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COMPARISON OF RANKINE-CYCLE POWER SYSTEMS:
EFFECTS OF SEVEN WORKING FLUIDS

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by

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PREFACE

This report has been prepared as part of the Industrial Cogeneration Program at Argonne National Laboratory. The program is sponsored by the Energy Cascading Branch, Office of Industrial Programs, United States Department of Energy. Its main goals are to investigate the applicability of new technologies and systems for industrial cogeneration applications; help develop new technologies; demonstrate novel systems in industrial settings; and address and recommend solutions to economic, regulatory, or other institutional issues that may inhibit industrial cogeneration.

The cogeneration technology and system evaluation and development work at Argonne involves a broad range of technical expertise representing several divisions within the Laboratory. These divisions include Energy and Environmental Systems (EES), Components Technology (CT), Chemical Engineering (CEN), and Engineering (ENG).

Comments or questions should be referred to J.C. Bratis (EES).

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ABSTRACT

This study investigates the safety, health, technical, and economic issues surrounding the prime working-fluid candidates for industrial Rankine-cycle power systems in the range of 600-2400 kW. These fluids are water, methanol, 2-methyl pyridine/H₂O, Fluorinol 85, toluene, Freon R 11, and Freon R 113. Rankine-cycle power systems using water as a working fluid and boilers burning coal, refuse, oil, or gas -- or driven by nuclear energy -- have been the mainstay of power generation for about a century. Interest in energy conservation in the industrial sector is now encouraging the development of small Rankine power systems that use heat from a variety of waste streams. The temperature range of interest for industrial applications is from 500°F to 1100°F (260°C to 593°C) for gaseous streams and approximately 300°F (149°C) for condensing streams. At temperatures below about 700°F (371°C), steam systems become less efficient and too expensive to be used. However, other working fluids, usually organic compounds, can be economically attractive at the lower temperatures. This study shows that, at current and projected energy costs, Rankine-cycle power systems using any of the seven working fluids investigated here can exceed the minimum return on investment (ROI) criteria of most industries. The highest ROIs occur for those systems using a 300°F (149°C) condensing stream as the heat source. There appear to be no significant health or safety problems that would prevent the use of any of the candidate working fluids. The only limitation of an organic fluid is its maximum stability temperature, which may prevent its use with high-temperature waste-heat streams.

1 INTRODUCTION

Rankine-cycle power systems using boilers fired with coal, oil, gas, or refuse -- or driven by nuclear energy -- have been the mainstay of utility and industrial power generation for nearly a century. The cycle, devised for working fluids that are alternately vaporized and condensed, was developed by

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John MacQuorn Rankine (1820-1872), a Scottish engineer and a professor of civil engineering at the University of Glasgow. Over the years, as combustion and material technology has improved, those utility power-generation systems using the Rankine cycle with water as the working fluid have become very efficient. Now, large power-generation steam plants can attain efficiencies of about 40%. However, to achieve these efficiencies, turbine-inlet temperatures and pressures have been driven into the supercritical region, and the systems have increased in size to take advantage of economies of scale.

Recent industrial-sector interest in energy conservation has encouraged the development of small power systems that can use heat normally exhausted from various industrial processes.¹ Gaseous streams with temperatures in the range of 500°F-1100°F (260°C-593°C) and condensing streams at about 300°F (149°C) are quite common in a number of industries. These heat streams are not only generated by industrial plants but also by small utilities that use gas turbines or diesel engines as their primary source of power. The bulk of the energy, however, is lost at temperatures less than 700°F, the temperature at which Rankine-cycle power systems using water as the working fluid become significantly less efficient and capital costs increase. To overcome this disadvantage, and still exploit the basic Rankine-cycle technology developed over the years, systems based on working fluids other than water are being developed. These alternative fluids are usually organic compounds that have, for the most part, advantageous thermodynamic characteristics; however, they may also have disadvantageously high costs and present potential health and safety hazards.

The organic fluids that look most promising for Rankine-cycle power systems, because they can use heat normally rejected in the industrial sector, include several well known refrigerants as well as more exotic proprietary liquids. Working-fluid candidates of prime interest are water, methanol, 2-methyl pyridine/H₂O, Fluorinol 85, toluene, Freon R 11, and Freon R 113. Water is considered a base case, but its use is thought to be mainly confined to higher-temperature heat sources such as those found in gas-turbine and diesel-engine exhausts. The size of Rankine-cycle power systems is probably limited to a range of 600-2400 kW, mainly because of the mass-flow rates and temperatures normally encountered in industrial applications.

Because there are so many systems based on the various working fluids being developed, several questions arise. For example, is there a working fluid most appropriate for each temperature range? Are there significant economies of scale that would accrue to larger systems? Are there specific health and safety reasons that would prevent any of the fluids from being used, even if it proved to be economically attractive? Indeed, can systems using the newer fluids be economically attractive to industrial users and, if so, under what conditions? Answers to these questions are not only of interest to potential users of new fluids for the Rankine-cycle technology but also to those who are spending time and money to develop them. This study was intended to determine answers to at least some of the above questions so that a rational research and development plan can be devised for systems that look most promising.

More specifically, the study makes a comparative analysis of the several specific working fluids that have been proposed for use in a Rankine-cycle power system (RCPS). This comparison evaluates: (a) thermo-

physical properties of the various working fluids; (b) system performance as a function of size, heat-source temperature, and working fluid; (c) capital, operating, and maintenance costs; and (d) potential return on investment (ROI) of a Rankine-cycle system as a function of system size, heat-source temperature, working fluid, and various economic factors including electricity costs, load factor, and tax policies.

Figure 1.1 shows the general methodological steps used in the RCPS study. After the candidate working fluids were selected, we then proceeded along two paths. The first was to determine the health, safety, and thermophysical properties of each working fluid. This was done in order to reveal any characteristics that would either eliminate the fluid from further consideration or significantly affect system design and cost. Concurrently, system design and costing methods were devised. This effort included development of a small computer code to minimize the capital cost of each system. Also, capital cost data were developed for each component -- turbines, pumps, vaporizers, etc.; these data are presented in subsequent sections of this study.

Next, based on the working-fluid characteristics and design methodology, optimal system designs were devised. The prime purpose of this task was to determine the minimum capital cost of each fluid-based system so that an economic analysis could be made based on a range of economic parameters. The purpose here was to determine the ROI of each system and compare systems as a function of temperature and size. Lastly, each RCPS was ranked according to health, safety, thermophysical, and economic criteria.

The results of this study, especially those relating to economics, should be used to make general comparisons only. The data do not support specific comparisons of systems being developed by various industrial firms, either those working on their own or those whose projects are under government sponsorship. What is intended is a ranking of the working fluids based on physical parameters and economics. What is not addressed, at least not directly, is the question of whether unnecessary a market exists based on those economics. Given the uncertainty of the costing data and the differences in each manufacturer's design, the authors do not presume to resolve that question.

Fig. 1.1 Sequence of Issues Addressed in This Study

The report presents:

- A review of the history and development of the basic technology of Rankine-cycle power systems, along with the approach and guidelines used in this study (Sec. 2);
- An analysis of the health, safety, and thermophysical properties of each working fluid that was investigated (Sec. 3);
- Optimal-cost RCPS designs for each candidate working fluid (Sec. 4);
- An economic analysis of the expected return on investment from the various power systems, each using a different working fluid (Sec. 5); and
- A set of conclusions and recommendations, including the most suitable temperature or temperature range for each working fluid and recommendations for future RCPS developmental efforts (Sec. 6).

In addition, five appendixes amplify the technical and economic reasoning presented in the main body of the report and buttress the discussions with further data.

2 TECHNICAL REVIEW OF RANKINE CYCLE; APPROACH AND GUIDELINES USED IN THIS STUDY

2.1 DESCRIPTION OF THE RANKINE CYCLE AND ITS POWER-SYSTEM APPLICATION

The Rankine cycle is relatively simple conceptually. When applied to power systems, however, it can become quite complex, especially in operations at large utility plants. Figure 2.1 shows a temperature-enthalpy representation of the Rankine cycle superimposed on a phase diagram of a fluid that can exist as a liquid, vapor, or combination of the two. The working fluid depicted is a "non-wetting" type with a positively sloped saturated-vapor line. Figure 2.2 is a flowchart showing the major components of a Rankine-cycle power system: vaporizer, turbine, regenerator, condenser, pump, and cooling tower. The thermodynamic state points designated by numbers on the temperature-enthalpy diagram (Fig. 2.1) correspond to those shown on the flowchart (Fig. 2.2).

Beginning at state point 1, liquid from the condenser is pumped to state point 2, representing the pressure condition of the fluid as it enters the vaporizer if the system does not use a regenerator. If the system uses a regenerator, the fluid is then heated to the condition at state point 3 with heat extracted from the turbine exhaust. The fluid is heated, turned into vapor, and possibly superheated in a waste-heat boiler or vaporizer. The vaporizer would include three sections: (a) an inlet section, where the fluid is heated to boiling; (b) a boiler section; (c) and, if necessary, a superheater section. After leaving the vaporizer, and reaching the condition of state point 4, the fluid enters the turbine. Also represented in Fig. 2.1 is the waste-heat stream side of the vaporizer. The waste-heat stream is losing heat to the working fluid; heat lost by the one equals that gained by the other. The nearest approach in temperature between the waste-heat stream and the working fluid is called the "pinch point" of the vaporizer. As the pinch-point temperature difference becomes smaller, the cost of the vaporizer increases since more surface area (a larger heat exchanger) is required. A more detailed technical discussion of vaporizer design appears in Appendix A.3.

In passing through the turbine to state point 5, the heat energy contained in the working fluid is converted into mechanical work. In an ideal turbine, the work will be produced in an isentropic expansion, and the turbine exit state would then be 5_s . In a real turbine, however, irreversibilities such as fluid turbulence prevent the maximum work from being produced. Thus, expansion in the turbine does not allow a full recovery of heat energy from the fluid; the actual exit condition of the fluid from the turbine is state point 5. After leaving the turbine, the working fluid is then condensed back to state point 1, and the cycle is repeated. If there is sufficient superheat in the fluid as it exits the turbine, that heat may be passed through a regenerator to preheat the working fluid entering the vaporizer; this regeneration process recovers heat that would otherwise be lost.

In general, for the temperature ranges of typical industrial-sector waste-heat streams, efficiencies in the range of 10-20% are attained by Rankine-cycle systems. Higher efficiencies are possible at higher turbine-

inlet temperatures and pressures; these, in turn, can only occur in conjunction with high-temperature waste-heat streams. However, most industrial applications will probably be used in conjunction with streams having a maximum temperature of 700°F (371°C), the point below which systems using water as a working fluid have low efficiencies and high costs.

In order to overcome this cost-efficiency impasse, Rankine-cycle systems using fluids with thermodynamic properties more suitable for use with lower source temperatures have been under development. In general, these are organic fluids or combinations of fluids with properties considered desirable for such applications. Desirable characteristics include high molecular weight (along with a minimum number of atoms per molecule); a high fluid boiling point; a positive-slope, or zero-slope, saturated-vapor line; and good thermal stability and material compatibility. The suitability criteria for Rankine-cycle power systems have been addressed by several authors, especially with regard to thermodynamic properties.^{2,3,4} These findings are reviewed briefly in Appendix A.2.

Fig. 2.1 Temperature-Enthalpy Diagram of Rankine Cycle Superimposed on a Phase Diagram of Fluid

Fig. 2.2 Flowchart of a Rankine-Cycle Power System

Rankine-cycle power systems for use with low-temperature sources have been under development since the early 1960s. Initially, they were intended for aerospace applications. Beginning in the early 1970s, however, emphasis shifted to power systems that could benefit from the exhaust-gas streams of stationary engines such as diesels and gas turbines. This developmental effort, most of it supported by the U.S. Department of Energy, has resulted in several demonstrations of power systems at municipal power plants. These include plants in Beloit, Kan.; Easton, Md.; and Rockville Center, N.Y.⁵ Further demonstrations are scheduled for various industrial plants. Rankine-cycle systems using organic fluids are also being developed for use with solar systems,⁶ diesel trucks, and marine diesel engines.⁷

That industrial markets for Rankine-cycle power systems exist is evident, given the heat-stream temperature distribution and amount of heat available. In 1978, Resource Planning Associates (RPA) performed a study of the industrial market for organic Rankine-cycle engines.⁸ At that time, RPA found that the total technical market could be as high as 3220 MW and the economic market could be approximately 2100 MW. Penetrating this market is considered somewhat difficult because of a lack of technical and economic information about the technology. More data on reliability, safety, and operating costs are needed to enable industrial users to make rational decisions about the possibility of adopting this technology.

The market potential for Rankine-cycle power systems is large; the potential number of different Rankine-cycle systems, each applicable to a different temperature range, is also quite large. One system, based on only one working fluid, will not be suitable for all applications. The RPA study did not differentiate between systems using different working fluids and did not assess the impact of such differences on RCPS economics. The differences could affect RCPS market potential and market-penetration rates. In any case, systems are being developed by several manufacturers for a variety of working fluids -- the reason being that each fluid is thought of as specifically applicable to a certain temperature or temperature range. For example, Rankine-cycle power systems using refrigerants such as Freon R 11 or R 113 are intended for applications with waste-heat temperatures of less than 400°F (204°C). Systems using toluene or 2-methyl pyridine/H₂O are considered more suitable for higher temperatures -- about 800°F (427°C).

Even though a large number of organic working fluids have potential for Rankine-cycle power systems, most of them have been eliminated for various reasons, and only a few strong candidates remain. Of those that do remain, there is reason to believe that certain fluids are more economical than others at given temperatures. Whether that conclusion can legitimately be drawn will be tested in this study; in addition, the study will investigate each fluid's health and safety implications. As was pointed out in the introduction, we will not make direct comparisons of systems being developed by various organizations and using different working fluids. The study is more generic in character.

2.2 STUDY APPROACH AND GUIDELINES

The purpose of this Rankine-cycle study of working fluids, again, is to compare several aspects of the prime-candidate fluids proposed for use

in Rankine-cycle power systems. The aspects to be compared are: (a) thermo-physical, heat-transport, health and safety, and economic characteristics of the various working fluids; (b) power-system performance as a function of size, heat-source temperature, and working fluid; (c) capital, operating, and maintenance costs for each system as a function of size, source temperature, and working fluid; and (d) return on investment (ROI) for each system based on size, source temperature, working fluid, and a set of economic parameters including capacity factor, taxes, investment credits, and electricity costs. The emphasis is on performance and cost variances between the different fluid-based systems; if possible, the most appropriate temperature and size will be determined for each system. The report is not intended to compare specific systems being developed by several organizations. Rather, the study is normative in its approach. All systems are "designed" the same way, with only the working fluid changing.

System design parameters for this study can be classified into three categories: (a) candidate fluids, (b) waste-heat temperature and type, and (c) system power level.

Candidate fluids considered in this study are limited to those currently of prime interest for use in Rankine-cycle power systems. These fluids are:

- Water,
- Methanol,
- 2-methyl pyridine/H₂O,
- Fluorinol 85,
- Toluene,
- Freon R 11, and
- Freon R 113.

The waste-heat sources and temperature ranges considered are shown in Table 2.1. The condensing process stream is considered to be at a saturated condition, which corresponds to a pressure of 67 psia (0.462 MPa). To avoid condensation of corrosive fluids, the lower temperature limit of the waste-gas streams -- including boiler, furnace, and engine exhausts -- is to be 300°F (149°C). This gives a maximum ΔT range of 200-800°F (93-427°C). It should be noted that the mass-flow rate of the waste-heat streams has not been fixed. This results in a tacit assumption that no matter what the power system size or efficiency may be, the mass-flow rate of the waste-heat stream is adequate to accommodate it. This assumption is waived, however, in Sec. 4, where a fixed waste-heat stream is addressed.

The RCPS market is expected to require a reasonable "catalogue" of engine sizes. Although extensive market studies regarding engine size are not available in the open literature, it is felt that the range will be similar to that for medium-speed diesel engines. Thus, for this study, the engine sizes considered are 600, 1200, and 2400 kW. This range will allow a reasonable assessment of the economies of scale of the various systems.

Costs associated with each system include capital, operating, and maintenance costs. Capital costs for a system are determined by aggregating the costs for each system component, e.g., vaporizer, pump, turbine, condenser, regenerator, and heat-rejection equipment. Installed costs include

Table 2.1 Heat Sources and Their Temperatures

Source	Type	Inlet Temperature (°F)	Maximum Vaporizer ΔT (°F) ^a
Condensing process stream	Vapor/liquid	300	b
Process stream	Liquid	300	Variable
Boiler, furnace, engine, exhaust gas, and process streams	Gas	500	200
		700	400
		900	600
		1100	800

^aBased on 300°F outlet temperature for gas (can be lower for liquid).

^bNot applicable.

an allowance for auxiliary items such as buildings, control systems, and the labor required for system installation. Thus, the total installed cost of a system is the aggregate of equipment, labor, and other costs required for a complete, operating system. These costs will vary according to working fluid characteristics, unit size, and applicable temperature range.*

Based on the thermophysical data accumulated for each working fluid and on component costs and RCPS design characteristics, optimal systems are determined for each temperature and power output. An "optimal" system is one that requires the least capital cost for each operating condition. It should be noted, however, that this approach may not quite represent a truly optimal system. For example, when a system is optimized as above, it occurs at a specific system efficiency. It may be that the system efficiency could be improved at a marginal cost less than the expected marginal revenue. True optimization would then occur when marginal costs equal marginal returns. This degree of economic sophistication was not applied in the current analysis, but its implications are discussed in Sec. 4.

With regard to assumptions concerning the design of an RCPS and its resultant cost, a standardized approach was used. Several RCPS manufacturers have been quite innovative in their system designs. In order to solve sealing problems, moving components are sometimes enclosed in one casing. For example, one manufacturer has integrated the pump and turbine in that manner; another has done the same thing and added the generator as well. Given our limited data, it was difficult, and in some cases impossible, to determine the costs of such innovations. It was therefore assumed that all systems were built with more or less standard components. It is possible that such design innovations could become quite cost-effective, though only for a specific working fluid. This could distort the relative economic merits of a given fluid. There was no evidence that this is true, however, so the above normative approach seemed reasonable.

*Appendix D presents details of the costing methodology.

3 WORKING FLUIDS: CHARACTERISTICS, AVAILABILITIES, COSTS

3.1 GENERAL CONSIDERATIONS

Possible working fluids for use in Rankine-cycle power systems are almost countless. Factors that must be considered in any selection process include location and use of the power system; effects of a fluid on system availability, reliability, maintainability, and economics; and environmental effects attributable to the various fluids. For the most part, the methodology of fluid selection developed for heat-pump systems by Snyder and Moreland⁹ applies here, with appropriate modifications made for the nature of an RCPS application.

For the purposes of this study, the necessary and desirable characteristics for a Rankine-cycle power system working fluid can be divided into four main categories. These are: (a) thermophysical properties; (b) material compatibility and working-fluid stability limits; (c) health and safety characteristics; and (d) fluid availability and cost. Each category includes criteria for determining whether or not a particular fluid is desirable for use in a Rankine-cycle power system. Failure to meet one or more of these criteria could well eliminate a fluid from further consideration. It should be noted, however, that the working fluids analyzed in this study -- water, methanol, toluene, 2-methyl pyridine/H₂O, Fluorinol 85, Freon R 11, and Freon R 113 -- have already been screened by others for most, if not all, of these criteria.

The following general considerations apply with respect to the above criteria.

Thermophysical Properties^{2,3,9,10,11}

A Rankine-cycle working fluid should have a relatively high boiling point with respect to the maximum temperature of waste-heat sources. This will ensure low system operating pressures and will probably result in relatively low condenser vapor pressures -- low pressures will cut down friction losses and system sealing problems. The fluid should have a relatively high molecular weight (> 75 g/gmole) and a small number (5-10) of atoms. These parameters control the slope of the saturated-vapor line on a temperature-entropy chart. The slope of the saturated-vapor line, $(\partial T / \partial s)$, can be given in terms of a related quantity called the I-factor (see Appendix E.1). The I-factor should be in the range of 0.65-1.5. Fluids should also have good heat-transport properties in order to minimize heat-exchanger surface areas.

Material Compatibility and Fluid-Stability Limits

The working fluids selected must be noncorrosive to the more common engineering materials used in RCPS construction. This includes materials for pipes, turbines, pumps, heat exchangers, seals, and lubricants at both room and operating temperatures. The fluids should also be thermally and chemically stable, at all operating temperatures, in the presence of common

engineering materials and in the presence of air. Further, if the fluid should decompose or combine with other system materials, the resulting compounds should also be compatible, and decomposition should occur at a slow enough rate to avoid a drastic change in fluid characteristics.

Health and Safety Characteristics

A working fluid, as well as any decomposition products, should not be unacceptably toxic, carcinogenic, or unacceptably explosive. Nor should the fluid represent an unreasonable environmental hazard to either operating personnel or the general public. Fire and explosion hazards should be easily handled under fairly normal circumstances including consideration of the fluid's fire point, autoignition temperature, flash point, and explosion limits.

Fluid Availability and Cost

A fluid selected for use in Rankine-cycle power systems should be readily available, preferably from several suppliers, at a cost commensurate with its other attributes.

The subsections immediately below analyze each of the candidate working fluids with reference to the above criteria. Appendix E presents details regarding the effects that fluid characteristics have on RCPS performance; it also sets forth various significant parameters for each fluid-based system.

3.2 THERMOPHYSICAL PROPERTIES

In general, organic fluids chosen for use in Rankine-cycle power systems offer several thermodynamic advantages over steam. These include: (a) high molecular weight and low number of atoms per molecule, a combination of which results in little or no moisture in the turbine exhaust; (b) a small enthalpy change across the turbine, which results in lower turbine speeds and smaller diameters; (c) high vapor densities; and (d) higher cycle efficiencies, especially at the lower waste-heat temperatures usually encountered in industry.

One of the more important thermophysical properties of an ideal RCPS fluid is the slope of the saturated-vapor line. This is sometimes called the dew-point line. Organic fluids with a high molecular weight and low number of atoms per molecule have positively sloped saturated-vapor lines. That is, the I-factor is less than unity ($I < 1$). Thus, these fluids are usually superheated when exiting from a turbine and sometimes require a regenerator in order to increase system efficiency. Water and methanol, on the other hand, have negatively sloped saturated-vapor lines and I-factors much greater than unity ($I > 1$), resulting in a turbine exhaust that could be quite moist. The ideal fluid would have an I-factor near, but slightly greater than, unity; this would result in the fluid being near saturation at the turbine exhaust. Some fluids, like Freon R 11, have this as a natural property; other organic fluids can achieve a favorable I-factor by being mixed with water. A favorable I-factor is in the range of 0.65-1.5.

The main thermophysical properties of the seven fluids considered in this study are shown in Table 3.1. The fluids are arranged, left to right, from the smallest to the highest molecular weight. The 2-methyl pyridine/H₂O and Fluorinol 85 data represent the properties of the azeotropic fluid formed by a combination of two fluids. An azeotropic fluid is one with a single boiling point, even though it is composed of two fluids. From these data, favorable I-factors exist for 2-methyl pyridine/H₂O, Fluorinol 85, and Freon R 11. Toluene and Freon R 113 are at the lower edge of the allowable range, and they would probably require a regenerator because of high fluid superheat at the turbine exit. Water and methanol are "wetting" fluids and would require that care be taken to avoid excess moisture in the turbine exhaust. This would probably require a high superheat at the turbine inlet and perhaps a more costly vaporizer.

Suitability of a fluid for use in a power system is also governed by its stability temperature. The maximum fluid temperature in a waste-heat-recovery system will occur in the vaporizer. This maximum temperature should not exceed the stability temperature of the fluid. As shown in Table 3.1, the Freons are limited to temperatures below 400°F (204°C); other organics, notably toluene and 2-methyl pyridine/H₂O, are suitable for temperatures up to about 750°F (399°C). Steam, of course, is suitable for any temperature and in fact is the working fluid of choice in combined-cycle power plants.

Fluid heat-transfer coefficients affect the size of the various heat exchangers in a Rankine-cycle power system. Heat-transfer coefficients of the organic fluids are less than those of water under the same operating conditions. As an indication of the relative magnitudes of the liquid heat-transfer properties of the various fluids, Table 3.1 shows the calculated film coefficients for liquids at 200°F (93°C) flowing in a pipe with a 1-in. nominal diameter at 5 ft/s (1.5 m/s). Water has a film coefficient almost a factor of three greater than the next highest fluid, methanol. The other fluids are in a rather narrow range, from 290 to 475 Btu/hr-ft²-°F; this indicates that heat exchangers using organic fluids will be larger, in general, than those using water. Depending upon operating pressures, this ranking of the organic fluids could change, primarily because of fluid-density differences.

Turbine design for Rankine-cycle power systems is affected by two factors listed in Table 3.1. These are the nominal enthalpy drop across the turbine and the saturated-vapor specific volume. The table indicates that as the molecular weight of a fluid increases, the isentropic enthalpy drop across the turbine generally decreases. Similarly, the specific volume of saturated vapor also decreases with increasing molecular weight. Because of the relatively low isentropic enthalpy drops characteristic of pure organic fluids, Rankine-cycle systems based on organic fluids can be designed using high-efficiency single-stage turbines. Further, the high vapor densities allow high mass-flow rates with small exit-flow areas. As the molecular weight decreases, or the organic fluid is used in an azeotropic mixture with water, the isentropic enthalpy drop increases, and it is necessary to increase the number of turbine stages as well as the flow areas to accommodate higher specific volumes of vapor. Steam systems, with the highest enthalpy drop, would require a multistage turbine, and this could have a strong economic impact. Details concerning the effect of isentropic enthalpy drop on turbine design can be found in Appendix A.2.

Table 3.1 Physical and Thermodynamic Properties of Prime-Candidate Working Fluids

Property	Water	Methanol	2-Methyl Pyridine/H ₂ O	Fluorinol 85	Toluene	Freon R 11	Freon R 113
Molecular weight	18	32	33	88	92	137	187
No. of atoms/molecule	3	6	14	9	15	5	8
Boiling point at 1 atmosphere (normal boiling point)							
(°F)	212	148	200	168	231	75	118
(°C)	100	64	93	75	110	24	48
Liquid density							
(lb _m /ft ³)	62.4	46.8	58.34	85.54	53.5	92.14	97.69
(kg/m ³)	999.5	749.6	934	1370	856.9	1476	1565
Specific volume of saturated vapor, nor- mal boiling point							
(ft ³ /lb _m)	27	12.8	14	5	5.4	2.7	2.2
(m ³ /kg)	1.69	0.80	0.87	0.31	0.34	0.17	0.14
Maximum stability temperature							
(°F)	-	350-450	700-750	550-625	750-800	300-350	350-450
(°C)	-	175-230	370-400	290-330	400-425	150-175	175-230
I-factor for normal boiling point	2.81	2.16	1.43	1.26	0.66	1.04	0.69
Liquid-film coefficient at 200°F							
(Btu/hr-ft ² -°F)	1640	585	475	425	375	315	290
(W/m ² -°C)	9300	3320	2700	2410	2130	1790	1650
Heat of vaporization, normal boiling point							
(Btu/lb _m)	970	472	378	190	157	78	63
(kJ/kg)	2256	1098	879	442	365	181	146
Isentropic enthalpy drop across turbine							
(Btu/lb _m)	150-500	70-130	80-230	30-80	50-100	10-20	10-20
(kJ/kg)	348-1160	162-302	186-534	70-186	116-232	23-46	23-46

3.3 MATERIAL COMPATIBILITY AND FLUID-STABILITY LIMITS

The suitability of an RCPS fluid, even one whose thermodynamic properties are attractive, can be strongly influenced by its material compatibility and thermal stability. The fluid should be noncorrosive to common engineering materials so that it does not dictate the use of more expensive materials. Further, the fluid should be stable under the varying temperatures encountered in a Rankine-cycle power system and also should not break down in the presence of RCPS materials or in the presence of air.

Table 3.2 lists stability temperatures and limits of material compatibility of the candidate working fluids. For the most part, all of the organic

Table 3.2 Material Compatibility and Thermal Stability
of Prime-Candidate Working Fluids

Fluid	Stability Temperature (°F)	Compatible Materials	Incompatible Materials	References
Water	-	Common engineering materials (with proper treatment)	--	--
Methanol	350-450	Carbon steel, copper	Tungsten-carbide seal, titanium-carbide seal	9,14,18
2-methyl pyridine/H ₂ O	700-750	Carbon steel, aluminum	Copper	9,19,21,22
Fluorinol 85	550-625	Carbon steel, low-carbon steel, stainless steel	Copper, aluminum ^a	4,12,15,20
Toluene	750-800	Carbon steel, copper	Nonmetallic, plaster paint, etc.	4,9,12,13,16
Freon R 11	300-350	Carbon steel, copper	Zinc, magnesium, aluminum	9,13,23
Freon R 113	350-450 ^b	Carbon steel, copper ^b	N/A	9,17

^aAccording to the manufacturer, Fluorinol 85 is compatible with copper or aluminum at condenser temperatures.

^bIn the presence of carbon steel, copper, or oil, Freon 113 decomposes at temperatures above 300°F.

candidates are compatible with such common engineering materials as carbon steel, copper, and aluminum. Exceptions include the incompatibility with copper and aluminum of 2-methyl pyridine/H₂O and Fluorinol 85; see, however, footnote "a," Table 3.2. Methanol is incompatible with the more exotic steels, such as tungsten-carbide steel and titanium-carbide steel. Only Freon R 113 appears to decompose, at temperatures above 300°F, in the presence of carbon steel and copper.

Material-compatibility data for Fluorinol 85 are incomplete, if not contradictory, with respect to steel. In at least one case,^{4,25} experiments showed that carbon steel was at least mildly attacked by Fluorinol 85 in the temperature range of 300°F to 550°F (149°C to 288°C). This would seem to indicate that stainless steel is required because of the higher-temperature portions of system operation (e.g., vaporization and piping to the turbine). If stainless steel is required, then the capital costs could be increased by a factor of 2. However, a recent design study indicates that carbon steel ASME SA-210A-1 is suitable.²⁶ Despite the uncertainty, it will be assumed here that the relatively inexpensive carbon steels are suitable for use in power systems using Fluorinol 85.

It should be noted that material incompatibilities usually exist at some high temperature for each given fluid. This usually occurs at temperatures approaching the thermal-decomposition stability limit of the fluid. If a fluid has suitable thermodynamic characteristics, it may be possible to design an economic system using temperatures much lower than the thermal-stability limit, i.e. below the limit of material incompatibility.

The thermal-stability limit for each fluid is shown in Table 3.2 as a range rather than a specific temperature. This range acknowledges the uncertainty reflected in the data and also the fact that decomposition is not a well defined state point. In a given Rankine-cycle power system, the fluid moves rapidly around the loop, reaching maximum operating temperatures in the vaporizer and minimum temperatures in the condenser. Since decomposition of a fluid depends on temperature and time, the amount of decomposition may vary as a function of system design -- including total fluid inventory, mass-flow rate, and the presence of locally trapped fluid. As a result, each fluid will have a range of applicability up to the maximum end of the stability-temperature range without suffering significant decomposition.

3.4 HEALTH AND SAFETY CHARACTERISTICS

The working fluids considered in this study have, in general, a broad range of toxicity, flammability, and explosion characteristics. These fluids range from the slightly toxic, essentially inert refrigerants Freon R 11 and Freon R 113 to the very toxic Fluorinol 85 and 2-methyl pyridine/H₂O fluids. A substance may be toxic if it is ingested, inhaled, or absorbed through the skin. Toxic effects may occur because of an acute, one-time, exposure or because of more subtle, long-term, continuous exposure to low levels.

Three situations involving a given working fluid may cause toxic hazards: (1) normal system operation; (2) system maintenance, or charging and storage of the working fluid; and (3) system boundary failure, resulting in fluid-inventory leakage at varying rates. Exposures during normal system operation will most likely be chronic and due mainly to leakage through seals in the pumps and turbines. However, this is easily controlled either by flushing the seal areas or by providing adequate forced ventilation to keep concentration levels at or below a threshold limit value (TLV) established for the substance. Products of decomposition must also be considered in the design of a system, especially if they are purged and vented on a continuous basis.⁹

During system charging and maintenance, and also in the storage of fluids, special precautions must be taken to protect operating personnel from spills or breathing air with concentrations greater than the TLV. Ventilation could be enhanced during this time, and special systems could be designed to preclude direct skin contact with the fluid.⁹

Particular care must be taken in the event of a system failure that releases a significant amount of the system's fluid inventory. Special ventilation precautions, detection systems, and deluge systems -- which are needed when any flammable liquid is present in a plant -- should be installed in conjunction with a Rankine-cycle power system. Toxic and other hazardous effects of each fluid, following a system-failure condition, merit special examination.

TLVs and other toxicity data for RCPS organic working fluids are shown in Table 3.3. The most toxic fluids are Fluorinol 85 and 2-methyl pyridine/H₂O. The least toxic are the refrigerants Freon R 11 and R 113. Only one of the fluids is suspected of being carcinogenic; the others either are not carcinogenic, or not enough is known about them to make that characterization.

Table 3.3 Toxicity Hazards of Organic Working Fluids

Fluid	TLV ^a (ppm)	OSHA ^b / NIOSH ^c Class	POC and POD ^e	Level of POC/POD Hazard ^f	Possible Carcinogenic Hazard	References
Methanol	200	Toxic	Varied	Toxic	Unknown	28,29,31,32
2-methyl pyridine/H ₂ O	5	Toxic	Cyanides	Toxic	No	9,12,19,22
Fluorinol 85	5	Toxic	Fluorides	Toxic	Unknown	15
Toluene	100	Toxic	Varied	Toxic	Suspected benzene content	28,31,32
Freon R 11	1000	Slightly toxic	Phosgene, hydro- fluoric acid ^d	Toxic	No	13,23,28,31
Freon R 113	1000	Slightly toxic	Phosgene, hydro- fluoric acid ^d	Toxic	No	17,24,28,31

^aThreshold limit value, 8-hr exposure; the value defines maximum daily exposure at which no adverse effects appear for nearly all workers.

^bOccupational Health and Safety Administration.

^cNational Institute for Occupational Safety and Health.

^dDecomposition products when exposed to fire.

^eProducts of combustion; products of decomposition.

^fRefs. 27,30,33,34.

Not surprisingly, the products of combustion of all the fluids are toxic. The Freons generate toxic phosgene gas as a result of direct flame exposure, and particular care must be taken in a system based on these fluids to ensure proper ventilation in case of fire. It should be noted that most of the working fluids are routinely used in industry. No fluid examined here would be precluded from an RCPS application; according to data presently available.

Table 3.4 lists the flammability and explosion characteristics of the various working fluids. Flammability is indicated by several criteria: flash point, autoignition point, fire point, and explosion limits. Definitions of these quantities are:

- Flash point: The lowest temperature at which a liquid will have sufficient flammable vapor near or at its surface so that an intimate mixture with air and a spark or flame will ignite it.
- Fire point: The lowest temperature at which a mixture of vapor and air continues to burn in an open container when ignited.
- Autoignition point: The temperature at which a material will self-ignite and sustain combustion in the presence of air even in the absence of a spark or flame.

Table 3.4 Flammability and Explosion Characteristics of Organic Working Fluids

Fluid	Flash Point (°F)	Fire Point (°F)	Auto-ignition Point (°F)	Explosion Limits (% of Vapor in Air)		References
				Lower	Upper	
Methanol	54	N/A	867	6.7	36.5	9
2-methyl pyridine/H ₂ O	130	145	1060	N/A	N/A	9
Fluorinol 85	109	160	896	6.3	38.5	9
Toluene	40	40	947	1.27	7	9
Freon R 11	None	None	None	None	None	23
Freon R 113	None	None	None	None	None	9,17,24,31

- Lower explosion limit (LEL): The minimum vapor concentration in air at which an explosion can occur. (Below the LEL, vapor concentration in the air is too lean to cause an explosion.)
- Upper explosion limit (UEL): The maximum vapor concentration in air at which an explosion can occur. (Above the UEL, vapor concentration in the air is too rich to cause an explosion.)

Flammability of the various working fluids becomes an important issue in three situations: (1) normal operation; (2) charging, maintenance, and storage; and (3) system failure, the conditions of which were discussed at length by Snyder and Moreland.⁹ They considered a large number of working fluids for use in large industrial heat-pump applications and arrived at six desirable flammability characteristics that should be applied to an RCPS candidate fluid.

1. Autoignition temperature is higher than the maximum cycle temperature (usually the stability point for Rankine-cycle power systems).
2. Flash or fire point is higher than 100°F (38°C).
3. The fluid has a high value for its lower explosion limit, with a narrow range between its LEL and UEL.
4. The fluid has a low vapor density, preferably less than air density, which promotes dissipation.
5. The fluid has a high boiling point at standard conditions.
6. Nonflammable liquids are most desirable, because they would remove any operating restrictions that might otherwise be necessary.

An RCPS candidate fluid should have most of these characteristics. Autoignition points are higher than the stability limits for all the applicable organic fluids. Toluene and methanol have flash and fire points of less than 100°F. The explosion range of toluene is the least, while the ranges of methanol and Fluorinol 85 are quite large. The vapor densities of all but methanol and steam are higher than the vapor density of air; fluids other than methanol and steam will not dissipate easily in the atmosphere and therefore will have a tendency to collect at low points near leaks -- increasing their explosion potential.

All in all, however, the flammability hazard posed by each of the fluids is controllable. Special vapor detection, fire detection, and suppression systems may be required for those fluids exhibiting the greatest potential hazard. Still, it is not expected that hazard-control requirements will significantly affect the basic economics of Rankine-cycle systems using the fluids considered in this study.

3.5 FLUID AVAILABILITY AND COST

The cost of a working fluid affects the overall economic performance of a Rankine-cycle power system. This economic effect occurs not only because of the cost of the initial fluid inventory of a system, but also as a result of fluid leakage and makeup during operation. Depending upon system size, the fluid inventory could be as high as several thousand gallons and represent as much as 15% of the initial system cost.

Table 3.5 shows the cost and availability of the organic working fluids considered. The price ranges from a low of \$1/gal for methanol to a high of \$112/gal for Fluorinol 85. "Excellent" availability means that the fluid can be obtained from a number of suppliers in a competitive-market situation. "Good" availability essentially means that the fluid is available from a limited number of suppliers, either because it is protected by patents or because it has a small market. "Limited" availability means essentially that the fluid is proprietary and is therefore supplied by only one manufacturer.

It is worth noting the sensitivity of the prices to exogenous market influences. The price of toluene has a tendency to track the price of unleaded gasoline. To a certain extent, this explains the increase in price since the Snyder and Moreland report of 1978⁹ -- toluene is used as a constituent of unleaded gasoline, specifically for increasing its octane rating. If methanol were also to be used as a motor fuel, as has been proposed, its cost might also increase significantly. However, given the fluid requirements for Rankine-cycle power systems, it does not appear that this will significantly affect system economics.

Fluorinol 85, however, is a different story. This azeotropic fluid is supplied by only one manufacturer, and its current demand is low. A significant demand increase could substantially reduce the manufacturing cost of the fluid. If this were accompanied by a substantial reduction of its price, systems based on Fluorinol 85 would become much more economically attractive. At present prices, the cost of a charge of working fluid for a Fluorinol 85 system could be as high as 15% of the initial system cost. Fluid-makeup

requirements could significantly affect system operating and maintenance costs. Also, it appears that use of Fluorinol 85 as a power-system working fluid is protected by a patent held by the manufacturer.

3.6 CONCLUSIONS

Based on the foregoing data, a working fluid is considered desirable when it exhibits: (a) low toxicity; (b) reasonably controllable flammability characteristics; (c) low cost and high availability; (d) good thermophysical properties, resulting in high-efficiency, low-cost systems; and (e) an ability to cover the range of waste-stream temperatures. Unfortunately, no one fluid will meet all of these requirements. It appears from a review of the pertinent data that toluene, methanol, and the Freons meet most of the requirements. Toluene would be suitable for high-temperature streams, up to about 800°F (427°C). Methanol and Freon R 113 would appear to be good for the mid-range temperatures, to about 450°F (232°C). Both Freons would seem suitable for low-temperature streams, of less than about 350°F (177°C).

Each fluid considered here has survived a series of selection processes by various manufacturers. Thus, none of them has strongly undesirable thermophysical, health, safety, or economic characteristics. Economic issues, on which this study focuses, are addressed in subsequent sections.

Table 3.5 Cost and Availability of Organic Fluid Candidates

Fluid	Cost ^a (\$/gal)	Availability
Methanol	1	Excellent
2-methyl pyridine/H ₂ O	9	Good
Fluorinol 85	112	Limited
Toluene	1.70	Excellent
Freon R 11	15	Good
Freon R 113	20	Good

^aPrices as of June 1981.

4 SYSTEM PERFORMANCE AND COST ANALYSIS

4.1 SYSTEM CAPITAL COSTS

The capital costs of Rankine-cycle power systems are presented in this section. In obtaining these costs, a thermodynamic analysis of each fluid-based system was first performed. Performance and cost data were then used to determine minimum system costs over a range of heat-source temperatures and power outputs. The system costs, in turn, were needed to calculate the corresponding return on investment (ROI) for the various working fluids. The ROI calculations and results are discussed in Sec. 5.

RCPS capital costs for three power outputs -- 600 kW, 1200 kW, and 2400 kW -- are given, respectively, in Tables 4.1, 4.2, and 4.3. The capital cost of each system is the lowest that could be calculated. Mass-flow rates of the waste-heat streams were not held constant for these calculations. The methodology followed in analyzing the cycle performance and determining the minimum costs is described in Appendixes A.1 and B.

The cost data in Tables 4.1-4.3 are displayed graphically in Figs. 4.1-4.5. These figures show system capital costs (\$/kW) for 300°F condensing steam and for exhaust-gas streams with maximum temperatures of 500-1100°F. As discussed in Sec. 3, some of the working fluids considered in this report cannot be used with higher-temperature heat sources -- they are therefore omitted from the following figures where they would be inappropriate.

Capital costs decrease as heat-source temperatures and power outputs increase because of increased thermodynamic efficiency and economies of scale. Predicted costs for a system using 300°F saturated steam, however, are much smaller than those for one using a 500°F gaseous stream. This is primarily due to a reduction in vaporizer costs, in turn due to a high heat-transfer

Table 4.1 Fluid-Based Variations in RCPS Capital Costs, by Temperature of Waste-Heat Stream: 600-kW Output

Fluid	Waste-Heat Temperature									
	300°F ^a		500°F		700°F		900°F		1100°F	
	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW
Water	410	683	875	1458	690	1150	607	1012	560	933
Methanol	395	658	797	1328	b	b	b	b	b	b
2-methyl pyr./H ₂ O	409	682	851	1418	663	1105	572	953	519	865
Fluorinol 85	501	835	927	1545	699	1165	606	1010	b	b
Toluene	489	815	899	1498	701	1168	601	1002	549	915
Freon R 11	546	910	955	1592	b	b	b	b	b	b
Freon R 113	555	925	950	1583	b	b	b	b	b	b

^aCondensing vapor stream.

^bStability-temperature limitations exclude use of fluid.

Table 4.2 Fluid-Based Variations in RCPS Capital Costs, by
Temperature of Waste-Heat Stream: 1200-kW Output

Fluid	Waste-Heat Temperature									
	300°F ^a		500°F		700°F		900°F		1100°F	
	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW
Water	548	457	1356	1130	1037	864	893	744	813	678
Methanol	545	454	1249	1041	b	b	b	b	b	b
2-methyl pyr./H ₂ O	572	477	1344	1120	1019	849	830	692	739	616
Fluorinol 85	750	625	1500	1250	1099	916	938	782	b	b
Toluene	702	585	1421	1184	1080	900	907	756	816	680
Freon R 11	809	674	1524	1270	b	b	b	b	b	b
Freon R 113	821	684	1514	1262	b	b	b	b	b	b

^aCondensing vapor stream.

^bStability-temperature limitations exclude use of the fluid.

Table 4.3 Fluid-Based Variations in RCPS Capital Costs, by
Temperature of Waste-Heat Stream: 2400-kW Output

Fluid	Waste-Heat Temperature									
	300°F ^a		500°F		700°F		900°F		1100°F	
	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW	\$10 ³	\$/kW
Water	787	328	2221	925	1659	691	1405	585	1263	526
Methanol	809	337	2065	860	b	b	b	b	b	b
2-methyl pyr./H ₂ O	855	356	2237	932	1632	680	1292	538	1131	471
Fluorinol 85	1204	502	2555	1065	1837	765	1551	646	b	b
Toluene	1073	447	2357	982	1758	733	1455	606	1294	539
Freon R 11	1278	533	2554	1064	b	b	b	b	b	b
Freon R 113	1293	539	2534	1056	b	b	b	b	b	b

^aCondensing vapor stream.

^bStability-temperature limitations exclude use of the fluid.

coefficient in condensation. Film coefficients for the exhaust-gas streams are typically 8-12 Btu/hr-ft²-°F.³⁵ Film coefficients for condensing steam are on the order of 1000-3000 Btu/hr-ft²-°F.³⁶ Because of the higher heat-transfer coefficients, the vaporizer size (and cost) will be reduced for the condensate stream. Because the vaporizer is an expensive system component, its very high efficiency and correspondingly low cost result in great savings to the system as a whole.

Fig. 4.1 RCPS Capital Costs, Based on 300°F Condensing Stream

Fig. 4.2 RCPS Capital Costs, Based on 500°F Gaseous Stream

4.2 CAPITAL-COST VARIATIONS

To examine the variation in predicted costs with power output and heat-source temperatures, the overall capital costs were broken down into components. For example, the cost breakdowns for a toluene-based system at two different exhaust-gas temperatures are shown, respectively, in Figs. 4.6 and 4.7. The total system costs were divided into the respective costs for the vaporizer, power conversion module (PCM), PCM building, cooling tower system, working fluid, and installation costs. The PCM costs included costs of the pump, turbine, condenser, regenerator, electrical generator, and gearbox. The vaporizer and PCM costs included both equipment costs and manufacturing costs, as discussed in Appendix D.

For a given power output, the reduction in capital costs with increasing heat-source temperatures is due primarily to decreases in the costs of the heat-transfer equipment. At the higher heat-source temperatures, the maximum operating temperatures and pressures of the RCPS can be increased, thus improving system efficiency. Systems with higher efficiencies require lower flow rates of the working fluid to produce a specified power output. The lower flow rates, in turn, result in smaller heat-exchanger surface areas and lower costs.

Fig. 4.3 RCPS Capital Costs, Based on 700°F Gaseous Stream

Fig. 4.4 RCPS Capital Costs, Based on 900°F Gaseous Stream

At a fixed heat-source temperature, the system capital costs (\$/kW) decrease with increasing power output. The vaporizer costs decrease only slightly over the range of 600-2400 kW. PCM and installation costs show more significant economies of scale. The various components of the PCM have different economies of scale; for the most part, however, their costs are dominated by heat-exchanger costs. (See Appendix D.)

Cost curves for an RCPS based on Fluorinol 85, with a 500°F heat source, are shown in Fig. 4.8. As noted in Sec. 3, the cost of Fluorinol 85 is significantly higher than that of the other working fluids. The fluid costs in Fig. 4.8 constitute about 10-15% of the total system capital costs. Despite these high fluid costs, the predicted capital costs of a Fluorinol 85 system are comparable to those of a toluene system of the same heat-source temperature, as shown in Figs. 4.6 and 4.8. This cost equivalence is due to more favorable thermophysical properties in the Fluorinol 85 system; thermophysical properties affect PCM component costs and the need for a regenerator.

For example, although regenerators were included in all of the toluene-based systems, some of those based on Fluorinol 85 did not need a regenerator. For certain turbine-inlet pressures and temperatures, the Fluorinol 85 expansion resulted in a slightly wet vapor (less than 5% moisture) at the turbine exhaust. A regenerator was therefore eliminated in these cases, and the subsequent savings in capital costs tended to offset the high fluid costs of Fluorinol 85. As noted in Appendix B, however, eliminating the regenerator

Fig. 4.5 RCPS Capital Costs, Based on 1100°F Gaseous Stream

Fig. 4.6 Component Capital Costs, Toluene RCPS Based on 500°F Gaseous Stream

could increase the potential for corrosion problems in the economizer, due to the lower inlet temperature of the working fluid -- even though the exit temperature of the waste-gas stream is maintained at 300°F.

There are many factors to consider when comparing the capital costs for systems based on the various fluids. Four of the more important factors that affect capital costs are listed in Table 4.4. As the table shows, Fluorinol 85 is the most expensive organic fluid, although the calculated capital costs for Fluorinol 85 systems are not significantly higher than those for the other fluids. The producer of Fluorinol 85, Halocarbon Products Corp., has indicated that a larger scale of production could lower the cost by somewhat less than a factor of two. The cost of water was taken to be zero in this study. Even if a nominal fluid cost for water had been included, however, the changes in the calculated capital costs would have been negligible.

The heat-transfer coefficients of a fluid affect the configurations and therefore the sizes of the heat exchangers. Because the gas-stream heat-transfer coefficient controls the heat-transfer process, a common overall heat-transfer coefficient for the vaporizer was used for all fluids. However, condenser and regenerator heat-transfer coefficients did vary between fluids. The overall heat-transfer coefficients for the condenser are shown in Table 4.4. Water has the highest overall coefficient, the Freons have the lowest values, and the remaining organic fluids fall between these limits. The liquid film coefficients presented in Appendix E show a similar ranking for

Fig. 4.7 Component Capital Costs, Toluene RCPS Based on 1100°F Gaseous Stream

Fig. 4.8 Component Capital Costs, Fluorinol 85 RCPS Based on 500°F Gaseous Stream

Table 4.4 Fluid Costs and Other Fluid-Dependent Factors That Affect RCPS Capital Costs

Fluid	Fluid Cost (\$/gal)	Condenser Heat-Transfer Coefficient at 150°F (Btu/hr-ft ² -°F)	Need for a Regenerator in the RCPS	Turbine Configuration
Water	(negligible)	475	Not needed	Three stages (or more)
Methanol	1	265	Not needed	Single-stage or three-stage
2-methyl pyridine/H ₂ O	9	230	Needed only for dry expansion	Single-stage or three-stage
Fluorinol 85	112	175	Needed only for dry expansion	Single-stage
Toluene	1.70	165	Needed	Single-stage
Freon R 11	15	125	Needed	Single-stage
Freon R 113	20	100	Needed	Single-stage

the various fluids. The regenerator heat-transfer coefficients show a somewhat different ranking because of the different shell-side (i.e., vapor-side) conditions for each organic fluid. (See Appendix A.3.)

In performing the system analysis, a regenerator was included for fluids with superheated turbine-exhaust conditions. Thus, regenerators were included in all cost calculations for Freon R 11, Freon R 113, and toluene. Design and cost calculations for systems using methanol and water, since these are wetting fluids, did not include regenerators. Because of the nature of the saturated-vapor curves for Fluorinol 85 and 2-methyl pyridine/H₂O, the turbine expansion could be either wet or dry, depending upon the turbine-inlet temperature and pressure. Typically, the minimum-cost systems for these two fluids did not have regenerators.

Turbine size and number of stages also affected the system capital costs. The turbine staging requirements were selected on the basis of the fluid isentropic enthalpy drops, as discussed in Appendixes A.2 and B. Table 4.4 shows that four of the fluid-based systems require only single-stage turbines for the conditions of this study. Systems based on methanol and 2-methyl pyridine/H₂O require either single-stage or three-stage turbines, depending upon the cycle conditions. Turbines with three or more stages are required for systems based on water. Because different cost curves were developed for single-stage, three-stage, and multistage turbines with more than three stages, turbine-design requirements are reflected in the overall capital costs.

As the data in Table 4.4 indicate, it can be difficult to determine specific factors responsible for the cost differences of systems based on different organic fluids. Moreover, the additional properties discussed in Sec. 3, such as toxicity and stability-temperature limitations, also affect the applicability of the working fluids. All of these characteristics should be considered when selecting a working fluid for a particular heat-recovery application.

4.3 LEAST-COST VARIATIONS

As noted earlier in this section, the methods used here identify the system of "least cost" for each working fluid at a specific power output and waste-heat temperature. For example, results for several fluids at 600 kW and a heat-source temperature of 700°F are shown in Fig. 4.9. The minimum cost and related efficiency of each fluid-based system corresponds to the low point of its cost curve. System efficiency is defined as the electrical power output divided by the energy available (Btu/hr) from the heat source -- including inefficiencies in the turbine, pump, electrical generator, and gearbox. In Fig. 4.9 a methanol-based system, using a single-stage turbine, is the least-cost RCPS that can produce 600 kW using 700°F waste gas; it has an installed cost of \$1060/kW. Efficiency of the methanol-based system is 12%, which means a waste-gas flow rate of 171,000 lb/hr is required.

Systems based on other working fluids, although higher in cost, exhibit higher efficiencies. This means they can produce the required 600 kW using either less waste gas or, conversely, produce more than 600 kW when supplied with the same waste-gas flow rate required by the least-cost

Fig. 4.9 RCPS Cost Vs. Efficiency, Six Fluids
(Heat Source, 700°F; Power, 600 kW)

system. The cost curves in Fig. 4.10 are based on the waste-gas flow rate of the methanol system; however, they reflect higher efficiencies of systems using fluids other than methanol and economies of scale due to larger system size. From these curves it is possible to estimate the marginal cost of each additional kilowatt of power and determine the least-cost system for a fixed flow rate of gas.

Improved system efficiency will lower the cost per kilowatt when waste-heat source is fixed. As shown in Fig. 4.10, a 2-methyl pyridine/H₂O-based system will produce 50% more power at 95% of the cost per kilowatt of a single-stage-turbine, methanol-based system. A three-stage methanol system or a single-stage toluene system can also produce more power at less capital cost per kilowatt than the methanol system with a single turbine stage. These alternatives are also presented in Table 4.5. It is apparent that, given a fixed waste stream, there are systems other than methanol that can produce more power. Because of economies of scale, these systems can produce power at a lower cost per kilowatt than methanol.

These results show that selecting the least-cost system and corresponding power output does not guarantee that the chosen system will be the most competitive. The heat-source flow rate must also be considered. In very few cases will that flow rate be unlimited as compared to the quantity of electrical power desired. In fact, under requirements of the Public Utility Regulatory Policy Act, a user would have an incentive to maximize power output from a given source.

In order to make one size of Rankine-cycle power system the standard size for an industry, an integrator of RCPS components must have at his disposal a market survey that addresses the typical waste-gas flow rates of

Fig. 4.10 RCPS Cost Vs. Output for a Least-Cost System
(Fixed Waste Stream of 171,000 lb/hr at 700°F)

Table 4.5 RCPS Alternatives for a Waste-Stream Flow of
171,000 lb/hr, at 700°F

	Number of Turbine Stages	System Efficiency (%)	Power (kW)	Cost (\$/kW)	Marginal Cost, Each Additional kW (\$/kW)
Methanol	One	12	600	1060	Base Case
Methanol	Three	15.6	780	1040	970
2-methyl pyridine/H ₂ O	Three	18	900	990	850
Toluene	One	18	900	1040	1000
Toluene	One	19.8	990	1080	1110

the industry he wishes to serve. As suggested by the example, a 600-kW unit, or multiples thereof, would not be purchased if more efficient 900-kW units, with lower unit costs, were available.

An RCPS working fluid that can cover a range of efficiencies -- with minimum change in price per kilowatt, to maximize power output for several waste-gas flow rates -- can be highly competitive. This is illustrated in Fig. 4.9 by the cost-versus-efficiency curve of a 2-methyl pyridine/H₂O-based system. Such a system could be used with a variety of waste-gas flow rates, and it could result in the least \$/kW variation.

5 ECONOMIC ANALYSIS

5.1 GENERAL CONSIDERATIONS

To make an economic evaluation of a Rankine-cycle power system, an internal rate of return, or return on investment (ROI), is used. The cost of the RCPS represents the capital investment; the cost of the electricity produced (at the price that would be paid to a utility if the power were purchased) -- after the cost for operating and maintaining the RCPS is subtracted -- represents savings to a user. It is assumed that electricity produced by an RCPS displaces only electricity normally bought from the local utility. Assuming that the amount of electricity produced by the RCPS does not vary from year to year, the amount of savings attributable to a system varies only with the price of electricity.

Estimating possible RCPS savings over the system's anticipated 20-yr life requires a forecast of electricity prices for that period of time. For all the calculations that follow, the national average electricity prices predicted by the Energy Information Administration are used.³⁷ Projections for both electricity prices and RCPS operating and maintenance costs for the next 20 yr are given in Appendix C. Since average prices are used in these calculations, the results obtained underestimate the ROI for those regions of the country with higher-than-average electricity prices. Conversely, they overestimate the ROI for those regions with lower-than-average electricity prices. An economic ranking of RCPS systems, however, would remain unaffected by these regional price variations.

Along with the capital cost, expected system life (20 yr), and electricity price projections, other parameters can also affect the return on investment. These are capacity factor, tax rate, depreciation schedule, life of the system for tax purposes, and tax credits. Representative values of these parameters are given in Table 5.1.

If an RCPS were to operate at full load for a year, it would produce a maximum of 8760 kWh of energy per installed kilowatt. The ratio of the actual energy produced by the system per installed kW to the maximum possible energy is referred to as the capacity factor. As the capacity factor of an RCPS increases, the savings from it, and therefore its rate of return, also increase.

Figure 5.1 shows the combined effect of the capacity factor and taxes on the ROI calculated for a Rankine-cycle power system. The abscissa represents the installed cost of the RCPS, per kilowatt of power, in 1981 dollars; the ordinate represents the ROI in percent. The solid lines in Fig. 5.1 indicate the variation of the ROI with installed cost in the absence of tax considerations -- these lines correspond to capacity factors of 1, 0.8, and 0.5. As an example of the capacity-factor effect on ROI, for an RCPS with an installed cost of \$1400/kW, the ROI decreases from 34.7% for a capacity factor of 1, to 29.2% for a capacity factor of 0.8, and to 20.6% for a 0.5 capacity factor.

The dashed lines in Fig. 5.1 represent the variation of the ROI with installed cost when taxes are taken into account. To derive these curves, it

is assumed that the savings from on-site electricity production are taxed at a rate of 50% after the operating and maintenance costs and the depreciation of the RCPS are deducted. A straight-line depreciation schedule was assumed, with zero salvage value. In addition, it was assumed that a 10% tax credit is applicable. The effect of the taxes is seen in Fig. 5.1. The after-tax ROI that was found for an RCPS costing \$1400/kW, and with a capacity factor of unity, is reduced to 23.8%. When the capacity factor is 0.5, the ROI becomes 14.7%.

The effect of depreciation method and tax credits on the ROI can also be significant. For all cases shown in Fig. 5.2, it is assumed that the life of the system is 20 yr, capacity factor is 0.9, and tax rate is 50%. The solid line represents the ROI variation with installed cost due to straight-line (SL) depreciation and a 10% tax credit. If the tax credit is increased from 10% to 20%, the ROI is increased. These results are represented by the short-dash line. Tax-credit increases change the ROI of the \$1400/kW-system from 22.2% to 24%. If SL depreciation is replaced by sum-of-years-digits (SYD) depreciation, the ROI increases, as indicated in Fig. 5.2. The ROI value of 22.2% that corresponds to SL depreciation with 10% tax credit for an RCPS costing \$1400/kW becomes 23.4% when SYD depreciation is used.

Also shown in Fig. 5.2 is the ROI when both a 20% tax credit and the SYD depreciation method are assumed. In this case, the ROI for the \$1400/kW system becomes 25.2%. Fig. 5.2 shows that an increase of the tax credit to 20% and a change in the depreciation method shift the ROI curve upwards. For example, an increase of the tax-credit rate to 20% increases the ROI by 3-5 percentage points when the system cost is between \$800 and \$2000 per kW. Changing the depreciation method from SL to SYD increases ROI by 2-3 percentage points. The combination represents a 5-8 percentage-point increase, which could be significant in terms of market penetration. It should be pointed out that the allowed credit is specified by the government, while the depreciation method is at the discretion of the user.*

Figure 5.3 shows the effect on ROI of a shorter depreciation period. As was mentioned earlier, the life of an RCPS is assumed to be 20 yr. Nevertheless, a shorter life can be assumed for depreciation purposes. For reference, Fig. 5.3 includes the base case of Fig. 5.2 -- the assumptions being a

Table 5.1 Economic Assumptions Used for ROI Calculations

Item	Economic Assumption
RCPS-related	
Capacity factor	0.5 to 1
Owner's tax rate (%)	50
System life (yr)	20
Investment tax credit (%)	10 and 20
Depreciation period (yr)	10 and 20
Depreciation method	SL ^a or SYD ^b
Operating and maintenance cost (\$/kWh)	Estimates in Appendix C
Utility-related	
Electricity cost (\$/kWh)	Estimates in Appendix C

^aStraight-line.

^bSum-of-years-digits.

*The recently passed tax bill would decrease the depreciation life of Rankine-cycle power systems. The impact of this on RCPS economics, however, is not expected to be substantial. See subsequent discussion.

life of 20 yr; a capacity factor of 0.9; a tax credit of 10%; and a tax rate of 50%, with linear depreciation over the 20-yr life of the plant and zero salvage value. Shown in the figure is the variation in ROI when the system is linearly depreciated in 10 yr. This reduction of the depreciation time to 10 yr increases the ROI by approximately 4 percentage points for RCPS installed costs between \$800 to \$2000 per kW. If, in addition to shortening the depreciation time to 10 yr, the depreciation method is changed from SL to SYD, the ROI increases by an additional 2-3.5 percentage points. Finally, the ROI can be increased by another 4 percentage points if, in addition to the previous changes, the tax credit is increased from 10% to 20% -- as shown in Fig. 5.3.

Fig. 5.1 Effect of RCPS Capacity Factor on ROI, with and w/o Taxes Added (1981 \$)

Figures 5.2 and 5.3 can be used to determine the allowable cost of an RCPS in order to obtain a required return on investment, or hurdle rate. If a company requires an ROI of at least 25% to consider an investment, then Fig. 5.2 indicates that the industry cannot pay more than \$1180/kW for an RCPS if the capacity factor is 0.9, the tax credit is 10%, and straight-line depreciation is used for the 20-yr life of the system. If the tax credit is increased to 20%, then the same company could pay up to \$1320/kW and still meet its required ROI. A maximum allowable investment of this size is not uncommon in specific industries where either of the tax-credit figures can be applied. As an example, refer to Table 5.2, where the hurdle rates for four representative industries are given. The maximum allowable installed cost in the chemical industry would be more than \$2000/kW under the most favorable conditions. However, for the steel industry, with a hurdle rate of 37.5%, the maximum installed cost would be only \$770/kW. Given these limits and the capital-cost data shown in Tables 4.1 through 4.3, it is possible to find RCPS units that can meet these hurdle-rate requirements.

5.2 RETURN ON INVESTMENT FOR SPECIFIC FLUID-BASED SYSTEMS

The results that follow assume a useful RCPS life of 20 yr, a 0.9 plant capacity factor, a 50% tax rate, a 10% investment tax credit, an SYD depreciation method, and a depreciation life of 10 yr.

Table 5.3 shows the ROI for Rankine-cycle systems that use a condensing vapor heat source at 300°F. As was pointed out earlier, the cost of these systems is rather low because of the high heat-transfer coefficient of the condensing medium. The result of this relatively low capital cost is a high return on investment. Among those units having a power output of 600 kW, it

Fig. 5.2 Effect on ROI of RCPS Investment Tax Credit and Depreciation Schedule

is seen that the methanol-based system has the highest ROI, 41.2%. This is followed by the systems using 2-methyl pyridine/H₂O and straight water, each of which has an ROI of 40.2%. Of the seven fluids considered, Freon R 113 has the lowest ROI, 32.5%.

Because of economies of scale, 1200-kW systems have somewhat higher ROIs than do smaller systems. The highest ROIs are attained by 2400-kW units, because their cost per installed kilowatt is least. At 2400 kW, the methanol-based system exhibits an ROI of 69.5%; it is followed by the water-based system, with 67%, and the system based on 2-methyl pyridine/H₂O, with 63%. In general, the ROIs of all systems using a condensing source are fairly high and compare favorably with the required ROIs of the four industries listed in Table 5.2. Although these rates apply to 1978, they are indicative of general ROI requirements. Current rates should be considered when determining specific industrial applications, however, because they vary from time to time.

Figure 5.4 shows ROI variations for a 600-kW RCPS with gaseous-stream inlet temperatures ranging from 500°F to 1100°F. As expected, the ROI increases as waste-stream temperature increases. For 500°F and 700°F stream

Fig. 5.3 Effect on ROI of RCPS Depreciation Method and Life

Table 5.2 Hurdle Rates and Corresponding RCPS Capital Costs Allowed for Four Industries

Industry	Hurdle Rate ^a (%)	Allowable Installed Cost (\$/kW) to Meet the Hurdle Rate	
		20% Tax Credit, SYD Depreciation ^b	10% Tax Credit, SL Depreciation ^b
Chemicals and allied products	18	>2000	1810
Petroleum and coal products	18	>2000	1810
Blast-furnace and steel-mill products	37.5	770	670
Glass products	25	1350	1060

^aReference 38.

^bRefer to Fig. 5.2.

Table 5.3 RCPS Return on Investment, Assuming a 300°F Condensing Vapor Source, for Three Outputs

RCPS Working Fluid Used	ROI (%)		
	600 kW	1200 kW	2400 kW
Water	40.2	53	67
Methanol	41.2	53.5	69.5
2-methyl pyridine/H ₂ O	40.2	51.5	63
Fluorinol 85	35.8	42.5	49.5
Toluene	36.5	44.5	53.5
Freon R 11	33.5	40.5	47.5
Freon R 113	32.5	40.2	47.1

temperatures, the methanol-based system offers the highest ROI: 26% and 30%, respectively. For stream temperatures of 900°F and 1100°F, the system based on 2-methyl pyridine/H₂O offers the highest ROI: 32.5% and 34.6%, respectively. At these high exhaust-gas temperatures, no ROI values are given for the methanol system because of its relatively low stability temperature. Similar limitations and exclusions apply to the Freons.

An important point to be made with reference to Fig. 5.4 is that the spread of the ROIs for each waste temperature is rather small. For example, at a waste-heat temperature of 500°F the highest ROI is 26% (for the methanol system) while the lowest is 23% (for the Freon R 11 system). With such a small difference in ROI, other factors, such as fluid toxicity, would heavily influence the selection process.

Figure 5.5 presents the ROI as a function of waste-stream temperature for 1200-kW systems. Although these results are similar to those for 600-kW systems, the 1200-kW ROIs are higher due to economies of scale. For example, at a waste-stream temperature of 500°F the highest ROI is 30.7%, for the methanol-based system; the Freon R 11 system has the lowest ROI, 27%. As was the case for the 600-kW systems, the highest ROI for the 500°F and 700°F waste-stream temperatures is achieved with a methanol-based system and for the 900°F and 1100°F waste-stream temperatures with a system based on 2-methyl pyridine/H₂O.

Finally, Fig. 5.6 shows the ROIs of 2400-kW systems for the seven fluids examined. Comments made about the smaller systems are also applicable here. Because of economies of scale, the 2400-kW ROIs are higher than those of the smaller systems.

For systems of all sizes, the methanol-based RCPS yields the highest ROI for waste streams of 500°F and 700°F, and the 2-methyl pyridine/H₂O RCPS yields the highest ROI for waste-stream temperatures of 900°F and 1100°F.

Fig. 5.4 Effect on ROI of Waste-Stream Temp: 600-kW RCPS

Fig. 5.5 Effect on ROI of Waste-Stream Temp: 1200-kW RCPS

ROI rankings of the other fluids vary with size and waste-stream temperature -- with both of those factors dependent on varying economies of scale, specific system transport, and each fluid's thermodynamic characteristics.

In general, all the systems considered have ROIs that can significantly exceed the hurdle rates of the most energy-intensive industries. Results can be affected by both waste-heat temperature and available mass-flow rate of the rejected heat stream. Although the steel industry has high hurdle rates, it also has high-temperature streams of waste heat and large mass-flow rates. With Rankine-cycle power systems it is possible to take advantage not only of the high waste-heat temperatures available in the steel industry but also the economies of scale inherent in large systems.

Fig. 5.6 Effect on ROI of Waste-Stream Temp: 2400-kW RCPS

6 CONCLUSIONS AND RECOMMENDATIONS

The main conclusions of this study -- concerning thermophysical, health, and safety characteristics; optimal system designs; and return on investment -- can be summarized as follows.

The working fluids studied do not possess any characteristics that would preclude their use in Rankine-cycle power systems. Generally, any safety or health problems that may be caused by the use of these working fluids can be controlled by careful system design, fabrication, and operation. These requirements will not affect system economics to such an extent as to cause one fluid to be superior to another. All of the working fluids considered have survived screening processes that could have eliminated them from further consideration as a Rankine-cycle power system working fluid; none was eliminated. Selection of a working fluid, all else being equal, will favor less toxic substances.

Barring any material-compatibility problems that may yet be discovered (and some are expected to exist), the systems studied have comparable returns on investment. The range of ROIs is so narrow as to preclude any definite preference for a working fluid in a given temperature range. The fluid that apparently has the most uncertainty in this regard is Fluorinol 85. If stainless steel is required in the hotter sections of the system -- e.g., the vaporizer and piping to the turbine -- the capital cost of the system could increase by as much as a factor of two. It is recommended that this problem be studied further if it is believed that a system based on Fluorinol 85 offers important advantages other than economics. Smaller system size for a given power output may be one such advantage, especially for RCPS applications on trucks and tug boats.

Based on fluid-stability temperatures and system ROIs, certain fluids appear most desirable for certain inlet temperature ranges. The fluids by temperature range are:

- < 400°F methanol, 2-methyl pyridine/H₂O
- 500°F to 700°F methanol, 2-methyl pyridine/H₂O
- 700°F to 900°F 2-methyl pyridine/H₂O, toluene, water
- 900°F to 1100°F 2-methyl pyridine/H₂O, toluene, water

The spread of ROIs is limited, however, and design innovations or other system considerations could influence these choices. For example, a lower cost of Fluorinol 85 coupled with a low fluid-inventory design could change its currently somewhat unfavorable position. Based on ROIs alone, the selection is not well defined; the range is only about three percentage points. The highest ROIs occur with a 300°F condensing heat source, and any of the working fluids can be used at that temperature.

Relatively strong economies of scale exist for organic Rankine-cycle systems over the power range of 600-2400 kW. Economies of scale are realized mainly from total installation costs and the cost of a power conversion module. The vaporizer has small economies of scale; it constitutes about 25% of the cost of a 600-kW system, but almost 40% of a 2400-kW system.

ROIs for Rankine-cycle power systems can exceed the investment hurdle rates for energy-intensive industries, as demonstrated by national projections of future electricity prices. The market would be more attractive in those regions with higher-than-average electricity prices over the next 20 years. Aside from expectations about future electricity prices, capacity factors are most critical in determining system ROIs. It does not appear that tax credits or accelerated-depreciation schemes, except in marginal cases, will have a strong effect on investment decisions.

Although the ROIs for Rankine-cycle power systems are attractive, it is not clear that they represent the best choice as an energy-conservation measure for all industrial waste-heat streams. For example, a 1200-kW RCPS based on 2-methyl pyridine/H₂O and attached to a furnace with a flue-gas outlet temperature of 1100°F (593°C) has an ROI of 43%. This exceeds the hurdle rate for most industries. However, a heat exchanger installed to preheat combustion air would have an ROI of 105% and would thus be the obvious free-market choice. It is recommended that a closer look at RCPS markets be undertaken, especially to determine in which industries Rankine-cycle power systems are competitive with other uses of waste streams.

APPENDIX A

SYSTEM THERMODYNAMIC ANALYSIS

A.1 ANALYSIS OF RANKINE-CYCLE SYSTEM

In order to compare the performance of the various working fluids, a thermodynamic analysis of the heat-recovery system was required. A computer code was developed for that purpose. The code was used to calculate the necessary system characteristics (e.g. temperatures, enthalpies, flow rates) over a range of operating pressures and temperatures. The resulting information was then used with the economic models to determine system costs.

The computer code can be used to perform a system analysis for any fluid, wetting or drying, provided that thermodynamic data for the fluid are available. The analysis can be done for an idealized cycle or for prescribed values of component efficiencies. Either a specific cycle-operating temperature or a range of operating temperatures could be specified for a particular computer run.

To grasp the methodology followed in the thermodynamic analysis, consider Figs. 2.1 and 2.2 in this report. Figure 2.2 illustrates the major hardware components in a Rankine-cycle power system. The corresponding h-s diagram for the cycle is given in Fig. 2.1. Although this design shows a regenerator (or recuperator), one would not be included for wetting fluids. The heat source in Fig. 2.1 is assumed to be a gaseous stream. The pinch point is the minimum temperature difference between the exhaust gas and the working fluid.

At state point 1 in the cycle, the working fluid has left the condenser and is in the liquid state. The pressure of the working fluid is then increased as it passes through the pump. For an incompressible fluid, the ideal work per pound of fluid that is done in the pump, w_{pi} , is determined from

$$w_{pi} = v_1 (P_2 - P_1) \quad (A.1)$$

where:

v_1 = specific volume of the working fluid, and

P = fluid pressure.

Inefficiencies would be present so that the actual pump work, w_{pa} , would be given by

$$w_{pa} = w_{pi} / \eta_p \quad (A.2)$$

where:

η_p = specified pump efficiency.

When a regenerator is included in the system, the working fluid is preheated prior to entering the vaporizer. In order to calculate the fluid conditions at state point 3, the amount of heat transferred in the regenerator

needs to be determined. This was done by specifying the regenerator effectiveness and an overall heat-transfer coefficient. Calculation of the heat-transfer coefficients for the regenerator, as well as those for the vaporizer and condenser, is discussed in Sec. A.3 below.

In the vaporizer, the working fluid is heated, vaporized, and sometimes superheated. In order to determine the vaporizer cost, its heat-transfer surface area must be obtained. For that purpose, the vaporizer was divided into three sections: an economizer, a boiler, and a superheater. Log mean temperature differences and heat-transfer rates were calculated for each section. The sectional surface areas, A , were then determined from

$$A = \frac{Q}{U \Delta T_{lm}} \quad (A.3)$$

where:

Q = heat-transfer rate,

U = overall heat-transfer coefficient, and

ΔT_{lm} = the corresponding log mean temperature difference.

The heat exchangers in the system were assumed to be of the counterflow type; hence, T_{lm} is given³⁹ by

$$\Delta T_{lm} \equiv \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \left(\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}} \right)} \quad (A.4)$$

where:

T_{ci} and T_{co} = inlet and outlet temperatures of the cold fluid, and

T_{hi} and T_{ho} = inlet and outlet temperatures of the hot fluid.

The surface areas for the three sections of the vaporizer were added together to obtain the total heat-transfer surface area. For exhaust-gas streams, it was assumed that finned tubes were used in the vaporizer.

At the vaporizer exit, the working fluid will be saturated or superheated vapor. The fluid properties at this point were specified in the input data. They included the fluid pressure, P_4 ; temperature, T_4 ; enthalpy, h_4 ; and entropy, s_4 . The fluid now expands through a turbine and produces work. Ideally, the expansion would be isentropic, with the turbine work per pound of fluid, w_{ti} , given by

$$w_{ti} = h_4 - h_{5s} \quad (A.5)$$

where:

h_{5_s} = fluid enthalpy at the turbine exit.

Because of turbine inefficiencies the actual turbine work, w_{t_a} , would be less, i.e.,

$$w_{t_a} = h_4 - h_5 \quad (A.6)$$

State points h_5 and h_{5_s} are located on the isobar corresponding to the condensing pressure. In the computer code, the ideal turbine work, w_{t_i} , was calculated from the fluid thermodynamic properties. The actual turbine work was then obtained from

$$w_{t_a} = \eta_t w_{t_i} \quad (A.7)$$

where:

η_t = a specified turbine efficiency.

The turbine exit enthalpy for an isentropic expansion, h_{5_s} , was calculated by one of two techniques, depending on the conditions at the turbine exit. If the fluid is superheated at state point 5_s (as shown in Fig. 2.1), h_{5_s} would be determined by interpolation from superheated enthalpy data at the condensing pressure. For a wet expansion, the fluid would not be superheated at state point 5_s ; rather, it would contain a certain percentage of moisture. Here the vapor mass fraction or quality, x , would be calculated from

$$s_{5_s} = x s_g + (1-x) s_f \quad (A.8)$$

where:

s_g and s_f = the saturated vapor and liquid entropies, respectively, at the condensing temperature.

Rearranging Eq. A.8 into another form gives

$$s_{5_s} = s_g - (1-x) s_{gf} \quad (A.9)$$

where:

$$s_{gf} = s_g - s_f \quad (A.10)$$

For an isentropic turbine expansion,

$$s_{5_s} = s_4 \quad (A.11)$$

In the computer code, s_4 , s_g , and s_{gf} were known from the input data. Hence, the turbine exit quality, x , could be calculated from Eq. A.9 and h_{5_s} would be given by

$$h_{5_s} = h_g - (1-x) h_{gf} \quad (A.12)$$

where:

h_g = the known saturated-vapor enthalpy, and

h_{gf} = the known heat (enthalpy) of vaporization, both at the condensing temperature.

Once h_{5s} had been determined for either a wetting or drying fluid, w_{t_i} was calculated from Eq. A.5 and w_{t_a} from Eq. A.7. Then the turbine exit enthalpy for the actual expansion process, h_5 , could be obtained from Eq. A.6.

The corresponding temperature at the turbine exit, T_5 , was also evaluated in one of two ways. If the fluid at state point 5 is superheated, as shown in Fig. 2.1, T_5 would be determined by interpolation from temperature data at the condensing pressure. For a wetting fluid, T_5 is the condensing temperature, specified as an input parameter.

In a wet expansion, the working fluid enters the condenser directly after leaving the turbine. If the fluid is superheated at the turbine exit, a regenerator could be added, as shown in Fig. 2.1. Here, heat is transferred from the superheated vapor to the liquid working fluid. Unfortunately, the amount of heat transferred in the regenerator is not known a priori. The regenerator heat-transfer surface area, and the fluid-exit temperatures (T_3 , T_6) are also unknown. In order to calculate these quantities, the regenerator effectiveness was specified.

Consider a heat exchanger in which

$$(mc_p)_h < (mc_p)_c \quad (A.13)$$

where:

$(mc_p)_h$ = the product of the mass flow rate and specific heat of the hot stream, and

$(mc_p)_c$ = the corresponding quantity for the cold stream.

In this case, the heat-exchanger effectiveness, ϵ , is given by

$$\epsilon = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}} \quad (A.14)$$

These temperatures were defined in terms of Eq. A.4. The regenerator effectiveness thus becomes

$$\epsilon = \frac{T_5 - T_6}{T_5 - T_2} \quad (A.15)$$

If the temperature rise through the pump is neglected (this temperature increase was exaggerated in Fig. 2.1, for clarity), ϵ can also be expressed

in terms of enthalpies as

$$\epsilon = \frac{h_5 - h_6}{h_5 - h_g} \quad (\text{A.16})$$

where:

h_g = the saturated vapor enthalpy at the condensing temperature, as defined earlier.

In the code, h_g was known from the input data, and was specified as an input parameter. Since $T_1 \cong T_2$ is the known condensing temperature, T_6 can be calculated from Eq. A.15 and h_6 from Eq. A.16. The liquid exit temperature, T_3 , was calculated by an interpolation scheme from saturation temperatures and saturated liquid enthalpies. The regenerator heat-transfer surface area was determined from Eq. A.3.

After leaving the regenerator, the working fluid enters the condenser. The log mean temperature difference was calculated from the known condensing temperature and the cooling water temperatures. Eq. A.3 was used to obtain the condenser area.

Having determined the necessary thermodynamic variables, the cycle thermal efficiency, η_{th} , could be calculated. The cycle efficiency was calculated from

$$\eta_{th} = \frac{w_{ta} - w_{pa}}{q} \quad (\text{A.17})$$

where:

q = the heat added (Btu/lb_m) during the cycle.

The heat input was determined from

$$q = h_4 - h_3 \quad (\text{A.18})$$

A.2 TURBINE DESIGN AND FLUID PROPERTIES

Rankine-cycle power systems use turbines to convert thermal energy in a working fluid to mechanical energy. The mechanical energy can be used directly to drive pumps and compressors or electrical generators. The design of turbines for use with fluids other than steam is based, essentially, on well known thermodynamic and fluid dynamic theory and practice.^{2,40,41,42,43} The purpose here is to explain, briefly, the basis for the turbine efficiencies used in this study as well as the number of required stages -- both turbine efficiency and staging exert strong influence on turbine performance and cost determinations.

In general, two types of turbines are used: impulse and reaction. In an impulse turbine, the pressure drop occurs in the inlet nozzle, or stationary stage, in which the thermal energy in the fluid is converted into kinetic

energy. The fluid pressure remains essentially constant in passing through the moving blades of the stage, while the change in direction and magnitude of the fluid velocity causes the blades to move. In a reaction turbine, the pressure drop and change in fluid velocity occur both in the moving and stationary blades. Purely reaction turbines are rarely used and then usually in the latter stages of a multistage turbine. The various types of turbine staging and staging arrangements are shown in Fig. A.1.

Of particular interest in turbine design is the total enthalpy drop or pressure drop across the turbine. These quantities, particularly the enthalpy change, will help determine the number of stages required as well as the turbine-wheel diameter and rotational speed. Figure 2.2 shows a typical schematic of a Rankine-cycle power system. Figure 2.1 shows the corresponding Rankine cycle on a temperature-enthalpy diagram. It should be noted that a regenerator will only be required if there is a significant amount of fluid superheat at the turbine exit, due either to the fluid's basic thermodynamic properties or inefficiency in the turbine. Heat is added to the working fluid from state points 2 to 4, through a combination of a waste-heat boiler and a regenerator, as required. Ideally, work is done by the turbine as the fluid expands isentropically from state point 4 to 5_s. Taking into account the turbine inefficiency, the expansion actually occurs from state point 4 to 5. Heat is rejected from the fluid, from state point 5 to 1, with a combination of a regenerator and a condenser as required.

The total work done by the turbine is given by

$$W_T = \eta_T (h_4 - h_{5s}) \quad (\text{A.19})$$

where:

W_T = work done by turbine,

η_T = turbine efficiency,

h_4 = turbine inlet enthalpy, and

h_{5s} = isentropic outlet enthalpy.

This equation can also be written as

$$W_T = h_4 - h_5 \quad (\text{A.20})$$

Fig. A.1 Types of Turbine Staging (Adapted from Ref. 42)

where:

h_5 = the actual turbine outlet enthalpy.

In order to design a turbine that can operate at its maximum efficiency, it is necessary to know the ratio of the turbine blade speed to the theoretical fluid velocity entering the stage. Using the steady flow heat balance, it follows that

$$h_4 + \frac{v_4^2}{2gJ} = h_{5s} + \frac{v_5^2}{2gJ} \quad (\text{A.21})$$

where:

V_4 and V_5 = the inlet and outlet fluid velocities,
respectively, at the nozzle;

g = gravitational acceleration; and

J = thermal-to-mechanical energy-conversion factor.

The inlet velocity, as a first approximation, is considered to be zero. Thus, the theoretically attainable exit velocity from the nozzle is

$$V_5 = \sqrt{2gJ (h_4 - h_{5s})} \quad (\text{A.22})$$

Since, in engineering units, g is 32.2 ft/s² and J is 778 ft-lb/Btu, then the theoretical fluid velocity is

$$V_5 = 223.8 \sqrt{h_4 - h_{5s}} \quad (\text{A.23})$$

This equation holds for all fluids.

Experience has shown that to obtain maximum power from a turbine, the ratio of the blade pitch diameter velocity to the theoretical fluid velocity must be optimized. This relationship between turbine efficiency and the blade/fluid velocity ratio is shown in Fig. A.2. As an example, consider a small, single-stage steam turbine, with inlet conditions of 560°F at 150 psia exhausting to atmospheric pressure. The isentropic enthalpy change is 198.7 Btu/lb, so that the theoretical fluid velocity is 3154 ft/s. To reach the optimal velocity ratio of 0.45 for a single-stage impulse turbine, the blade pitch diameter velocity should be about 1419 ft/s. To achieve a rotational speed of 3600 rpm, the wheel pitch diameter would have to be 90.3 in., which is quite large and would require an expensive casing. Conversely, if the pitch diameter were held to a more reasonable 24 in., the speed would have to be 13,550 rpm, which could put severe stresses on the blade roots. In most cases, small steam turbines have efficiencies in the range of 45 to 50% mainly because of these economic and mechanical design considerations.

Organic fluids offer significant advantages over water as a Rankine-cycle working fluid. Their thermodynamic properties, due to their high molecular weight, allow low superheat at the turbine inlet and low moisture content at the turbine exhaust. Further, the organics have, in general, very

Fig. A.2 Turbine Average Efficiency Vs. Velocity Ratio
(Adapted from Ref. 42)

low enthalpy changes, and this property allows more reasonable turbine designs at the operating temperatures of interest. However, they do have such disadvantages as potentially high cost, need for tighter seals, the possibility of being toxic, and possible instability at operating temperatures.

Figure A.3 shows the effect that molecular weight has on the thermodynamic characteristics of the working fluid. Of particular interest here is the slope of the saturated-vapor line. A high-molecular-weight fluid, as shown in (a), has a positive slope. Thus, it is possible to heat the fluid so that it has no superheat at the turbine inlet but at the same time will, in general, have superheat at the turbine exit. Fluids such as the Freons and toluene have this characteristic. Wetting fluids such as water and methanol, shown in (b), could require significant superheat at the turbine inlet to avoid excessive moisture at the exit point. These fluids exhibit a negative slope of the saturated-vapor line. Some fluids can exhibit a nearly vertical saturated vapor line, as shown in (c). To achieve this, it is possible to combine "wetting" and "drying" fluids that have this characteristic. Fluids such as the 2-methyl pyridine/ H_2O mixture and proprietary fluids such as Fluorinol 85 have this characteristic.

In the temperature ranges of interest in this study, the pure organic working fluids have relatively low enthalpy changes in the turbine. In general, the enthalpy changes are less than 100 Btu/lb. Mixtures that contain a significant amount of water, such as the 2-methyl pyridine/ H_2O fluid (which contains 80% water, by volume) could have enthalpy changes on the order of 100-200 Btu/lb.

As noted above, this enthalpy change is crucial in turbine design, especially if we wish to maintain turbine efficiencies as high as possible.

Fig. A.3 Rankine Cycle with Various Working-Fluid Characteristics

Thus, we have decided, somewhat arbitrarily but nevertheless based on turbine-design requirements, that the number of stages required will be a function of the enthalpy drop. These are shown in Table A.1. In general, the pure organic fluids will require only a single turbine stage while the organic fluid/water mixtures will require three turbine stages. The higher-efficiency steam systems will require a larger, multistage turbine. With this breakdown, we are then able to assume that it is possible to design a turbine with an efficiency of about 70-80% for all systems. However, due to the turbine staging requirements, the capital costs would vary, perhaps considerably, for each system.

A.3 HEAT-EXCHANGER ANALYSIS

The costs of the heat-transfer components (vaporizer, regenerator, condenser) are based on heat-transfer surface area. In order to determine heat-exchanger costs, therefore, the total surface area of each unit must be calculated. This requires a knowledge of the overall heat-transfer coefficients of the various heat exchangers. Calculation of the overall heat-transfer coefficients is described in this section.

The condenser was taken to be a shell-and-tube heat exchanger, with the condensing fluid on the shell side and cooling water on the tube side. The tube diameter was 0.75 in., and the length was taken as 20 ft. These are the tube dimensions of the condenser whose cost data were used for this study. The cooling water was assumed to enter the condenser at 85°F and exit at 110°F. A velocity of 7 ft/s was selected for the cooling water.⁴⁴ The tube-side film coefficient was calculated from the relation given by McAdams³⁹ for fluids being heated in a tube:

Table A.1 Turbine Stages Required for a Given Enthalpy Change

Enthalpy Change (Btu/lb)	Required Number of Turbine Stages
< 100	One
100-200	Three
> 200	More than three

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (A.24)$$

where:

$$\text{the Reynolds number, } Re = \frac{\rho V D}{\mu}, \text{ and} \quad (A.25)$$

$$\text{the Prandtl number, } Pr = \frac{\mu c_p}{k}. \quad (A.26)$$

Here, ρ is the fluid density, V the fluid velocity, D the tube diameter, μ the fluid viscosity, c_p the fluid specific heat, and k the fluid thermal conductivity. The tube-side film coefficient, h_i , can then be determined from

$$Nu = \frac{h_i D}{k} \quad (A.27)$$

The cooling-water properties were evaluated at an average fluid temperature (100°F).

The condensing coefficient, h_c , was calculated from the expression in Kern⁴⁵

$$h_c = 1.51 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} \left(\frac{4W}{\mu L} \right)^{-0.333} \quad (A.28)$$

where:

g = the acceleration due to gravity,

L = the tube length, and

W = the condensate mass-flow rate per tube.

Equation A.28 applies to condensation on the outside of horizontal tubes. The condensate properties (ρ , k , μ) were calculated at 125°F.

The overall heat-transfer coefficient for each heat exchanger, U , was determined from

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + R_i + R_o \quad (A.29)$$

where:

h_i and h_o = tube-side and shell-side film coefficients, respectively; and

R_i and R_o = tube-side and the shell-side fouling resistances, respectively.

Typically, the recommended values of R_i and R_o range from 0.0005-0.001 hr-ft²-°F/Btu for organic fluids and water.⁴⁶ However, caution must be used

when applying fouling resistances in situations where h_i and h_o are large, as the use of high fouling resistances will result in unrealistically low values of U . This is particularly true of water, where the heat-transfer coefficients are much higher than those for the organic fluids. Generally, a fouling resistance of $0.001 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$ is satisfactory in most cases.⁴⁴ For this study, a total fouling resistance ($R_i + R_o$) of $0.001 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$ was used in calculating U . The tube-wall resistance, which typically is quite small, was neglected. The overall heat-transfer coefficients for the condenser, U_c , are given in Table A.2.

The regenerator was taken to be a shell-and-tube heat exchanger with the relatively high-pressure liquid on the tube side. The tubes were 0.75 in. in diameter and 20 ft long. The film coefficient on the tube side was calculated from Eq. A.24. The tube-side fluid velocities were assumed to be 5 ft/s.⁴⁴ The shell-side film coefficient, h_o , was determined from

$$\frac{h_o D}{k} = 0.33 (\text{Pr})^{0.333} \left(\frac{DG_{\max}}{\mu} \right)^{0.6} \quad (\text{A.30})$$

where:

D = the tube diameter (0.75 in.), and

$G_{\max}$ = the maximum mass-flow rate ($\text{lb}_m/\text{ft}^2\text{-hr}$)
on the shell side.³⁵

The maximum acceptable vapor velocity on the shell side must be chosen using both heat-transfer and pressure-drop considerations. Higher velocities improve the heat-transfer coefficient, but they also increase the pressure drop. Ludwig gives the maximum recommended shell-side vapor velocity as a function of the fluid molecular weight and operating pressure.³⁶ Here, the operating pressure is the condensing pressure at a saturation temperature of 150°F , which will be different for each fluid. The maximum recommended velocities from Ref. 36 were used to calculate $G_{\max}$.

The average tube-side and shell-side temperatures in the regenerator will vary for each fluid, depending on the cycle. The primary emphasis of this report, however, is on the relative differences between fluids, rather than on detailed design calculations for a specific system. Hence, it was decided to evaluate the tube-side fluid properties at 200°F and the shell-side properties at 250°F for all fluids. These values represent reasonable averages over a range of cycle conditions.

The overall heat-transfer coefficients for the regenerator, U_R , are given in Table A.3. Because water and methanol are wetting fluids (the saturated-vapor slopes in the temperature-entropy diagrams are negative), regenerator heat-transfer coefficients were not calculated for these fluids.

The determination of an overall heat-transfer coefficient for the vaporizer is more difficult. The fluid on the shell side will either be exhaust gas or 300°F condensing steam. The working fluid will enter the vaporizer as a liquid, evaporate, and then possibly be superheated. Thus, three distinct regimes must be considered when evaluating the heat-transfer coefficients of the working fluids.

Table A.2 Overall Heat-Transfer Coefficients for a Condenser, U_c

Rankine-Cycle Working Fluid	U_c (Btu/hr-ft ² -°F)
Water	475
Methanol	265
2-methyl pyridine /H ₂ O	230
Fluorinol 85	175
Toluene	165
Freon R 11	125
Freon R 113	100

Table A.3 Overall Heat-Transfer Coefficients for a Regenerator

Rankine-Cycle Working Fluid	U_R (Btu/hr-ft ² -°F)
2-methyl pyridine /H ₂ O	20
Fluorinol 85	30
Toluene	25
Freon R 11	40
Freon R 113	35

Fortunately, some simplification results when the heat source is exhaust gas. This is due to the low film coefficient of the exhaust gas, which is of the order of 10 Btu/hr-ft²-°F.³⁵ Generally, finned tubes are used in vaporizers because of their low gas-film coefficients. Since the film coefficients on the tube-side are one-to-two orders of magnitude larger, the heat transfer is controlled by the resistance on the gas side. Thus, the overall heat-transfer coefficient is relatively insensitive to the film coefficient on the tube-side, regardless of the working fluid. Typically, these overall heat-transfer coefficients, with a 1/8"-thick steel tube wall, range from 5-6 Btu/hr-ft²-°F.³⁵ A value of 5.5 Btu/hr-ft²-°F was used in this study.

When the heat source is condensing steam at 300°F, the situation is more complex. Because the film coefficient for condensing steam is quite high (a value of 1500 Btu/hr-ft²-°F was used here), the overall heat-transfer coefficient is more sensitive to the film coefficients of the working fluid. Therefore, heat-transfer coefficients were calculated in each section of the vaporizer (economizer, boiler, superheater) for all fluids.

The film coefficients for the economizer and superheater were calculated from Eq. A.24. A velocity of 5 ft/s and a temperature of 200°F were used for the liquid in the economizer. For steam boilers, the recommended vapor velocities at the boiler exit were in the 30-60 ft/s range.⁴⁷ A value of 50 ft/s was used for the vapor velocity in the superheater section. The vapor properties were calculated at 250°F.

Boiling-film coefficients are difficult to predict accurately. Kern recommended a maximum value of 300 Btu/hr-ft²-°F for organics and 1000 Btu/hr-ft²-°F for water.⁴⁵ More recently, Kern and Kraus recommended somewhat higher boiling-film coefficients: 300-500 Btu/hr-ft²-°F for organics and

1000-2000 Btu/hr-ft²-°F for water.⁴⁸ In this study, a film coefficient of 400 Btu/hr-ft²-°F was selected for the organic fluids and 1500 Btu/hr-ft²-°F was used for water.

Based on these assumptions, heat-transfer coefficients were calculated for the various fluids. Table A.4 shows the overall heat-transfer coefficients for each section of the vaporizer. The relatively low heat-transfer coefficients in the superheater are due to the pressure constraints imposed by the 300°F heat source. The densities of superheated vapors were calculated at saturation pressures for 200°F. The vapor pressure of steam at 200°F is 11.5 psia. For Freon R 11, the corresponding pressure would be 105 psia. At these pressures, the density of the superheated vapor is quite low and this, in turn, reduces the heat-transfer coefficient in the superheater.

Table A.4 Overall Vaporizer Heat-Transfer Coefficients
for the 300°F Heat Source

Rankine-Cycle Working Fluid	U(Btu/hr-ft ² -°F)		
	Economizer	Evaporator	Superheater
Water	440	430	7.5
Methanol	295	240	30
2-methyl pyridine/H ₂ O	265	240	15
Fluorinol 85	250	240	40
Toluene	230	240	20
Freon R 11	205	240	85
Freon R 113	195	240	75

APPENDIX B

METHODOLOGY FOR DETERMINING MINIMAL-COST SYSTEMS

For each set of specified Rankine-cycle operating conditions (i.e., turbine inlet pressure and temperature), the thermodynamic analysis described in Appendix A.1 was conducted. The resulting information, together with the component cost data, was then used to determine the system cost at those conditions. Similar calculations were carried out over a range of turbine inlet temperatures and pressures. From these data, minimal-cost systems were obtained for all working fluids at given power outputs and waste-heat temperatures.

The component cost models were incorporated into the same computer code used for the thermodynamic analysis. The general methodology followed in calculating system costs is illustrated in Fig. B.1. After a working fluid had been selected, certain input data were required. These included the heat source and cooling water temperatures, component efficiencies, condensing temperature, and regenerator effectiveness (if a regenerator was included). In addition, the necessary fluid thermodynamic data were automatically read by the code.

The actual calculation sequence started when the code read a turbine inlet pressure and temperature from the available data. A thermodynamic analysis was then performed for this cycle, as described in Appendix A.1. Next, the compatibility of the cycle with the waste-heat stream was checked. This was done by calculating the pinch-point temperature difference (i.e., the minimum temperature difference) between the waste-heat stream and the working fluid. When this temperature difference was less than or equal to zero, i.e., when there was a crossover point in the vaporizer, a new set of turbine inlet conditions was selected. If the cycle was acceptable, the component costs and the overall system cost were then calculated. The above process was repeated for each set of turbine inlet conditions in the input data.

The aim of the present study was not to perform a detailed design analysis of a specific Rankine-cycle power system but rather to compare the relative performance of potential working fluids on a consistent basis. Consequently, a series of assumptions regarding component behavior and system design constraints was required in developing the model. One of our major assumptions was to neglect the temperature rise across the pumps. The other major assumptions are listed in Table B.1. Several of these are discussed below.

A maximum turbine inlet pressure of 500 psia was specified. Also, in calculating the pump cost, the pressure delivered by the pump was assumed to exceed the turbine inlet pressure by 100 psia to allow for pressure drops within the system. This pressure drop was selected based on previous designs.^{4,20} The maximum turbine inlet pressure of 500 psia (maximum system pressure of 600 psia) was chosen to avoid the increased costs associated with constructing and sealing high-pressure systems.

The turbine size limits were based on design considerations. With organic fluids, maximum efficiencies (70-80%) can be obtained using single-stage turbines if the isentropic enthalpy drop is approximately 100 Btu/lb_m

Fig. B.1 Computer-Code Scheme for Calculating RCPS Costs

or less.⁴⁹ For steam systems to achieve high overall efficiencies, however, the turbine isentropic enthalpy drops will exceed 200 Btu/lb_m. These larger enthalpy drops require the use of multistage steam turbines to maintain reasonable wheel diameters and tip speeds. Thus, it was assumed that single-stage turbines were used for isentropic enthalpy drops up to 100 Btu/lb_m and multistage steam turbines were used for enthalpy drops of more than 200 Btu/lb_m.

Table B.1 Major Model Assumptions and Parameters

Maximum turbine-inlet pressure (psia)	500
Overall system pressure drop (psia)	100
Turbine size limits based on isentropic enthalpy drops (Btu/lb _m)	
Single-Stage	$\Delta h \leq 100$
Three-Stage	$100 < \Delta h \leq 200$
More than three stages	$\Delta h > 200$
Turbine moisture limit (%)	10
Condensing temperature (°F)	150
Regenerator effectiveness (%)	80
Pump efficiency (%)	75
Gearbox efficiency (%)	97
Generator efficiency (%)	95
Turbine efficiency, both single-stage and three-stage (%)	75
Multistage steam-turbine efficiency (%)	65-70
Cooling water temperature range (°F)	85-110
Exhaust-gas exit temperature (°F)	300
Exhaust-gas specific heat (Btu/lb _m -F)	0.25

Cost data were collected for both single-stage turbines and multistage steam turbines, including a three-stage turbine. If the isentropic enthalpy drop was in the intermediate range of 100-200 Btu/lb_m, the system was assumed to include a three-stage turbine, and the cost of the turbine was calculated accordingly.

Based on the above criteria, the working fluids fell into several categories for the conditions of this study. For systems using Freon R 11, Freon R 113, toluene, and Fluorinol 85, single-stage turbines were used. Systems based on methanol and 2-methyl pyridine/H₂O as working fluids included either single-stage or three-stage turbines, depending on the enthalpy drop. Steam systems used either a three-stage turbine or a more expensive multistage turbine with more than three stages. For wetting fluids, a moisture limit of 10% in the turbine exhaust was imposed to prevent excessive blade erosion.⁴⁹

A condensing temperature of 150°F was used in all cases. A detailed system design analysis, which was beyond the scope of the present study, could result in a different optimal condensing temperature for each working fluid. Lowering the condenser temperature will improve the cycle efficiency. However, the possibility of corrosion problems in the economizer could then

increase, particularly for systems with little or no regeneration. Typically, the performance of various organic fluids has been compared at a single condensing temperature, such as 100°F⁴⁹ or 150°F.³⁵ A similar approach was adopted here.

An exit temperature of 300°F was assumed for all exhaust-gas streams. A lower temperature was not selected because of potential corrosion problems, caused primarily by sulfuric acid condensing on the vaporizer tubes. Because the lowest tube-wall temperatures occur in the economizer, this is the area of greatest concern. The minimum recommended metal temperature to avoid corrosion is a function of the fuel type and the sulfur content of the exhaust gas.⁵⁰

Heat-recovery systems with regenerators have an advantage in corrosion prevention. A regenerator will raise the temperature of a working fluid that enters the vaporizer. The resulting temperature increase of the economizer tubes could be beneficial in reducing or eliminating corrosion problems.

Using the assumptions listed in Table B.1, overall system costs were calculated for the various working fluids. In order to determine the maximum return on investment (ROI) for each fluid, system performance and cost need to be optimized. The particular system selected as "optimal" could vary, depending on the nature of the heat source. In this study, the system power outputs and the heat-source temperatures were both stipulated. System costs were calculated and compared at specific power output levels. This requires a waste-heat stream with a large enough flow rate to provide the necessary power.

Alternatively, the systems could be optimized for a fixed heat source with a specified flow rate. In that case the power output would vary, depending upon the system efficiency. The minimum system cost (\$/kW) for each fluid would then probably occur at different power output levels.

The capital costs tabulated in Sec. 4 were determined for specific power outputs. The cost listed for each working fluid is the minimal calculated system cost at a given power output. To illustrate the procedure used in obtaining minimal system costs, consider Figs. B.2 and B.3. These figures show the installed costs plotted as a function of overall system efficiency. The system efficiency is defined as the electrical power output divided by the available energy (Btu/hr) from the heat source. The system efficiency was calculated by accounting for inefficiencies in the turbine, pump, electrical generator, and gearbox.

The calculated costs for a 600-kW toluene-based system bottoming a 700°F heat source are shown in Fig. B.2. The curve corresponds to the minimal installed costs at each value of system efficiency. The system efficiency improves with higher operating temperatures and pressures. The minimal installed cost occurs for a system with an overall efficiency of 15%. To the left of this point, the systems become less efficient and require higher flow rates of the working fluid to produce 600 kW. These higher flow rates result in larger installed costs, due primarily to cost increases in the heat-transfer equipment. For system efficiencies of more than 15%, the installed costs again increase, in this case principally due to higher vaporizer costs. This is caused by the decreasing pinch-point temperature difference in the

Fig. B.2 Cost Vs. Efficiency for Toluene-Based RCPS

Fig. B.3 Cost Vs. Efficiency for 2-MP/H₂O-Based RCPS

vaporizer, which occurs at the higher operating temperatures and pressures of the more efficient systems. The lower log mean temperature differences in the vaporizer lead to a larger heat-transfer area and, therefore, to higher vaporizer costs.

A single-stage turbine was used for all of the toluene-based systems. In addition, the minimal-cost systems in Fig. B.2 were those with little or no superheat.

Similar results for a system based on 2 methyl pyridine/H₂O are shown in Fig. B.3. In this case, a break in the curve occurs at the transition point ($\Delta h = 100 \text{ Btu/lb}_m$) between a single-stage and a three-stage turbine. The minimal calculated cost occurs, for a system with an efficiency of 11.5%, at the upper limit of the single-stage turbine regime. Systems with higher efficiencies than 11.5% use three-stage turbines.

Although regeneration was specified for both the toluene and 2-methyl pyridine/water fluids, some systems using the latter did not have a regenerator. Due to the nature of the saturated vapor curve of the 2-methyl pyridine/H₂O mixture, the fluid undergoes a wet turbine expansion at certain inlet pressures and temperatures. For these cases, a regenerator was not included in the system. The minimal-cost curves in Fig. B.3 correspond to systems without regeneration.

APPENDIX C
ECONOMIC DATA FOR MAKE-OR-BUY
DECISIONS ABOUT ELECTRICITY

In the economic analysis presented in Sec. 5, the national average electricity prices for the next 20 years were used. These prices were predicted by the Energy Information Administration (EIA).³⁷ The EIA gives the cost of electricity in the industrial sector for the years 1985, 1990, and 1995 in 1979 dollars. Using these values, the price of electricity was estimated for all the years in the period 1981-2000. However, it is necessary for the analysis to use the price of electricity not in constant dollars, but in current dollars. By "current dollars," it is commonly understood that prices of each year are expressed in dollars of that year, e.g., the price of electricity in 1988 is expressed in 1988 dollars. Prices in current dollars are needed in the analysis in order to account correctly for depreciation.

To convert future electricity prices into current dollars, the Consumer Price Index (CPI), as projected by Data Resources Inc. (DRI),⁵¹ was used. The CPI and the estimated cost of electricity in current dollars for the next 20 yr are presented in Table C.1. The cost of operating and maintaining a Rankine-cycle system was assumed to be \$0.0016/kWh, in 1981 dollars. Since a large portion of this cost is labor, the future cost was estimated using the index of compensation per hour for nonfarm workers, as projected by Data Resources, Inc. (DRI).⁵¹

Table C.1 Projected Future Costs of
Electricity and RCPS
Operation and Maintenance

Year	CPI ^a	Cost of Electricity (\$/kWh)	Compensation Index for Nonfarm Workers ^b (\$/h)	Cost of O&M (\$/kWh)
1981	2.68	0.0386	2.97	0.0016
1982	2.93	0.0441	3.27	0.0018
1983	3.15	0.0502	3.59	0.0020
1984	3.39	0.0573	3.94	0.0022
1985	3.65	0.0653	4.33	0.0024
1986	3.95	0.0717	4.75	0.0026
1987	4.27	0.0787	5.22	0.0029
1988	4.62	0.0864	5.70	0.0032
1989	5.00	0.0949	6.20	0.0034
1990	5.41	0.1042	6.75	0.0037
1991	5.81	0.1115	7.35	0.0041
1992	6.25	0.1190	7.98	0.0044
1993	6.72	0.1280	8.65	0.0048
1994	7.23	0.1370	9.36	0.0052
1995	7.77	0.1460	10.12	0.0056
1996	8.30	0.1581	10.93	0.0061
1997	8.86	0.1711	11.80	0.0065
1998	9.46	0.1851	12.73	0.0071
1999	10.10	0.2000	13.73	0.0076
2000	10.79	0.2170	14.81	0.0082

^aConsumer Price Index, as predicted by Data Resources, Inc., (1967 = 1).

^bAs predicted by Data Resources, Inc. (1967 = 1).

APPENDIX D

COMPONENT AND SYSTEM COST DATA

The installed capital costs of the Rankine-cycle systems are based on the assumption that the end user purchases a packaged system from a system "integrator." Installation is to be done by a subcontractor under the technical supervision of the system integrator.

A system integrator is a manufacturing entity that purchases all the components, provides the engineering to complete the system design and integration, and has the manufacturing facilities to integrate the components into skid-mounted subsystems. This approach will result in minimal installation time and least expense to the end user.

D.1 COMPONENT COSTS

Component costs are based on recent vendor quotes to industrial clients, discounted to the approximate prices charged by an original equipment manufacturer (OEM) to the system integrator. The components constitute standard industrial equipment of carbon-steel construction or that of equivalent cost. It is considered that carbon steel produces an acceptable degradation rate to a working fluid over the temperature range for which the fluid is competitive with the other candidate fluids. If stainless steel construction is desired, the cost ratio for 321 or 316 welded pipe is in the range of 4.25 times that of carbon steel. The costs of the individual components were converted to equation form, covering the range of sizes needed for direct use in the computer program. Some equipment cost curves are shown in Figs. D.1 and D.2. The various components are described below in terms of design and cost.

- Vaporizer (for clean waste gas)

Two-drum, bent finned-tube type; equipped with divertor valve, stack and fire detection/suppression system, in addition to standard-trim items

$$\text{Cost (\$)} = 9.2 \cdot 12,000 \cdot (A/12,000)^{0.93}$$

(A = area in ft²)

- Vaporizer (300°F source)

Kettle-type, shell-and-tube

$$\text{Cost (\$)} = 8.8 \cdot 2000 \cdot (A/2000)^{0.65}$$

- Condenser and regenerator

Shell-and-tube

Condenser cost (\$) =
 $11.4 \cdot 2000 (A/2000)^{0.82}$

Regenerator cost (\$) =
 $13.8 \cdot 2000 (A/2000)^{0.85}$

- Vaporizer feed pump

Low NPSH multistage turbine

Cost (\$) 800-ft total head =
 $2253 + 35.88 \cdot \text{gpm}$

Cost (\$) 1600-ft total head =
 $3815 + 64.41 \cdot \text{gpm}$

(Interpolate to obtain pump cost for other gallon-per-minute flow rate and pump head.)

- Cooling tower system

Forced draft, with temperature control

Installed cost (\$) = $26,500 (\text{gpm}/1045)^{0.92}$

Fig. D.1 Heat-Exchanger Cost Curves

Fig. D.2 Cost Curves of Turbine, Including Base and Governor

- Turbine, including base and governor

Single-stage axial-flow:

$$\text{Cost (\$)} = 31,167 + 23.58 P - 3.66 \cdot 10^{-3} \cdot p^2$$

Three-stage axial-flow:

$$\text{Cost (\$)} = 48,333 + 36.5 P - 5.64 \cdot 10^{-3} \cdot p^2$$

Multistage axial-flow:

$$\text{Cost (\$)} = 147,999 - 119.0 P + 7.39 \cdot 10^{-2} \cdot p^2$$

(P = power in kW)

- Generator and gear-reducer

Synchronous, with controls and switch gear

$$\text{Cost (\$)} = 10,133 + 28.0 P + 2.96 \cdot 10^{-3} \cdot p^2$$

- Building for power conversion module (PCM)

Modular-steel construction with fire detection and suppression system and draw-through ventilation system.

$$\text{Installed cost (\$)} = 14,867 + 9.08 P - 8.8 \cdot 10^{-4} \cdot p^2$$

D.2 MANUFACTURING COSTS

The investment cost plus working capital for a system integrator to produce 60 units/yr, each rated 600 kW, is estimated at \$3,500,000. The minimum acceptable return on investment before taxes for a high-risk industrial enterprise is assumed to be 40%.⁵² Presented as an add-on percentage to a single fabricated 600-kW unit, this corresponds to 6%. The overhead and profit as a percentage of fabricated cost is as follows:

Marketing, administration, engineering ⁵²	6%
Profit	<u>6%</u>
Total add-on percentage for overhead and profit	12%

This 12% is assumed to be a constant percentage of the fabricated cost of all the power-conversion systems. Labor and material cost to integrate the component parts into a skid-mounted system by the system integrator has been estimated at \$57,000 for a 600-kW unit. This fabrication cost will be increased by the 0.4 exponent of the ratio of the larger-capacity unit to the 600-kW unit.

D.3 INSTALLATION COSTS

The cost to install a base-case 600-kW energy-conversion system was estimated by using a modified Lang Method, which applies factors for direct and indirect installation costs to the cost of individual equipment items.^{53,54,55} This is done by multiplying an integrator's selling price of an item by an appropriate factor from Table D.1. Separate multipliers are used for the two major components because each has specific installation requirements. Interfacing with the utility and installation of the automatic sequential-start system inflate the electrical and engineering factors for the PCM. Interconnecting piping is included with the installation cost of the vaporizer.

All the organic fluids except Fluorinol 85 have a long history of safe use in the chemical industry. Fluorinol 85 has not had extensive industrial use; however, its reaction products and physical data indicate that no additional safety-related construction costs are required beyond those of the other fluids.

Safety-related equipment is included in the installation cost factors. Any plant requires this equipment, however, if it uses industrial chemicals that are flammable or create potentially toxic products of decomposition. The equipment required consists essentially of fire detection/suppression systems, vapor sniffers, and ventilation systems.

The cooling tower system and building to house the PCM are the responsibility of the plant. These have been priced as if they were being installed by a subcontractor.

The installation costs for 1200-kW and 2400-kW units do not increase proportionally with the increase in power output. There will be little significant difference between engineering costs of the 600-kW and the 2400-kW unit. Similarly, field expenses will not be significantly greater for the larger-capacity unit. Therefore, indirect installation costs will be a constant dollar value for all units using the same working fluid.⁵⁶

The direct installation costs will not be directly proportional to the increase in the size of the unit. The equipment is set with cranes and hoists, which are not appreciably affected by the size or weight of what is being lifted. Direct installation costs will vary according to the Williams "six-tenth factor,"⁵⁷ an exponent of the ratio of capacities, r , expressed as:

$$C_{d2} = r^{0.6} C_{d1} \quad (D.1)$$

where:

C_{d2} = direct cost of the larger unit,

C_{d1} = direct cost of the 600-kW unit, and

r = ratio of larger unit's capacity to that of the 600-kW unit.

Table D.1 Installation-Cost Multipliers Applied to Equipment
Sales Prices for the Two RCPS Components

Item	Vaporizer	Power Con- version Module
• Direct-Cost:		
Taxes and freight	0.06	0.06
Foundation and supports	0.04	0.01
Handling and erection	0.21	0.14
Electrical	0.04	0.12
Piping	0.04	0.02
Insulation	0.02	----
Site preparation	0.02	0.03
Construction fee	0.11	0.11
Contingencies	<u>0.03</u>	<u>0.03</u>
Total multiplier for direct-cost items	0.57	0.52
• Indirect-Cost:		
Engineering and supervision	0.09	0.20
Construction and field	0.03	0.03
Startup	0.02	0.02
Performance test	<u>0.01</u>	<u>0.02</u>
Total multiplier for indirect-costs items	0.15	0.27

The value of the exponent for the vaporizer is 0.5, and that for the PCM is 0.6. These were derived from labor and material costs for similar equipment.⁵⁸

D.4 COSTS OF ORGANIC WORKING FLUIDS

The cost of Fluorinol 85 was obtained from its producer, Halocarbon Products Corp. Halocarbon indicated that a larger scale of production would lower its cost by somewhat less than a factor of two. Toluene is used as an octane booster in unleaded premium gas and has therefore experienced a large price increase in recent months. Working-fluid costs are given in Table D.2.

D.5 WORKING-FLUID INVENTORY REQUIREMENTS

The inventory of a working fluid in an RCPS is contained in the tube side of the economizer, regenerator, piping and hot well, and vaporizer. The fluid inventory in the economizer and regenerator is proportional to the surface-area requirements of those components, and that of the piping and hot well is proportional to the circulation rate. The vaporizer inventory is dependent on the circulation rate and vapor-disengagement requirements. The vapor-disengagement space is a function of the specific volume and, therefore, of the fluid molecular weights. The following vaporizer inventory is based on requirements of similar waste-heat steam boilers.

Working Fluid Inventory (WFI)

$$\begin{aligned} \text{WFI (gal)} = & 0.02 \cdot \text{economizer ft}^2 + 0.08 \cdot \text{regenerator ft}^2 \\ & + 1.2 \cdot \text{flow rate (gpm)} \\ & + 23 \cdot \text{flow rate (gpm)} \cdot 18/\text{mw} \end{aligned}$$

(mw = molecular weight of working fluid)

The inventory requirement is based on a two-drum, natural-circulation vaporizer. Natural circulation is caused by the density difference between the vapor/liquid mixture in the steam tubes and the cooler liquid in the unheated downcomers. A circulation ratio of 1:15 or more can be maintained for systems operating below the critical point. This vigorous circulation eliminates vapor blanketing and its potentially catastrophic results.

The use of a carefully designed forced-circulation vaporizer can reduce the inventory requirements of a high-cost organic working fluid by one-half. Forced-circulation vaporizers achieve circulation with pumps. Their required drive and control represents an added power consumption. Higher maintenance is typical because of additional moving parts. To minimize the size, cost, and power consumption of circulating pumps, the circulation ratio is normally limited to one-half that of natural circulation. Circulation stops upon power failure. For these reasons, natural-circulation vaporizers are more reliable and have a greater longevity than forced-circulation vaporizers.

D.6 INSTALLED COSTS OF A TYPICAL RANKINE-CYCLE SYSTEM

Individual items of equipment cost were combined by the computer program to obtain the installed cost of a system. The sum of the equipment, manufacturing (integration), and installation costs for a typical system is shown in Table D.3. The relationship between the selling price by an integrator and the installed cost to the purchaser is also shown.

Table D.2 Working-Fluid Costs

Fluid	Price ^a (\$/gal)
Methanol	1
2-methyl pyridine /H ₂ O	9
Fluorinol 85	112
Toluene	1.70
Freon 11	15
Freon 113	20

^aAs of June 1981.

Table D.3 Rankine-Cycle Power System Costs^a

Item	Capital Costs		Sales Price (\$10 ³)
	Cost (\$10 ³)	% of Total Capital Cost	
• Vaporizer module			
Equipment cost	208.8	23	208.8
Mfg. cost	25.1	2.8	25.1
Installation cost	129.3	14.2	
• PCM module			
Equipment cost			
Regenerator	48.8	5.4	48.8
Pump	8.8	0.9	8.8
Turbine	54.2	6	54.2
Generator	48	5.3	48
Condenser	24.1	2.6	24.1
Mfg. cost	106.3	11.7	106.3
Installation cost	192.7	21.2	
• Cooling tower system	36	3.9	
• PCM building	24.5	2.7	
• Fluid inventory	2.8	0.3	2.8
Totals	\$909.4	100	\$526.9
	(\$760/kW)		(\$440/kW)

^aBased on: Fluid: toluene
Power output: 1200 kW
Waste-gas inlet temperature: 900°F
System efficiency: 18.1%

APPENDIX E

WORKING-FLUID PROPERTIES

E.1 DEFINITION OF WORKING FLUID I-FACTOR

In deciding the appropriateness of a working fluid for use in a Rankine-cycle power system, the slope of the saturated-vapor line in the temperature range of interest is important. If the line has a positive slope, the working fluid in exiting the turbine will probably be superheated. This can affect capital cost because a regenerator would then be required to boost system efficiency. If the saturated-vapor line is negative (as it is for steam, e.g.), then the fluid will have some percentage of moisture in the last stages of the turbine -- and this could erode the blades. The ideal saturated-vapor line is vertical to slightly negative in slope in a temperature-entropy chart. Thus, it would be possible to have the working fluid pass through a real turbine at nearly saturated conditions throughout. A fluid's I-factor is a measure of the slope of its saturated-vapor line in a given temperature range.

For a simple, compressible fluid, the entropy may be written as

$$ds = C_p \frac{dT}{T} - \left(\frac{\partial v}{\partial T} \right)_P dP \quad (E.1)$$

Along the saturated vapor line (sometimes called the dew-point curve) this may be represented by

$$\left(\frac{\partial s}{\partial T} \right)_D = \frac{C_p}{T} - \left(\frac{\partial v}{\partial T} \right)_P \left(\frac{\partial P}{\partial T} \right)_D \quad (E.2)$$

where:

D = a point on the saturated-vapor line.

Equation E.2 can be inverted to give

$$\left(\frac{\partial T}{\partial s} \right)_D = \frac{(T/C_p)}{1 - \frac{T}{C_p} \left(\frac{\partial v}{\partial T} \right)_P \left(\frac{\partial P}{\partial T} \right)_D} \quad (E.3)$$

The use of Eq. E.3 in calculating the slope of the saturated-vapor line is not convenient because it can assume a value of infinity for an isentropic fluid. However, a factor may be defined to avoid this problem, by rearranging Eq. E.3 to give

$$\frac{T}{C_p} \left(\frac{\partial v}{\partial T} \right)_P \left(\frac{\partial P}{\partial T} \right)_D = 1 - \frac{(T/C_p)}{\left(\frac{\partial T}{\partial s} \right)_D} \quad (E.4)$$

From this equation, the I-factor is defined as:

$$I(T) = 1 - \frac{\left(\frac{T}{C_p}\right)}{\left(\frac{T}{\partial s}\right)_D} = \frac{T}{C_p} \left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial T}\right)_D \quad (E.5)$$

For points on the saturated-vapor line far below the critical temperature, the following values of the I-factor are indicated.¹²

Vapor Isentropic Expansion	Slope of Saturated-Vapor Line	I-Factor
Wetting	< 0	> 1
Isentropic	∞	1+
Drying	> 0	< 1

If the gas (vapor) can be considered an ideal gas, then

$$Pv = RT \quad (E.6)$$

and

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P} \quad (E.7)$$

and hence

$$I(T) = \left(\frac{T}{C_p}\right) \left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial T}\right)_D = \frac{v}{C_p} \left(\frac{\partial P}{\partial T}\right)_D \quad (E.8)$$

If, on the other hand, we use the Clapeyron equation, then

$$\left(\frac{dP}{dT}\right)_D = \frac{h_{fg}}{T(v_g - v_f)} \quad (E.9)$$

The I-factor therefore becomes

$$I(T) = \frac{v_g}{TC_p} \frac{h_{fg}}{(v_g - v_f)} \quad (E.10)$$

For most real fluids, at conditions far away from the critical point $v_f \ll v_g$, the I-factor can be simplified to

$$I(T) \cong \frac{h_{fg}}{C_p T} \quad (E.11)$$

where:

P = pressure in absolute units (lb_f/ft^2 , Pa)

C_p = specific heat at constant pressure $\left(\frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}}, \frac{\text{kJ}}{\text{kg } ^\circ\text{C}} \right)$,

T = temperature ($^\circ\text{R}$, $^\circ\text{K}$),

h_{fg} = fluid heat of vaporization (Btu/lb_m , kJ/kg),

v_g = saturated-vapor specific volume (ft^3/lb_m , m^3/kg), and

v_f = saturated-liquid specific volume (ft^3/lb_m , m^3/kg).

For vapor conditions far from the critical temperature, the enthalpy of vaporization, h_{fg} , and the specific heat at constant pressure of the vapor are essentially constant. Therefore the I-factor is approximately inversely proportional to the absolute temperature.

The I-factor range of 0.65-1.5 was chosen because of the suggestion and results of the investigation reported in Ref. 12. Values of the I-factor for the specific fluids considered in this study are given in Table 3.1.

E.2 HEAT-TRANSFER CHARACTERISTICS

The ability of a high-temperature fluid to transfer thermal energy to a lower-temperature fluid is characterized by an overall heat-transfer coefficient. This value depends on the heat-transfer coefficients of each fluid as well as on the thermal conductivity of the solid wall between them. Moreover, fouling of the heat-exchanger tubes can significantly contribute to heat-transfer resistance.

In general, the heat-transfer coefficient of liquid water is higher than that of other liquids. The best means of comparing the heat-transfer coefficient of any organic liquid with that of water is by plotting their ratio as a function of temperature; Figs. E.1 and E.2 do this.

The heat-transfer characteristics of the organic vapors are compared with those of air as a function of temperature in Figs. E.3 and E.4. The heat-transfer coefficient of any fluid can be easily calculated for a given temperature by using Figs. E.1 and E.2 for liquids and Figs. E.3 and E.4 for gases. As shown in Fig. E.2, methanol has a higher heat-transfer coefficient than 2-methyl pyridine/ H_2O , toluene, Freon 11, or Freon 113. On the other hand, toluene vapor has a higher heat-transfer coefficient than the other organic fluids, as shown in Fig. E.4. These values can have a significant impact on the surface-area requirements of a system's heat exchanger and therefore its subsequent costs.

Fig. E.1 Heat-Transfer Coefficient Vs. Mass Velocity of 200°F Water

E.3 IDEAL EFFICIENCIES OF THE FLUID-BASED SYSTEMS

As discussed in Sec. 3, the critical characteristics of a Rankine-cycle working fluid include toxicity, thermal stability, chemical corrosivity with common metals, flammability, molecular weight, boiling point, number of atoms per molecule, and cycle efficiency.^{2,11} The following discussion bears only on cycle efficiency.

A comparison of the ideal cycle efficiencies of systems based on toluene, Fluorinol 85, 2-methyl pyridine/H₂O (35% 2-methyl pyridine, 65% water, by mole), and water indicates that, at maximum operating temperatures of 600°F, water has a rather high ideal cycle efficiency compared with Fluorinol 85 and 2-methyl pyridine/H₂O. However, when actual system efficiencies are calculated, a water-based system ranks lower than systems based on toluene and 2-methyl pyridine/H₂O because of turbine-design constraints and pinch-point behavior.^{2,3,11} Furthermore, the shape of the liquid-vapor equilibrium curve for water causes a reduced "heat addition efficiency;" for this reason also, efficiencies of a water-based system are rather low compared to those based on organic fluids.

As defined by Niggemann, et al.,⁴ "heat addition efficiency" is the "fraction of the heat-source energy that is recoverable by the working fluid at the boiler." This definition is based on the practical temperature limit

Fig. E.2 Ratio of Heat-Transfer Coefficient of Organic Liquid to That of 200°F Water (Calculated from data in Refs. 59,60,61)

imposed on the temperature of gas exiting from the boiler in order to avoid condensation of acid from the gas stream. This is usually 300°F; specifically, the heat addition efficiency is:

$$\eta_{HA} = \frac{T_{\text{gas in}} - T_{\text{gas out}}}{T_{\text{gas in}} - 300} \quad (\text{E.12})$$

"System efficiency" is defined to be the product of "heat addition efficiency" multiplied by "actual cycle efficiency." Thus:

Fig. E.3 Heat-Transfer Coefficient Vs. Mass Velocity of 200°F Air

$$\eta_{\text{sys}} = \eta_{\text{HA}} \eta_{\text{th},a} \quad (\text{E.13})$$

where:

η_{HA} = heat addition efficiency,
 $\eta_{\text{th},a}$ = actual thermodynamic cycle efficiency, and
 η_{sys} = system efficiency.

"Actual cycle efficiency" is the thermodynamic efficiency of the Rankine cycle. Here, it is assumed that both the turbine efficiency and the pump efficiency are 80% and the regenerator heat-exchanger effectiveness is 0.8. Because of steam's thermodynamic properties, it is assumed that steam turbines have an isentropic efficiency of 70%. On the basis of these assumptions, the system efficiency was determined for several working fluid candidates.

The results of these calculations are shown in Tables E.1 and E.2. The ideal cycle efficiencies of several fluid-based systems are shown in Figs. E.5 to E.9. As the tables and figures show, even when the ideal cycle efficiency for a steam-based system is higher than that of organic fluids, it can be less efficient than some other systems based on organic fluids; this occurs because of turbine and heat-addition inefficiencies. Further, because of large steam-turbine enthalpy drops, an unacceptable ratio of exhaust speed to

bucket speed is attained with single-stage turbines, which results in excessive turbine-blade stresses. Thus, single-stage turbines cannot be used. Use of multistage turbines, on the other hand, can result in a very uneconomical design.

Fig. E.4 Ratio of Heat-Transfer Coefficient of Organic Fluid Vapor to That of 200°F Air (Calculated from Data in Refs. 59,60,61)

Table E.1 Maximum Temperature and Efficiencies of the Fluid-Based Systems

	Water	Methanol	2 Methyl Pyridine/H ₂ O	Fluorinol 85	Toluene	Freon 11	Freon 113
Maximum system temp. (°F)	600	450	600	550	600	250	250
Ideal cycle efficiency (%)	31.4	24.7	28.7	27.7	34.8	13.6	13.7
Actual cycle efficiency (%)	21.3	19.7	22.9	21.1	28.8	10.7	10.9
Heat-addition efficiency (%)	67.5	79.2	100	100	64.5	-	-
Systems efficiency (%)	14.4	15.6	22.9	21.1	18.6	10.7	10.9

Table E.2 Ideal-System Performance Comparison of Four Working Fluids, Based on Two Temperature/Pressure Combinations at Turbine Inlet

Condition	Water	2 MP/H ₂ O	Fluorinol 85	Toluene
Turbine inlet temperature (°F)	600	600	600	600
Turbine inlet pressure (psia)	500	500	500	500
Turbine exhaust temperature (°F)	140	155	338	372
Turbine exhaust pressure (psia)	2.9	5	7	3
Turbine enthalpy drop (Btu/lb _m)	362	131	65	73
Condenser dew temperature (°F)	140	155	139	145
Condenser bubble temperature (°F)	140	145	137	145
Condenser pressure (psia)	2.9	5	7	3
Regenerator heat recovery (Btu/lb _m)	0.0	0.0	45	52
Ideal cycle efficiency (%)	31.4	28.7	30.5	34.8
System efficiency (%)	14.4	22.9	23.3	18.6
Turbine inlet temperature (°F)	600	600	600	600
Turbine inlet pressure (psia)	200	200	200	200
Turbine exhaust temperature (°F)	140	155	400	438
Turbine exhaust pressure (psia)	2.9	5	7	3
Turbine enthalpy drop (Btu/lb _m)	316	110	56	67
Condenser dew temperature (°F)	140	155	139	145
Condenser bubble temperature (°F)	140	145	137	145
Condenser pressure (psia)	2.9	5	7	3
Ideal cycle efficiency (%)	27	23.7	27.5	33.3
System efficiency (%)	16.1	18.9	20.6	18.5

Fig. E.5 Ideal Efficiency of System Based on Freon 11 Vs. Turbine Inlet Conditions

Fig. E.6 Ideal Efficiency of Toluene-Based System
Vs. Turbine Inlet Conditions

Fig. E.7 Ideal Efficiency of System Based on Fluorinol 85
Vs. Turbine Inlet Conditions

Fig. E.8 Ideal Efficiency of System Based on 2-Methyl Pyridine/H₂O Vs. Turbine Inlet Conditions

Fig. E.9 Ideal Efficiency of Water-Based System
Vs. Turbine Inlet Conditions

REFERENCES

- 1 Sternlicht, B., and D. Colosimo, *The Rebirth of the Rankine Cycle*, Mechanical Engineering, 103(1) (Jan. 1981).
- 2 Tabor, H., and L. Bronicki, *Establishing Criteria for Fluids for Small Vapor Turbines*, Society of Automotive Engineers Transactions, 73:561-575 (1965).
- 3 Palmer, David A., and Bruce E. Sirovich, *Selection of a Rankine Cycle Fluid for Recovery of Work from Heat at Moderate Temperature*, Society of Automotive Engineers, paper 789606 (1978).
- 4 Niggeman, R.E., W.J. Greenlee, and P.D. Lacey, *Fluid Selection & Optimization of an Organic Rankine Cycle Waste Heat Power Conversion System*, American Society of Mechanical Engineers, paper 78-WA/Ener-6 (1978).
- 5 Gunn, Marvin E., Jr., *Organic Rankine Cycle: Waste Heat Recovery System Tests are Supported by Department of Energy*, Public Power, p. 44 (March-April 1978).
- 6 Barber, R.E., et al., *Solar Powered Rankine Cycle Engines - Characteristics and Costs*, 11th Intersociety Energy Conversion Engineering Conf., paper No. 769200 (1976).
- 7 *Final Report for the Study of the Validation of the Application of Rankine Bottoming Cycle Technology to Marine Diesel Engines*, General Electric Co., Advanced Energy Programs Dept., G.E. Report No. AED-EO-17 (July 31, 1980).
- 8 *Industrial Markets for Organic Rankine Cycle Bottoming Systems*, Resource Planning Associates, Cambridge, Mass., RPA Report RA-78-1402(3) (Dec. 1978).
- 9 Snyder, P.H., and W.C. Moreland, *The Selection of Working Fluid Candidates for Use in High Temperature Industrial Heat Pumps*, DOE Contract No. EC-77-C01-5026, Westinghouse Electric Corp. (Dec. 1978).
- 10 Barber, R.E., *Rankine Cycle Systems for Waste Heat Recovery*, Chemical Engineering, pp. 101-104 (1974).
- 11 Bronicki, L.Y., *The Ormat Rankine Power Unit*, 7th International Energy Conversion Engineering Conf., pp. 327-334 (1972).
- 12 Miller, D.R., H.R. Null, and Q.E. Thompson, *Optimum Working Fluids for Automotive Rankine Engines*, Vol. I, II, III, and IV, APTD-1563, APTD-1564, APTD-1565, and APTD-1566 (June 1973).
- 13 Lewandowski, G.A., and R.B. Strus, *Rankine Bottoming Cycle Safety Analysis*, Vector Engineering Inc., DOE/ET/15445 (Feb. 1980).
- 14 Gainer, G.C., R.M. Luck, and W.C. Moreland, *Thermal Stability and Materials Compatibility Tests of Five High Temperature Working Fluid Candidates*, DOE Contract No. EC-77-C-01-5026 (Sept. 1977 - Dec. 1978).

- 15 Ajmera, Pravin V., Halocarbon Products Corp., Hackensack, N.J., personal communication (Aug. 1, 1980).
- 16 Prasad, Anil, Sundstrand Energy Systems, Sundstrand Corp., Rockford, Ill., personal communication (July 2, 1980).
- 17 Benning, A.F., and R.C. McHarness, *Thermodynamic Properties of Freon 113, Trichlorotrifluoroethane CCl₂F - CClF₂, with Addition of Other Physical Properties*, Organic Chemicals Division, E.I. DuPont de Nemours & Company, Inc., Wilmington, Del., T-113A and T-113B (1938).
- 18 Smith, J.M., *Thermodynamic Properties of Methyl Alcohol*, Chemical Engineering Progress, 44(7):521-528 (July 1948).
- 19 Miller, D.R., Monsanto Industrial Chemicals Co., St. Louis, personal communication (July 8, 1980).
- 20 Soini, H.E., et al., *Combined Diesel-Organic Rankine Cycle Power Plant*, 12th International Energy Conversion Engineering Conf., pp. 1100-1107 (1977).
- 21 Somekh, G.S., *Water-Pyridine Azeotrope is an Excellent Rankine Cycle Fluid*, J. Engineering for Power, American Society of Mechanical Engineers, pp. 583-588 (Oct. 1975).
- 22 Blair, E.K., et al., *Multi-Megawatt Organic Rankine Power Plant (MORE)*, DOE/ET/12434-T1 (Jan. 31, 1980).
- 23 *Thermodynamic Properties of Freon 11 Refrigerant*, Organic Chemicals Division, E.I. DuPont de Nemours & Co., Inc., Wilmington, Del., T-11 (1965).
- 24 *Thermophysical Properties of Refrigerants*, American Society for Heating, Refrigeration, and Air Conditioning Engineers (1973).
- 25 *Toluene, an Organic Working Fluid*, Sundstrand Aviation (4/29/70), revised (5/8/70).
- 26 *For the Study of the Validation of the Application of Rankine Bottoming Cycle Technology to Marine Diesel Engines*, Vol. 1 - Executive Summary, Vol. 2 - Final Technical Report, General Electric Co., Cincinnati, Report No. AED-EO-17 (July 31, 1980).
- 27 Blake D.A., et al., *Toxicology and Applied Pharmacology*, 15, 83 (1969).
- 28 Christensen, H.E., E.J. Fairchild, and R.J. Lewis, *Suspected Carcinogens*, 2nd Ed., *A Subfile of NIOSH Registry of Toxic Effects of Chemical Substances*, U.S. Department of Health and Human Services (formerly HEW) (Dec. 1976).
- 29 Fassett, D.W., and D.D. Irish, ed., *Industrial Hygiene and Toxicology*, Interscience Publishers, 2nd Ed., II:1409-1422 (1963).

- 30 Leonardos, G., D. Kendall, and N. Banard, *Odor Threshold Determination of 53 Odorant Chemicals*, Journal of Air Pollution Control Assn. 19:91-95 (1969).
- 31 Proctor, N.H., and J.P. Jughes, *Chemical Hazards of the Work Place*, J.B. Lippincott Co., p. 482 (1977).
- 32 Sax, N. Irving, *Dangerous Properties of Industrial Materials*, 5th Ed., Van Nostrand Reinhold Co., New York City (1979).
- 33 *Initial Report of the TSCA Interagency Testing Committee to the Administration*, Environmental Protection Agency, NTIS, publication PB-275, p. 237 (Oct. 19).
- 34 *Guide to Precautionary Labeling of Hazardous Chemicals*, 7th Ed., Manufacturing Chemist Assn., Washington, D.C. (1970).
- 35 Boyen, J.L., *Thermal Energy Recovery* (2nd Ed.), John Wiley and Sons, Inc., New York City (1980).
- 36 Ludwig, E.E., *Applied Process Design for Chemical and Petrochemical Plants/Vol. 3*, Gulf Publishing Co., Houston (1965).
- 37 *Energy Information Administration Annual Report to Congress 1979*, DOE/EIA-0173(79)/3, 3.
- 38 *Industrial Markets for Organic Rankine Cycle Bottoming Systems*, prepared for U.S. Department of Energy by Resource Planning Associates. Inc., RA-78-1402(3) (Dec. 1978).
- 39 Chapman, A.J., *Heat Transfer*, MacMillan Publishing Co., New York City (1974).
- 40 Mooney, D.A., *Mechanical Engineering Thermodynamics*, Prentice-Hall, Inc., Englewood Cliffs, N.J. (1953).
- 41 Van Wylen, G.J. and Richard E. Sonntag, *Fundamentals of Classical Thermodynamics*, 2nd Ed., John Wiley & Sons, Inc., New York City (1973).
- 42 *DeLaval Engineering Handbook*, H. Gartmann, ed., 3rd Ed., McGraw-Hill Book Co., New York City (1971).
- 43 Nurken, Richard E., *Use Steam Turbines as Process Drivers*, Chemical Engineering, pp. 63-78 (Aug. 25, 1980).
- 44 Frank, O., *Simplified Design Procedures for Tubular Heat Exchangers*, in Practical Aspects of Heat Transfer, American Institute of Chemical Engineers, pp. 1-25 (1978).
- 45 Kern, D.Q., *Process Heat Transfer*, McGraw-Hill Book Co., New York City (1950).
- 46 *Standards of Tubular Exchanger Manufacturers Association*, Tubular Exchanger Manufacturers Assn., Inc., New York City (1968).

- 47 Fraas, A.P., and M.N. Ozisik, *Heat Exchanger Design*, John Wiley and Sons, Inc., New York City (1965).
- 48 Kern, D.Q., and A.D. Kraus, *Extended Surface Heat Transfer*, McGraw-Hill Book Co., New York City (1972).
- 49 *Industrial Application Study*, Drexel University, United Technologies Research Center, and Mathematica, Inc., III, DOE Contract No. C00/2862-3 (1978).
- 50 *Steam, Its Generation and Use*, (38th Ed.), Babcock and Wilcox Co., New York City (1972).
- 51 *Energy Review*, Data Resources, Inc. (Winter 1980-1981).
- 52 Aries, R.S., et al., *Chemical Engineering Cost Estimation*, McGraw-Hill Co., New York City (1955).
- 53 Vatabok, W.M., et al., *Factors for Estimating Capital and Operating Costs*, Chemical Engineering (Nov. 3, 1980).
- 54 Holland, F.A., et al., *How to Estimate Capital Costs*, Chemical Engineering (April 1, 1974).
- 55 Cran, J., *Improved Factored Method Gives Better Preliminary Cost Estimates*, Chemical Engineering (April 6, 1981).
- 56 Bauman, H.C., *Fundamentals of Cost Engineering in the Chemical Industry*, Van Nostrand Reinhold Co., New York City (1965).
- 57 Chilton, C.H., *Cost Engineering in the Process Industries*, McGraw-Hill, New York City (1960).
- 58 *Process Plant Construction Estimating Standards*, Richardson Engineering Services, Inc., Solana Beach, Cal. (1976-77 Ed.).
- 59 Touloukian, Y.S., P.E. Liley, and S.C. Saxena, *Thermal Conductivity Nonmetallic Liquids and Gases, Thermophysical Properties of Matter*, 3, Plenum Press (1970).
- 60 Touloukian, Y.S., and T. Makita, *Specific Heat of Nonmetallic Liquids and Gases, Thermophysical Properties of Matter*, 6, Plenum Press (1970).
- 61 Touloukian, Y.S., S.C. Saxena, and P. Hestermans, *Viscosity, Thermophysical Properties of Matter*, II, Plenum Press (1970).

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