of water) and a power recovery system using the balance (i.e., 90,400 gpm) to generate 17.3 MW of electric power. This combination would achieve a rate-of-return in excess of 28 percent (before tax) with a payback period of less than 4-1/2 years.

The rate of return is only one of several criteria that must be considered in selecting the optimum configuration. Reliability and flexibility of operation, for example, may necessitate the selection of a system that does not yield the highest rate of return. The final selection must take into account these and other criteria; some may not be quantifiable. Results of analyses shown in Fig. III-10 indicate that above a capacity of about 150×10^6 Btu/hr the ROI of a single heat pump system is relatively constant. Maximum ROI (before tax) is achieved for a heat pump size of about 400×10^6 Btu/hr, but a single heat pump system would provide no back-up capability and could impair reliability requirements of the CEMS. Substantially sized separate systems should be utilized. Smaller units might also allow a more orderly growth in CEMS capacity as the various industrial plants are phased into the industrial park complex. Therefore, the size of the heat pump systems for the Portsmouth CEMS application was selected as 200×10^6 Btu/hr. Two systems

FIGURE V-1

EFFECT OF FLOW SPLIT ON THE ECONOMICS OF A CENTRAL ENERGY MANAGEMENT SYSTEM AT THE PGDP

TOTAL FLOW 128,000 gpm
DEMAND 400×10^6 Btu/hr + 29 MW_e

would be required to meet the total 300°F steam thermal requirements.

E. DESCRIPTION OF HEAT PUMP AND POWER GENERATION SYSTEMS

The CEMS for the Portsmouth GDP would consist of two separate facilities: the power generation system located near the existing GDP cooling towers and the heat pump system housed 1-1/2 miles away at the hub of the industrial park energy distribution system. The pertinent features of the heat pump and power generation systems are described in the following paragraphs. Conceptual layout drawings of these systems are also presented. Descriptions of the required energy distribution system buildings and site characteristics are not provided since the effort to define them was beyond the scope of this preliminary study.

1. Heat Pump System

The economic analysis conducted to define the optimum heat pump size showed that maximum ROI is achieved with a single unit providing the entire 400×10^6 Btu/hr required by the industrial park. Even though the use of multiple, smaller units to provide this same total output would result in slightly lower ROI, such an approach was considered preferable on the basis of maintenance and reliability considerations. A reasonable compromise between ROI, maintenance, load growth, etc., is the use of two 200×10^6 Btu/hr units. Each unit will use approximately 18,800 gpm of GDP water at 156°F (and recover 96×10^6 Btu/hr from the GDP residual water). Conceptual layout drawings showing pertinent physical dimensions are presented in Figs. V-2 and V-3 for each unit. Table V-2 summarizes the unit specifications. Each heat pump system can be constructed from commercially available components built to order for the industrial market. However, the compatibility of the working fluids and other features used in commercial hardware at elevated temperatures would have to be verified prior to final equipment selection.

FIGURE V-2

ELEVATION VIEW OF HEAT PUMP SYSTEM

FIGURE V-3

END VIEW OF HEAT PUMP SYSTEM

TABLE V-2

HEAT PUMP SYSTEM MAJOR COMPONENT SPECIFICATIONS AND AVAILABILITY

200 x 10⁶ Btu/hr Output (Two Required for Portsmouth CEMS)

<u>Component</u>	<u>Capacity</u>	<u>Approximate Size</u>	<u>Availability</u>
Gas Turbine	18,000 hp	7 x 7 x 22 ft	Select from available models
Waste Heat Boiler	63 x 10 ⁶ Btu/hr Steam Pressure $\geq$ 50 psig	10 x 18 x 40 ft	Built to order
Two-Stage R11 Compressor (Interstage side stream nozzles)	29,000 cfm	7 x 6 x 6 ft	Built to order
Two-Stage Steam Com- pressor	30,000 cfm	7 x 6 x 6 ft	Built to order (many con- ventional designs available)
Gear Box	10,000 hp 3,000 rpm in/12,000 rpm out	6 x 6 x 6 ft	Catalog item
R11 Evaporator (Flooded, 2 tube passes)	96 x 10 ⁶ Btu/hr, 50 psig max, ~18,800 gpm	8 dia. x 40 ft (32 ft tubes)	Conventional, built to order by industrial refrigerant equipment mfgs.
R11-Steam Condenser/ Boiler (2 tube passes)	138 x 10 ⁶ Btu/hr 300 psig max	8 dia x 5 ft (40 ft tubes)	Conventional reboiler type built to order
R11 Flash Collector Tank	--	4 dia x 10 ft	Standard chiller component

The physical arrangement of the heat pump components, shown in Figs. V-2 and V-3, indicates that each unit occupies a plan area of 40 ft x 70 ft (2800 ft²) exclusive of access and work areas. The maximum height above grade is approximately 40 ft and occurs at the top of the residual heat boiler. The height required over the heat pump heat exchangers is approximately 27 ft. The major components of each unit include two large heat exchangers (i.e., a residual-water-to-Rll evaporator and an Rll-to-steam condenser/boiler), two compressors (one Rll and one steam), a gas turbine driver and a gas turbine residual heat boiler. An additional number of auxiliary components are utilized, including an Rll flash tank, a gearbox and lubrication systems for the compressors.

Rll vapor is generated at 130°F in a horizontal flooded type evaporator by the heat recovered from the 156°F GDP residual water. A refrigerant vapor from the evaporator is partially compressed in the first stage of a two-stage centrifugal compressor which is directly driven by the gas turbine engine. Additional vapor from the flash tank (or economizer) is introduced to the compressor between stages; the combined flow is compressed and then directed to the condenser/boiler. The condensing process at 268°F generates steam at 258°F on the shell side. Compression of the saturated steam in a two stage compressor provides steam with a 300°F saturation temperature. A step-up gear between the two compressors matches speeds. The hot exhaust gases from the gas turbine driver are directed horizontally into the bottom of a residual-heat steam boiler capable of producing steam at 300°F or higher. The boilers are a once-through, water-tube type, although other commercially available configurations might be selected in the final analysis.

All of the major components are predicated on the use of commercially available hardware designs. Industrial equipment, although listed as a catalog item, is usually built one at a time and can often be modified and designed to satisfy customer specifications. For example, special lubrication systems could

be utilized to ensure fluid compatibility. In addition, the use of steel tubing would be specified for the condenser/boiler.

Two heat pump systems, of the design described above, would be required in the Portsmouth CEMS to provide the full 400×10^6 Btu/hr of heat for the industrial park. Two such units placed along side each other would occupy a total floor area approximately 85 ft x 70 ft. The CEMS building area would have to be somewhat larger to provide suitable access to the units space for normal maintenance operations, space for storage of spare parts and a control room.

2. Power Systems

Power requirements for the industrial park controlled by the CEMS are projected to be 29 MW, exceeding the power generation capability of the residual heat source. The use of all of the available residual heat not employed for heat pumping is desirable so long as power from the residual heat is more economical than purchased power. The economic analysis of Section IV showed that the power from the 156°F residual heat source could be produced more economically than purchased power. Thus, after using 37,600 gpm for heat pumping, the remaining 90,400 gpm can be used for power generation in a dual-pressure ammonia system.

Efficiency and cost considerations indicate there is an advantage in generating all of the power in a single large unit. When the cost of test facilities and the flexibility in location and scheduling are considered, however, it may be desirable to use several smaller units rather than a single large unit. The analysis of Section IV shows that only a modest sacrifice in cost is suffered if the power plant is reduced to 1/4 size. Thus, four units, each using about 22,600 gpm of hot water were selected. Each of these units uses ammonia as the working fluid and produces 4,250 kW net power for a total power output of about 17 MW.

To prevent ammonia leakage into the process water providing the residual heat, either an isolation intermediate heat transfer loop is used, the process water is pumped to a pressure higher than that of the ammonia, or a pressurized buffer fluid layer is interposed between the process water and the ammonia in the evaporators. The first method of preventing leakage--the use of an intermediate loop--is the least favorable approach from the standpoint of economics and maximum power recovery, but is also the most conservative and readily acceptable approach. This is the assumed approach in the proposed power plant design described below. One of the other two approaches is, after further study, likely to replace it.

It was also shown in Section IV that the use of two stages of evaporation allows the recovery of a greater fraction of the available energy than the use of a single stage. The use of more than two stages slightly increases power recovery but has economic advantages only in cases where pipeline costs are high. Thus, a two-stage system is proposed.

The most favorable boiler temperatures in a two-stage system to minimize installed cost per kW were found to be 130°F and 105°F. This choice of temperatures causes a large unbalance in the heat transferred in the two boilers. In a real power plant, an economic advantage can be gained if the designs of the two boilers are similar or identical, which would not be possible at the optimum temperatures. Section IV also shows that the installed cost is quite flat with respect to departures from the optimum as far as 5°F, such as to 125°F and 110°F. The latter temperatures permit the use of boilers of identical cross section but only slightly different height; thus, these temperatures were selected for the proposed power plant.

Other parameters including intermediate heat exchanger temperature differential (10°F) and pinch temperature differential (5°F) were chosen to be what the analysis of Section IV showed to be optimum.

The power plant (see Fig. V-4) is a two-stage ammonia Rankine cycle, employing an intermediate heat exchanger to isolate the ammonia from the process water, and rejecting heat to a mechanical-draft cooling tower. The process water at 156°F enters two intermediate heat exchangers transferring heat to counterflowing water from the intermediate loop and raising its temperature to 146°F. The intermediate loop water flows successively through high and low-pressure ammonia evaporators and an ammonia feed heater. The evaporators are large vertical natural circulation boilers with water flowing through the tubes. One quarter of the water is diverted to flow through the ammonia feed heater which is of counterflow shell-and-tube design with the water on the shell side. Downstream of the feed heater the water flow mixes and the intermediate loop water temperature is reduced to 113°F. The pressure drop around the water loop is made up by two centrifugal circulating pumps which discharge into the intermediate heat exchangers.

Ammonia vaporizes in the pressure evaporator at a saturation temperature of 125°F and a pressure of 308 psia. A pinch point temperature differential of 5°F is employed. In the low pressure evaporator, ammonia vaporizes at a saturation temperature of 90°F and a pressure of 243 psia. Vapor from both evaporators flows to separate radial inflow turbines, expanding down to a pressure of 158 psia and a saturation temperature of 82°F. The radial turbines drive ac generators through gear reducers, producing about 6.2 MW of gross power each. The turbines are mounted directly over large horizontal shell-and-tube surface condensers. A condensate pump at each hot well pressurizes the condensate sufficiently to avoid cavitation and discharges it into a feed pump. The feed heater provides hot fluid to the high pressure evaporator and, through a throttling valve, to the low pressure evaporator.

Cooling water at 67°F is pumped through the condenser tubes by the cooling water pumps from the cooling tower supply duct. The cooling water, heated to 77°F, discharges into the cooling

SCHEMATIC FLOW DIAGRAM OF THE CEMS POWER GENERATION SYSTEM

FIGURE V-4

tower return line. Cooling tower circulating pumps provide the head required to flow the water over the tower surface. The physical arrangement of the power plant, shown in Figs. V-5 and V-6, indicates that it occupies a plan area of approximately 80 ft by 110 ft. The maximum height above grade is 57 ft and the cooling water duct at the pump intake is approximately 28 ft below grade. The intermediate heat exchangers and loop circulating pumps are grouped on one side of the area; the evaporators are located side by side in the upper corner and the two condensers are side by side in the center of the area. The turbogenerators are mounted on top of the condensers.

Major features of the power plant are listed in Table V-3. The gross generator output is approximately 6.2 MW. Heat input is 365×10^6 Btu/hr. Hot process water flow is 22,600 gpm and cooling water flow is 72,000 gpm to each unit.

FIGURE V-5

ELEVATION VIEW OF POWER GENERATION SYSTEM

FIGURE V-6

PLAN VIEW OF POWER GENERATION SYSTEM

TABLE V-3

POWER GENERATION SYSTEM SUMMARY

Source of heat - 22,600 gpm of hot water normally routed to existing cooling towers (each unit)

Location - Adjacent to existing pump house to cooling towers

Type of system - Staged evaporation ammonia Rankine cycle with isolating intermediate heat exchange loop

Hot water inlet temperature to intermediate loop - 156°F

Hot water inlet temperature to ammonia evaporator - 146°F

Gross generator output - 6,200 kW

Net output not considering parasitic losses to intermediate loop and cooling towers - 5,600 kW

Net output considering parasitic losses to intermediate loop and cooling towers - 4,250 kW

Cooling water flow - 72,000 gpm

Cooling water inlet temperature - 66°F

Ammonia condensing temperature - 82°F

Ammonia flows from boilers - 106 lb/sec HP, 97 lb/sec LP

Type of boilers - Vertical tubular natural circulation

Type of turbines (2) - Radial inflow with variable area inlet nozzle

Type of condensers (2) - Shell and tube surface condenser

Type of cooling water pumps - Axial flow, submerged inlet

F. ENERGY SAVINGS WITH CEMS

The construction of a CEMS to utilize the residual heat rejected by the Portsmouth GDP to supply the thermal and electric energy requirements of a nearby industrial park can provide significant fuel savings relative to conventional alternatives. Although the economic viability of the CEMS is a function of the prevailing fuel and electric energy prices and their projected escalation rates during the lifetime of the CEMS, energy savings can be accurately estimated once the detailed performance characteristics of the competing systems are identified. The approximate energy savings made possible by generating electricity and process steam using the GDP residual heat were defined using representative competing systems characteristics. These potential savings are listed in Table V-4 in terms of barrels of oil and cubic feet of natural gas.

As indicated in Table V-4, approximately 17 MW_e of the total 29 MW_e needed is generated by the power recovery systems. The remainder (12 MW_e) is purchased from the grid or generated on site by conventional means. A fuel savings equivalent to 1.55×10^{12} Btu/yr is achieved by generating the major part of the required electric power in the CEMS. The heat pump system provides all of the process heat required (400×10^6 Btu/hr) through the recovery of the residual heat from the GDP and from the driver gas turbine exhaust gases. This saves the fuel equivalent of an additional 1.49×10^{12} Btu/yr relative to a conventional industrial boiler. Thus, a total of 3.04×10^{12} Btu/yr of fuel energy can be saved by implementing both the power recovery and the heat pump systems in the CEMS facility. This would translate into an equivalent annual savings of 0.52×10^6 barrels of oil or 3.04×10^9 SCF of natural gas that would be consumed should the conventional approach rather than the CEMS concept be adopted.

TABLE V-4

FUEL SAVINGS ACHIEVABLE USING PORTSMOUTH CEMS

Energy Form	Amount Needed	Annual Fuel Consumption		SAVINGS (Conventional-CEMS)
		Conventional Method	CEMS	
		10 ¹² Btu/yr	10 ¹² Btu/yr	10 ¹² Btu/yr
Power	29 MWe	2.64* (29 MW _e)	1.09* (12 MW _e)	1.55
Process Heat	400 x 10 ⁶ Btu/hr	3.76**	2.27	1.49
Total		6.40	3.36	3.04

Total Equivalent Primary Fuel Savings

Oil, 10 ⁶ bbl/yr	0.52
Gas, 10 ⁹ scf/yr	3.04

* Based on a fuel-to-electric energy conversion efficiency of 30 percent

** Based on a boiler efficiency of 85 percent

SOUTH CAROLINA
Energy Research Institute
SUITE 670,
FIRST NATIONAL BANK BUILDING
COLUMBIA, SC 29201