THE MODELING OF A RANKINE CYCLE ENGINE FOR USE IN A RESIDENTIAL SOLAR ENERGY COOLING SYSTEM

BY

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ABSTRACT

The topic of this thesis is the use of solar energy with a Rankine cycle heat engine and vapor compression air conditioner to supply cooling for residential appli- cation. In the study, a mathematical model of the Ran- kine engine was formulated and a computer program was written. This program enabled studies of the effects of various engine designs and long term performance simula- tions of solar cooling systems when used in conjunction with the program TRNSYS.

A design study was undertaken to determine for a given set of design restrictions an optimal set of engine components. The engine components include, a boiler feed pump, a boiler, a shut-off valve, an expander, a con- denser, and a working fluid. A control strategy for the matching of engine output to the required load was deve- loped based on the engine design. The various alternatives for the methods of adding auxiliary energy were discussed. An electric motor generator and a variable speed boiler feed pump were selected as the best method of maintain- ing the desired constant expander speed while matching engine output to the load.

The engine model was used to compare working fluids and to study the effects of expander and boiler feed pump efficiency on engine performance.

A solar system consisting of a flat plate solar collector and thermal storage tank, Rankine engine, vapor compression air conditioner, cooling tower, and residential cooling load was modeled using components from TRNSYS and other sources. Performance for this system for a cooling season in Albuquerque, New Mexico was studied for various system configurations. The effect of engine size for three engines (1.49, 2.23 and 2.98 kw) and collector area (10, 20, 30, 40, and 50 m^2) was investigated to determine the percent cooling load and excess energy generated by solar means. The effect of thermal storage tank size on percent cooling and energy generated by solar involved varying tank size for one collector area and engine size. The sizes considered were 37.5, 75, and 150 kg of water per m^2 of collector area. ΔT_{MIN}, an engine control parameter which limits when the engine may operate by thermal constraints, was studied to determine its effect on total system performance.

I. INTRODUCTION

The use of solar energy from flat plate collectors for space and water heating in residential and commercial buildings is an increasingly popular concept. Certain areas of the country may show less favorable economic potential for such systems because they are unused during the summer months. Use factors on collectors have been increased by the use of absorption cycle air conditioners to supply summer cooling. Recently, several studies have investigated the feasibility of a Rankine cycle engine, using heat input from solar energy, connected to a conventional vapor compression air conditioner in a system as shown in Figure 1. The purpose of this thesis is to describe the component selection and the numerical modeling of such an engine and the resultant system performance over a cooling season.

A Rankine cycle is depicted in Figure 2. It consists of an adiabatic compression (or pumping process), a constant pressure heat addition in a boiler, an adiabatic expansion process which produces the work output, and a constant pressure heat rejection in a condenser. A constant enthalpy process is indicated at the boiler outlet, which is the usual position for such control devices as throttles and shut off valves. The Rankine engine is to be operated using heat input from flat plate collectors,

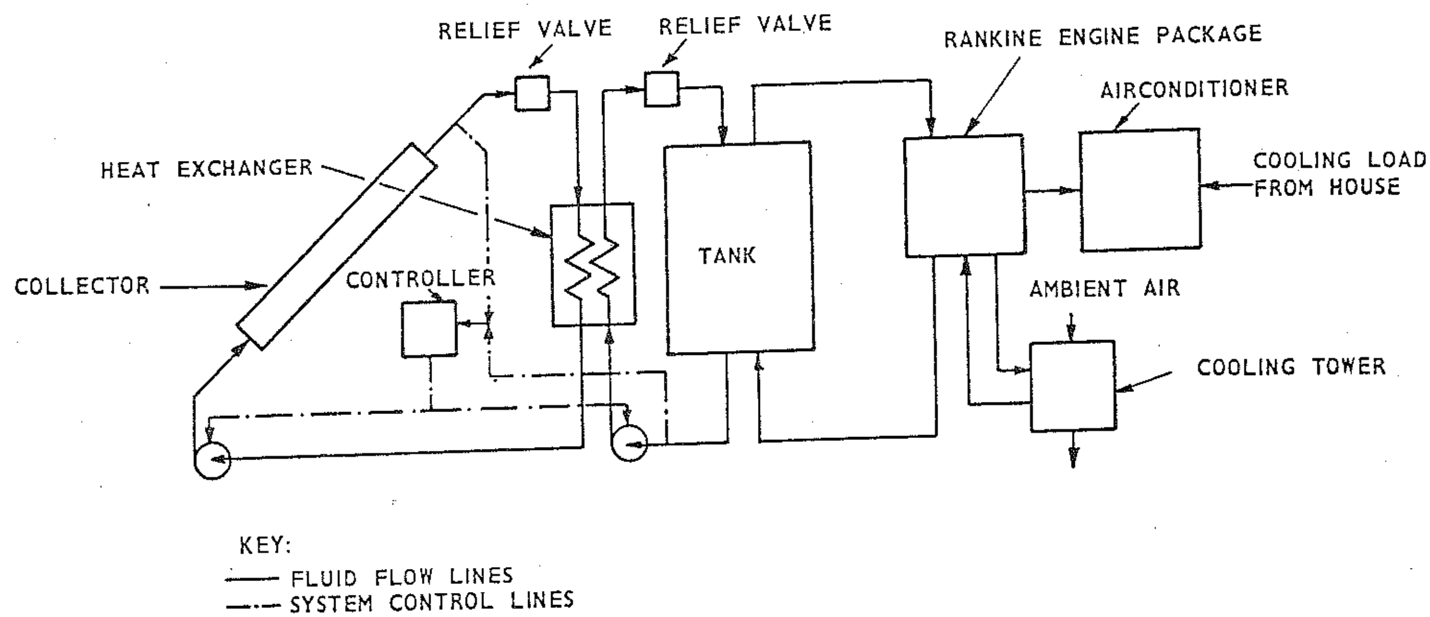
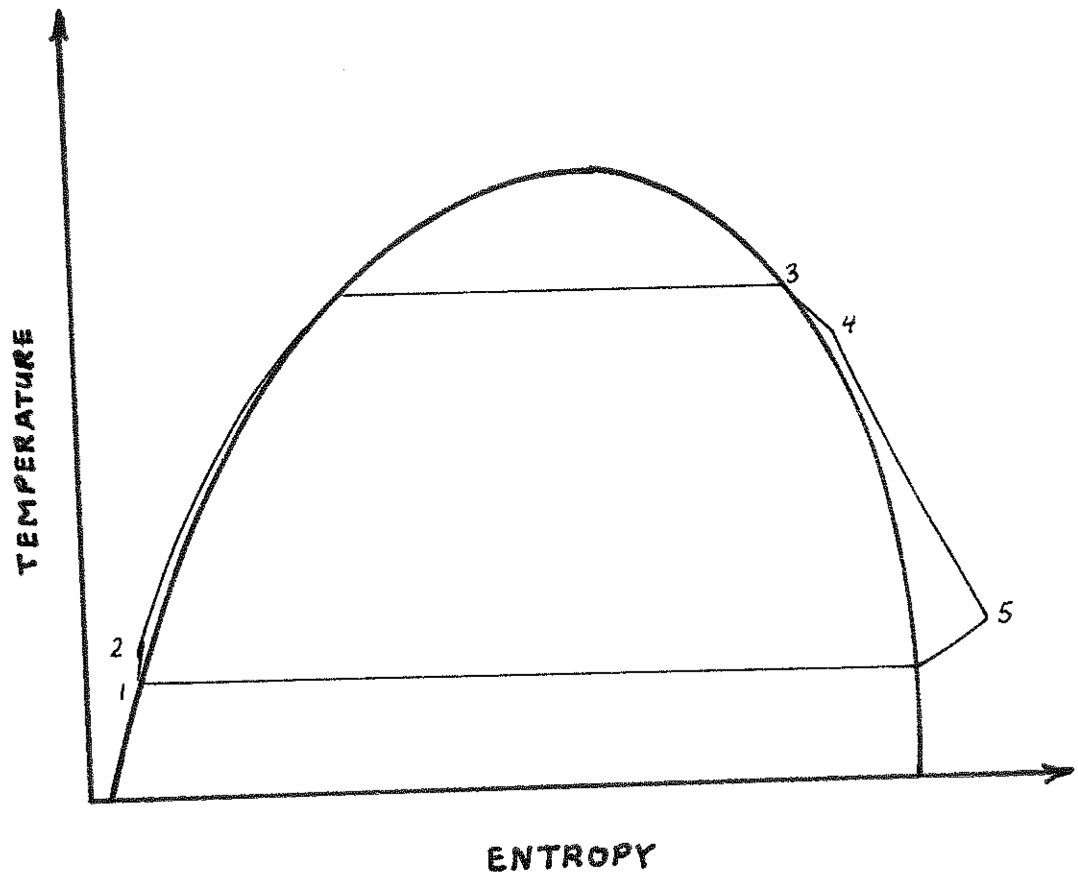


Figure 1. The Solar Rankine Engine Cooling System



KEY:

1-2 ADIABATIC COMPRESSION PROCESS

2-3 CONSTANT PRESSURE HEAT ADDITION PROCESS

3-4 CONSTANT ENTHALPY THROTTLING PROCESS

4-5 ADIABATIC EXPANSION PROCESS

5-1 CONSTANT PRESSURE HEAT REJECTION PROCESS

Figure 2. A Temperature Entropy Diagram for a Rankine Cycle

making it compatible with conventional solar systems using water as the heat transfer medium.

A Rankine engine-air conditioner unit is presently under development and test by Honeywell and the Barber-Nichols Engineering Company (1,2). United Aircraft Research Laboratories (3) is involved in the development of a similar machine under a contract from the National Science Foundation. Sargent and Teagan, at the University of Maryland and the Thermo-Electron Company, proposed such a unit in 1973 (4). A study of solar Rankine engines for small building application has been carried out by Hittman Associates, Inc. (5). The STEPS project (6) investigated the use of a solar powered Rankine engine for large scale power generation using focusing collectors.

II. RANKINE ENGINE COMPONENT SELECTION AND MODELING

2.1 Introduction

The Rankine engine's performance is directly related to the performance of each individual component. The selection of components must therefore include a consideration of the whole system and the conditions under which it must operate. The design of a Rankine engine operated with flat plate collectors becomes especially critical because of the low thermal efficiencies involved.

The conditions on which the Rankine engine design is to be based are:

1. The hot-side boiler fluid on the Rankine engine is water or a water-ethylene glycol mixture. The Rankine engine is to be used in a solar system which employs a flat plate collector with a working fluid of water or an ethylene glycol mixture and thermal storage using a tank of water. This requirement limits the inlet temperature to the engine boiler to a maximum of about 100°C.
2. The cold-side condenser fluid on the engine is to be water. The Rankine engine is connected to a cooling tower to meet its heat rejection requirements. This limits this inlet fluid temperature to a minimum of about 0°C.

3. The expander outlet shaft speed should be compatible with a conventional reciprocating compressor in a vapor compression air conditioner or heat pump.
4. Electrical energy is used to operate the system controls and the engine boiler feed pump.
5. The Rankine engine is intended to be coupled to small to medium capacity air conditioning units to be used in homes or small commercial buildings. The cooling capacity of the units at the A.R.I. 240 Standard is from 37,980 KJ/hour to 316,500 KJ/hour (3 to 25 tons).

Within these restraints, the following problems remain to be resolved before a Rankine engine may be designed and modeled:

1. Selection of a boiler and a condenser.
2. Selection of a boiler feed pump.
3. Selection of a system working fluid.
4. Selection of an expander.
5. Selection of an overall system control strategy including:
   a. The method of adding auxiliary energy to the engine.
   b. The method of matching expander output power to the required air conditioner input power.

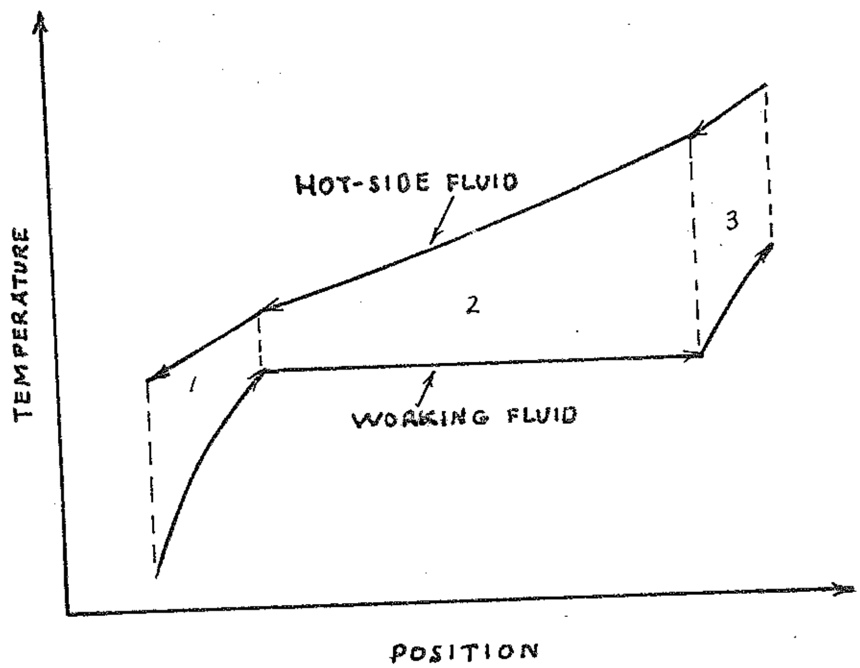
c. The control of expander shaft speed.

These topics are interrelated, i.e., the choice of system working fluid will affect whether the expander should be a reciprocating unit or not. The order indicated above is one of increasing component interdependence, boiler selection being relatively independent of other system components, while expander selection is directly related to the choice of other components. The selection procedure followed in this thesis will follow this general outline.

2.2 Boiler and Condenser

The boiler and condenser are the components least affected by other parts of the Rankine engine system and most affected by external constraints, thus the boiler and condenser are a starting point in the design of an engine. The selection of the boiler and of the condenser are done in a similar manner since they both use water as the heat transfer fluid. A detailed explanation will be done only for the boiler for simplicity.

The boiler can be considered to be two or three interconnected heat exchangers as shown in Figure 3. Using this representation, some fundamental concepts concerning the performance and economics of the boiler design may



KEY:
1 - LIQUID-TO-LIQUID HEAT EXCHANGER
2 - LIQUID-TO-MIXED PHASE HEAT EXCHANGER
3 - LIQUID-TO-VAPOR HEAT EXCHANGER

Figure 3. A Schematic Representation of a Boiler

be considered.

Superheating the working fluid in the boiler is preferable for the expander in terms of simplicity of design and operating performance. By superheating the inlet vapor, the need for liquid traps in the expander may be reduced. The expander will have less wear caused by liquid in the flow stream on internal parts. Its performance will be maintained at the design values for longer operating periods.

Although a deviation from the Carnot cycle (represented in Figure 4a), which offers the maximum thermal efficiency for any thermodynamic cycle operating between two temperatures T_A and T_R, superheating may be desirable. One such case is a fossil fueled steam power plant. The inlet hot-side fluid (air) has a very high temperature (1650 to 2200°C). The low fluid specific heat and density make it impractical to induce a small temperature drop in the air stream. The high air mass flow rates involved make operating costs (caused by the necessarily high fan input power) excessive. In order to make the best of this situation where pumping costs and not the hot-side fluid inlet temperature is the restriction, superheating is required to minimize the net temperature difference between fluids in the boiler. A small temperature drop between hot-side and cold-side fluids reduces

Figure 4a

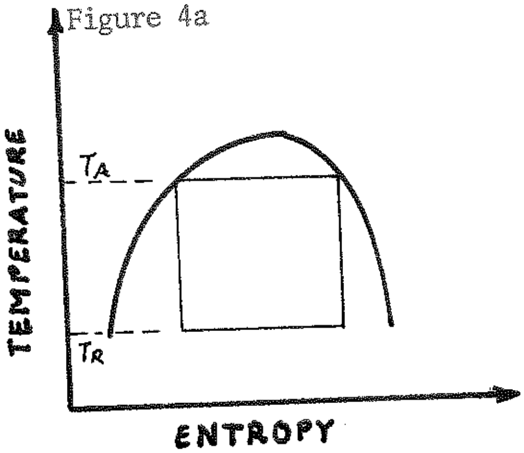


Figure 4b

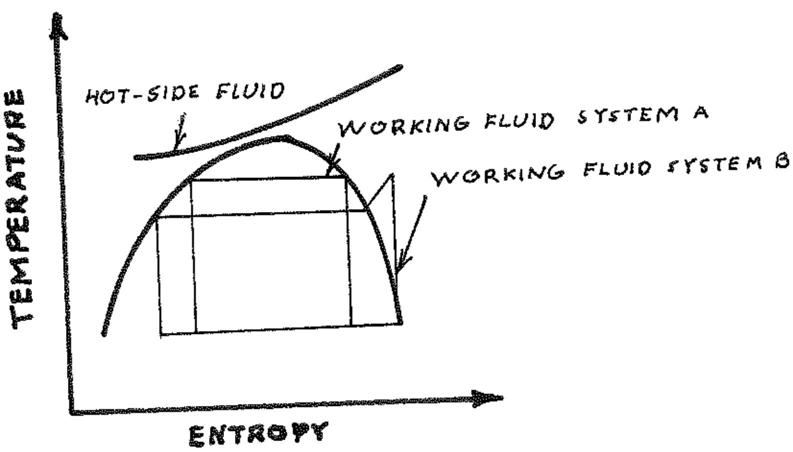


Figure 4. Temperature Entropy Diagrams for the Carnot and Other Cycles

the irreversibility in the heat transfer process.

In the boiler for the solar Rankine engine the conditions are different. The restriction for heat input from a flat plate collector using water is not pumping costs, but maximum hot-side fluid inlet temperature. A large temperature drop for the hot-side fluid is undesirable since it would force a reduction in the average heat addition temperature of the cycle. This would reduce the cycle thermal efficiency. As seen in Figure 4b, adding a superheat for system B with essentially the same heat addition as the unsuperheated system A forces system B to have a reduced temperature of heat addition. System A has less irreversibilities because of the smaller temperature drop between hot and cold fluids. It has a higher cycle efficiency than system B because it has a higher average temperature of heat addition. The boiler in the solar Rankine engine should not have a superheat because of the limited maximum temperature of the fluid heated by solar energy in a flat plate collector.

The foregoing analysis assumes the working fluid is undergoing heat addition isothermally in mixed phase. A system which operates in such a manner offers a reduction in heat transfer surface (and cost) because of the high heat transfer coefficients involved in a boiling process.

Various trade-offs can be made in the design of a boiler which affect cost and performance. From Figure 5, if the hot-side mass flow rate is increased, ΔT_1 will decrease, and pumping costs will increase, for a constant heat transfer rate and heat transfer area. Alternatively, the amount of heat transfer surface may be decreased, but the hot-side mass flow rate is increased to maintain a constant heat transfer rate. This is a common trade-off; initial cost for heat transfer surface for increased operating costs. Similarly, a decrease in ΔT_2 results in a decrease in heat transfer rates. For a constant heat transfer rate an increase in heat exchanger area (and cost) would be required. Although the economics involved in such a decision is beyond the scope of this thesis, the implications are of interest for real systems.

The design of the condenser proceeds in a similar fashion, except that the working fluid is the hot-side fluid and the cold-side fluid is water as seen in Figure 6. Subcooling is not used because it reduces the engine's efficiency.

Modeling of the boiler and condenser uses the following assumptions:

1. The analysis is quasi-steady state with thermal capacitance effects ignored and each operating

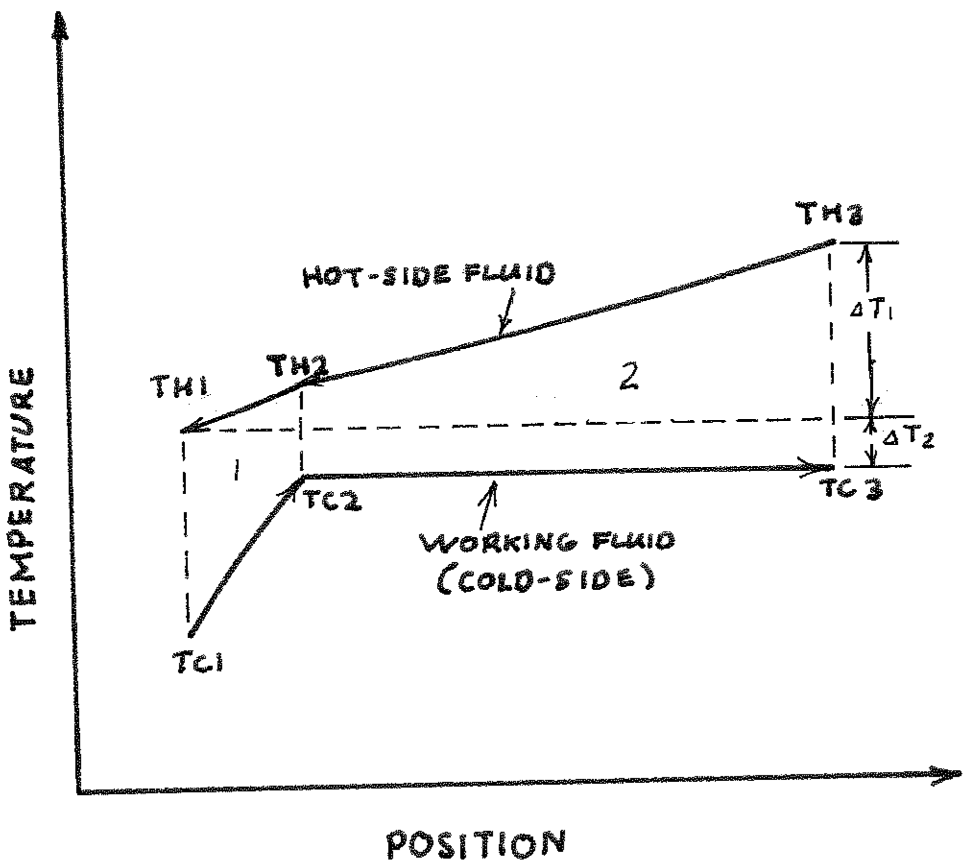


Figure 5. An Unsuperheated Counterflow Boiler

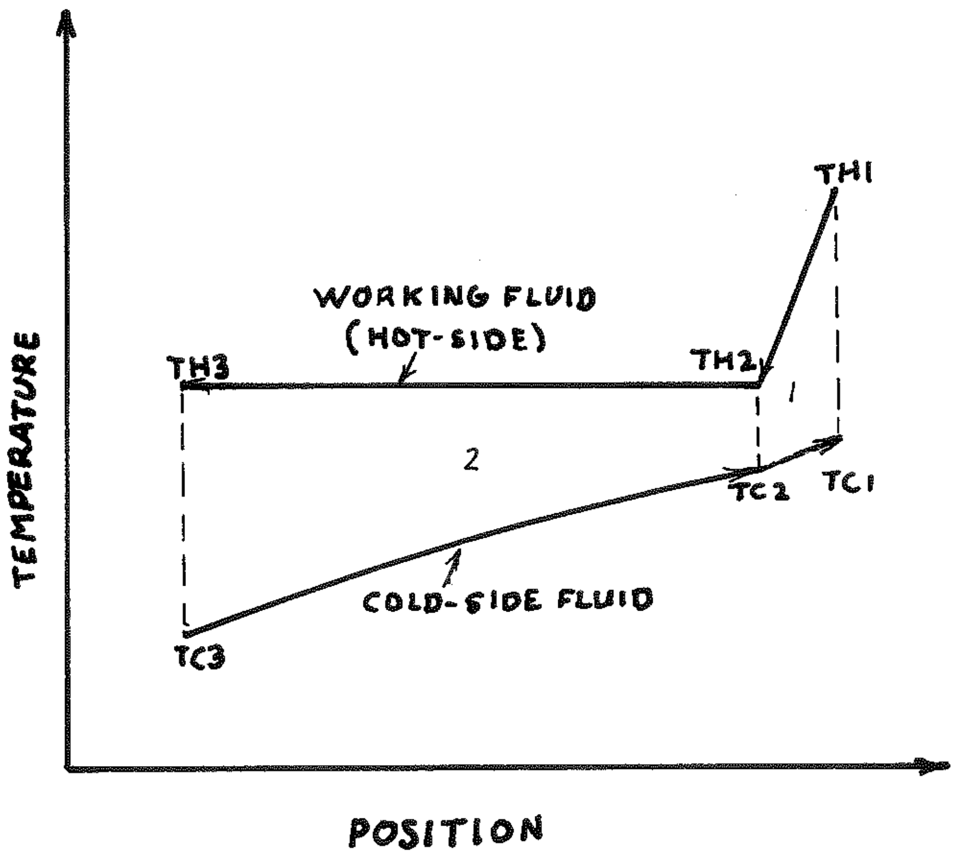


Figure 6. An Unsubcooled Counterflow Condenser

state is assumed to be the steady state solution.

2. The outlet state of the boiler working fluid is at saturated vapor. The outlet state of the condenser working fluid is at saturated liquid.
3. Pressure drops in the boiler and condenser are assumed to be negligible.
4. Heat transfer coefficients are assumed to be some average value for each section of the boiler and condenser. They are assumed to vary from design values only as a function of the fluid mass flow rates.

The boiler and condenser are essentially flooded type single pass units as depicted in Figure 7.

Referring to Figure 5 the boiler is defined to be two heat exchangers coupled together. The heat transfer area in each exchanger may vary, but the total heat transfer area is constant, i.e.,

$$A_T = A_1 + A_2 \qquad (2.2\text{-}1)$$

We define the capacity rates as follows:

$C_{MIN} = \dot{m}c_p$ for the minimum capacity rate fluid
$C_{MAX} = \dot{m}c_p$ for the maximum capacity rate fluid
$C_{HOT} = \dot{m}c_p$ for the hot-side fluid
$C_{COLD} = \dot{m}c_p$ for the cold-side fluid

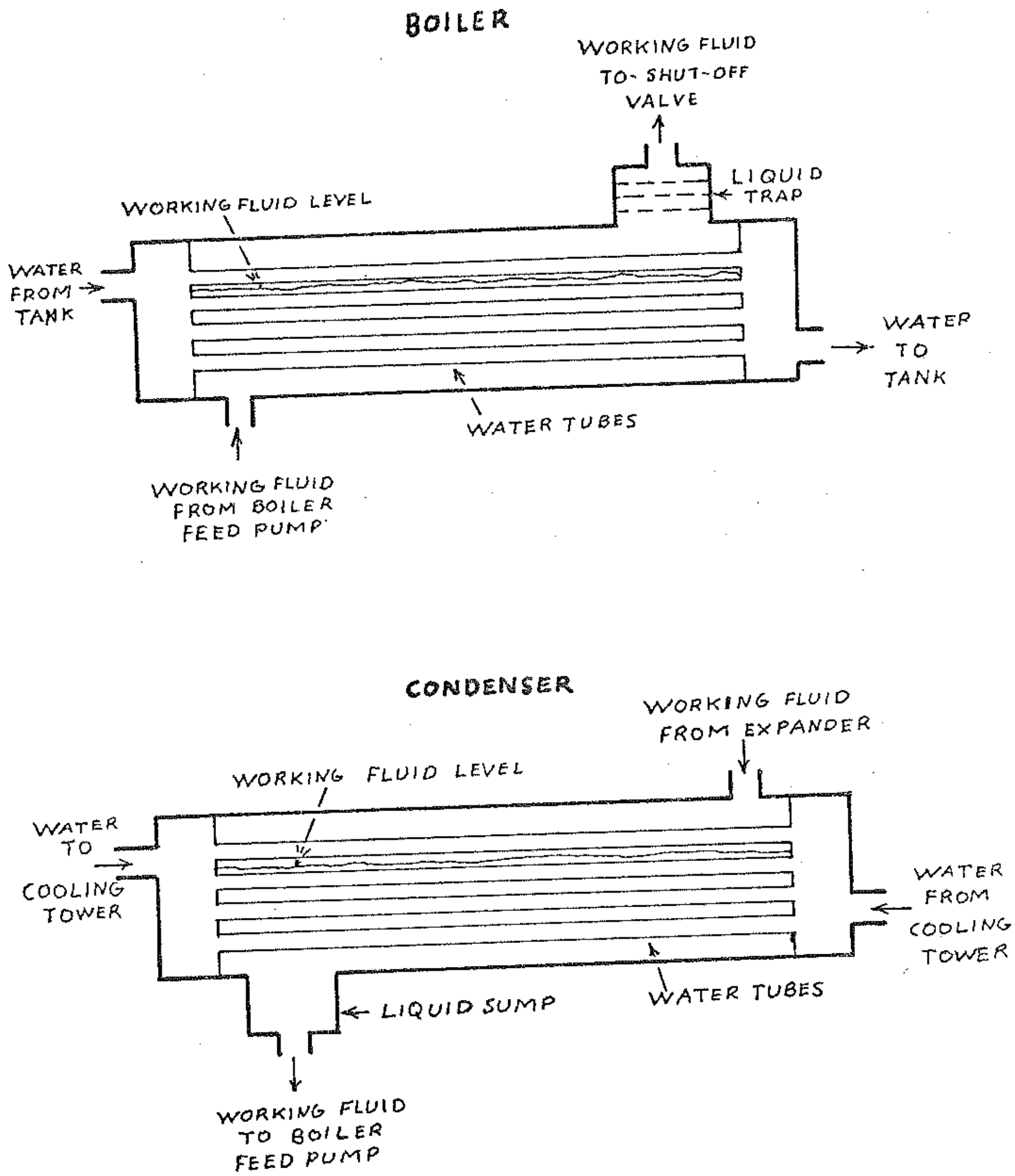


Figure 7. Diagrams of Boiler and Condenser Construction

From Kays and London (7) for a counterflow heat exchanger we may define the effectiveness (E) as

$$E = \frac{1-\mathrm{EXP}[-\frac{UA}{C_{MIN}}(1-\frac{C_{MIN}}{C_{MAX}})]}{1-(\frac{C_{MIN}}{C_{MAX}})\mathrm{EXP}[-\frac{UA}{C_{MIN}}(1-\frac{C_{MIN}}{C_{MAX}})]} \qquad (2.2\text{-}2)$$

Where U is the overall heat transfer coefficient and A is the effective heat transfer area. This relation may be simplified for a liquid to mixed phase heat exchanger by observing that for a fluid in mixed phase the specific heat is infinite. The maximum capacity rate fluid is the working fluid with $C_{MAX} = \infty$. Introducing this into equation 2.2-2 the relation reduces to:

$$E_2 = 1-\mathrm{EXP}(\frac{-U_2A_2}{C_{HOT}}) \qquad (2.2\text{-}3)$$

The solution of the counterflow boiler problem is obtained iteratively in the following manner. Given as knowns: T_{H3}, T_{C1}, P_B, A_T, the hot side mass flow rate and specific heat ($\dot{m}_{HOT}$ and c_{pHOT}, respectively), the cold-side mass flow rate and liquid specific heat ($\dot{m}_{COLD}$ and c_{pCOLD}), working fluid property data, U_1 and U_2, we may define:

$$\dot{Q}_2 = \dot{m}_{COLD}(h_{C3} - h_{C2})$$

$$T_{C2} = T_{C3}$$

$$E_2 = \dot{Q}_2/[C_{HOT}(T_{H3}-T_{C2})] \qquad (2.2\text{-}4)$$

Solving for A_2 in terms of U_2, C_{HOT} and E_2,

$$E_2 = 1\text{-}EXP(\frac{-U_2A_2}{C_{HOT}})$$

$$\ell n(1-E_2) = -\frac{U_2A_2}{C_{HOT}}$$

$$A_2 = -(C_{HOT}/U_2)\ \ell n(1-E_2) \qquad (2.2\text{-}5)$$

Thus we may obtain the surface area of the heat exchanger in section 1 as:

$$A_1 = A_T - A_2$$

Now we can solve for the effectiveness of the liquid heat exchanger in section 1 using equation 2.2-2. C_{MIN1} and C_{MAX1} are determined from the fluid capacity rates. Writing several energy balances we solve for the hot-side outlet temperature, T_{H1}.

$$C_{HOT}(T_{H1}-T_{H2}) = E_1C_{MIN1}(T_{H2}-T_{C1})$$

$$T_{H1}-T_{H2} = -E_1(C_{MIN1}/C_{HOT})(T_{H2}-T_{C1})$$

$$T_{H1} = T_{H2}-E_1(C_{MIN1}/C_{HOT})(T_{H2}-T_{C1}) \qquad (2.2\text{-}6)$$

And $\dot{Q}_1 = C_{HOT}(T_{H2}-T_{H1})$

From this we may calculate the boiler heat transfer rate as:

$$\dot{Q}_A = \dot{Q}_1 + \dot{Q}_2$$

The iterative nature of this solution is due to the fact that P_B is an unknown. As a check to determine if the guess for the boiler pressure is correct, the boiler heat transfer may also be calculated as:

$$\dot{Q}_1 = C_{COLD}(T_{C2}-T_{C1})$$

$$\dot{Q}_2 = \dot{m}_{COLD}(h_{C3}-h_{C2})$$

$$Q_A = \dot{Q}_1 + \dot{Q}_2$$

When the correct value of P_B is chosen, the heat transfer rates for both methods of calculations will agree.

For a parallel flow boiler as depicted in Figure 8, the solution follows a similar procedure. Given as knowns T_{H1}, T_{C1}, P_B, A_T, $\dot{m}_{HOT}$, c_{pHOT}, $\dot{m}_{COLD}$, U_1, U_2, and property data for the cold-side fluid, we obtain the effectiveness

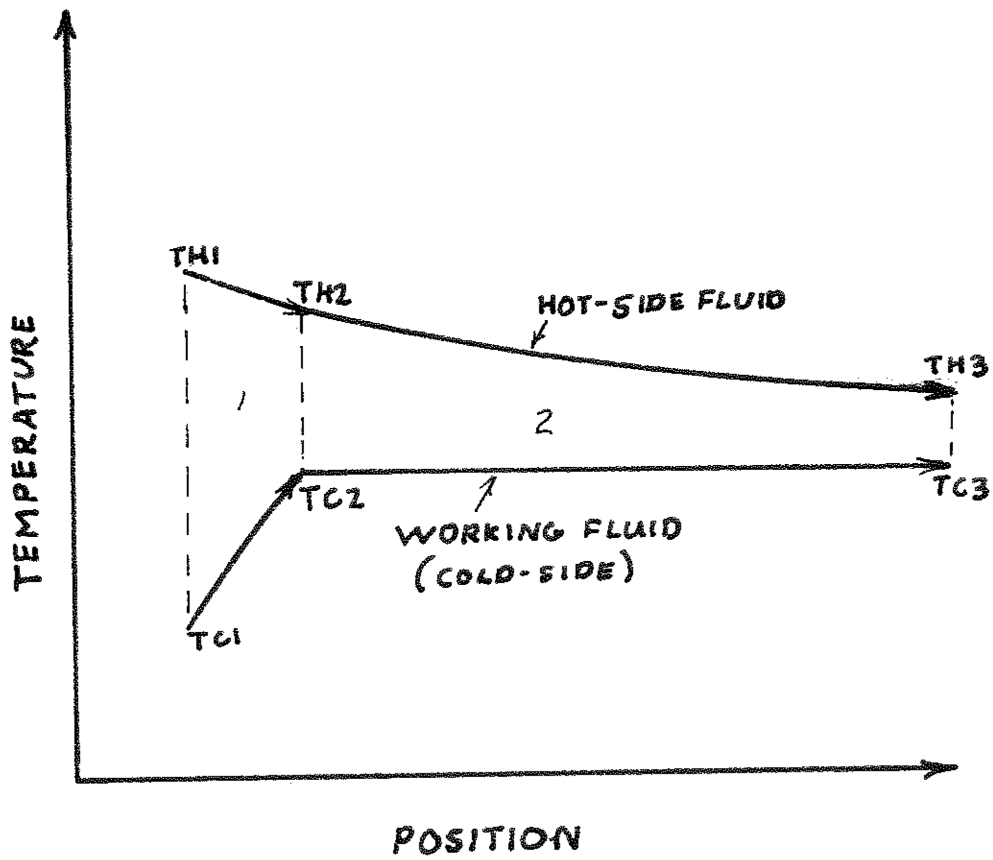


Figure 8. An Unsuperheated Parallel Flow Boiler

(E_1) of a parallel flow heat exchanger as defined by Kays and London (7) as:

$$E_1 = \frac{1-EXP[\frac{-U_1A_1}{C_{MIN1}}(1+\frac{C_{MIN1}}{C_{MAX1}})]}{1 + \frac{C_{MIN1}}{C_{MAX1}}} \qquad (2.2-7)$$

C_{MIN1} and C_{MAX1} are the minimum and maximum fluid capacity rates for section 1 of the boiler. Once more we assume that:

$$A_T = A_1 + A_2$$

We now define by energy balances, E_1:

$$\dot{Q}_2 = C_{COLD}(T_{C2}-T_{C1})$$

$$T_{H2} = T_{H1} - \dot{Q}_1/C_{HOT}$$

$$E_1 = \dot{Q}_1/[C_{MIN1}(T_{H1}-T_{C1})] \qquad (2.2-8)$$

Solving for A_1 in terms of U_1, C_{MIN1}, C_{MAX1}, and E_1 by rearranging equation 2.2-7,

$$E_1 = \frac{1-EXP[\frac{-U_1A_1}{C_{MIN1}}(1+\frac{C_{MIN1}}{C_{MAX1}})]}{1 + \frac{C_{MIN1}}{C_{MAX1}}}$$

$$1-E_1(1+\frac{C_{MIN1}}{C_{MAX1}}) = EXP[\frac{-U_1A_1}{C_{MIN1}}(1+\frac{C_{MIN1}}{C_{MAX1}})]$$

$$\left(\frac{-U_1A_1}{C_{MIN1}}\right)\left(1+\frac{C_{MIN1}}{C_{MAX1}}\right) = \ell n\left[1-E_1\left(1+\frac{C_{MIN1}}{C_{MAX1}}\right)\right]$$

$$A_1 = \frac{-\left(\frac{C_{MIN1}}{U_1}\right)\left\{\ell n\left[1-E_1\left(1+\frac{C_{MIN1}}{C_{MAX1}}\right)\right]\right\}}{1 + \frac{C_{MIN1}}{C_{MAX1}}} \qquad (2.2\text{-}9)$$

We now obtain the effective area for section 2 of the boiler as:

$$A_2 = A_T - A_1$$

The effectiveness of section 2 of the boiler may be obtained by using equation 2.2-7. Noting that, as before, $C_{MAX} = \infty$ for a fluid in mixed phase equation 2.2-7 simplifies to a familiar form:

$$E_2 = 1\text{-}EXP\left(\frac{U_2A_2}{C_{HOT}}\right) \qquad (2.2\text{-}10)$$

Using the value for E_2, we again write several energy balances and solve for the hot-side outlet temperature, T_{H3}:

$$C_{HOT}(T_{H2} - T_{H3}) = E_2C_{HOT}(T_{H2}-T_{C1})$$

$$T_{H3} = T_{H2} - E_2(T_{H2} - T_{C2})$$

The heat transfer rate in section 2 of the boiler is simply:

$$\dot{Q}_2 = C_{HOT}(T_{H2}-T_{H3})$$

From this we may calculate the heat addition rate in the boiler as:

$$\dot{Q}_A + \dot{Q}_1 + \dot{Q}_2$$

This solution is also iterative because P_B is a guess for each calculation until the energy balances close as described for the counterflow case.

In calculating the overall heat transfer coefficients in the liquid to liquid and liquid to mixed phase sections (U_1 and U_2, respectively), the effective heat exchanger surface area seen by each flow stream in each section is assumed to be equal. Given a value at design conditions for U_1 and knowing the design mass flow rates for both streams, the values of U_1 and U_2 may be calculated at off design conditions. The thermal resistance of the tubing in the heat exchangers is assumed to be negligible. For the liquid to liquid heat exchanger we assume it is well designed, thus at design conditions:

$$(h_D A)_{HOT1} = (h_D A)_{COLD1} \tag{2.2-11}$$

And since: $A_{HOT1} = A_{COLD1}$

$$h_{D\ HOT1} = h_{D\ COLD1}$$

Defining the overall heat transfer coefficient for section 1,

$$U_1 = 1/(1/h_{HOT1} + 1/h_{COLD1})$$

Solving for h_{HOT1} on design conditions,

$$U_{D1} = 1/(2/h_{D\ HOT1})$$

$$h_{D\ HOT1} = 2U_{D1} \qquad (2.2\text{-}12)$$

For off-design operation, the value of the hot side fluid heat transfer coefficient (h_{HOT1}) may be assumed to be constant throughout the boiler, so:

$$h_{HOT2} = h_{HOT1}$$

Furthermore, the value of h_{HOT1} will remain constant since the water side mass flow rate is constant and property value related changes are small.

The value of h_{COLD1} must be calculated for off design conditions since the working fluid mass flow rate will not remain a constant. The flow path of the working fluid as seen in Figure 7 is essentially perpendicular to the water for a single-pass flooded evaporator. Using a crossflow forced convection relation for liquid flow perpendicular to a number of cylinders from Holman (8) for constant properties and a high Reynolds number we obtain the general relation:

$$h_{COLD1} = C(\dot{m}_{COLD})^{0.52} \qquad (2.2\text{-}13)$$

Where C is some constant. Now solving for the off-design value of h_{COLD1}, given $\dot{m}_{D\ COLD}$, $\dot{m}_{COLD}$, and $h_{D\ COLD1}$:

$$C = \frac{h_{D\ COLD1}}{(\dot{m}_{D\ COLD})}0.52 = \frac{h_{COLD1}}{(\dot{m}_{COLD})}0.52$$

$$h_{COLD1} = h_{D\ COLD1}\ (\dot{m}_{COLD}/\dot{m}_{D\ COLD})^{0.52} \qquad (2.2\text{-}14)$$

With this value we can solve for U_1 at off design conditions.

In the mixed-phase heat transfer section of the boiler we assume the average boiling heat transfer coefficient to remain relatively constant (9). Having previously defined h_{HOT2} we can define the overall heat transfer coefficient as:

$$U_2 = 1/(1/h_{HOT2} + 1/h_{COLD2})$$

The value of h_{COLD2} is actually dependent on the fluid being boiled, the temperature gradients present, the tube diameter, the number of tubes, and the tube material and surface finish. The boiling mode for the temperature gradients encountered in this unit should be nucleate. Data for this type of mode is somewhat scattered and the available analytical methods somewhat questionable, so changing the value of h_{COLD2} was not felt to be a worthwhile alternative.

The modeling of the condenser in parallel and counterflow modes follows a similar analysis as the boiler, so only the differences will be noted here.

The condenser model consists of a vapor to liquid heat exchanger linked to a mixed-phase to liquid heat exchanger as shown in Figure 6. The unit modeled is a simple pass flooded type condenser as in Figure 7. The working fluid is assumed to be in a laminar film mode of condensation. Utilizing the Nusselt correlation for laminar film condensation on horizontal tubes from (8), values of the heat transfer coefficient were calculated for various working fluids. It was found that for the range of temperatures expected to be encountered, the heat transfer coefficient did not vary sufficiently to warrant any assumption but that of a constant heat transfer coefficient for the working fluid condensation.

The overall heat transfer coefficients U_1 and U_2 were obtained in a manner similar to that of the boiler. Given U_1 at design conditions, the remaining design values were calculated as before. For off-design operation McAdam's (10) relation for gases flowing normal to unbaffled staggered tubes in turbulent flow was used. Making the assumption of constant properties for small changes in operating conditions we obtain the general relation:

$$h_{HOT1} = C(\dot{m}_{HOT})^{0.6} \tag{2.2-15}$$

Where C is a constant. Solving for the off-design value of h_{HOT1} given: $\dot{m}_{D\ HOT}$, $\dot{m}_{HOT}$, and $h_{D\ HOT1}$,

$$C = \frac{h_{D\ HOT1}}{(\dot{m}_{D\ HOT})^{0.6}} = \frac{h_{HOT1}}{(\dot{m}_{HOT})^{0.6}}$$

$$h_{HOT1} = h_{D\ HOT1}\ (\dot{m}_{HOT}/\dot{m}_{D\ HOT})^{0.6} \tag{2.2-16}$$

Finally solving for U_1 with the assumption that the cold-side heat transfer coefficient h_{COLD1} is constant since the water mass flow rate is to be held constant:

$$U_1 = 1/(1/h_{HOT1} + 1/h_{COLD1})$$

2.3 Boiler Feed Pump

The boiler feed pump is the interface between the low pressure level in the condenser and the high pressure in the boiler. It must produce a large change in pressure (preferably in a single stage), yet be sturdy enough to pump liquids which are boiling. It is desirable that the feed pump operate at a high efficiency at a wide range of operating speeds. Variable pump speed is a desirable option, since it is possible to control the operation of the Rankine engine at varying loads by changing the working fluid mass flow rate with such a pump. Low pump speed is an advantage in that the drive motor may be connected directly to the unit removing the need for a power wasting speed increaser. As with the rest of the system, the feed pump should be of conventional design.

Centrifugal pumps are probably not desirable. It is difficult to produce such a pump for the requirements indicated cheaply. Large scale centrifugals used in steam power plants are specially built to precise specifications, yet they are one of the weak links in generating plant reliability. These pumps are especially vulnerable to cavitation because of their close tolerance design which is required to reduce fluid slippage. The high rotational speed of such a unit requires a speed increaser between the drive motor and the pump. Finally, centrifugal pumps suffer from poor operating performance at off design speeds.

Reciprocating pumps with spring loaded valves are superior in that they are low speed machines compatible with direct coupling to the drive motor. Although the design adiabatic efficiency may not be as high as for a centrifugal unit, this type of pump offers improved off-design speed operation over the centrifugal pump. Unfortunately, the spring loaded valves are particularly vulnerable to damage caused by the pumped fluid cavitating. The required net positive suction head (9) needed to prevent cavitation (by pressurizing the pumped fluid before entering the pump) may prove difficult to obtain in a small home or business situation. A reciprocating pump with valves would probably also be a poor selection.

A rotating positive displacement is felt to be the type most suitable for this application. This unit offers valveless construction with a minimum of moving parts, high efficiency, and good operating characteristics at off design speed operation. This type of pump has proven to be satisfactory in long term operation(11) and is compatible with direct coupling to the drive motor. Cavitation difficulties are minimal.

The assumptions made in the modeling procedure are:

1. The pumping process involves a constant density fluid.
2. The adiabatic efficiency (η_c) and the volumetric efficiency (η_{vc}), which is defined as:

   $$\eta_{vc} = \frac{\text{volume flow actually displaced}}{\text{pump displacement}}$$

   are assumed to be constant. The value for η_{vc} is assumed to be 0.90 as suggested by Potter (12).
3. Inlet and outlet pressure drops caused by flow obstructions are assumed to be negligible.
4. The unit net positive suction head is assumed to be negligible.

Writing the energy balance for an adiabatic constant density pumping process,

$$\Delta w_{PUMP} = -v_{INLET}(\Delta P)/\eta_c \qquad (2.3\text{-}1)$$

Where ΔP is the pressure rise in the unit and v_{INLET} is the working fluid inlet specific volume. The engine working fluid mass flow rate is proportional to the rotational speed of the pump. Potter gives it as:

$$\dot{m}_{SYS} = R\ D\ \eta_{vc}/v_{INLET} \qquad (2.3\text{-}2)$$

With R being the pump rotational speed and D the pump displacement per revolution. The pump input power is calculated as:

$$\dot{W}_{INPUT} = \dot{m}_{SYS} \quad \Delta w_{PUMP} \tag{2.3-3}$$

Note that the thermodynamic sign convention is upheld and the pump work is considered a negative quantity since it is an input to the system.

2.4 System Working Fluid

The choice of system working fluid is one of the most critical, in terms of Rankine engine performance, of all of the system components. Following the method of Obert and Gaggioli (13) for determining the optimum fluid we first establish the temperature limitations of the cycle.

1. The maximum cycle temperature, the boiler temperature, is chosen to be 100°C (212°F), which is consistent with the capabilities of non-pressurized flat plate solar collectors.
2. The minimum cycle temperature, the condenser temperature, is chosen to be 0°C (32°F) which is consistent with using a cooling tower as the heat rejection mechanism.

Given these two main criteria we seek to find a superior system working fluid.

Obert and Gaggioli give the following requirements for a fluid to be used in a power cycle.

1. The critical point should be above the highest temperature which the system will reach. This is advantageous since:

   a. Heat transfer rates are maximized in the mixed phase which allows smaller hardware at a lower cost.

   b. A constant temperature heat addition reduces the irreversibility involved in the process if the change in temperature of the fluid giving off heat is small.

2. The vapor pressure in the boiler should be moderate (in this case under 2500 k Pa).

   a. A low boiler pressure allows manufacturing costs to be reduced.

   b. Lower boiler pressure results in greater operating safety and reduced maintenance, a plus for a home system where the operator would probably do little of his own maintenance.

3. The vapor pressure in the condenser should be above atmospheric at the lowest operating temperature.

a. Air leakage into the condenser raises the mixture pressure and reduces the work output of the expander (the pressure of the condenser is the exhaust pressure of the expander).

b. A condenser pressure level higher than atmospheric prevents non-condensibles from entering the system. Non-condensibles are undesirable since they reduce heat transfer rates and increase the danger of cavitation to the pump.

4. The specific volume of the fluid at the condenser inlet temperature should be small. This will minimize the size of the expander.

5. The saturated vapor entropy should be essentially constant with pressure, i.e., the saturated vapor line on a T-S diagram should be nearly vertical. A result of this is to reduce the amount of moisture in the expander which:

   a. Reduces the amount of erosion in the expander.

   b. Increases the expander efficiency.

6. The latent heat of vaporization should be large in relation to the specific heat. In such a fluid the majority of the heat added in the boiler will be in the boiling section. Then the heat may be added across a small temperature

drop to an isothermal source minimizing losses.

7. The fluid properties should be conducive to high heat transfer rates. This will minimize the heat exchanger area.
8. The fluid should be cheap, readily available, stable, nonflammable, nonexplosive, noncorrosive, and nonpoisonous.

Additionally, the pressure ratio between boiler and condenser pressure levels should be low. In a reciprocating or other type of positive displacement expander, a high pressure ratio, generally above 10 (9), will require multistage expansion with the resultant increase in cost. A high pressure ratio will also affect the design, at least making the design more complex for a single stage unit and probably will require a second stage of expansion.

The theoretical work available from an isentropic expansion is desired to be high. A large value of this specific work from expansion will allow a low system mass flow rate in order to meet the desired power output. Lower system mass flow rates tend to decrease pumping power and result in smaller system components. This reduces both operating and initial costs.

In Table 1 eight organic fluids which are possible candidates for the system working fluid are compared. Using this data plus the information from Figure 9, a

Table 1. A Comparison of Working Fluids

T [°C]	P [kPa] ν [m^3/kg]	R-11	R-12	R-21	R-22	R-113	R-114	R-216	R-C318
121.1	P	1255	Above Critical P	2074	Above Critical P	710	2102	960	Above Critical P
93.3	P	707	2965	1147	4733	377	1230	524	1799
65.6	P	361	1719	600	2546	179	666	257	971
	P	162	909	276	1452	72.3	316	109	465
37.8	ν	0.11	0.02	0.08	0.02	0.19	0.04	0.10	0.02
	P	60.6	423	106	630	23.6	128	37.4	188
10.0	ν	0.28	0.04	0.21	0.04	0.53	0.10	0.26	0.06
	P	17.7	164	31.9	267	6.15	41.0	9.53	60.1
-17.8	ν	1.54	0.10	0.64	0.09	1.84	0.30	1.00	0.17
Slope of Sat. Vapor Line on T-S Diagram		(-)	(-)	(-)	(-)	(+)	(+)	(+)	(+)
$\frac{P(93.3\ °C\ Sat.\ Vap.)}{P(10°C\ Sat.\ Vap.)}$		11.7	7.01	10.8	6.95	16.0	9.63	14.0	9.59
Δh Isentropic From 93.3°C Sat. Vapor to 10°C	$\frac{KJ}{KG}$	37.2	30.5	65.2	33.4	37.9	33.4	33.0	27.1
	$\frac{BTU}{Lbm}$	16.0	13.2	28.0	14.3	16.3	14.3	14.2	11.6

(Table 1 continued on following page)

Table 1. continued

Data from Ref. 14

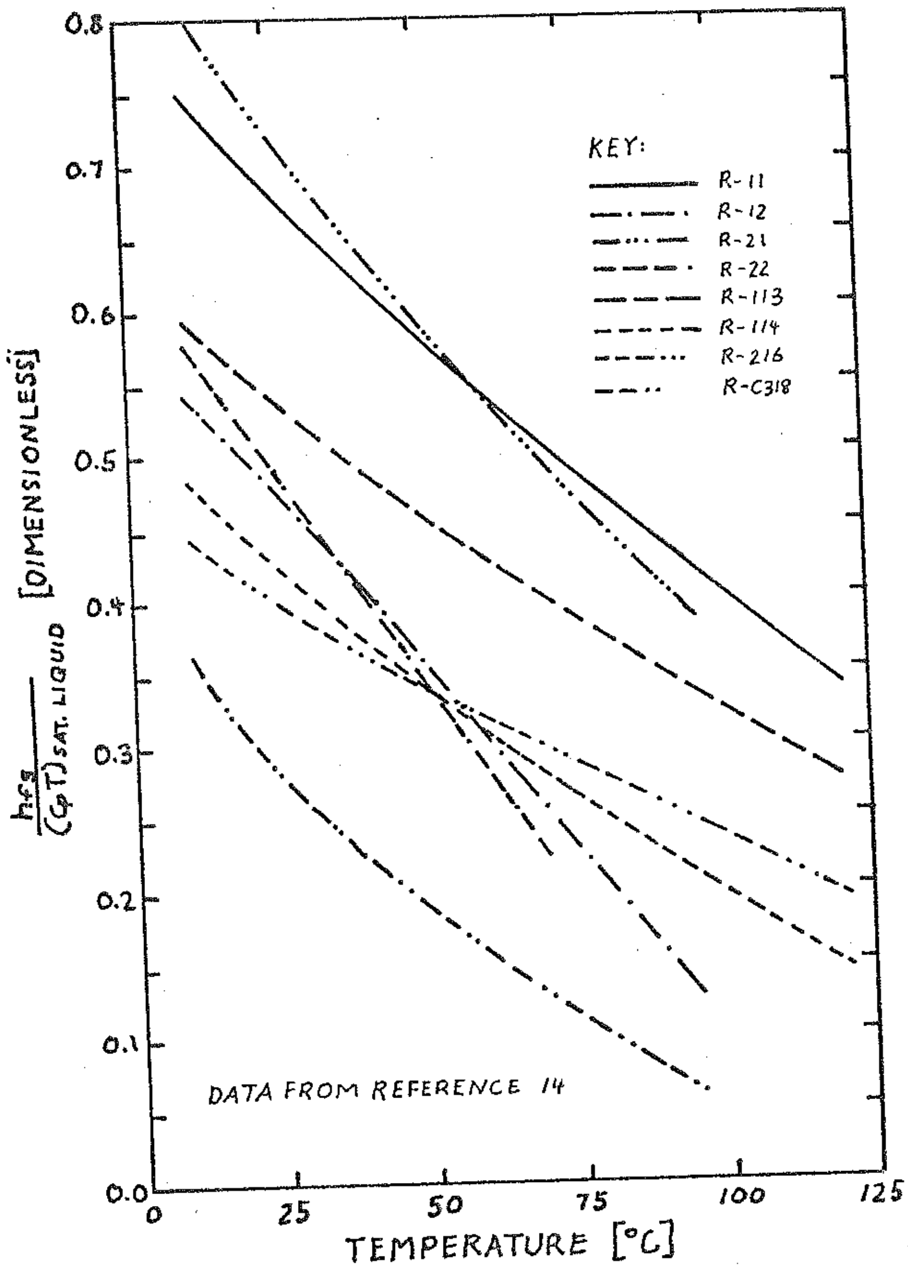
All properties are for saturated vapor unless otherwise noted.

Conversions: $T°F = 1.8°C + 32$

$1\ kPa = 0.145\ PSIA$

$1\ m^3/kg = 16.0\ ft^3/Lbm$

Figure 9. A Comparison of $h_{fg}/(CpT)_{SAT.\ LIQUID}$ Versus Temperature of Various Working Fluids



superior operating fluid is to be chosen.

1. R-11. Thermodynamically, R-11 is the best choice available for low temperature operation of a Rankine engine. As shown by Figure 9, the ratio of h_{fg} over C_pT is very high so boiler irreversibilities are minimized. The expander specific work is large, thus the system mass flow rate will be small. On the negative side, the condenser pressure is below atmospheric and the expander outlet specific volume is large. The slope of the saturated vapor line is negative indicating that moisture in the expander may be a problem.

2. R-12. This freon has a low pressure ratio and small condenser inlet specific volume. It is one of the more commercially available refrigerants today, so costs for a charge of freon are small. R-12 suffers from a low critical temperature and high pressures in the boiler so that system costs would rise. The slope on the T-S diagram of the saturated vapor line is negative so moisture in the expander would be a problem. R-12 is not a good choice for a fluid.

3. R-21. R-21 is interesting because of its high values of expander specific work and h_{fg}/C_pT. The pressure in the boiler is moderate and the condenser pressure is above atmospheric. The disadvantages are possible moisture damage and high specific volumes at the expander out-

let.

4. R-22. A very common medium temperature refrigerant, R-22 has the lowest operating pressure ratio of any fluid. The application of this fluid is not recommended, however, since it has a very low critical temperature and high boiler pressure levels.

5. R-113. Another fluid which appears to be favorable thermodynamically, R-113 has a high value of h_{fg}/C_pT so boiler irreversibilities should be lessened. Expander specific work is relatively high and no moisture difficulties should be encountered. It suffers in that condenser pressures are less than atmospheric, a tendency which could decrease performance and increase maintenance. The pressure ratio indicates that two stages of expansion will probably be required for efficient system operation. Expander exit specific volumes are high. R-113 is probably suitable only for multi-stage expanders.

6. R-114. R-114 is a medium performance refrigerant with moderate pressure levels in the boiler. Condenser pressure levels are above atmospheric and expander exit specific volumes are moderate. Moisture problems in the expander will likely not arise.

7. R-216. This fluid is another one which is moderately good thermodynamically. The slope of the saturated vapor line on the T-S diagram is positive, hence mois-

ture should tend to not form in the expander. Boiler pressure levels are low, but the condenser pressure levels are below ambient. The operating pressure ratio is high enough that two stages of expansion are required. This may make this refrigerant undesirable.

8. R-C318. A probable lack of expander moisture problems, a moderate boiler pressure level, and acceptable operating pressure ratio are the only favorable points that R-C318 has. It has a small h_{fg}/C_pT product indicating high irreversibilities in the boiler. R-C318 is limited in high temperature applications because of its low critical point. This refrigerant is not suitable for a solar Rankine engine working fluid.

Based on the criteria established the system working fluid to be used in the modeling process is R-114. While not optimum from the performance standpoint, this refrigerant may be the best from a practical view with the idea of constructing a unit which requires minimal maintenance over a long operating life. For an optimal fluid with performance as the only criterion, R-11, R-21 or R-113 would be the choice, but they have some cost and operating difficulties. The probability of moisture damage in the expander was the deciding factor against R-11 and R-21. Additionally, both R-11 and R-113 have condenser pressure levels lower than atmospheric which would entail either a much costlier design to vent non-condensibles or periodic (weekly) maintenance. R-11 and R-113 would both need two stage expanders to operate efficiently in the extreme operating conditions noted for the pressure ratios of Table 1. R-114 is acceptable only because although it is not optimal, it has no glaring weakness.

2.5 Expander

The assets desirable in the expander for a solar Rankine engine are simplicity, high efficiency and compatibility with the load, in this case a conventional, low speed reciprocating compressor in a vapor compression unit. Various types of expanders have been developed during the past few years in the rated power output range which is of interest (from 2500 to 80,000 KJ per hour). Few, however, have passed into the production phase, hence performance data is scarce. The main types of expanders are turbine, rotary, and reciprocating. Each has its own merits for application in a solar Rankine engine.

The turbine obtains mechanical energy by the conversion of thermal energy into kinetic energy and then shaft work. Two basic types of turbines exist: impulse and reaction (15).

The impulse turbine expands the inlet vapor through a nozzle and directs the vapor stream onto blades attached to a shaft. The shaft revolves as a reaction to the force of the vapor stream. Pressure drop across the blades is minimal. Vapor inlet to the turbine is via a set of nozzles which are attached to a stationary ring. When these nozzles occupy the entire ring area the turbine is termed a full periphial admission turbine. Impulse turbines do not require full admission, so they operate at a higher part load efficiency. This is accomplished by throttling only some of the nozzles, while leaving others unthrottled. Impulse turbines are capable of higher enthalpy drops per stage than reaction turbines. The reaction turbine causes the shaft to rotate by a pressure drop across each moving stage of blading. It uses a stationary set of blades between the rotating stages as nozzles to direct the flow stream. There is little pressure drop across these stationery blades in a reaction turbine. Reaction turbines cannot use partial admission but have higher design efficiencies than do impulse turbines. Often a combination of both types is employed to obtain the advantages of both. The reaction turbine, because it has a lower enthalpy drop per stage than the impulse turbine, must have more stages than an impulse turbine.

The turbines used in small Rankine engines and using fluoronated hydrocarbons as the working fluid generally operate at a high design rotational speed. Reduction gearing on the output shaft is required to couple it to low speed generators and reciprocating equipment. High speed operation generally entails special treatment of bearings and seals to minimize frictional losses and wear which adds to the complexity of the system.

A reciprocating expander obtains mechanical energy by expanding a vapor in a non-flow process against a piston in a cylinder. Vapor is admitted at the correct time during the stroke by a valve. The expansion process continues until an exhaust valve is opened at the desired time during the stroke.

The basic types of reciprocating expanders are: reverse-flow and uniflow. The reverse-flow type locates both its intake and exhaust valves on the same end of the cylinder. The fluid flow is forced to reverse its direction in this type of expander. The uniflow type locates the intake valves at one end of the cylinder and the exhaust valves at the other. The flow stream is in one direction. The efficiency of this type is somewhat higher owing to less irreversibilities caused by heat transfer from the hot inlet fluid to a cold cylinder wall and vice versa.

Reciprocating expanders have their output controlled by either throttling of the inlet stream or variable cut-off governing.Throttling changes the work output by changing the inlet pressure for a fixed inlet valve open time for each stroke. Variable cut-off governing changes the time that the inlet valve is open for each stroke with an essentially constant inlet pressure. In this way it varies the mass flow rate through the system and the expander output.

Rotary expanders come in a wide variety of designs, but all follow essentially the same concept. A rotor which is connected to a shaft is installed within a larger diameter cylinder or stator. The stator is separated into high and low pressure sections via oscillating flappers, sliding vanes, or the construction of the rotor. The pressure differences force the rotor to move which causes a shaft to rotate. The rotation of the rotor is also used to promote the intake and exhaust of the working fluid.

A reciprocating expander with throttle governing was chosen to be modeled. The turbine is possibly a superior device, but it includes the difficulties of adding a speed reducer (a substantial source for power losses) and probably pressure lubrication for the bearings to deal with the high rotational speeds encountered. A rotary expander is an interesting idea (one which is currently under study at the Battelle Labs (16), but insufficient data is available to model such a unit.

Modeling the reciprocating expander involves the basic assumption that it performs similar to the way that a reciprocating compressor functions (9,17,18), but power is received from the fluid rather than delivered to it.

We define the volumetric efficiency of a reciprocating expander:

$$\eta_{VA} = \frac{\text{volume flow entering the expander}}{\text{displacement rate of the expander}}$$

Relating this to the expander rotational speed R,

$$R = \frac{\dot{m}_{SYS}\ \nu_{INLET}}{D\ \eta_{VA}} \tag{2.5-1}$$

Where $\dot{m}_{SYS}$ is the system mass flow rate, D is the expander displacement per stroke, and ν_{INLET} is the working fluid inlet specific volume.

The expansion is assumed to occur adiabatically between the shut-off valve outlet state to a state which is on the condenser pressure level. Thus for a given inlet state and the outlet pressure known for an isentropic expansion we may determine the isentropic outlet state. Then the isentropic work is defined as:

$$\Delta W_{E\ ISEN} = h_I - h_o' \tag{2.5-3}$$

Where h_I is the inlet state specific enthalpy and h_o' is the outlet specific enthalpy for an isentropic expansion. The actual work is determined via an adiabatic efficiency (η_E), assumed to be constant, as:

$$\Delta W_{E\ ACT} = \Delta W_{E\ ISEN}\ \eta_E \qquad (2.5\text{-}3)$$

The actual outlet state may now be defined:

$$h_o = h_I - \Delta W_{E\ ACT} \qquad (2.5\text{-}4)$$

The outlet state has the same pressure as the condenser. The power output of the expander is:

$$\dot{W}_{OUT} = \Delta W_{E\ ACT}\ \dot{m}_{SYS} \qquad (2.5\text{-}5)$$

The value of the volumetric efficiency as a function of operating pressure ratio was taken from the Trane Company representative data for reciprocating compressors (19). Chlumsky (17) defines a theoretical clearance volumetric efficiency:

$$\eta_{vc} = 1.0 - m\left(\frac{\nu_{INLET}}{\nu_{DISCHARGE}} - 1\right) \qquad (2.5\text{-}6)$$

Modifying this relation to take into account the reversal in the direction of flow:

$$\eta_{vc} = 1.0 - m\left(\frac{\nu_{DISCHARGE}}{\nu_{INLET}} - 1\right) \qquad (2.5\text{-}7)$$

Where m is the ratio of clearance volume in the cylinder to minimum cylinder volume. Chlumsky further suggests that for reciprocating compressors circulating a fluoronated hydrocarbon refrigerant and where m is between 0.04 to 0.05 the actual volumetric efficiency may be obtained from the clearance volumetric efficiency by the relation:

$$\eta_{VA} = \eta_{vc} - 0.06 \qquad (2.5\text{-}8)$$

Where both efficiencies are expressed as a fraction. Comparing the volumetric efficiency obtained by equations 2.5-7 and 2.5-8 and a curve fit taken of the Trane Company data which is of the form:

$$\eta_{VA} = 1.0893 - 0.12725\ P_R + 1.6036 \times 10^{-2}\ P_R^{\ 2} - 1.1709 \times 10^{-3}\ P_R^{\ 3} + 3.3379 \times 10^{-5}\ P_R^{\ 4} - 3.0405 \times 10^{-7}\ P_R^{\ 5} \qquad (2.5\text{-}9)$$

We find that there is good agreement for most refrigerants and pressure ratios between the two methods. Equation 2.5-9 was used to obtain the expander volumetric efficiency as a function of P_R, the ratio of inlet over outlet expander pressures, in the model.

2.6 Shut-Off Valve

A shut-off valve, positioned at the inlet to the expander, is used to stop the Rankine engine in case of expander overspeed or other system failure. The presence of such a device would also account for some pressure drop between the boiler outlet and the expander thus improving the model.

We employ the assumptions of fully developed flow with complete turbulence and an incompressible flow to obtain a rough value of pressure drop at off-design conditions. For such assumptions we find that the orifice coefficient C is relatively constant for wide ranges of Reynolds number. At design conditions as denoted by the subscript D:

$$\Delta P_D = C \frac{\rho_D}{g_c} \frac{{V_D}^2}{2} \tag{2.6-1}$$

Where ΔP_D is the design pressure drop, ρ_D is the inlet fluid density, and V_D is the fluid inlet velocity.

Employing the continuity equation and solving for V,

$$V = \dot{m}/\rho A \tag{2.6-2}$$

Where A is some constant cross sectional area. Substituting:

$$\Delta P_D = C \frac{\rho_D}{g_c} \left(\frac{\dot{m}_D}{\rho_D A} \right)^2$$

$$\Delta P_D = \frac{C}{g_c A^2} \frac{(\dot{m}_D)^2}{\rho_D} \tag{2.6-3}$$

The shut off valve is going to be unmodulated so the values of part of equation 2.6-3 are relatively constant and can be consolidated as:

$$k = \frac{C}{g_c A^2} = \frac{\Delta P_D \rho_D}{\dot{m}_D^2} = \frac{\Delta P \rho}{\dot{m}^2} \tag{2.6-4}$$

Solving for the off-design pressure drop ΔP we obtain:

$$\Delta P = \Delta P_D \left(\frac{\rho_D}{\rho}\right)\left(\frac{\dot{m}}{\dot{m}_D}\right)^2 \tag{2.6-5}$$

Expressing equation 2.6-5 in terms of the inlet specific volume:

$$\Delta P = \Delta P_D \left(\frac{\nu}{\nu_D}\right)\left(\frac{\dot{m}}{\dot{m}_D}\right)^2 \tag{2.6-6}$$

This expression was used to predict the off-design drop of the shut-off valve.

2.7 System Control Strategy

At this point, the Rankine engine is configured to meet a static load at the design conditions. In reality, the engine will be forced to continually match its output which is determined by the solar system performance and ambient conditions to the required load input power. A control strategy is required to accomplish this. The options for control of the expander (20) include operation at a constant or variable expander speed with a throttled or unthrottled expander (possibly matching loads with an additional electric generator). Auxiliary energy may be added as heat to the boiler or work to the expander output shaft.

Constant speed expander operation with no throttling was chosen as the best control option for the expander. The amount of controls required is increased for this option over that of a variable speed expander, but it offers certain advantages:

1. Operation at constant speed is consistent with current operating practice for reciprocating compressors.
2. Data is lacking to make an accurate estimate of off-design speed expander and compressor performance.
3. Constant speed operation is consistent with the use of an electric motor as an auxiliary power source.
4. Constant expander speed may be used to control the operation of the Rankine engine.

No throttling is consistent with minimizing the losses in the unit.

An electric motor-generator is used as a motor to supply auxiliary energy as mechanical energy to the air conditioner, or as a generator to use the excess work of the expander and thus match engine output to required load. Heat addition in the boiler, while using a cheaper energy source (natural gas or oil), offers some marked disadvantages:

1. The fuel fired auxiliary must be modulated in order to match engine output to load.
2. Low engine efficiency (about 10%) means relatively high fuel consumption.
3. Heat addition to the water stream entering the boiler would raise the solar system temperatures and reduce its efficiency.
4. Heat addition to the working fluid at the exit of the boiler may cause local hot spots in the fluid stream and degradation of the fluid. Under a high heat input chlorinated hydrocarbons decompose to phosgene gas, a highly toxic substance. DuPont (21) suggests a maximum use temperature for R-114 of 377°C, but it is as low as 249°C for other refrigerants. Prolonged exposure to temperatures exceeding these limits results in decomposition of the fluid.

Despite the low part-load motor efficiency, electric motor auxiliary is suited to constant speed expander and compressor operation.

The generator option allows the engine to generate power when its output is greater than the input required by the air conditioner. In this manner, the generator matches the engine output to the required load while generating power rather than throttling the engine and wasting the energy. The value of this power is uncertain, but it is conceivable it could be used in supplying the household electrical demand.

The Rankine engine package, consisting of engine components, auxiliary, and controls is depicted in Figure 10. This engine would be placed in the solar energy system shown in Figure 1. The expander speed is held constant by varying the system mass flow rate by changing the speed of the boiler feed pump, as determined by the controller. A thermal cut-off in the controller shuts down the engine and the pump if the boiler and condenser waterside inlet temperatures fall out of the desired operating range. The controller also determines if the engine, the boiler-tank loop pump, and the condenser-cooling tower loop pump should operate, based on load power requirements.

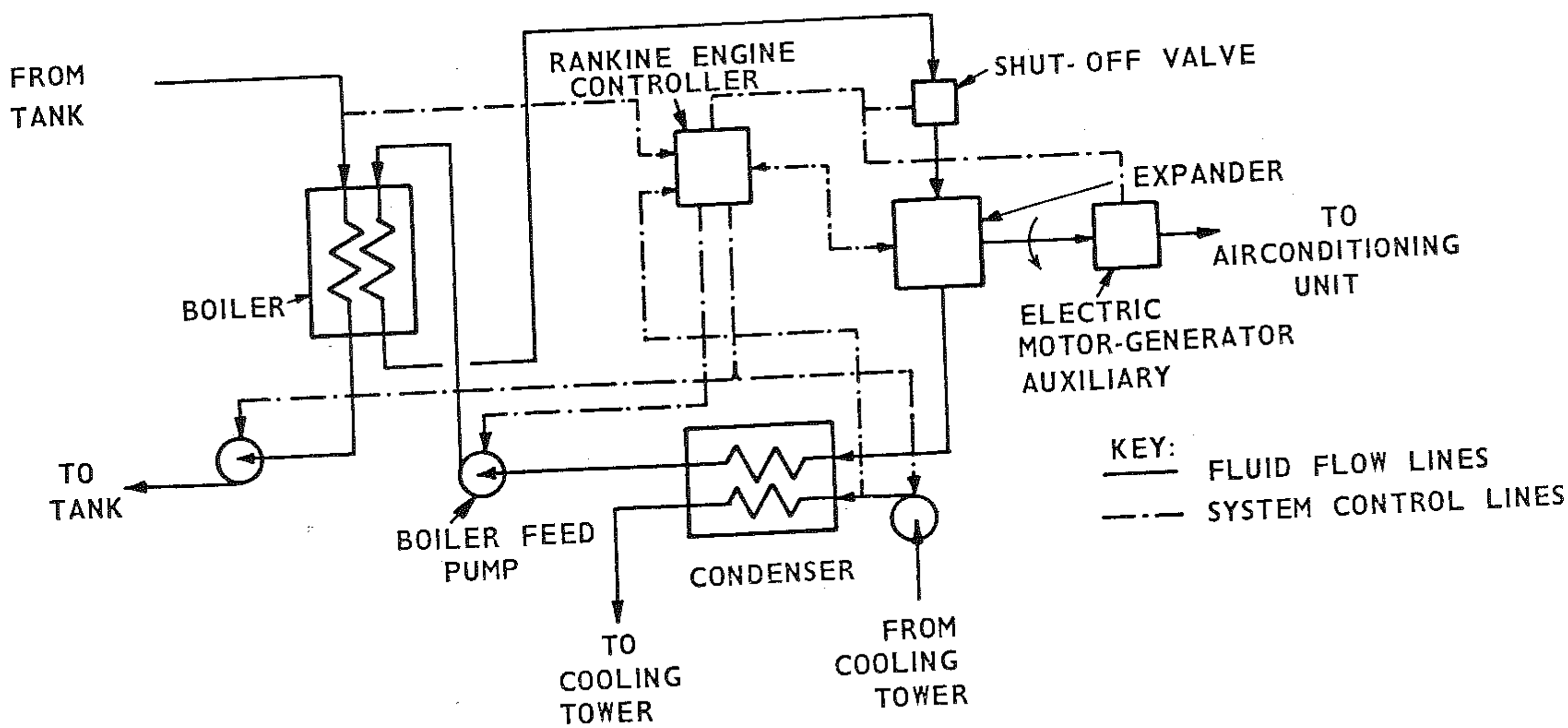


Figure 10. The Rankine Engine Package.

III. SOLAR SYSTEM COMPONENTS

3.1 Introduction

In order to be of value, the Rankine engine model should be operated over a variety of conditions which approximate those encountered by a real system. The performance of a solar Rankine engine is dependent on the solar system performance; thus without modeling an entire system, the characterization of the engine at design conditions can only be used to predict trends of operation. With this in mind, the Rankine engine was combined with the systems program TRNSYS (22,23). TRNSYS is a transient simulation program which contains many of the components (collector, water storage tanks, pumps, heat exchangers) that are commonly used in the simulation of solar heating and cooling systems in modular form. A house heating load model developed in modeling Colorado State University Solar House I by R. L. Oonk et al. (24) was included to provide a verified load model. Finally, a modified form of the vapor compression air conditioner and heat pump model developed for the GE Phase 0 Report to the National Science Foundation (20) was incorporated into the simulation. The following sections discuss the models of each major component used in the solar cooling system simulation.

3.2 Solar Collector

The solar collector model used in TRNSYS is a model of a flat-plate solar collector with forced circulation. The Hottel, Whillier, Bliss model is used since its results agree excellently with more elaborate models. This model expresses the rate of energy collection, $\dot{Q}_u$ as

$$Q_u = AF_R[H_T\tau\alpha - U_L(T_i - T_a)]$$

A = the collector area

F_R = a collector efficiency factor, see reference 25 for further details

H_T = the rate of total radiation per unit area incident on the tilted collector surface

τ = the transmittance of the glass cover(s)

α = the absorbtance of the collector plate for solar radiation

U_L = the collector overall energy loss coefficient per unit collector area

T_i = the inlet fluid temperature

T_a = the ambient temperature

The weather information used to drive the simulations is actual recorded data.

3.3 Thermal Storage Tank

The thermal storage tank is modeled as a fluid-filled sensible energy storage tank as described in Duffie and Beckman (25). The tank may be assumed to be fully-mixed and no stratification is to be considered. The heat rejection rate includes losses to the environment.

3.4 House Load

The house heating and cooling load used is one developed by Oonk (24) in the design studies for the CSU Solar I House project. This model was compared to the actual house performance for verification purposes and was found to give good representations of house performance.

The model consists of components which include an exterior wall, a roof, and a combined interior room and basement. These components are then used to "build" a house of the desired configuration. Exterior walls are modeled as having multiple nodes each with thermal capacitance. Each node is connected in thermal network with the appropriate value of thermal resistance for the given construction between nodes. The user decides on the type of construction and then assigns pertinent design information for each wall (eg. wall area, percent glass area, percent shading of the glass, orientation of the wall, absorbtance for solar radiation, wall construction, and the latitude of the house). The roof component requires the design information of: roof construction, the roof pitch, the house latitude, solar absorbtance, area, orientation, and optionally for pitched roofs, the collector area. The room and basement component requires design information of house volume, infiltration rate, internal generation, house perimeter, basement area and depth, and ground water temperature. Given this data plus ambient temperature, incident solar, wind speed, and room temperature setting the heating or cooling load may be calculated.

3.5 Air Conditioner

The model of a vapor compression air conditioner used in the simulation is the model developed by the General Electric Space Division for the NSF Phase 0 Report on Solar Heating and Cooling of Buildings (20). It is based upon GE data for air-to-air heat pumps in both the heating and cooling mode. The COP of the cooler was assumed independent of size, when used with air-to-air heat pump systems with design capacities between 8.79 kw (2.5 tons) and 35 kw (10 tons). A description of the computer program used and a listing of the program is included in

Appendix B.

3.6 Cooling Tower

In establishing the design criteria for the Rankine engine, the assumption was made that the condenser would use a liquid as the heat rejection fluid. This requires that the heat eventually be rejected to a cooling tower since the use of ground water or municipal water sources for this purpose is prohibited in most locations. A comparison may be drawn with the performance of an absorption air condition, which also uses a cooling tower for heat rejection purposes if a cooling tower is also used for the Rankine engine. The cooling tower is modeled as a constant approach device (eg. the return water temperature is the ambient wet bulb temperature plus a constant).

3.7 Rankine Engine

The Rankine engine model uses user-supplied input parameters to size an engine on a specified design point. Given the input parameters, refrigerant property data, and the sized engine, the simulation of the engine may be accomplished in a variety of manners.

The user may choose to execute the engine model in a design study in which individual operating states are considered. In this case the control option IMAP is set equal to 1 and the model is configured as in Figure 11a. (A detailed explanation of each subroutine is contained in Appendix A). Such a configuration would only be used for a study which involves a small number of calls to the model, generally less than 100. For long term simulations the model can produce a "black box" model by first generating performance information for all operating states within a desired range. This information is stored as a performance map for this machine as sized by the user. The performance map is then used to provide a black box model of the engine's operation by interpolating for the desired operating state in the map rather than solving the required equations which model the engine for each state. Computational accuracy is unaffected by the use of the black box model, but computational time is substantially reduced. Figure 11b depicts the engine model as

Figure 11a

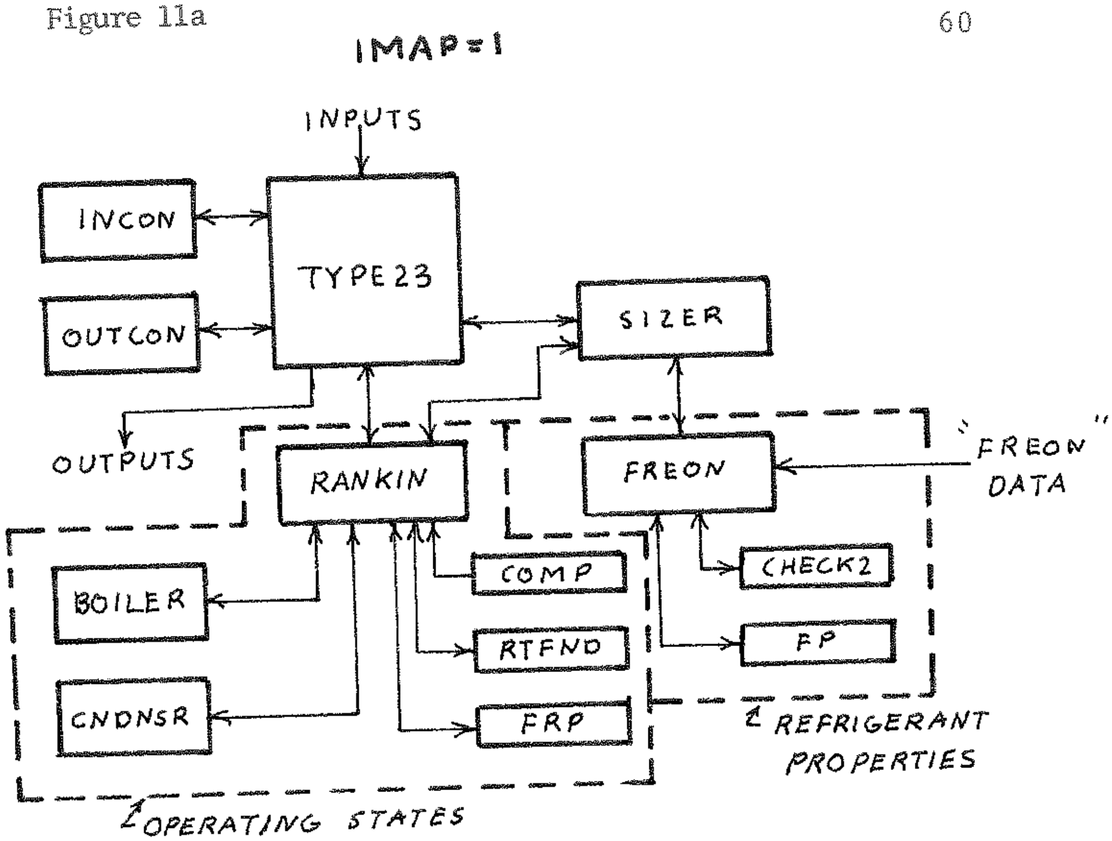


Figure 11b

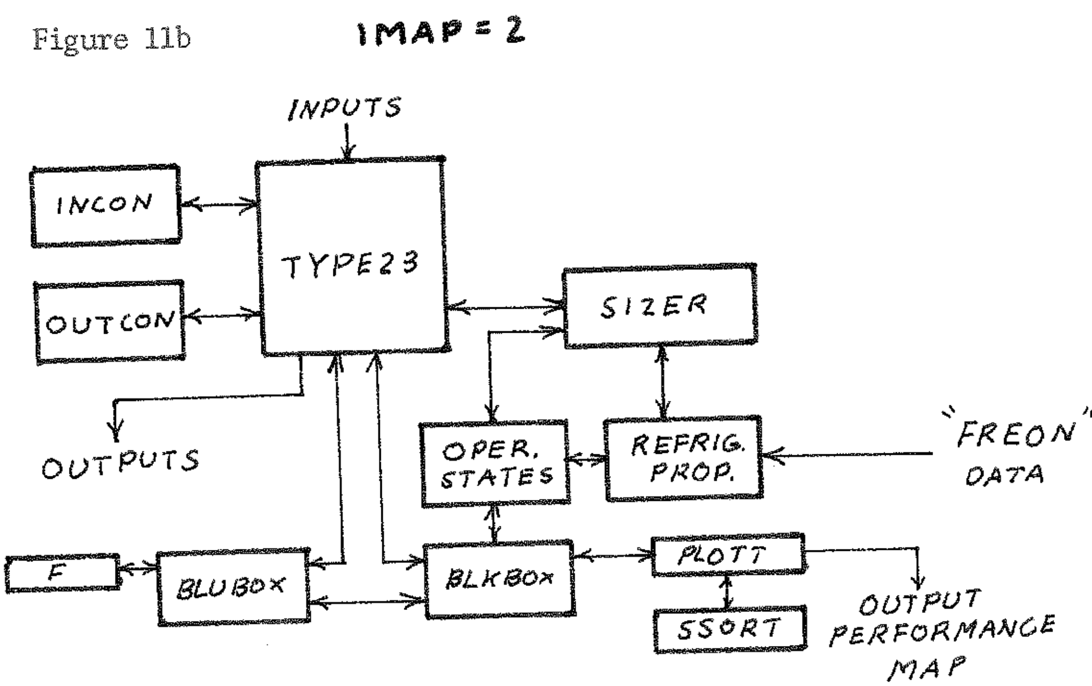


Figure 11. Information Flow Diagrams for the Control Modes of the Rankine Engine Model

Figure 11c

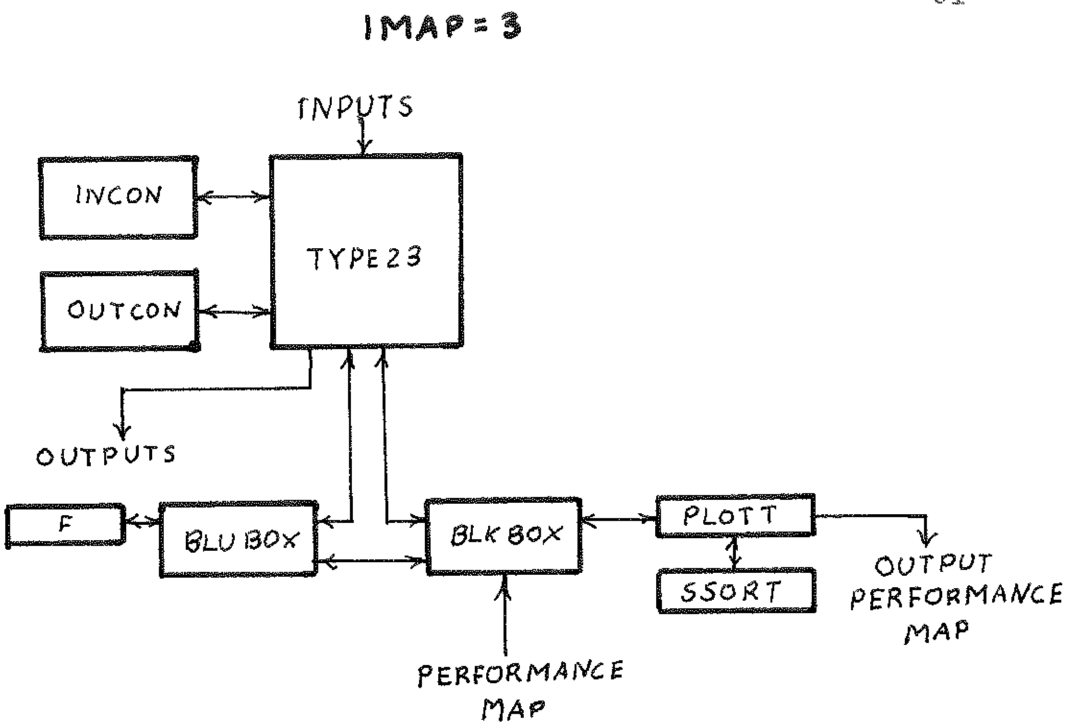


Figure 11d

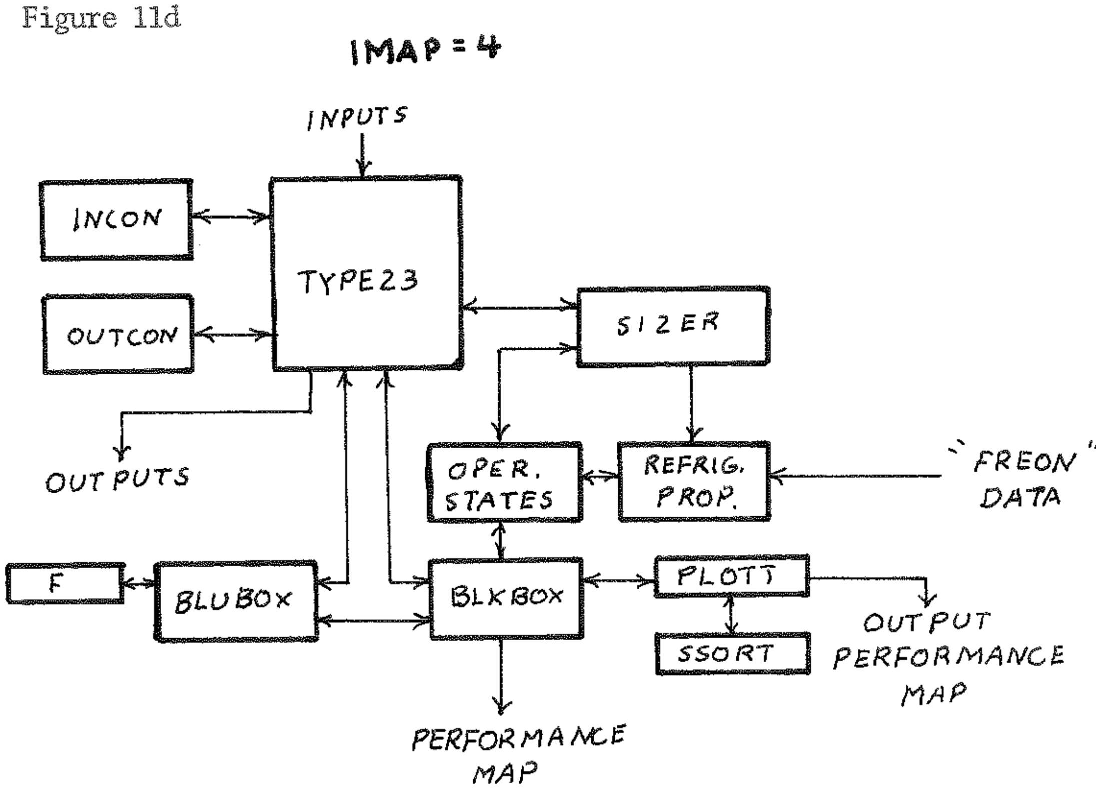


Figure 11. Information Flow Diagrams for the Control Modes of the Rankine Engine Model

configured for such a simulation for IMAP=2. A sample performance map for an engine using R-114 as a working fluid is illustrated in Figures 12, 13 and 14. With the design assumptions in the selection of engine components discussed in Chapter 2, the engine was designed with the following design condition parameters:

1. Expander work output = 2.23 kw (3 HP)
2. Boiler waterside inlet temperature = 101.8°C.
3. Boiler liquid-to-liquid section mean overall heat transfer coefficient = 0.454 kw $/m^2$°C
4. Boiler inlet to outlet waterside temperature drop = 5.56°C
5. Boiler outlet temperature difference, waterside minus working fluid outlet = 2.78°C
6. Condenser waterside inlet temperature = 26.7°C
7. Condenser liquid-to-vapor section mean overall heat transfer coefficient = 0.397 kw/m^2°C
8. Condenser inlet to outlet waterside temperature rise = 5.56°C
9. Condenser outlet temperature difference, working fluid minus waterside outlet = 2.78°C
10. Expander shaft speed = 3600 RPM
11. Expander adiabatic efficiency = 80%
12. Boiler feed pump shaft speed = 1800 RPM
13. Boiler feed pump adiabatic efficiency = 70%

Figure 12. Engine Thermal Efficiency As A Function of the Waterside Inlet Temperatures.

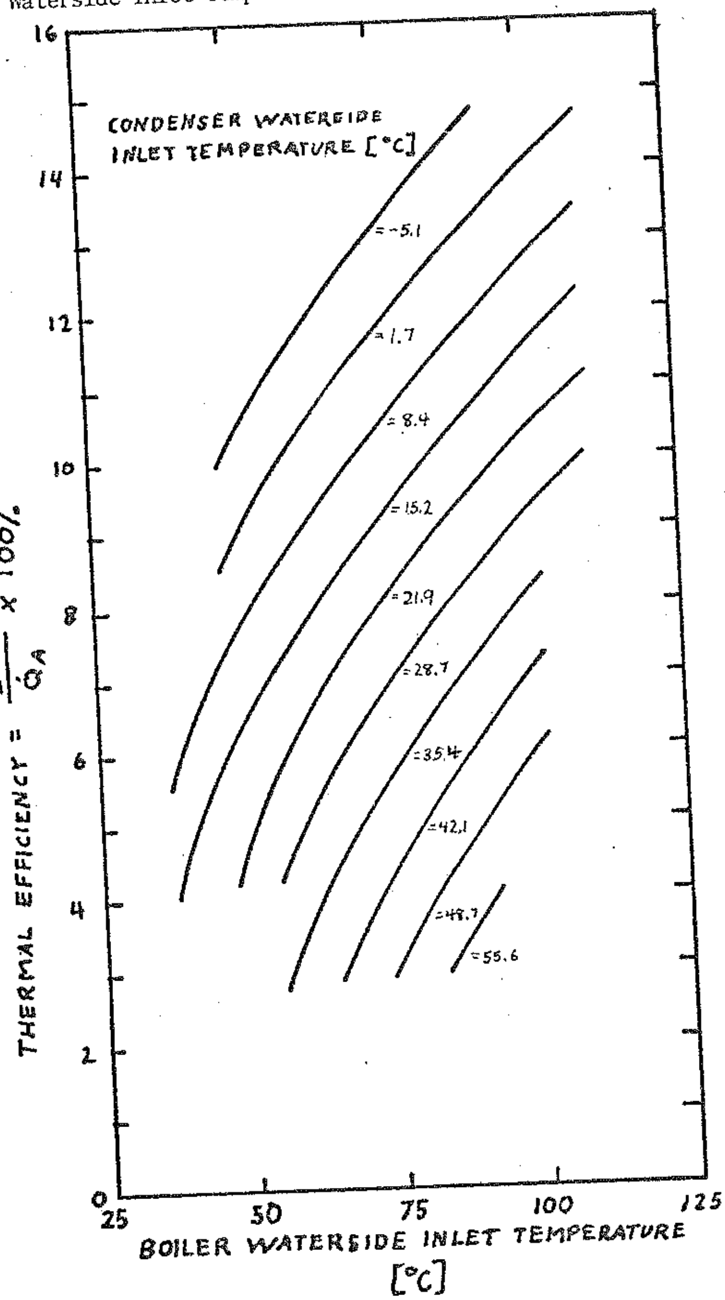


Figure 13. $\dot{W}_{OUT}/\dot{W}_{OUT\ DESIGN}$ for the Expander As A Function of the Waterside Inlet Temperatures

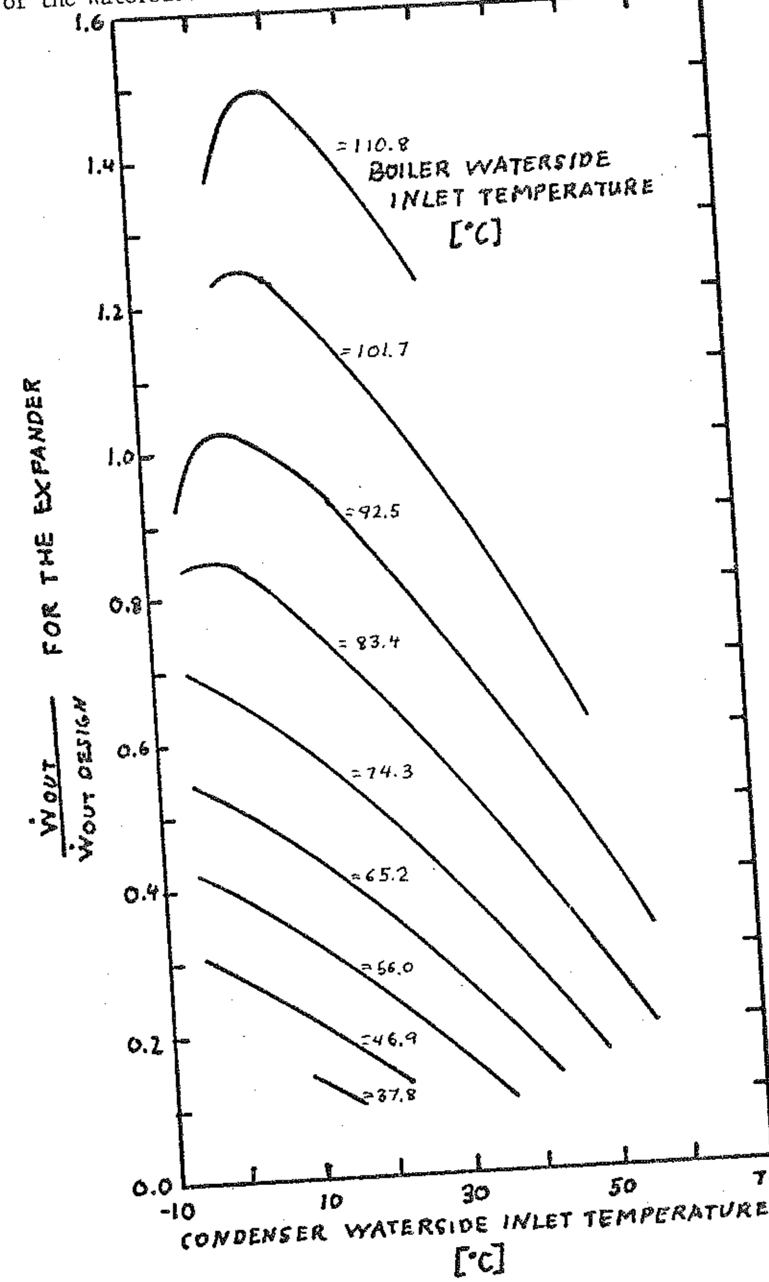
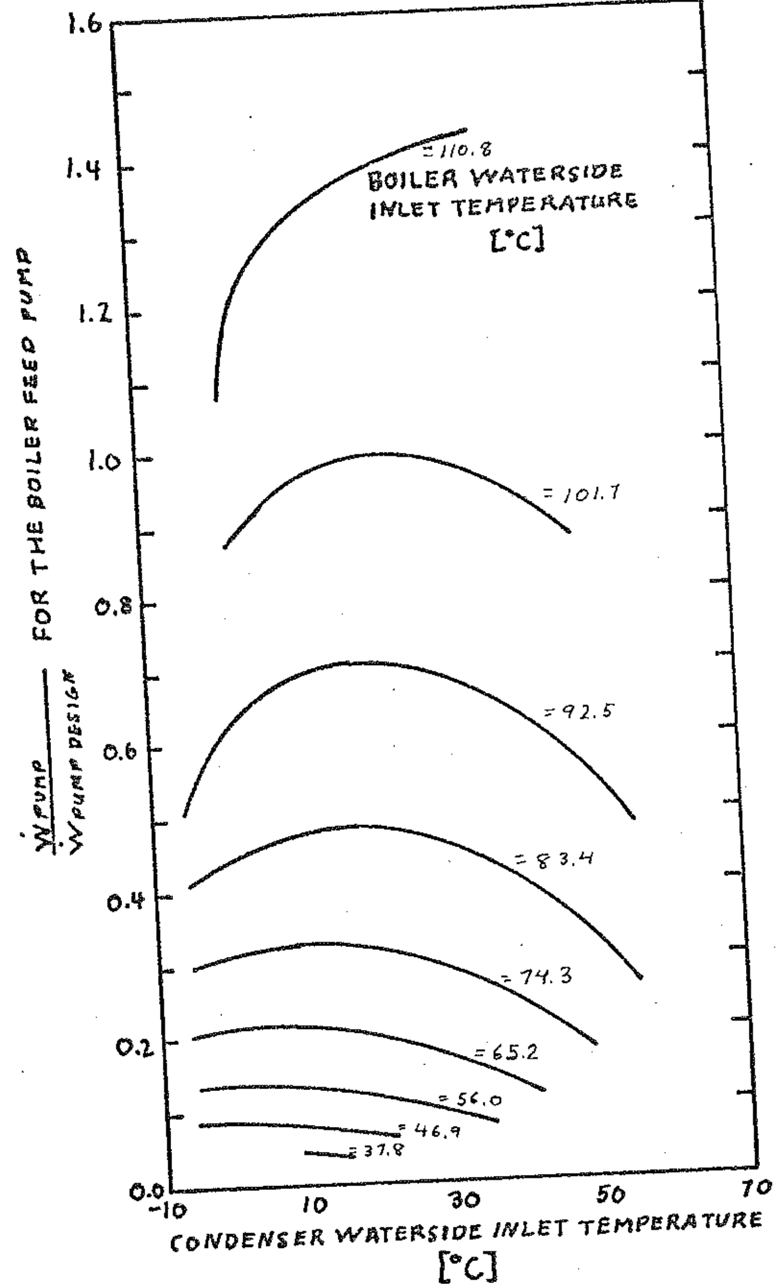


Figure 14. $\dot{W}_{PUMP}/\dot{W}_{PUMP\ DESIGN}$ for the Feed Pump As A Function of the Waterside Inlet Temperatures.



14. Shut-off valve pressure drop = 34.5 kPa

A parallel flow boiler and condenser configuration was selected for this unit.

The performance map produced for this engine consists of three separate functions of boiler waterside inlet and condenser waterside inlet temperature: engine thermal efficiency as depicted in Figure 12, the ratio of expander output power to design expander output power as shown in Figure 13, and the ratio of pump input power to design pump input power as given in Figure 14. Given the input temperatures to the boiler and condenser, subroutine BLUBOX interpolates in the performance map using a Lagrangian function and determines the engine performance for that operating state as follows:

Given $T_{BOIL\ IN}$ and $T_{COND\ IN}$, the functions; η_{TH} $(T_{BOIL\ IN}, T_{COND\ IN})$, WEXR $(T_{BOIL\ IN}, T_{COND\ IN})$, and WPR $(T_{BOIL\ IN}, T_{COND\ IN})$, the waterside boiler mass flow rate $(\dot{m}_{BOIL})$, the waterside condenser mass flow rate $(\dot{m}_{COND})$, the design expander power $(\dot{W}_{EXP\ D})$, and the design pump input power $(\dot{W}_{PUMP\ D})$:

$$\dot{W}_{EXP} = \text{WEXPR}\ \dot{W}_{EXP\ D} \qquad (3.7\text{-}1)$$

$$\dot{W}_{PUMP} = \text{WPR}\ \dot{W}_{PUMP\ D} \qquad (3.7\text{-}2)$$

$$\dot{Q}_A = \frac{\dot{W}_{EXP} + \dot{W}_{PUMP}}{\eta_{TH}} \qquad (3.7\text{-}3)$$

$$T_{BOIL\ OUT} = T_{BOIL\ IN} - \dot{Q}_A/(\dot{m}c_p)_{BOIL} \tag{3.7-4}$$

$$\dot{Q}_R = -\dot{Q}_A + (\dot{W}_{EXP} + \dot{W}_{PUMP}) \tag{3.7-5}$$

$$T_{COND\ OUT} = T_{COND\ IN} - \dot{Q}_R/(\dot{m}c_p)_{COND} \tag{3.7-6}$$

The performance at any operating state within the user specified limits can be easily defined in this manner.

When the same engine is to be used in a variety of simulations the model can write and save the performance on mass storage (IMAP=4 as shown in Figure 11d) or read from mass storage without resizing the unit (IMAP=3 as in Figure 11c). The Rankine engine model also incorporates a plotter which prints out the performance map on a standard line printer as an aid to the design of the engine.

IV. SYSTEM PERFORMANCE

4.1 Introduction

Numerous studies (1,2,3,4,5,11,20) have proposed or attempted to evaluate the performance of a Rankine cycle engine connected to a vapor compression air conditioner in order to supply cooling by solar energy. At this time, available design operation data is sketchy and data for the operation of such a system under actual operating conditions is not available. Two different types of numerical experiments will be discussed in the following paragraphs. The first set of results discuss design options for the Rankine engine. The second set of results discuss options in the system, excluding the Rankine engine.

4.2 Working Fluid Comparison

Based on the design assumptions in Chapter 2, a Rankine engine was designed using the parameters described in Section 3.7. The design shut-off valve pressure drop was assigned to be equal to 0.0367 x (P_{BOILER} - $P_{CONDENSER}$). Assigning a constant pressure drop for all working fluids would have penalized those fluids which have a small difference between boiler and condenser pressures and helped those with large differences. For R-114 the value of this pressure drop is 34.5 kPa.

A parallel flow boiler and condenser configuration was selected for computational simplicity. There is essentially no performance difference between the parallel flow and counter-flow modes for this design. This can be explained by examining the boiling process where most of the heat is transferred isothermally. Referring to Kays and London (7), it can be shown that for a heat exchanger with one fluid having an infinite thermal capacitance rate (a fluid which is evaporating or condensing and is isothermal), the concept of parallel flow or counterflow is meaningless. Additionally, the water experiences only a small temperature drop so that the liquid-to-liquid sections in the boiler see essentially the same temperature difference for either flow configuration.

Using refrigerant property data from the DuPont Company (26,27) for R-C318, R-11, R-114, R-113, and R-12 and the Rankine engine as specified previously, a comparison was made to determine thermal efficiency for varying boiler and condenser waterside outlet temperatures. Examining Figure 15, which is for condenser waterside inlet temperatures of 40°C, 30°C and 20°C, we note that R-11 has the best thermal efficiency while R-C318 has the worst. As predicted, an engine using R-114 has an intermediate performance, better than R-12 but poorer than R-11 and R-113.

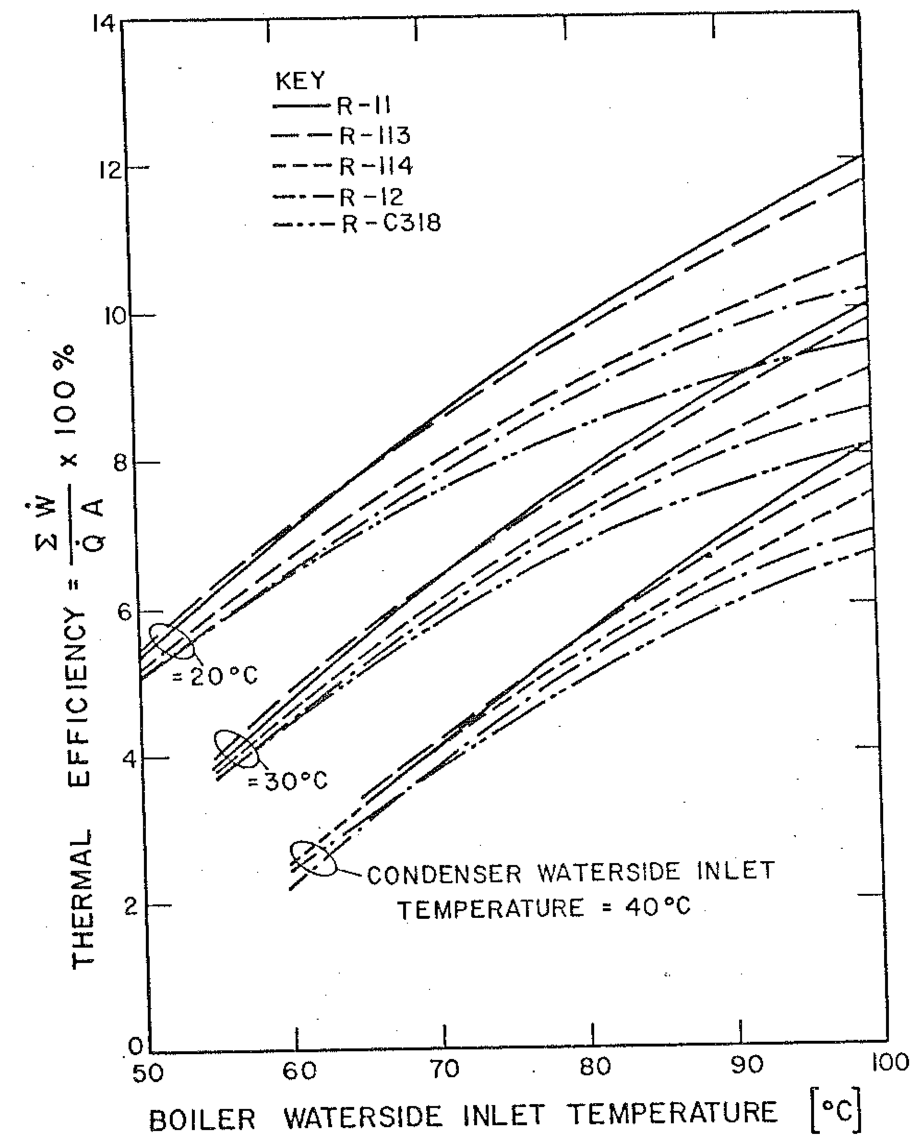


Figure 15. Engine Thermal Efficiency Versus Boiler Waterside Inlet Temperature For Various Working Fluids

A Rankine engine which is designed for the optimum performance over the range of operating conditions encountered by a solar system will use either R-11 or R-113. Under the temperature extremes indicated in Figure 15 these fluids are superior to the others tried in the model. However, as pointed out previously, thermal performance may not be the deciding factor on such a system. In general, the model used will include R-114 as the working fluid since it is felt this refrigerant offers the best compromise on the choice of working fluid.

4.3 Expander Performance

In investigating the performance of the Rankine engine components, the effect of expander efficiency on Rankine engine design point thermal efficiency was considered. The engine used was a 1.85 kw (2.5 HP) expander design output unit using R-114 as the working fluid. The design boiler feed pump efficiency was 80% and the shut-off valve pressure drop was 0.10 x $(P_{BOILER}-P_{CONDENSER})$ or 93.8 kPa for this case. All of the remaining parameters for the engine were the same as used in the working fluid comparison.

Figure 16 shows that cycle thermal efficiency is a strong function of expander adiabatic efficiency. Current

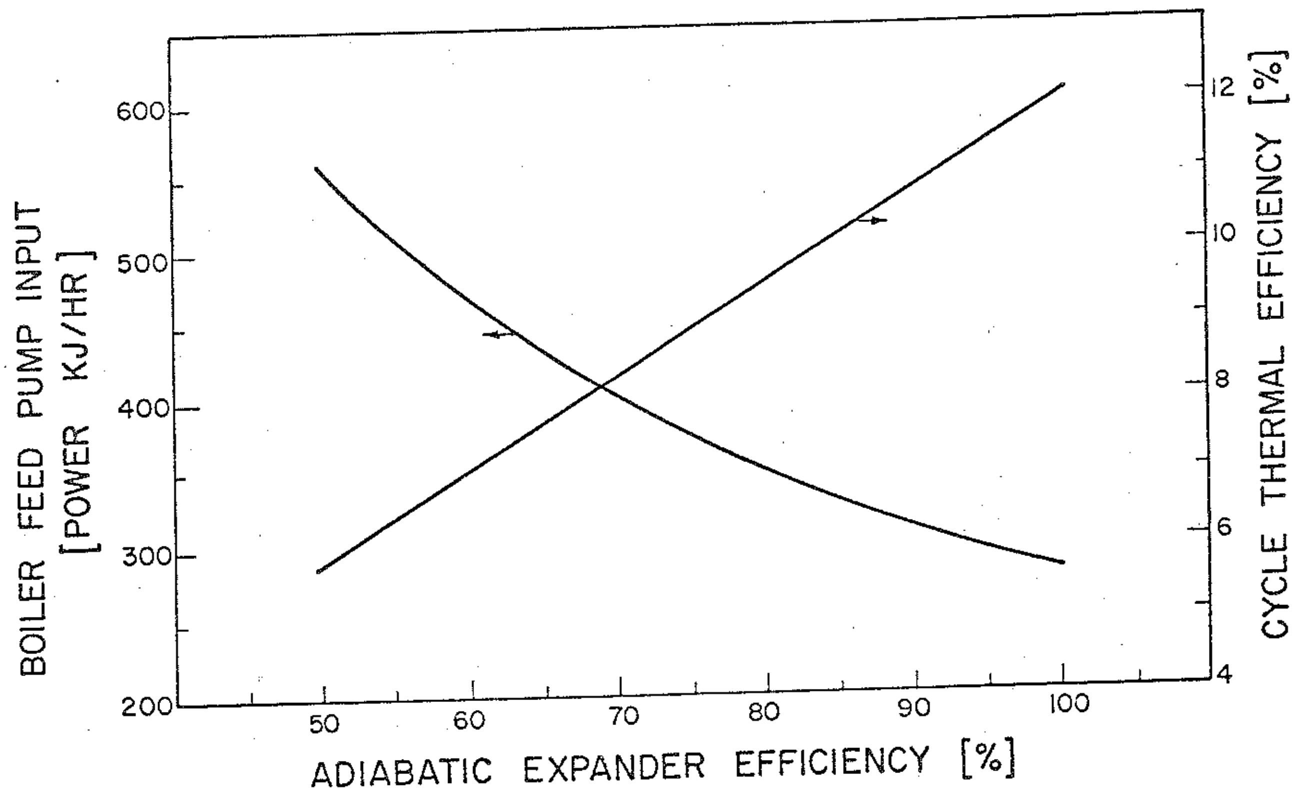


Figure 16. Expander Adiabatic Efficiency Versus Engine Thermal Efficiency and Feed Pump Input Power

expanders are in the 60 to 80% adiabatic efficiency range. The improvement of expander efficiency from 70 to 85% would result (for the machine designed here) in a 1.9% increase in cycle efficiency (from 8.3 to 10.2%). This represents a substantial (23%) increase in engine thermal efficiency.

Figure 16 also shows the required Rankine engine feed pump input power versus expander adiabatic efficiency. Feed pump power increases with decreasing expander efficiency. The system mass flow rate must increase as expander efficiency decreases in order to meet the desired power output. Increased mass flow rate in the system increases the power input required and results in increased operating costs.

4.4 Boiler Feed Pump Performance

The effect of boiler feed pump efficiency on Rankine engine thermal efficiency parallels the study on the effect of expander efficiency. The engine was the same as the one used in the expander tests with a design expander output of 1.85 kw (2.5 HP) and a working fluid of R-114. The design expander adiabatic efficiency is 80% and the shut-off valve design pressure drop was 93.8 kPa. All the remaining parameters were the same as previously used.

It is shown in Figure 17 that cycle thermal efficiency is weakly dependent on pump efficiency. Varying the pump efficiency from 50 to 100% increased the Rankine engine efficiency by 0.38%.

As expected, input pump power is strongly affected by pump adiabatic efficiency. For power inputs of this low level, fractional horsepower motors with the resulting poor efficiencies will be necessary. Variable speed coupling was used which results in even more losses. Fortunately, the values of power involved are small and the net power lost is small. The performance of the boiler feed pump is not critical to the performance of the Rankine engine.

4.5 Solar System Performance

The Rankine engine model was added to a solar system using water as the heat transfer and thermal storage medium. This system was made up of components from the simulation program TRNSYS (22,23). The system was set up as shown in Figure 1. The design parameters common to all of the systems modeled are listed in Table 2.

With this system, simulations were run over a cooling season (April through September) with meteorological data

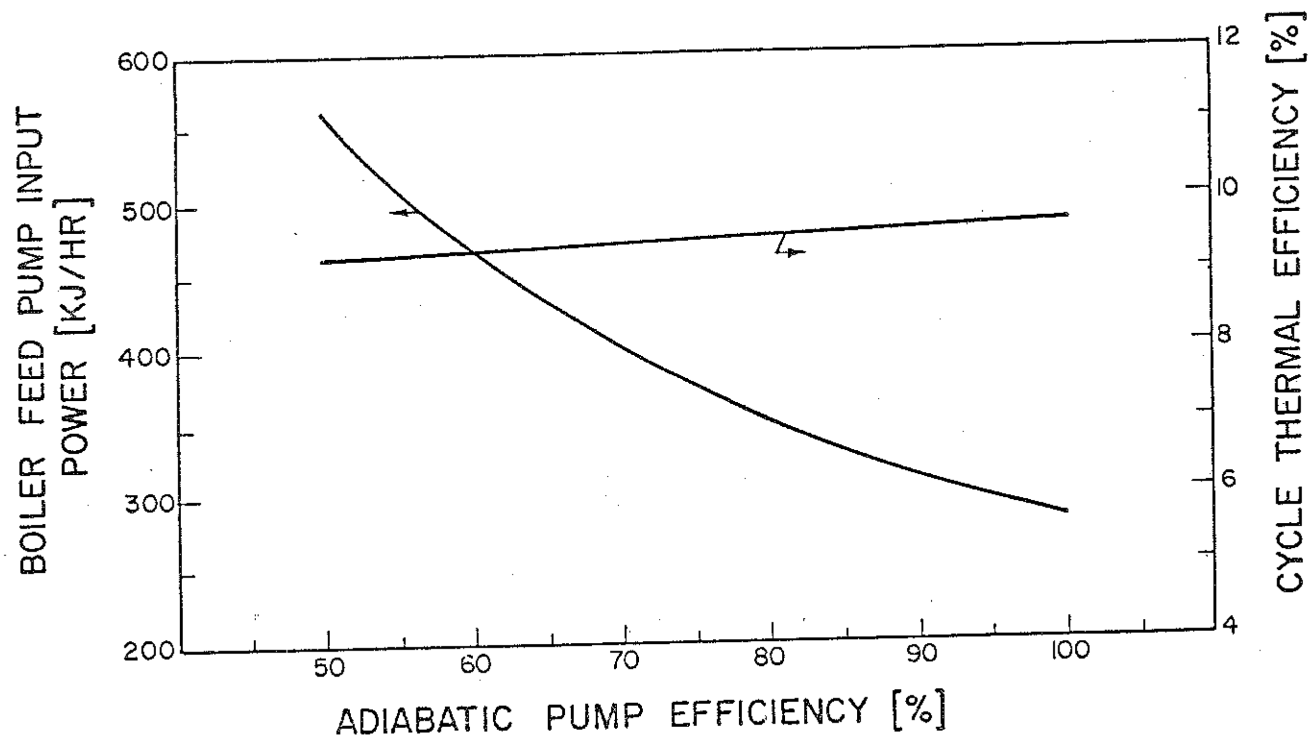


Figure 17. Boiler Feed Pump Adiabatic Efficiency Versus Engine Thermal Efficiency and Feed Pump Input Power

Table 2. Solar Cooling System Design Parameters

1. Collector (essentially a 2 cover selective surface design).
   a. Geometry efficiency factor (F') = 0.95
   b. Plate absorptance for solar radiation (α) = 0.9
   c. Numbers of covers = 2
   d. Plate emittance for infrared radiation (ε_p) = 0.10
   e. Back edge loss coefficient (U_{be}) = 1.5 kJ/hr m^2 °C
   f. Tilt from horizontal ≈ 40°
   g. Transmittance of the 2 covers (τ) = 0.82
   h. Front losses calculated as a function of conditions as discussed in reference 25.
2. Collector loop to tank loop heat exchanger effectiveness to set to 0.70
3. Pump flow rates on both loops are set to be 50 kg/hr for each m^2 of collector area. Pumps are turned on when collector fluid temperature rise is greater than 6°C
4. Tank
   a. Volume is set to be 75 kg for each m^2 of collector area except as specified.
   b. Height is 2m

c. Overall loss coefficient is 1.5 kJ/hr $m^2 °C$

5. Rankine engine, with design parameters similar to those described in previously except:
   a. Expander design output is 1.49, 2.23 or 2.98 kw as specified (2 HP, 3 HP, and 4 HP, respectively)
   b. Working fluid is R-114
   c. Boiler feed pump efficiency is 70%.
   d. Expander efficiency is 80%.
   e. Shut-off valve design pressure drop is 34.5 KP_a.
   f. The controller is set to not run the Rankine engine unless a work load is required and ΔT_{MIN}= (boiler waterside inlet-condenser waterside inlet) is greater than 45°C unless otherwise specified.
   g. The mass flow rate in the Rankine engine water pumps is the amount required to meet the design waterside temperature drop in the boiler and rise in the collector.
6. Cooling tower with a constant approach to ambient wet-bulb temperature of 5.56°C.
7. Vapor compression air conditioner
   a. Cooling capacity at the American Refrigeration Institute 240 design standards = 10.6 kw (3 tons).
   b. Design coefficient of performance excluding fan power input = 4.0

c. Fan power input at design = 0.85 kw (1.20 HP).

8. House Load

   a. Design heating requirements are for a house with an approximate overall loss coefficient UA = 0.33 kw/°C

   b. Internal generation is set to 0.67 kw

   c. Environmental losses from the tank are considered as extra internal generation for Albuquerque, New Mexico. The data used included incident solar radiation on a horizontal surface for the past hour, average wind speed, relative humidity, and ambient wet and dry bulb temperature. Albuquerque was chosen because of the relatively high heating and cooling loads encountered. With the cooling requirements of that location, the introduction of a solar cooling unit would presumably make the use of solar energy more feasible for year round application. The tabulated results with the energy quantities involved for all of the simulations are included in Tables 3a and b. The following discussion is a summary of the major results of those simulations.

As expected, the percent cooling by solar increases with increasing collector area, as shown in Figure 18.

An unexpected result is that for small collector areas, the smallest Rankine engine-air conditioner unit supplies the highest percent of the cooling load by solar. The situation is this; for small collector areas any Rankine engine has heat addition requirements which are larger than the amount which can be supplied by solar. The largest Rankine engine pulls the system temperature down to the lowest possible operating temperature allowed by the controller and then maintains that condition. The collector efficiency is high as a result of the low tank

Table 3a. Energy Quantities for the Cooling Season Simulations for Albuquerque*

ENGINE	A_c [1,3]	η_{RC} [2]	W_{OUT}	W_{MTR}	W_{GEN}	W_{COMP}	W_{AUX}	Q_{TANK}	Q_{CAPY}	Q_{DUMP}	Q_u	η_{COLL} [2]	% Cooling [2]
1.49 kw (2 HP)	10	7.25	1.85	3.15	0.29	4.70	1.61	23.6	21.1	0.0	24.8	53.7	33.1
	20	8.13	3.68	1.90	0.78	4.80	1.64	45.0	21.5	0.1	47.3	51.1	60.4
	30	8.86	5.72	1.08	1.88	4.91	1.68	62.5	22.0	1.1	66.8	48.1	78.1
	40	9.41	7.39	0.60	2.96	5.03	1.72	75.8	22.6	4.2	84.5	45.7	88.1
	50	9.84	8.79	0.31	3.95	5.15	1.76	83.0	23.1	12.3	101.2	43.8	94.0
2.23 kw (3 HP)	10	7.13	1.55	3.52	0.36	4.70	1.61	23.3	21.1	0.0	25.0	54.0	25.1
	20	7.49	3.55	2.04	0.81	4.78	1.63	46.9	21.4	0.0	49.2	53.2	57.4
	30	8.13	3.36	1.01	1.71	4.68	1.66	68.0	21.8	0.2	70.9	51.1	79.1
	40	8.64	7.68	0.44	3.16	4.96	1.69	86.3	22.3	0.9	90.8	49.1	91.0
	50	9.05	9.56	0.20	4.70	5.06	1.73	102.2	22.7	2.5	109.3	47.3	96.1
2.98 kw (4 HP)	10	7.11	1.67	3.36	0.33	4.70	1.61	23.6	21.1	0.0	25.0	54.0	24.3
	20	7.26	3.74	2.19	1.16	4.77	1.63	47.7	21.4	0.0	49.6	53.6	54.0
	30	7.71	5.51	1.13	1.79	4.84	1.65	70.0	21.7	0.0	73.0	52.6	76.7
	40	8.18	7.63	0.48	3.19	4.92	1.68	90.8	22.1	0.2	94.3	51.0	90.2
	50	8.53	9.63	2.18	4.85	5.00	1.71	109.6	22.5	0.9	114.6	49.6	95.6

* All Rankine engine design parameters as in Section 3.7
$\Delta T_{MIN} = 45°C$
Tank Size = 75 kg per meter^2 of collector area
All other solar system parameters as in Table 2.

1. Collector area in meter^2
2. Expressed as a percent; all other energy quantities KJ x 10^{-6}
3. These terms are defined on page 81a.

Table 3b. Energy Quantities for the Cooling Season Simulations for Albuquerque*

SIMULATION[2]	$\eta_{RC}^{1,3}$	W_{OUT}	W_{MTR}	W_{GEN}	W_{COMP}	W_{AUX}	Q_{TANK}	Q_{CAPY}	Q_{DUMP}	Q_u	η_{COLL}^{1}	% Cooling[1]
1 - 75 2 - 45	8.13	5.56	1.01	1.71	4.68	1.66	68.0	21.8	0.17	70.9	51.1	79.1
1 - 37.5 2 - 45	8.37	5.70	0.89	1.82	4.76	1.63	67.4	21.4	0.49	69.8	50.3	81.3
1 - 150 2 - 45	7.87	5.62	1.24	1.90	5.00	1.71	67.4	22.5	0.02	71.8	51.8	74.3
1 - 75 2 - 25	7.27	5.38	1.26	1.82	4.82	1.65	71.9	21.6	0.17	73.9	53.3	73.9
1 - 75 2 - 35	7.61	5.52	1.09	1.78	4.84	1.65	70.6	21.7	0.17	72.9	52.5	77.4
1 - 75 2 - 65	9.56	6.03	1.47	2.54	4.96	1.70	59.5	22.3	0.22	63.7	45.9	70.3

* All Rankine engine design parameters as in Section 3.7.
2.23 kw Rankine engine
30 m^2 collector area
All other solar system parameters as in Table 2

1 Expressed as a percent; all other energy quantities KJ x 10^{-6}

2 1 - Tank size m kg/m^2 collector area; 2 - ΔT_{MIN} in °C

3. These terms are defined on page 81a.

Table 3 (continued)

A_c - the collector area

η_{RC} - the Rankine engine thermal efficiency

W_{OUT} - the net engine output energy

W_{MTR} - the net auxiliary energy added by the electric motor

W_{GEN} - the net excess energy produced by the engine

W_{COMP} - the net input energy required for the air conditioning compressor

W_{AUX} - the net fan energy required for the air conditioner

Q_{TANK} - the net energy supplied as heat input to the Rankine engine from the solar system

Q_{CAPY} - the net house cooling load including environmental losses from the thermal storage tank

Q_{DUMP} - the net energy dumped by the collector relief valves

Q_u - the net useful energy collected

η_{COLL} - the collector efficiency which is defined as Q_u divided by the incident solar on the tilted surface

% Cooling - the percent of the cooling load met by solar means defined as:

$$= \frac{W_{OUT} - W_{GEN}}{W_{COMP}}$$

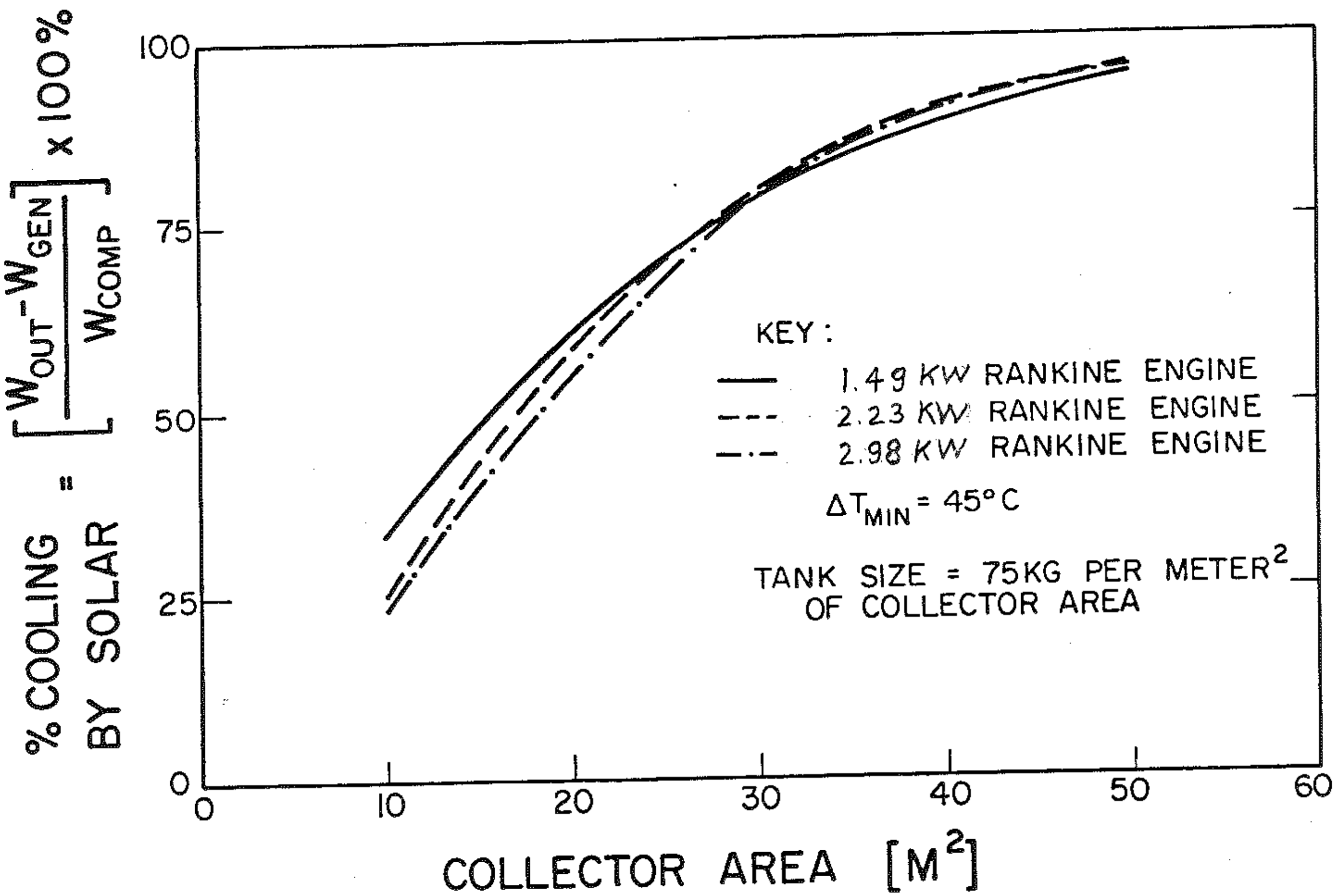


Figure 18. Collector Area Versus Percent Cooling Load Supplied By Solar Means

temperatures, but the large engine thermal efficiency is much lower. The result is that the overall operating efficiency of the large engine system is less than that of the small engine system. The small engine system allows the tank temperature to float above the controller minimum.

The smallest Rankine engine uses less of its output on generating electricity as shown in Figure 19 and has its output more evenly matched to the power requirements of the air conditioner a greater percentage of the time.

As the collector area is increased the solar system is able to supply more of the energy requirements of the engine. The solar system temperatures remain higher, and while collector efficiency falls, the Rankine engine efficiency increases. With a collector area of 30 m^2, the 1.49 kw (2 HP) engine supplies less of the percent cooling load by solar than does the larger 2.23 kw (3 HP) engine. Note that the largest engine, the 2.98 kw (4 HP) machine, is still limited in its performance by an undersized collector. The performance of the 2.98 kw (4 HP) unit approaches that of the 2.23 kw (3 HP) unit as collector area approaches 40 m^2.

The control strategy only allows the Rankine engine to run when the air conditioner operates. Figure 19 indicates how much the engine output was not matched to

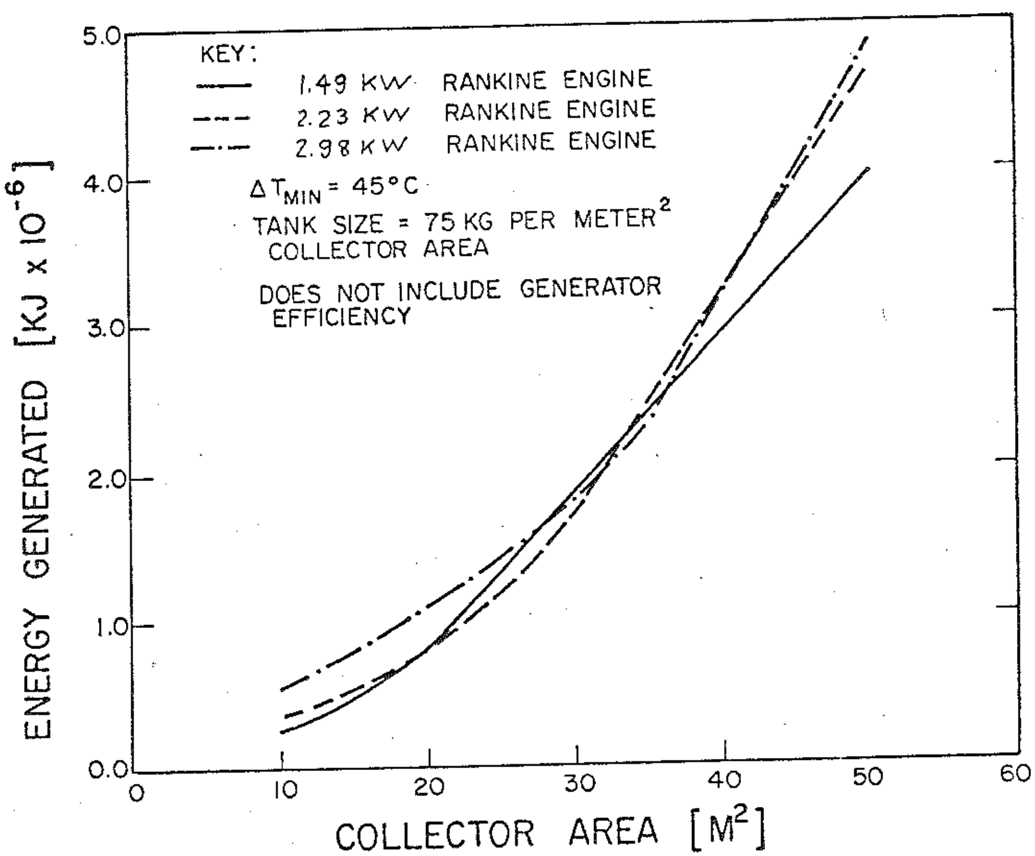


Figure 19. Collector Area Versus Excess Energy Generated By Solar Means

the load requirements. The air conditioner design compressor input power was 2.64 kw (3.43 HP); the 1.49 kw and 2.23 kw machines were undersized while the 2.98 kw machine was oversized at design conditions. From the results of this simulation it is apparent that the Rankine engine design output should be less than the load design power requirements if an auxiliary energy source is used. Like all solar systems it is inefficient to design the system to meet a very high percentage of the load with solar energy, since the uncertainty of the solar input will make this a difficult if not impossible goal to achieve. With small collectors the smallest Rankine engine provided the best performance. In larger collector areas the difference in performance is so small that the smallest engine remains the preferred choice since its initial and operating costs will be less.

Another parameter which was investigated was the effect of thermal storage tank size on system cooling performance. From the previous discussion on Rankine engine operation in a solar system with a well designed collector and well insulated tank, it was desirable to operate the solar system at high temperatures. A small tank will be heated up quicker than a large tank and the resulting higher average temperatures supplied to the Rankine engine may more than compensate for the reduced collector efficiency. The small tank may also increase the amount of energy dumped by the collector relief value.

Table 5 gives the percent solar for a 30 m^2 collector with a 2.23 kw (3 HP) engine and 3 different storage tank sizes. The percent cooling load supplied by solar is highest for the design of 37.5 kg of storage per m^2 of collector area and is the least for the design of 150 kg of storage per m^2 of collector area.

It is interesting to note that a definite minimum exists in energy generated for different tank volumes. The increase in energy generated as shown in Table 4 at the lowest tank volume is due to the Rankine engine operating at a higher output when it can run. This is possible with the small tank's higher operating temperatures. The increase in energy generated at the highest tank volume is due to the increased amount of storage in that system. Greater thermal storage means that the tank temperatures are lower and less energy is dumped by the collector relief valve. This increase in collection efficiency plus the increase in thermal storage allows the engine to operate more than in a system with a smaller tank.

A control parameter which was investigated in its relation to total solar system performance was ΔT_{MIN}, the minimum temperature difference between the boiler waterside and condenser waterside inlet temperatures allowed

Table 4. Performance of the Rankine Engine For Varying Values of Tank Size*

	TANK SIZE		
	37.5 kg/m^2	75 kg/m^2	150 kg/m^2
% COOLING BY SOLAR = $\frac{WOUT-WGEN}{WCOMP}$ x 100%	81.30	79.13	74.33
EXCESS ENERGY GENERATED $kJx10^{-6}$	1.824	1.713	1.895

* Storage Medium is Water ρ = 1000 kg/m^3

Collector area = 30 m^2

2.23 kw Rankine Engine

for Rankine engine operation. Holding all other system parameters constant it is expected that the curve of ΔT_{MIN} versus percent cooling by solar would have a maximum at some intermediate value. At low values of ΔT_{MIN} the Rankine engine operating efficiency is low because a low ΔT_{MIN} allows the engine to operate a large amount of the time and reduce the solar system temperature. The solar collector operates at a higher efficiency but this improvement is insufficient to overcome the degradation in engine performance. The percent cooling by solar for a low value of ΔT_{MIN} would be low. A high value of ΔT_{MIN} will allow the Rankine engine to run at high efficiencies, but also forces the tank to operate at higher temperatures. Consequently the collector will operate at low efficiencies and in addition energy may be dumped by the collector relief value.

The results of several simulations over the cooling season for Albuquerque are shown in Table 5. With a ΔT_{MIN} equal to 25°C, the percent cooling load supplied by solar is due to inefficient operation of the Rankine engine. A ΔT_{MIN} equal to 65°C causes the collector performance to fall sufficiently to reduce the percent of cooling load provided by solar. At intermediate values of ΔT_{MIN} (eg. 45-50°C) this particular system-load configuration shows an optimum. A different system would have an

Table 5. Performance of the Rankine Engine for Varying Values of ΔT_{MIN}*

	ΔT_{MIN}			
	25°C	35°C	45°C	65°C
% COOLING BY SOLAR = $\frac{WOUT-WGEN}{WCOMP}$ x 100%	73.91	77.42	79.13	70.31
EXCESS ENERGY GENERATED $kJ x 10^{-6}$	1.819	1.779	1.713	2.541

*$\Delta T_{MIN} = T_{BOILER\ WATERSIDE\ INLET} - T_{CONDENSER\ WATERSIDE\ INLET}$

Collector Area = 30 m^2

Tank size = 2.25 m^3 (water) = 2250 kg

2.23 kw Rankine Engine

optimum which was shifted (eg. a poorer collector design would shift the optimum to a lower value of ΔT_{MIN}). The value of ΔT_{MIN} required for optimum system performance is also a function of engine size and tank size. It is possible that if each system in Figure 18 had its own optimum value of ΔT_{MIN}, the largest system may have had the best performance.

ΔT_{MIN} could be a function of time because the size of the load (which has a general seasonal change) affects the way in which the cooling system operates. For optimal performance, the value of ΔT_{MIN} may vary in a seasonal manner. The optimal value for ΔT_{MIN} will be difficult to obtain because it will be different for each year.

Table 5 shows the amount of excess energy generated as a function of ΔT_{MIN}. Systems with a small ΔT_{MIN} run with low tank temperatures. The collector performance of such systems is high, but engine performance is low. The Rankine engine generates more electricity because the low ΔT_{MIN} control requirement is more easily fulfilled, and the engine runs a greater portion of the possible time. Systems with a high ΔT_{MIN} run with high tank temperatures. The collector performance of such systems is low, but the Rankine engine performance is high. The Rankine engine generates more electricity because it operates at a higher efficiency during the reduced periods it may run. Operating periods are reduced because of the reduced time when the solar system can exceed the high ΔT_{MIN} requirements.

V. CONCLUSIONS AND RECOMMENDATIONS

The Rankine engine cooling system is a feasible alternative to the absorbtion cycle in supplying cooling for residential use. The Rankine engine cooling system which uses an auxiliary energy source should be undersized relative to the design load power requirements. Design loads for cooling (and heating) are peak loads; sizing the engines of the type of system studied here to meet these loads results in less than optimum thermal performance when integrated over the cooling season.

There is an optimum tank size which maximizes the performance of the solar Rankine cooling system. This size is smaller than that recommended for heating systems (25). Finally, the engine may be controlled using some value of ΔT_{MIN} to optimize the system performance.

Although this system has been evaluated as uneconomical at this time by Hittman Associates, Inc. (5) for the National Science Foundation, it is believed that rising energy costs in the future could change this evaluation to a more favorable one. The Hittman analysis was much simpler in concept, using the performance of one design day at four different locations and a sinosoidal solar input to base its results on. It is possible that a more detailed analysis such as the one presented in this thesis may improve the understanding of the operation of a solar Rankine cooling system and change the results of the analysis.

The engine model still has several weaknesses. The expander, as presently modeled, is a single stage unit effectively limiting its useful operation to a ratio of inlet to outlet pressures of about 10. In extreme operating conditions or for some working fluids this may be exceeded. A second stage of expansion should be added to the engine model. It also may be of interest to add a turbine type expander as an option.

The condenser model may not be satisfactory for simulations of units in areas where evaporative cooling towers may be banned because of prevalent water shortages. A crossflow air condenser should be added as an option to the model to deal with such situations.

It is believed the engine model may be of more use to larger scale solar systems such as in commercial buildings. Simulation of solar cooling systems with the Rankine engine in large scale buildings may be of considerable interest.

Finally, a study should be made of the effects of the control parameter, ΔT_{MIN} on the performance of cooling systems with varying engine, collector, and tank sizes. The investigation into a seasonal variation of ΔT_{MIN} for optimal performance should also prove of considerable interest.

Appendix A. The Rankine Engine Model and A Program Listing

This section briefly describes the general purpose of each subroutine or function and how that routine interfaces with the remainder of the engine model. A listing of the Rankine engine model is included. An information flow diagram which indicates the configuration of the model subroutines is included as Figure 11.

Subroutine TYPE23

This routine is the interface with TRNSYS or any main program and also serves as the control section for the remainder of the model. TYPE23 determines whether the engine may run for the input variables on each step based on input control information and operation condition limits. Based on input information it decides if the users inputs are in the unit system KJ, Hour, m, °C or BTU, Hour, foot, °F. The dimensional system used exclusively in the model internally is the latter, and conversion to this system is accomplished by the routines INCON and OUTCON. Input variables and parameters (the arrays AXIN(I) and APAR(I), respectively) are unmodified, the arrays XIN(I) and PAR(I) convey the modified input to the remainder of the model. The parameter IMAP determines how the simulation should be executed as described in Section 3.7. Subroutine SIZER designs a Rankine engine on design conditions. Subroutine RANKIN produces performance data for each operating state. Subroutine BLXBOX produces the performance map when desired and subroutine BLUBOX uses the performance map to produce the black box model used in prolonged simulations.

```
      SUBROUTINE TYPE23(TIME,AXIN,OUT,DEP,DTDT,APAR,INFO)
C     TYPE23 IS THE CONTROL ROUTINE FOR THE RANKINE ENGINE
C     MODEL. THIS IS THE INTERFACE BETWEEN THE ENGINE MODELING
C     ROUTINES AND TRNSYS OR MAIN1.
C     THE ARRAYS AXIN AND APAR CONTAIN TRNSYS VARIABLES WHICH
C     REMAIN UNMODIFIED. XIN AND PAR CONTAIN VALUES WHICH ARE
C     MODIFIED TO HAVE THE UNITS BTU, FT, HOUR, DEGREE F. THIS
C     DIMENSIONAL SYSTEM IS USED EXCLUSIVELY WITHIN THE
C     ENGINE MODELING ROUTINES.
      DIMENSION XIN(10),OUT(15),PAR(40),DPAR(20),INFO(8)
      DIMENSION AXIN(10),APAR(40)
      DATA ICT/1/,ISKIP/1/
      IRUN=AXIN(4)
C     IRUN IS AN EXTERNAL CONTROL FUNCTION.
      IF(IRUN) 100,100,9
  100 CONTINUE
      DO 101 J=1,12
      OUT(J)=0.0
  101 CONTINUE
      OUT(3)=AXIN(1)
      OUT(4)=AXIN(2)
      OUT(7)=AXIN(3)
      GO TO 12
    9 CONTINUE
C     THIS IS A CHECK TO DETERMINE IF THE INPUT VARIABLES ARE
C     WITHIN THE USER SUPPLIED LIMITS AS DEFINED BY PARAMETERS 27-30.
      IF(AXIN(1).GT.APAR(27).OR.AXIN(1).LT.APAR(28)) GO TO 100
      IF(AXIN(2).GT.APAR(29).OR.AXIN(2).LT.APAR(30)) GO TO 100
      GO TO (14,15),ICT
   14 CONTINUE
C     IMAP DETERMINES HOW THE SIMULATION SHOULD BE EXECUTED, I.E.
C     IF A BLACKBOX SHOULD BE CREATED (IMAP=2&4), OR IF THE BLACKBOX
C     SHOULD BE READ FROM MASS STORAGE VIA A LOGICAL UNIT (IMAP=3)
C     OR WROTE ONTO MASS STORAGE AND SAVED VIA A LOGICAL UNIT (IMAP=4).
      IMAP=APAR(25)
C     ISI INDICATES TO THE SUBROUTINES INCON AND OUTCON IF THE INPUTS
C     ARE IN KJ, M, HOUR, DEGREES C (ISI=POSITIVE), OR BTU, HR, FT,
C     DEGREES F (ISI=NEGATIVE).
      ISI=APAR(24)
      APAR(24)=ABS(APAR(24))
   15 CONTINUE
      CALL INCON(AXIN,XIN,APAR,PAR,ICT,ISI)
      GO TO (5,5,4,5),IMAP
    5 CONTINUE
      GO TO (1,2),ICT
C     SIZER MAY BE CALLED ONLY ONCE IN A SIMULATION.
    1 CONTINUE
      CALL SIZER(PAR,DPAR)
      ICT=2
    2 CONTINUE
      GO TO (3,4,4,4),IMAP
    3 CONTINUE
```

```
      CALL RANKIN(XIN,OUT,PAR,DPAR)
      GO TO 8
    4 CONTINUE
      GO TO (6,7),ISKIP
    6 CONTINUE
C     BLKBOX MAY BE ONLY CALLED ONCE IN A SIMULATION.
      CALL BLKBOX(PAR,DPAR)
      ICT=2
      ISKIP=2
    7 CONTINUE
      TBOIL=XIN(1)
      TCON=XIN(2)
      WOUT=XIN(3)
      CALL BLUBOX(TBOIL,TCON,WOUT,OUT,DPAR,PAR)
    8 CONTINUE
      IF(ISI) 12,13,13
   13 CONTINUE
      CALL OUTCON(OUT)
   12 CONTINUE
      RETURN
      END
```

Subroutine INCON

Subroutine INCON converts the input variables and parameters from the users dimensional system of KJ, Hour, m, °C or BTU, Hour, foot, °F to the internal system of units (BTU, Hour, foot, °F). The users inputs are contained in the arrays AXIN(I) and APAR(I) which contain variables and parameters. The values of these arrays are modified according to the value of the variable ISI and transferred to the arrays XIN(I) and PAR(I). The values of the parameters are only modified on the first call to INCON.

```
      SUBROUTINE INCON(AXIN,XIN,APAR,PAR,ICT,ISI)
C     THIS ROUTINE MODIFIES THE INPUT VARIABLES AND PARAMETERS
C     FROM AXIN AND APAR AND INSERTS THEM IN THE ARRAYS XIN AND PAR.
C     ALLOWED INPUT DIMENSIONS ARE: KJ, M, HOUR, DEG C (ISI=+), OR
C     BTU, FT, HOUR, DEG F (ISI=-). INTERNAL UNIT SYSTEM IS BTU, FT,
C     HOUR, DEG F.
      DIMENSION XIN(10),AXIN(10),APAR(40),PAR(40)
      IF(ISI) 4,3,3
    3 CONTINUE
      XIN(1)=AXIN(1)*1.8+32.
      XIN(2)=AXIN(2)*1.8+32.
      XIN(3)=AXIN(3)/1.055
      XIN(4)=AXIN(4)
      GO TO (1,2),ICT
    1 CONTINUE
      PAR(1)=APAR(1)/1.055
      PAR(2)=APAR(2)*1.8+32.
      PAR(3)=APAR(3)/4.1867
      PAR(4)=APAR(4)/20.4425
      PAR(5)=APAR(5)*1.8
      PAR(6)=APAR(6)*1.8
      PAR(7)=APAR(7)
      PAR(8)=APAR(8)*1.8+32.
      PAR(9)=APAR(9)/4.1867
      PAR(10)=APAR(10)/20.4425
      PAR(11)=APAR(11)*1.8
      PAR(12)=APAR(12)*1.8
      PAR(13)=APAR(13)
      PAR(14)=APAR(14)/1.055
      PAR(15)=APAR(15)
      PAR(16)=APAR(16)
      PAR(17)=APAR(17)
      PAR(18)=APAR(18)
      PAR(19)=APAR(19)/4.1867
      PAR(20)=APAR(20)
      PAR(21)=APAR(21)
      PAR(22)=APAR(22)
      PAR(23)=APAR(23)
      PAR(24)=APAR(24)
      PAR(25)=APAR(25)
      PAR(26)=APAR(26)
      PAR(27)=APAR(27)*1.8+32.
      PAR(28)=APAR(28)*1.8+32.
      PAR(29)=APAR(29)*1.8+32.
      PAR(30)=APAR(30)*1.8+32.
      PAR(31)=APAR(31)
      PAR(32)=APAR(32)
      GO TO 2
    4 CONTINUE
      DO 5 I=1,4
      XIN(I)=AXIN(I)
    5 CONTINUE
      GO TO (6,2),ICT
```

```
    6  CONTINUE
       DO 7 J=1,32
       PAR(J)=APAR(J)
    7  CONTINUE
    2  CONTINUE
       RETURN
       END
```

Subroutine OUTCON

Subroutine OUTCON converts the output values contained in the array OUT(I) from the internal dimensional system of BTU, Hour, ft, °F only if the user is in the system KJ, Hour, m, °C.

```
      SUBROUTINE OUTCON(OUT)
C     THIS ROUTINE CONVERTS THE VALUES OF THE ARRAY OUT FROM
C     THE UNITS USED INTERNALLY OF BTU, FT, HOUR, DEG F TO THE USERS
C     DIMENSIONAL SYSTEM. IT IS ONLY CALLED IF THE USER IS IN THE SYSTEM
C     KJ, M, HOUR, DEG C.
      DIMENSION OUT(15)
      OUT(1)=OUT(1)*1.055
      OUT(2)=OUT(2)*1.055
      OUT(3)=(OUT(3)-32.)/1.8
      OUT(4)=(OUT(4)-32.)/1.8
      OUT(5)=OUT(5)*1.055
      OUT(6)=OUT(6)*1.055
      OUT(7)=OUT(7)*1.055
      OUT(8)=OUT(8)*0.45359
      OUT(9)=OUT(9)*0.45359
      RETURN
      END
```

Subroutine SIZER

This routine sizes and designs the Rankine engine based on the user supplied design condition parameters. This information on the designed engine is used in the engine simulation. The arrays T, H, S, and V contain the state of each of the points in the engine. T(1) corresponds to the inlet temperature to the feed pump. T(2) is the inlet temperature to the boiler. T(3) is a point on the saturated liquid locus at the boiler working fluid pressure, while T(4) is the boiler working fluid outlet temperature. T(5) is the shut-off valve outlet temperature and expander inlet temperature. T(6) is the expander outlet temperature. T(7) is the saturated vapor temperature of the condenser working fluid at the pressure in the condenser.

Print control information (IPRT) is transferred in the labeled common block SNAP to subroutines RANKIN, BOILER, and CNDNSR.

```
      SUBROUTINE SIZER(PAR,OUT)
C     THE ARRAY PAR CARRIES THE USER SUPPLIED INPUT PARAMETERS IN THE
C     CORRECT DIMENSIONAL SYSTEM, OUT IS AN OUTPUT ARRAY CONTAINING
C     RANKINE ENGINE PARAMETERS WHICH ARE CALCULATED IN SIZER.
      DIMENSION PAR(40),OUT(20)
      DIMENSION T(8),H(8),S(8),V(8)
C     THE COMMON BLOCK SNAP BRINGS THE VALUE OF IPRT WHICH IS A
C     PRINT OPTION FOR THE RANKINE SIMULATION.
      COMMON /SNAP/ IPRT
      REAL MDS,MBOIL,MCON
      WDOUT=PAR(1)
      TBOIL=PAR(2)
      CPBOIL=PAR(3)
      ULL=PAR(4)
      DELTB=PAR(5)
      DELTOB=PAR(6)
      IBHX=PAR(7)
      TCON=PAR(8)
      CPCON=PAR(9)
      ULV=PAR(10)
      DELTC=PAR(11)
      DELTOC=PAR(12)
      ICHX=PAR(13)
      REVE=PAR(15)
      ENE=PAR(16)
      REVC=PAR(17)
      ENC=PAR(18)
      CPSL=PAR(19)
      PLOSS=PAR(20)
      IPRT=PAR(24)
C     BEGIN SIZING PROCEDURE FOR THE BOILER.
      THO=TBOIL-DELTB
      T(4)=THO-DELTOB
      Q=1.0
      ITYPE=15
      CALL FREON(T(4),PMAX,H(4),S(4),Q,V(4),ITYPE)
C     BEGIN SIZING PROCEDURE FOR CONDENSER.
      TCO=TCON+DELTC
      T(1)=TCO+DELTOC
      Q=0.0
      ITYPE=15
      CALL FREON(T(1),PMIN,H(1),S(1),Q,V(1),ITYPE)
      Q=0.0
      ITYPE=25
      CALL FREON(T(3),PMAX,H(3),S(3),Q,V(3),ITYPE)
C     BEGIN THE SIZING PROCEDURE FOR THE PUMP-COMPRESSOR
      WC=-V(1)*(PMAX-PMIN)*144./(778.16*ENC)
      H(2)=H(1)-WC
C     FINAL SIZING FOR THE THROTTLE
      PT=PMAX-PLOSS*(PMAX-PMIN)
      DPD=PMAX-PT
      H(5)=H(4)
```

```
      ITYPE=23
      CALL FREON(T(5),PT,H(5),S(5),Q,V(5),ITYPE)
      S6S=S(5)
      ITYPE=24
      CALL FREON(T6S,PMIN,H6S,S6S,Q,V6S,ITYPE)
      WES=H(5)-H6S
      WED=WES*ENE
      H(6)=H(5)-WED
      ITYPE=23
      CALL FREON(T(6),PMIN,H(6),S(6),Q,V(6),ITYPE)
C     DETERMINATION OF SYSTEM DESIGN MASS FLOW RATE
      MDS=WDOUT/WED
C   DETERMINATION OF THE DESIGN PUMP POWER
      WPD=MDS*WC
C     FINAL SIZING FOR THE EXPANDER
      ANVA=FRP(PT,PMIN)
      VOLE=MDS*V(5)*(1./ANVA)/(REVE*60.)
C     FINAL SIZING FOR THE BOILER FEED PUMP.
C   PUMP VOLUMETRIC EFFICIENCY IS ASSUMED TO BE 90%.
      ENVCP=0.90
      VOLC=MDS*V(1)/(ENVCP*REVC*60.)
C     FINAL SIZING FOR BOILER
      QA=MDS*(H(4)-H(2))
      MBOIL=QA/(CPBOIL*10.)
      CCLS=MDS*CPSL
      CBOIL=MBOIL*CPBOIL
C    THIS DETERMINES THE MAXIMUM CAPACITY RATE FLUID IN THE
C    LIQUID TO LIQUID SECTION OF THE BOILER.
  102 IF(CCLS.GT.CBOIL) GO TO 103
      CMIN1=CCLS
      CMAX1=CBOIL
      GO TO 104
  103 CMIN1=CBOIL
      CMAX1=CCLS
  104 CR1=CMIN1/CMAX1
C    THIS DETERMINES THE DESIGN HEAT TRANSFER COEFICIENTS IN
C    THE BOILER.
      HRB=ULL*2.0
      HWB=HRB
      UTB=1./(1./HWB+1./300.)
      Q1=(H(3)-H(2))*MDS
      T(2)=T(3)-Q1/CCLS
      Q2=(H(4)-H(3))*MDS
      QAB=Q1+Q2
      GO TO (200,210),IBHX
  200 CONTINUE
C   SIZING FOR THE PARALLEL FLOW BOILER
      E1=Q1/(CMIN1*(TBOIL-T(2)))
      A1=((-CMIN1/ULL)*ALOG(1.-E1*(1.+CR1)))/(1.+CR1)
      TH2=TBOIL-Q1/CBOIL
      E2=Q2/(CBOIL*(TH2-T(3)))
```

```
      A2=-((CBOIL/UTB)*ALOG(1.-E2))
      GO TO 220
  210 CONTINUE
C  THIS IS THE SIZING FOR THE COUNTERFLOW BOILER
      E2=Q2/(CBOIL*(TBOIL-T(3)))
      A2=-(CBOIL/UTB)*ALOG(1.-E2)
      TH2=TBOIL-Q2/CBOIL
      E1=Q1/(CMIN1*(TH2-T(2)))
      A1=(CMIN1/ULL)/(1.-CR1)*ALOG((CR1*E1-1.)/(E1-1.))
  220 CONTINUE
      ATB=A1+A2
C   THE FINAL SIZING FOR THE CONDENSER.
      QR=MDS*(H(1)-H(6))
      MCON=-QR/(CPCON*10.)
C   THIS DETERMINES THE DESIGN HEAT TRANSFER COEFFICIENTS IN THE
C   THE CONDENSER.
      HRC=ULV*2.0
      HWC=HRC
      UTC=1./(1./HWC+1./300.)
C   THIS DETERMINES THE WORKING FLUID SUPERHEAT SPECIFIC HEAT
C   IN THE CONDENSER.
      ITYPE=25
      Q=1.0
      CALL FREON(T(7),PMIN,H(7),S(7),Q,V(7),ITYPE)
      IF(T(6)-T(7)-0.01) 108,108,107
  107 CPCVC=(H(6)-H(7))/(T(6)-T(7))
  109 CCVCS=MDS*CPCVC
      CCON=MCON*CPCON
C   THIS DETERMINES THE MAXIMUM CAPCITY RATE FLUID IN THE
C   CONDENSER VAPOR TO LIQUID SECTION.
      IF(CCVCS.GT.CCON) GO TO 105
      CCMIN1=CCVCS
      CCMAX1=CCON
      GO TO 106
  105 CCMIN1=CCON
      CCMAX1=CCVCS
  106 CCR1=CCMIN1/CCMAX1
      QC1=(H(7)-H(6))*MDS
      QC2=(H(1)-H(7))*MDS
      GO TO (230,240),ICHX
  230 CONTINUE
C  SIZING FOR THE PARALLEL FLOW CONDENSER
      EC1=-QC1/(CCMIN1*(T(6)-TCON))
      AC1=((-CCMIN1/ULV)*ALOG(1.-EC1*(1.+CCR1)))/(1.+CCR1)
      TC2=TCON-QC1/CCON
      EC2=-QC2/(CCON*(T(7)-TC2))
      AC2=-((CCON/UTC)*ALOG(1.-EC2))
      GO TO 130
  240 CONTINUE
C  SIZING FOR THE COUNTERFLOW CONDENSER
      EC2=-QC2/(CCON*(T(7)-TCON))
```

```
      AC2=-(CCON/UTC)*ALOG(1.-EC2)
      TC2=TCON-QC2/CCON
      EC1=-QC1/(CCMIN1*(T(6)-TC2))
      AC1=(CCMIN1/ULV)/(1.-CCR1)*ALOG((CCR1*EC1-1.)/(EC1-1.))
      GO TO 130
C     IN THE FOLLOWING SECTION THE CONDENSER IS BEING DESIGNED
C     WITH THE OUTLET STATE OF THE EXPANDER IN 2-PHASE.
  108 CPCVC=0.0
      EC1=0.0
      AC1=0.0
      QC1=0.0
      CCON=MCON*CPCON
      TC2=TCON
      QC2=(H(1)-H(6))*MDS
      EC2=-QC2/(CCON*(T(6)-TC2))
      AC2=-((CCON/UTC)*ALOG(1.-EC2))
  130 ATC=AC1+AC2
      QRC=QC1+QC2
      GO TO (140,140,140,140,150),IPRT
  140 CONTINUE
      WRITE(-,500)
  500 FORMAT(1X,' THIS IS THE RANKINE CYCLE ENGINE AS SIZED ',/)
      WRITE(-,501)
  501 FORMAT(1X,' THE UNITS ARE DEGREES F, BTU PER LBM, AND PSIA ')
      WRITE(-,601)
  601 FORMAT(1X,' T H P AT THE COMPRESSOR INLET ')
      WRITE(-,-)T(1),H(1),PMIN
      WRITE(-,602)
  602 FORMAT(1X,' T H P AT THE BOILER INLET ')
      WRITE(-,-)T(2),H(2),PMAX
      WRITE(-,603)
  603 FORMAT(1X,' T H P AT THE SHUT OFF VALVE INLET ')
      WRITE(-,-)T(4),H(4),PMAX
      WRITE(-,6031)
 6031 FORMAT(1X,' T H P AT THE EXPANDER INLET ')
      WRITE(-,-)T(5),H(5),PT
      WRITE(-,604)
  604 FORMAT(1X,' T H P AT THE CONDENSOR INLET ')
      WRITE(-,-)T(6),H(6),PMIN
      WRITE(-,502)
  502 FORMAT(1X,/,1X,' PUMP DESIGN ')
      WRITE(-,503) MDS
  503 FORMAT(1X,' THE DESIGN MASS FLOW RATE ',F9.3,' LBM PER HOUR ')
      WRITE(-,504) VOLC
  504 FORMAT(1X,' THE EFFECTIVE DISPLACEMENT ',F10.8,' FEET**3')
      WRITE(-,505) ENC
  505 FORMAT(1X,' THE ADIABATIC EFFICIENCY ',F6.3)
      WRITE(-,506) REVC
  506 FORMAT(1X,' THE DESIGN SPEED ',F9.3,' RPM')
      WRITE(-,507) WPD
  507 FORMAT(1X,' THE DESIGN WORK ',F9.3,' BTU PER HOUR ')
      WRITE(-,508)
  508 FORMAT(1X,/,1X,' BOILER DESIGN ')
      WRITE(-,509) TBOIL,THO
```

```
509 FORMAT(1X,' THE HOT WATERSIDE INLET ',F9.3,' AND OUTLET ',F9.3,
   1' TEMPERATURE DEGREES F ')
    WRITE(-,510) MBOIL
510 FORMAT(1X,' THE HOT WATERSIDE MASS FLOW RATE ',F10.4,' LBM PER
   1HOUR ')
    WRITE(-,511) ATB
511 FORMAT(1X,' THE HEAT TRANSFER AREA ',F10.4,' FEET **2')
    WRITE(-,512) ULL
512 FORMAT(1X,' THE DESIGN LIQUID-TO-LIQUID HEAT TRANSFER COEFFICIENT
   1',,F10.4,' BTU PER HOUR FEET**2')
    WRITE(-,513) UTB
513 FORMAT(1X,' THE DESIGN LIQUID-TO-2-PHASE HEAT TRANSFER COEFFICIEN
   1',F10.4,' BTU PER HOUR FEET**2 ')
    WRITE(-,514) QAB
514 FORMAT(1X,' THE DESIGN HEAT TRANSFER RATE ',F12.4,' BTU PER HOUR
   1')
    WRITE(-,515)
515 FORMAT(1X,/,1X,' THE SHUT OFF VALVE DESIGN ')
    WRITE(-,516) DPD
516 FORMAT(1X,' THE DESIGN PRESSURE DROP ',F10.4,' PSIA ')
    WRITE(-,517)
517 FORMAT(1X,/,1X,' THE EXPANDER DESIGN ')
    WRITE(-,518) VOLE
518 FORMAT(1X,' THE EFFECTIVE DISPLACEMENT ',F10.8,' FEET**3 ')
    WRITE(-,519) FNE
519 FORMAT(1X,' THE ADIABATIC EFFICIENCY ',F6.3)
    WRITE(-,520) ANVA
520 FORMAT(1X,' THE DESIGN VOLUMETRIC EFFICIENCY ',F6.3)
    WRITE(-,521) REVE
521 FORMAT(1X,' THE DESIGN SPEED ',F9.3,' RPM ')
    WRITE(-,522) WDOUT
522 FORMAT(1X,' THE DESIGN WORK ',F9.3,' BTU PER HOUR ')
    WRITE(-,523)
523 FORMAT(1X,/,1X,' THE CONDENSER DESIGN ')
    WRITE(-,524) TCON,TCO
524 FORMAT(1X,' THE COLD WATERSIDE INLET ',F9.3,' AND OUTLET ',F9.3,
   1' TEMPERATURE DEGREES F ')
    WRITE(-,525) MCON
525 FORMAT(1X,' THE COLD WATERSIDE MASS FLOW RATE ',F10.4,' LBM PER
   1HOUR ')
    WRITE(-,526) ATC
526 FORMAT(1X,' THE HEAT TRANSFER AREA ',F10.4,' FEET**2 ')
    WRITE(-,527) ULV
527 FORMAT(1X,' THE DESIGN LIQUID-TO-VAPOR HEAT TRANSFER COEFFICIENT'
   1,F10.4,' BTU PER HOUR FEET**2 ')
    WRITE(-,528) UTC
528 FORMAT(1X,' THE DESIGN LIQUID-TO-2-PHASE HEAT TRANSFER COEFFIENT'
   1,F10.4,' BTU PER HOUR FEET**2 ')
    WRITE(-,529) QRC
529 FORMAT(1X,' THE DESIGN HEAT TRANSFER RATE ',F12.4,' BTU PER
   1HOUR ')
```

```
150 CONTINUE
    OUT(1)=PMAX
    OUT(2)=ATB
    OUT(3)=MDS
    OUT(4)=HWB
    OUT(5)=HRB
    OUT(6)=MBOIL
    OUT(7)=PMIN
    OUT(8)=ATC
    OUT(9)=HWC
    OUT(10)=HRC
    OUT(11)=MCON
    OUT(12)=VOLE
    OUT(13)=VOLC
    OUT(14)=WPD
    RETURN
    END
```

Subroutine RANKIN

Subroutine RANKIN solves the thermodynamic and heat transfer equations used to fix each operating state for the Rankine engine. This routine bases each state on the user supplied input parameters and the design engine parameters defined by subroutine SIZER and contained in the array DPAR(I). The equations are solved by successive approximation using a three-dimensional Newton's method. Newton's method is obtained numerically in subroutine RTFND. Subroutines BOILER and CNDNSR contain the boiler and condenser models, respectively. The boiler feed pump model is contained in COMP. No specific subroutine describes the expander, but the expander volumetric efficiency correlation is referenced in function FRP. Refrigerant property data is obtained by calling subroutine FREON. The labeled common block SNAP carries the print control variable IPRT to and from subroutines SIZER, BOILER, and CNDNSR.

Subroutine RANKIN is referenced by TYPE23 for simulations when only a few operating states need be evaluated. For longer simulations RANKIN is used by subroutine BLKBOX in the production of a performance map.

```
      SUBROUTINE RANKIN(XIN,OUT,PAR,DPAR)
C   THIS ROUTINE CALCULATES THE PERFORMANCE OF THE RANKINE ENGINE
C   GIVEN THE BOILER INLET WATERSIDE TEMPERATURE XIN(1), THE CONDENSER
C   INLET WATERSIDE TEMPERATURE XIN(2), AND THE LOAD POWER
C   REQUIREMENTS, XIN(3). XIN CONTAINS INPUTS, OUT CONTAINS OUTPUTS AS
C   CALCULATED INTERNALLY, PAR CONTAINS USER INPUT PARAMETERS, DPAR
C   CONTAINS CALCULATED PARAMETERS FROM SIZER, AT,AH,AS,AV CONTAIN
C   PROPERTY VALUES FOR EACH OF THE 7 STATE POINTS IN THE RANKINE
C   ENGINE, BPAR CONTAINS BOILER PARAMETERS, CPAR CONTAINS CONDENSER
C   PARAMETERS, AND CMPAR CONTAINS FEED PUMP PARAMETERS.
      DIMENSION XIN(10),OUT(15),PAR(40)
      DIMENSION AT(7),AH(7),AS(7),AV(7)
      DIMENSION BPAR(10),CPAR(10),DPAR(20),CMPAR(10)
C   IPRT IS A PRINT CONTROL VARIABLE.
      COMMON /SNAP/ IPRT
      REAL MDS,MCON,MBOIL,MDSD
      DATA ICT/1/
      GO TO (1,2),ICT
C   INITIALIZE PARAMETERS ONCE ONLY.
    1 CONTINUE
      WDOUT=PAR(1)
      CPBOIL=PAR(3)
      DELTB=PAR(5)
      DELTOB=PAR(6)
      IBHX=PAR(7)
      CPCON=PAR(9)
      DELTC=PAR(11)
      DELTOC=PAR(12)
      ICHX=PAR(13)
      WFRIC=PAR(14)
      REVE=PAR(15)
      ENE=PAR(16)
      REVCD=PAR(17)
      ENC=PAR(18)
      CPSL=PAR(19)
      PLOSS=PAR(20)
      ERR=PAR(21)
      ERD=ERR*0.01
      ERV=PAR(22)
      ERF=ERV*0.001
      IOSCL=PAR(23)
      IPRT=PAR(24)
      IMAP=PAR(25)
C   ERR IS THE ERROR OF CLOSURE  LIMIT FOR DELHB IN THE BOILER AND DELHC
C   IN THE CONDENSER. ERV IS THE CLOSURE LIMIT FOR DELR IN THE
C   EXPANDER.
      PMAX=DPAR(1)
      ATB=DPAR(2)
      MDSD=DPAR(3)
      HWB=DPAR(4)
      HRB=DPAR(5)
      PMIN=DPAR(7)
      ATC=DPAR(8)
      HWC=DPAR(9)
```

```
      HRC=DPAR(10)
      VOLE=DPAR(12)
      VOLC=DPAR(13)
      WPD=DPAR(14)
C     THIS IS THE CALCULATION OF THE DESIGN SHUT-OFF VALVE PRESSURE
C     DROP AND THE SHUT-OFF VALVE DESIGN INLET STATE.
      DPD=PLOSS*(PMAX-PMIN)
      Q=1.0
      ITYPE=25
      CALL FREON(TO,PMAX,HD,SD,Q,VD,ITYPE)
      ICT=2
        REVC=REVCD
    2 CONTINUE
C     THIS TEST IS FOR STABILITY OF THE SOLUTION METHOD.
        IF(ENTH.LE.0.02) REVC=0.5*REVCD
      TBOIL=XIN(1)
      TCON=XIN(2)
      WOUT=XIN(3)
      MBOIL=DPAR(6)
      MCON=DPAR(11)
      DO 6 J=1,7
      AT(J)=0.0
      AH(J)=0.0
      AS(J)=0.0
      AV(J)=0.0
    6 CONTINUE
      GO TO (3,4,4,4),IMAP
C     THE INPUT VALUE OF WOUT IS IGNORED UNLESS IMAP=1.
C       FOR IMAP=2,3,4 WOUT IS SET EQUAL TO WDOUT.
    4 CONTINUE
      WOUT=WDOUT
    3 CONTINUE
C     THIS TEST ARBITRARILY TURNS OFF THE MACHINE IF THE INPUT
C     WATER STREAM TEMPERATURES DIFFER BY 30 DEG F OR LESS.
        IF((TBOIL-TCON).LE.30.0) GO TO 108
C     THIS INITIALIZES THE INPUTS TO THE BOILER WHICH ARE CONSTANT
      BPAR(2)=MBOIL
      BPAR(4)=CPBOIL
      BPAR(5)=CPSL
      BPAR(6)=HWB
      BPAR(7)=HRB
      BPAR(8)=ATB
      BPAR(9)=MDSD
C     THIS INITIALIZES THE INPUTS TO THE CONDENSER WHICH ARE CONSTANT
      CPAR(3)=MCON
      CPAR(4)=CPCON
      CPAR(5)=HRC
      CPAR(6)=HWC
      CPAR(7)=ATC
      CPAR(8)=MDSD
      AXW=0.0
      PB=0.0
      DELHOB=0.0
      PC=0.0
```

```
      DELHOC=0.0
      REVCO=0.0
      DELRO=0.0
C   THIS INITIALIZES THE COUNTERS.
      NCT=1
      JCT=1
      IVCT=1
      ISYS=0
      ISPD=0
      ICNTB=1
      ICNTR=1
      ICNTC=1
C   ICRSH IS THE MAXIMUM NUMBER OF TIMES THE EXPANDER MAY
C   ATTEMPT TO OVERSPEED OR UNDERSPEED FOR EACH OPERATING STATE.
      ICRSH=IOSCL/4
C   THIS IS THE FIRST GUESS FOR THE BOILER OUTLET TEMP FOR THE
C   WORKING FLUID.
      AT(4)=TBOIL-DELTB-DELTOB
      Q=1.0
      ITYPE=15
      CALL FREON(AT(4),PMAX,AH(4),AS(4),Q,AV(4),ITYPE)
C   THIS IS THE FIRST GUESS FOR THE CONDENSER OUTLET TEMP FOR THE
C   WORKING FLUID.
      AT(1)=TCON+DELTC+DELTOC
      ITYPE=15
      Q=0.0
      CALL FREON(AT(1),PMIN,AH(1),AS(1),Q,AV(1),ITYPE)
   21 CONTINUE
C   THIS INITIALIZES THE INPUTS INTO THE BOILER FEED PUMP.
      ICOMP=1
      CMPAR(1)=PMAX
      CMPAR(2)=PMIN
      CMPAR(3)=REVC
      CMPAR(4)=VOLC
      CMPAR(5)=ENC
      CMPAR(6)=CPSL
      CALL COMP(ICOMP,CMPAR,AH(1),AH(2),AT(2),MDS,AWC,AWCT)
C     THIS INITIALIZES THE INPUT INTO THE BOILER
      BPAR(1)=PMAX
      BPAR(3)=MDS
      CALL BOILER(IBHX,BPAR,AT(2),AH(2),AT(4),AH(4),TBOIL,TBOILO,QAB,DE
     1HB)
C   TEST FOR CLOSURE ON THE BOILER OUTLET STATE ENTHALPY.
      IF(ABS(DELHB).LE.ERR) GO TO 30
      STEP=2.
      CALL RTFND(PMAX,PB,DELHB,DELHOB,DERIVB,NCT,STEP,ERD)
      Q=1.0
      ITYPE=25
      CALL FREON(AT(4),PMAX,AH(4),AS(4),Q,AV(4),ITYPE)
      IF(AT(4).GE.TBOIL) GO TO 23
C ****************************************************************
C   IF INSTABILITY IS DEVELOPED WHEN USING A COUNTERFLOW BOILER
C   CHANGE THE NUMBER OF ARGUMENTS IN THE COMPUTED GO TO TO READ:
C     GO TO (21,21,21,21,21,21,21,21,21,21,21,21,21,21,30),NCT
```

```
C ***************************************************************
      GO TO (21,21,21,30),NCT
   23 CONTINUE
C     THE GUESS FOR THE BOILER WORKING FLUID TEMPERATURE WAS TOO HIGH.
      PMAX=PMAX-5.
C ***************************************************************
C     REMOVE THE FOLLOWING CARD IF INSTABILITY DEVELOPS WHEN USING A
C     COUNTERFLOW BOILER .
      NCT=2
C ***************************************************************
      ICNTB=ICNTB+1
      GO TO (21,21,21,25),ICNTB
   25 CONTINUE
      WRITE(-,24)ICNTB,PMAX,PR
   24 FORMAT(1X,' FAILURE TO CLOSE ON PMAX ',I4,2F10.4)
   30 CONTINUE
      NCT=1
      ICNTB=1
C     THE SHUT OFF VALVE IS A CONSTANT ENTHALPY DEVICE.
      AH(5)=AH(4)
   41 CONTINUE
      ICOMP=2
      CMPAR(1)=PMAX
      CMPAR(2)=PMIN
      CMPAR(3)=REVC
      CALL COMP(ICOMP,CMPAR,AH(1),AH(2),AT(2),MDS,AWC,AWCT)
C     DETERMINATION OF THE SHUT OFF VALVE PRESSURE DROP.
      OPD=DPD*(AV(4)/VD)*(MDS/MDSD)**2
      PT=PMAX-OPD
      ITYPE=23
      CALL FREON(AT(5),PT,AH(5),AS(5),Q,AV(5),ITYPE)
      ENVA=FRP(PT,PMIN)
      REVET=MDS*AV(5)/(VOLE*ENVA*60.)
      DELR=REVET-REVE
C     TEST FOR CLOSURE ON THE EXPANDER SHAFT SPEED.
      IF(ABS(DELR).LE.ERV) GO TO 50
      STEP=4.
      CALL RTFND(REVC,REVCO,DELR,DELRO,DERIVV,JCT,STEP,ERF)
C     THE FOLLOWING TEST IS AN ARBITRARY ONE FOR THE MINIMUM
C     ALLOWABLE EXPANDER SPEED.
      IF(REVC.LT.100.) GO TO 42
      GO TO (41,41,41,50),JCT
   42 CONTINUE
      REVC=0.15*REVCD
      JCT=2
      ICNTR=ICNTR+1
      GO TO (41,41,41,46),ICNTR
   46 CONTINUE
      ISPD=ISPD+1
      WRITE(-,47) ICNTR,REVC,REVCO
   47 FORMAT(1X,' FAILURE TO CLOSE ON FEED PUMP SPEED ',I4,2F10.4)
   50 CONTINUE
      JCT=1
      ICNTR=1
```

```
   51  CONTINUE
       ITYPE=24
       CALL FREON(T6S,PMIN,H6S,AS(5),Q,V6S,ITYPE)
       AH(6)=AH(5)-(AH(5)-H6S)*ENE
C     THIS IS THE CALCULATION OF THE POWER OUTPUT OF THE EXPANDER.
       WACT=(AH(5)-AH(6))*MDS
C     THIS IS THE DETERMINATION OF THE AMOUNT OF AUXILIARY NEEDED
C     TO MEET THE LOAD IF WOUT<WACT OR THE AMOUNT OF ENERGY TO BE
C     GENERATED IF WOUT>WACT.
       AXW=WOUT-WACT
       ITYPE=23
       CALL FREON(AT(6),PMIN,AH(6),AS(6),Q,AV(6),ITYPE)
       CPAR(1)=PMIN
       CPAR(2)=MDS
       CALL CNDNSR(ICHX,CPAR,AT(6),AH(6),AT(1),AH(1),TCON,TCONO,QRC,DELH
      1)
C     TEST FOR CLOSURE ON THE CONDENSER OUTLET STATE ENTHALPY.
       IF(ABS(DELHC).LT.ERR) GO TO 55
       STEP=2.
       CALL RTFND(PMIN,PC,DELHC,DELHOC,DERIVC,IVCT,STEP,ERD)
       Q=1.0
       ITYPE=25
       CALL FREON(AT(7),PMIN,AH(7),AS(7),Q,AV(7),ITYPE)
       IF(AT(7).LE.TCON) GO TO 52
C ****************************************************************
C     IF INSTABILITY IS DEVELOPED WHEN USING A COUNTERFLOW CONDENSER
C     CHANGE THE NUMBER OF ARGUMENTS IN THE COMPUTED GO TO TO READ:
C       GO TO (51,51,51,51,51,51,51,51,51,51,51,51,51,55),IVCT
C ****************************************************************
       GO TO (51,51,51,55),IVCT
   52  CONTINUE
C     THE GUESS FOR THE CONDENSER WORKING FLUID TEMPERATURE WAS TOO LOW.
C ****************************************************************
C     REMOVE THE FOLLOWING CARD IF INSTABILITY DEVELOPS WHEN USING A
C     COUNTERFLOW CONDENSER .
       IVCT=2
C ****************************************************************
       PMIN=PMIN+5.
       ICNTC=ICNTC+1
       GO TO (51,51,51,53),ICNTC
   53  CONTINUE
       WRITE(-,54)ICNTC,PMIN,PC
   54  FORMAT(1X,' FAILURE TO CLOSE ON PMIN ',I4,2F10.4)
   55  CONTINUE
       IVCT=1
       ICNTC=1
       GO TO (56,57,57,57,57),IPRT
   56  CONTINUE
C     THESE ARE FOR BEBUGGING ONLY.
       WRITE(-,-) PMAX,PB,DELHB,DELHOB,DERIVB
       WRITE(-,-) MDS
       WRITE(-,-) PMIN,PC,DELHC,DELHOC,DERIVC
       WRITE(-,-) TBOILO,TCONO,PT
       WRITE(-,-) REVC,REVCO,DELR,DELRO,DERIVV
```

```
  57  CONTINUE
      IF(ABS(DELHB).GT.ERR) GO TO 70
      IF(ABS(DELHC).GT.ERR) GO TO 70
      IF(ABS(DELR).LE. ERV) GO TO 100
  70  CONTINUE
C   THIS TEST DETERMINES IF THE OPERATING STATE IS SUCH THAT THE
C   EXPANDER CANNOT MAINTAIN CONSTANT SPEED OPERATION AT THE
C   DESIRED OPERATING SPEED.
      IF(ISPD.GE.ICRSH) GO TO 108
C   ISYS IS A COUNTER WHICH DETERMINES THE NUMBER OF TRYS AT FINDING
C   THE CURRENT OPERATING STATE WHICH HAVE BEEN MADE. UP TO IOSCL
C   NUMBER OF TRYS ARE ALLOWED.
      ISYS=ISYS+1
      IF(ISYS.LE.IOSCL) GO TO 21
      WRITE(-,60)
  60  FORMAT(1X,' FAILURE TO CLOSE ON SYSTEM STATES ')
      WRITE(-,61)
  61  FORMAT(1X,' ISYS DELHB DELHOB DELHC DELHOC DELR DELRO ')
      WRITE(-,-) ISYS,DELHB,DELHOB,DELHC,DELHOC,DELR,DELRO
 100  CONTINUE
C   CRVMX IS AN ARBITRARY MAXIMUM SPEED THAT THE BOILER FEED PUMP
C   CAN ATTAIN. CRVMN IS AN ARBITRARY MINIMUM SPEED THAT THE
C   BOILER FEED PUMP CAN ATTAIN.
      CRVMX=1.25*REVCD
      CRVMN=0.25*REVCD
      IF (REVC.GT.CRVMX.OR.REVC.LT.CRVMN) GO TO 108
C   WFRIC IS THE INPUT MINIMUM EXPANDER POWER OUTPUT
C   ALLOWED BEFORE THE MACHINE WOULD STOP DUE TO INTERNAL FRICTION.
      IF(WACT.GT.WFRIC) GO TO 110
 108  CONTINUE
C   THE MACHINE HAS BEEN SHUT OFF FOR THIS SET OF INPUTS .
      QAB=0.0
      QRC=0.0
      MBOIL=0.0
      MCON=0.0
      TBOILO=TBOIL
      TCONO=TCON
      AWCT=0.0
      WACT=0.0
      ENTH=0.0
      AXW=WOUT
      GO TO 109
 110  CONTINUE
      ENTH=(WACT+AWCT)/QAB
 109  CONTINUE
      GO TO (115,115,115,116,116),IPRT
 115  CONTINUE
      WRITE(-,101)ISYS
 101  FORMAT(2X,I4,2X,' ITERATIONS WERE REQUIRED ')
 116  CONTINUE
C   THE MACHINE CAN RUN FOR THESE INPUTS AND IT HAS FOUND A SOLUTION.
      OUT(1)=QAB
      OUT(2)=QRC
      OUT(3)=TBOILO
```

```
      OUT(4)=TCONO
      OUT(5)=AWCT
      OUT(6)=WACT
      OUT(7)=AXW
      OUT(8)=MBOIL
      OUT(9)=MCON
      OUT(10)=WACT/WDOUT
      OUT(11)=AWCT/WPD
      OUT(12)=ENTH
      GO TO (120,120,120,125,125),IPRT
  120 CONTINUE
      WRITE(-,-) DELHB,DELHOB,DERIVB,PMAX,DELHC,DELHOC,DERIVC,PMIN
      WRITE(-,-) DELR,DELRO,DERIVV
      WRITE(-,-)(OUT(I),I=1,12)
  125 CONTINUE
      GO TO (121,121,126,126,126),IPRT
  121 CONTINUE
      WRITE(-,500) TBOIL,TBOILO
  500 FORMAT(1X,' HOT WATERSIDE INLET ',F9.3,' AND OUTLET ',F9.3,
     1,1X,'DEGREES F')
      WRITE(-,501) TCON,TCONO
  501 FORMAT(1X,' COLD WATERSIDE INLET ',F9.3,' AND OUTLET ',F9.3,
     1,1X,'DEGREES F')
      WRITE(-,5000)
 5000 FORMAT(1X,' THE UNITS ARE DEGREES F, BTU PER LBM, AND PSIA ')
      WRITE(-,610)
  610 FORMAT(1X,'T H P AT THE COMPRESSOR INLET ')
      WRITE(-,-) AT(1),AH(1),PMIN
      WRITE(-,611)
  611 FORMAT(1X,' T H P AT THE BOILER INLET ')
      WRITE(-,-) AT(2),AH(2),PMAX
      WRITE(-,612)
  612 FORMAT(1X,' T H P AT THE SHUT OFF VALVE INLET ')
      WRITE(-,-)AT(4),AH(4),PMAX
      WRITE(-,6121)
 6121 FORMAT(1X,' T H P AT THE EXPANDER INLET ')
      WRITE(-,-) AT(5),AH(5),PT
      WRITE(-,613)
  613 FORMAT(1X,' T H P AT THE CONDENSER INLET ')
      WRITE(-,-) AT(6),AH(6),PMIN
      WRITE(-,502) MDS
  502 FORMAT(1X,' SYSTEM MASS FLOW RATE ',F9.3,' LBM PER HOUR ')
      WRITE(-,503) REVC
  503 FORMAT(1X,' PUMP SPEED ',F9.3,' RPM ')
      WRITE(-,504) AWCT
  504 FORMAT(1X,' PUMP WORK ',F9.3,' BTU PER HOUR ')
      WRITE(-,505) ENVA
  505 FORMAT(1X,' EXPANDER VOLUMETRIC EFFICIENCY ',F6.3)
      WRITE(-,506) REVE
  506 FORMAT(1X,' EXPANDER SPFED ',F9.3,' RPM ')
      WRITE(-,507) WACT
  507 FORMAT(1X,' EXPANDER WORK ',F9.3,' BTU PER HOUR ')
  126 CONTINUE
      RETURN
      END
```

Subroutines BOILER and CNDNSR

These routines model the boiler and condenser used in the Rankine engine as described in Section 2.2 for parallel and counterflow configurations. Subroutines BOILER and CNDNSR use the effectiveness correlation described by Kays and London (7). Each routine obtains working fluid property data from subroutine FREON. In subroutine BOILER, the hot-side fluid is denoted by the subscript H while the working fluid (the cold-side fluid) is denoted by the subscript C. Subroutine CNDNSR denotes the working fluid (the hot-side fluid) with the subscript H while the cold-side fluid is denoted by the subscript C.

These routines are referenced only by RANKIN. The common block SNAP transfers the value of the print control variable IPRT into BOILER and CNDNSR.

The boiling and condensing heat transfer coefficients are assumed to be constants with the value 300 BTU/Hour ft^2 °F.

```
      SUBROUTINE BOILER(ICTL1,PAR,TC1,HC1,TCO,HCO,THI,THO,QA,DELHO)
C     THIS ROUTINE SIMULATES A PARALLEL OR COUNTERFLOW BOILER USING
C     THE EFFECTIVENESS RELATIONS DEVELOPED BY KAYS AND LONDON.
C     THE ARRAY PAR CONTAINS THE INPUT PARAMETERS AND VARIABLES.
      DIMENSION PAR(10)
C     IPRT IS A PRINT CONTROL VARIABLE.
      COMMON /SNAP/ IPRT
      REAL MDOTH,MDOTC,MDCD
      PMAX=PAR(1)
      MDOTH=PAR(2)
      MDOTC=PAR(3)
      CPH=PAR(4)
      CPC=PAR(5)
      HWB=PAR(6)
      HCD=PAR(7)
      AT=PAR(8)
      MDCD=PAR(9)
      CH=MDOTH*CPH
      CC1=MDOTC*CPC
      Q=1.0
      ITYPE=25
      CALL FREON(TC3,PMAX,HC3,SC3,Q,V3,ITYPE)
C     THIS SECTION DETERMINES THE MAXIMUM AND MINIMUM CAPACITY FLUID
C     IN THE LIQUID TO LIQUID SECTION OF THE BOILER.
      IF(CC1.LE.CH) GO TO 3
      CMIN1=CH
      CMAX1=CC1
      GO TO 6
    3 CMIN1=CC1
      CMAX1=CH
    6 CR1=CMIN1/CMAX1
      Q=0.0
      ITYPE=25
      CALL FREON(TC2,PMAX,HC2,SC2,Q,V2,ITYPE)
C     THIS SECTION DETERMINES THE HEAT TRANSFER COEFFICIENTS.
      HF=HCD*(MDOTC/MDCD)**0.52
      ULL=1./(1./HF+1./HWB)
      U2P=1./(1./HWB+1./300.)
      GO TO (19,200),ICTL1
  200 CONTINUE
C     THIS IS THE COUNTERFLOW SECTION OF THE BOILER
      Q2=MDOTC*(HC3-HC2)
      E2=Q2/(CH*(THI-TC3))
      IF(E2.GT.1.0) GO TO 21
      TH2=THI-Q2/CH
      A2=-(CH/U2P)*ALOG(1.-E2)
      IF(A2.LE.AT) GO TO 22
   21 CONTINUE
      A2=AT
      E2=1.-EXP(-U2P*A2/CH)
      TCO=TC3
      THO=THI-E2*(THI-TC2)
```

```
      Q2=CH*(THI-THO)
      QA=Q2
      DELHO=(HC3-HCI)-QA/MDOTC
      HCO=HC3-DELHO
      Q1=0.
      E1=0.
      A1=0.
      GO TO 45
   22 CONTINUE
      A1=AT-A2
      E1=(1.-EXP((-ULL*A1/CMIN1)*(1.-CR1)))/(1.-CR1*EXP((-ULL*A1/CMIN1)*
     1(1.-CR1)))
      THO=TH2-E1*(CMIN1/CH)*(TH2-TCI)
      Q1=CH*(TH2-THO)
      TCO=TC3
      QA=Q1+Q2
      DELHO=(HC3-HCI)-QA/MDOTC
      HCO=HC3-DELHO
      GO TO 45
   19 CONTINUE
C  THIS IS THE PARALLEL FLOW SECTION OF THE BOILER
      IF(HCI.GE.HC2) GO TO 7
      Q1=CC1*(TC2-TCI)
      TH2=THI-Q1/CH
      E1=Q1/(CMIN1*(THI-TCI))
      EP=E1*(1.+CR1)
      IF(EP.GE.1.0) GO TO 11
      A1=(-CMIN1/ULL)*ALOG(1.-E1*(1.+CR1))/(1.+CR1)
      HCI=HC2-Q1/MDOTC
      IF(A1.LE.AT) GO TO 10
   11 A1=AT
      E1=(1.-EXP(-ULL*(A1/CMIN1)*(1.+CR1)))/(1.+CR1)
      TCO=TCI+E1*(CMIN1/CC1)*(THI-TCI)
      THO=THI-E1*(CMIN1/CH)*(THI-TCI)
      QA=CC1*(TCO-TCI)
      HCO=HCI+QA/MDOTC
      DELHO=HC3-HCO
      RETURN
    7 A1=0.0
      E1=0.0
      Q1=0.0
      HC2=HCI
      TC2=TCI
      TH2=THI
   10 A2=AT-A1
      E2=1.-EXP(-U2P*A2/CH)
      TCO=TC2
      THO=TH2-E2*(TH2-TC2)
      Q2=CH*(TH2-THO)
      HCO=HC2+Q2/MDOTC
      QA=Q1+Q2
```

```
      DELHO=HC3-HCO
   45 CONTINUE
      GO TO (50,51,51,51,51),IPRT
   50 CONTINUE
      WRITE(-,-) PMAX,DELHO,HCO,HC2,HC3,TC2,TC3,HCI,TCI,THI,TH2,THO
      WRITE(-,615)
  615 FORMAT(1X,' QA  AT  Q1  Q2  E1  E2  A1  A2  FOR THE BOILER ')
      WRITE(-,-)QA,AT,Q1,Q2,E1,E2,A1,A2
   51 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE CNDNSR(ICTL2,PAR,THI,HHI,THO,HHO,TCI,TCO,QR,DELHO)
C     THIS ROUTINE SIMULATES A PARALLEL OR COUNTERFLOW CONDENSER USING
C     THE EFFECTIVENESS RELATIONS DEVELOPED BY KAYS AND LONDON.
C     THE ARRAY PAR CONTAINS THE INPUT PARAMETERS AND VARIABLES.
      DIMENSION PAR(10)
C     IPRT IS A PRINT CONTROL VARIABLE.
      COMMON /SNAP/ IPRT
      REAL MDOTH,MDOTC,MDHD
      PMIN=PAR(1)
      MDOTH=PAR(2)
      MDOTC=PAR(3)
      CPC=PAR(4)
      HHD=PAR(5)
      HWC=PAR(6)
      AT=PAR(7)
      MDHD=PAR(8)
      CC=MDOTC*CPC
      Q=1.0
      ITYPE=25
      CALL FREON(TH2,PMIN,HH2,SH2,Q,V2,ITYPE)
      Q=0.0
      ITYPE=25
      CALL FREON(TH3,PMIN,HH3,SH3,Q,V3,ITYPE)
C     THIS SECTION DETERMINES THE HEAT TRANSFER COEFFICIENTS.
      HF=HHD*(MDOTH/MDHD)**0.6
      ULV=1./(1./HWC+1./HF)
      U2P=1./(1./HWC+1./300.)
      IF(THI.GT.TH2) GO TO 5
      GO TO (31,21),ICTL2
    5 CONTINUE
C     THIS DETERMINES THE WORKING FLUID SPECIFIC HEAT IN SUPERHEAT.
      CPHI=(HHI-HH2)/(THI-TH2)
      CHI=MDOTH*CPHI
C     THIS SECTION DETERMINES THE MAXIMUM AND MINIMUM CAPACITY FLUID
C     IN THE LIQUID TO VAPOR SECTION OF THE CONDENSER.
      IF(CHI.LE.CC) GO TO 3
      CMIN1=CC
      CMAX1=CHI
      GO TO 4
    3 CMIN1=CHI
      CMAX1=CC
    4 CR1=CMIN1/CMAX1
      GO TO (19,200),ICTL2
  200 CONTINUE
C     THIS IS THE COUNTERFLOW SECTION OF THE CONDENSER
      Q1=MDOTH*(HH2-HHI)
      Q2=MDOTH*(HH3-HH2)
      E2=-Q2/(CC*(TH2-TCI))
      IF(E2.GT.1.0) GO TO 21
      A2=-(CC/U2P)*ALOG(1.0-E2)
      IF(A2.LE.AT) GO TO 22
   21 CONTINUE
C     THE INPUT STATE OF THE WORKING FLUID IS IN THE MIXED PHASE.
      A2=AT
```

```
      E2=1.D-EXP(-U2P*A2/CC)
      TCO=TCI+E2*(TH2-TCI)
      QR=CC*(TCI-TCO)
      THO=TH3
      DELHO=(HH3-HHI)-QR/MDOTH
      HHO=HH3-DELHO
      Q1=0.0
      A1=0.0
      E1=0.0
      GO TO 45
   22 CONTINUE
      A1=AT-A2
      ARG=EXP((-ULV*A1/CMIN1)*(1.-CR1))
      E1=(1.-ARG)/(1.-CR1*ARG)
      TC2=TCI-Q2/CC
      TCO=TC2+E1*(CMIN1/CC)*(THI-TC2)
      QR=CC*(TCI-TCO)
      THO=TH3
      DELHO=(HH3-HHI)-QR/MDOTH
      HHO=HH3-DELHO
      GO TO 45
   19 CONTINUE
C  THIS IS THE PARALLEL FLOW SECTION OF THE CONDENSER
      Q1=MDOTH*(HH2-HHI)
      TC2=TCI-Q1/CC
      E1=-Q1/(CMIN1*(THI-TCI))
      EP=E1*(1.+CR1)
      IF(EP.GE.1.0) GO TO 11
      A1=(-CMIN1/ULV)*ALOG(1.-E1*(1.+CR1))/(1.+CR1)
      IF(A1.LT.AT) GO TO 10
   11 A1=AT
      E1=(1.-EXP((-ULV*A1/CMIN1)*(1.+CR1)))/(1.+CR1)
      THO=THI-E1*(CMIN1/CH1)*(THI-TCI)
      TCO=TCI+E1*(CMIN1/CC)*(THI-TCI)
      QR=CC*(TCI-TCO)
      HHO=HHI+QR/MDOTH
      DELHO=HH3-HHO
      GO TO 45
   31 CONTINUE
C  THE INPUT STATE OF THE WORKING FLUID IS IN THE MIXED PHASE.
   20 A1=0.0
      E1=0.0
      Q1=0.0
      HH2=HHI
      TC2=TCI
      TH2=THI
   10 A2=AT-A1
      E2=1.-EXP(-U2P*A2/CC)
      TCO=TC2+E2*(TH2-TC2)
      THO=TH2
      Q2=CC*(TC2-TCO)
```

```
      HHO=HH2+Q2/MDOTH
      QR=Q1+Q2
      DELHO=HH3-HHO
   45 CONTINUE
      GO TO (50,51,51,51,51),IPRT
   50 CONTINUE
      WRITE(-,-) PMIN,DELHO,HHO,HH2,HH3,TH2,TH3,HH1,TH1,TC1,TC2,TCO
      WRITE(-,617)
  617 FORMAT(1X,' QR AT Q1 Q2 E1 E2 A1 A2 FOR THE CONDENSER ')
      WRITE(-,-)QR,AT,Q1,Q2,E1,E2,A1,A2
   51 CONTINUE
      RETURN
      END
```

Subroutine RTFND

This subroutine uses Newton's method to solve for the roots of the equations in an iterative manner. It is referenced only by subroutine RANKIN.

Function FRP

Function FRP is a curve fit for volumetric efficiency for a reciprocating compressor as a function of pressure ratio. It is used to vary the expander volumetric efficiency as a function of the ratio of inlet to outlet pressures. The data is from the Trane Company.

```
      SUBROUTINE RTFND(X,XO,DELX,DELXO,DERIV,ICOUNT,STEP,ERR)
C   THIS SUBROUTINE USES TO NEWTONS METHOD TO FIND THE REAL ZEROS.
C     STEP IS THE FIRST GUESS FOR THE ITERATION PROCEDURE.
      GO TO (1,2,2,2,2,2,2,2,2,2,2,2,2,2,2,2,2,2,2,2,2),ICOUNT
    1 XO=X
      X=X+STEP
      DELXO=DELX
      DERIV=10.
      ICOUNT=ICOUNT+1
      GO TO 3
    2 IF(ABS(DELXO-DELX).LE.ERR) GO TO 4
      IF(ABS(X-XO).LT.0.01*ERR) GO TO 4
      DERIV=(DELX-DELXO)/(X-XO)
    4 XO=X
      DELXO=DELX
      X=X-DELX/DERIV
      ICOUNT=ICOUNT+1
    3 CONTINUE
      RETURN
      END
```

```
      FUNCTION FRP(PMAX,PMIN)
C     THIS FUNCTION DETERMINES VOLUMETRIC EFFICIENCY AS A FUNCTION
C     OF EXPANDER PRESSURE RATIO. IT IS BASED ON TRANE CO. DATA FOR
C     RECIPROCATING COMPRESSORS.
      PR=PMAX/PMIN
      ARG1=1.0893-.12725*PR+1.6036E-02*PR**2-1.1709E-03*PR**3
      ARG2=3.3379E-05*PR**4-3.0405E-07*PR**5
      FRP=ARG1+ARG2
       IF(FRP.LT.0.050) FRP=.050
      RETURN
      END
```

Subroutine COMP

Subroutine COMP models a variable speed valveless, positive displacement pump. It assumes constant volumetric and adiabatic efficiencies and treats the compression as that of a constant density fluid. Volumetric efficiency is set equal to 90%. Subroutine FREON is referenced only when ICOMP=2 and new values of the variables HOUT and TOUT are required. This routine is referenced only by subroutine RANKIN.

```
      SUBROUTINE  COMP(ICOMP,PAR,HIN,HOUT,TOUT,MDS,AWC,AWCT)
C     THIS ROUTINE SIMULATES A BOILER FEED PUMP OPERATING AT CONSTANT
C     VOLUMETRIC EFFICIENCY AND ADIABATIC EFFICIENCY FOR A CONSTANT
C     DENSITY COMPRESSION. THE VARIABLE ICOMP IS TO REDUCE COMPUTATION
C     BY NOT CALLING FREON UNLESS NEEDED.
      DIMENSION PAR(10)
      REAL MDS
      PMAX=PAR(1)
      PMIN=PAR(2)
      REVC=PAR(3)
      VOLC=PAR(4)
      ENC=PAR(5)
      CPSL=PAR(6)
      Q=0.0
      ITYPE=25
      CALL FREON(TIN,PMIN,HIN,SIN,Q,VIN,ITYPE)
      AWC=-VIN*(PMAX-PMIN)*144./(778.16*ENC)
C     PUMP VOLUMETRIC EFFICIENCY IS ASSUMED TO BE 90%.
      MDS=REVC*VOLC*60.*.90/VIN
      GO TO (10,100),ICOMP
   10 CONTINUE
      AWCT=AWC*MDS
      HOUT=HIN-AWC
      Q=0.0
      ITYPE=25
      CALL FREON(T3,PMAX,H3,S3,Q,V3,ITYPE)
      TOUT=T3-(H3-HOUT)/CPSL
  100 CONTINUE
      RETURN
      END
```

Subroutines FREON, CHECK2 and FP

Subroutine FREON supplies working fluid properties to the Rankine engine simulation. This is accomplished using a cubic interpolation scheme (function FP) which determines the desired properties by interpolating tabulated data. Any working fluid may be used providing the input data is in the correct format. Subroutine CHECK2 is used to ensure the interpolation uses the correct set of data points.

Given the independent properties temperature and/or pressure and any one other property (enthalpy, entropy, quality in mixed phase only, and specific volume), subroutine FREON will fix the state and return the values of the properties which were undefined. The value of ITYPE specifies the independent variables.

```
      SUBROUTINE FREON(TX,PX,HX,SX,Q,VX,ITYPE)
C     THIS ROUTINE INTERPOLATES TO FIND FREON PROPERTIES .
      LOGICAL FLAG
      DIMENSION PM(50),TM(50),HF(50),HG(50),SF(50),SG(50),VF(50),VG(50)
      DIMENSION P(50),T(4,50),H(4,50),S(4,50),V(4,50)
      DIMENSION X(4,50)
      DATA FLAG/.FALSE./
 700  IF(FLAG) GO TO 4
      READ,N,N2
      DO77 J=1,N
      READ(-,-) P(J)
      DO86 I=1,4
      READ(-,-) T(I,J),H(I,J),S(I,J),V(I,J)
 86   CONTINUE
 77   CONTINUE
      DO3 I=1,N2
      READ(-,-) TM(I),PM(I),HF(I),HG(I),SF(I),SG(I),VF(I),VG(I)
 3    CONTINUE
      FLAG=.TRUE.
 4    IP1=ITYPE/10
      IP2=ITYPE-IP1*10
      IF(IP2.LT.IP1) GO TO 311
      GO TO 312
 311  INT=IP2
      IP2=IP1
      IP1=INT
 312  IF(IP1.GT.2) GO TO 999
      IF(IP2.EQ.2.AND.IP1.EQ.1) GO TO 501
C     GO TO SUPERHEAT T-P
      MTYPE=10+IP2
C     MTYPE IS RECONSTRUCTED ITYPE WHERE IP1 IS ALWAYS LESS THAN IP2
      GO TO (109,110),IP1
 109  IC=0
C     *******  TEMPERATURE US ONE INDEPENDENT VARIABLE IN 2-PHASE
      DO112 I=1,N2
      DIFF=TX-TM(I)
      IF(DIFF.LT.0.0) GO TO 190
 112  CONTINUE
      WRITE(6,162)    TX,ITYPE
 162  FORMAT(1X,'SPECIFIED TEMPERATURE IS OUT OF THE RANGE OF THE
     1 PROPERTY TABLES',5X,'T = ',4X,F8.4,5X,'ITYPE = ',4X,I5)
C     ************************************
      STOP
 190  JTYPE=MTYPE-12
      CALL CHECK2(I,N2,I1,I2,I3,I4)
 105  IC=IC+1
      GO TO (101,102,104,106),IC
 106  GO TO (201,202,103,204),JTYPE
 101  HFX=FP(TX,TM(I1),TM(I2),TM(I3),TM(I4),HF(I1),HF(I2),HF(I3),HF(I4))
      HGX=FP(TX,TM(I1),TM(I2),TM(I3),TM(I4),HG(I1),HG(I2),HG(I3),HG(I4))
      PX=FP(TX,TM(I1),TM(I2),TM(I3),TM(I4),PM(I1),PM(I2),PM(I3),PM(I4))
      GO TO 105
 102  SGX=FP(TX,TM(I1),TM(I2),TM(I3),TM(I4),SG(I1),SG(I2),SG(I3),SG(I4))
      SFX=FP(TX,TM(I1),TM(I2),TM(I3),TM(I4),SF(I1),SF(I2),SF(I3),SF(I4))
```

```
      1H(4,I4))
       HX=FP(PX,P(I1),P(I2),P(I3),P(I4),HT1,HT2,HT3,HT4)
       GO TO 509
  511  ST1=FP(XX,X(1,I1),X(2,I1),X(3,I1),X(4,I1),S(1,I1),S(2,I1),S(3,I1),
      1S(4,I1))
       ST2=FP(XX,X(1,I2),X(2,I2),X(3,I2),X(4,I2),S(1,I2),S(2,I2),S(3,I2),
      1S(4,I2))
       ST3=FP(XX,X(1,I3),X(2,I3),X(3,I3),X(4,I3),S(1,I3),S(2,I3),S(3,I3),
      1S(4,I3))
       ST4=FP(XX,X(1,I4),X(2,I4),X(3,I4),X(4,I4),S(1,I4),S(2,I4),S(3,I4),
      1S(4,I4))
       SX=FP(PX,P(I1),P(I2),P(I3),P(I4),ST1,ST2,ST3,ST4)
       GO TO 509
  520  VT1=FP(XX,X(1,I1),X(2,I1),X(3,I1),X(4,I1),V(1,I1),V(2,I1),V(3,I1),
      1V(4,I1))
       VT2=FP(XX,X(1,I2),X(2,I2),X(3,I2),X(4,I2),V(1,I2),V(2,I2),V(3,I2),
      1V(4,I2))
       VT3=FP(XX,X(1,I3),X(2,I3),X(3,I3),X(4,I3),V(1,I3),V(2,I3),V(3,I3),
      1V(4,I3))
       VT4=FP(XX,X(1,I4),X(2,I4),X(3,I4),X(4,I4),V(1,I4),V(2,I4),V(3,I4),
      1V(4,I4))
       VX=FP(PX,P(I1),P(I2),P(I3),P(I4),VT1,VT2,VT3,VT4)
       GO TO 509
  513  TT1=FP(XX,X(1,I1),X(2,I1),X(3,I1),X(4,I1),T(1,I1),T(2,I1),T(3,I1),
      1T(4,I1))
       TT2=FP(XX,X(1,I2),X(2,I2),X(3,I2),X(4,I2),T(1,I2),T(2,I2),T(3,I2),
      1T(4,I2))
       TT3=FP(XX,X(1,I3),X(2,I3),X(3,I3),X(4,I3),T(1,I3),T(2,I3),T(3,I3),
      1T(4,I3))
       TT4=FP(XX,X(1,I4),X(2,I4),X(3,I4),X(4,I4),T(1,I4),T(2,I4),T(3,I4),
      1T(4,I4))
       TX=FP(PX,P(I1),P(I2),P(I3),P(I4),TT1,TT2,TT3,TT4)
       GO TO 509
  999  PRINT 998
  998  FORMAT(1X,'*********ILLEGAL ITYPE SPECIFICATION*************')
       STOP
  304  RETURN
  514  RETURN
       END
```

```
      SUBROUTINE CHECK2(I,N2,I1,I2,I3,I4)
      IF(I.EQ.N2) GO TO 3
      IF(I.GT.2) GO TO 2
      I1=1
      I2=2
      I3=3
      I4=4
      RETURN
2     I4=I+1
      I3=I
      I2=I-1
      I1=I-2
      RETURN
3     I4=N2
      I3=N2-1
      I2=N2-2
      I1=N2-3
      RETURN
      END
```

```
      FUNCTION FP(A,A1,A2,A3,A4,B1,B2,B3,B4)
      FP=B1*(A-A2)*(A-A3)*(A-A4)/((A1-A2)*(A1-A3)*(A1-A4))
     1+B2*(A-A1)*(A-A3)*(A-A4)/((A2-A1)*(A2-A3)*(A2-A4))
     1+B3*(A-A1)*(A-A2)*(A-A4)/((A3-A1)*(A3-A2)*(A3-A4))
     1+B4*(A-A1)*(A-A2)*(A-A3)/((A4-A1)*(A4-A2)*(A4-A3))
      RETURN
      END
```

```
      FUNCTION F(Y,Y1,Y2,X)
      F=Y*(X-1.)*(X-2.)/2.-Y1*X*(X-2.)+Y2*X*(X-1.)/2.
      RETURN
      END
```

Subroutines BLKBOX, PLOTT, and SSSORT

Subroutine BLKBOX is used to create a set of performance maps which contain the operating performance of the Rankine engine over a user specified operating range. These maps are functions only of hot-side boiler and cold-side condenser inlet temperature. The map is obtained by calling subroutine BLKBOX over an internally generated N x N range of these inlet temperatures (N is supplied by the user) within the user's operating limits. The common block ANUM contains WEXP (the ratio of expander power to design expander power), WPUMP (the ratio of pump input power to the design input power, and ENTH (engine thermal efficiency) which comprise the performance maps. The common block COORD contains the information needed to reread these arrays. Both common blocks transfer information to subroutine BLUBOX. This performance map may be written onto or read from mass storage as described in Section 3.8. Subroutine PLOTT is a printer plotting routine from the Engineering Computing Lab which is used to output the performance map for design purposes. PLOTT calls SSSORT internally.

```
      SUBROUTINE BLKBOX(PAR,DPAR)
C     PAR CONTAINS USER SUPPLIED INPUT PARAMETERS, DPAR CONTAINS INPUT
C     PARAMETERS AS CALCULATED BY SIZER.
C     THE MAXIMUM NUMBER OF POINTS WHICH MAY BE USED IN A BLACKBOX
C     IS NOW 20. IF MORE POINTS ARE DESIRED THE ARRAYS X,YENT,XC,
C     YEXP, AND YPMP SHOULD BE REDIMENSIONED, AND THE ARRAYS IN PLTTER
C     SHOULD BE CHANGED AS INDICATED THERE,
C     IF MORE THAN 40*40 POINTS ARE REQUIRED, THE COMMON BLOCK ARRAYS
C     ENTH,WEXP,WPUMP SHOULD BE REDIMENSIONED AS WELL.
      DIMENSION XIN(10),OUT(15),DPAR(20),PAR(40),X(401),YENT(401)
      DIMENSION XC(401),YEXP(401),YPMP(401)
C     THE COMMON BLOCKS ANUM AND COORD TRANSFER THE PERFORMANCE
C     MAP AND INFORMATION NEEDED TO INTERPOLATE IN IT TO BLUBOX.
      COMMON /ANUM/ ENTH(40,40),WEXP(40,40),WPUMP(40,40)
      COMMON /COORD/ DLTH,DLTC,THIH,THIC,N
      IMAP=PAR(25)
C     IMAP=1 WHEN THE RANKINE CYCLE IS OPERATED DIRECTLY, IE. WITHOUT A
C     BLACKBOX. IMAP=2 WHEN THE ENGINE IS TO OPERATED WITH A
C     BLACKBOX, BUT IT IS NOT TO BE SAVED. IMAP=3 WHEN THE BLACKBOX IS
C     READ FROM MASS STORAGE. IMAP=4 WHEN THE RANKINE CYCLE IS TO BE
C     OPERATED WITH A BLACKBOX WHICH IS ALSO TO BE SAVED ON MASS STORAGE.
      N=PAR(26)
      TMAXH=PAR(27)
      TMINH=PAR(28)
      TMAXC=PAR(29)
      TMINC=PAR(30)
      ISAVE=PAR(31)
      IPLOT=PAR(32)
      DLTH=(TMAXH-TMINH)/(N-3)
      DLTC=(TMAXC-TMINC)/(N-3)
      THIH=TMAXH+DLTH
      THIC=TMAXC+DLTC
      GO TO (10,2,100,2),IMAP
    2 CONTINUE
      DO 20 J=1,N
      XIN(1)=THIH-(J-1)*DLTH
      DO 20 I=1,N
      L=I
      IF(J.NE.2*(J/2)) GO TO 7
      L=N-L+1
    7 CONTINUE
      XIN(2)=THIC-(L-1)*DLTC
C     THIS TEST ARBITRARILY SHUTS OFF THE RANKINE ENGINE IF THE
C     WATERSIDE INLET TEMPERATURES DIFFER BY 30 DEG F.
      IF((XIN(1)-XIN(2)).LE.30.) GO TO 31
      XIN(3)=PAR(1)
      CALL RANKIN(XIN,OUT,PAR,DPAR)
      GO TO 30
   31 CONTINUE
      OUT(12)=0.0
      OUT(10)=0.0
      OUT(11)=0.0
```

```
      XMIN=50.
      XMAX=250.
      YMIN=0.0
      YMAX=0.20
      CALL PLOTT(X,YENT,NX,XMIN,XMAX,YMIN,YMAX,5)
      XMIN=0.
      XMAX=150.
      YMAX=2.0
      CALL PLOTT(XC,YEXP,NX,XMIN,XMAX,YMIN,YMAX,5)
      CALL PLOTT(XC,YPMP,NX,XMIN,XMAX,YMIN,YMAX,5)
  110 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE PLOTT (X,Y,N,XMIN,XMAX,YMIN,YMAX,K)
C     THIS IS A PLOTTING ROUTINE FROM ECL. IF MORE POINTS THAN
C     400 ARE DESIRED TO BE PLOTTED CHANGE THE DIMENSION OF THE
C     ARRAYS IX AND IY.
      DIMENSION X(1), Y(1), IUSED(101), IX(401), IY(401), AX(11)
      DATA IOOUT /6/
      DATA IIIIII /1HI/, IBLANK /1H /, IMINUS /1H-/, IISTAR /1H*/
C     FOR MORE THAN 400 POINTS THIS TEST MUST BE  CHANGED.
  400 IF (N .GT. 0 .AND. N .LE. 400) GO TO 405
  403 WRITE (IOOUT,8) N
      WRITE (IOOUT,404)
  404 FORMAT(27H IMPROPER NUMBER OF POINTS.  )
      GO TO 1001
  405 IF((51-K)*K-50)410,411,411
  410 KX=5
      WRITE (IOOUT,412) K
  412 FORMAT(12H IMPROPER K, /11H K INPUT AS I5)
      GO TO 402
  411 KX=K
  402 NX=N
      IXFLG=0
      IF(XMIN-XMAX)431,432,431
  432 IXFLG=1
      WRITE (IOOUT,435)
  435 FORMAT(10H XMIN=XMAX)
  431 IF(YMIN-YMAX)433,434,433
  434 WRITE (IOOUT,436)
  436 FORMAT(10H YMIN=YMAX)
      IXFLG=1
  433 IF(IXFLG)437,437,1001
 1001 WRITE (IOOUT,9) XMIN, XMAX, YMIN, YMAX
      GO TO 100
  437 AX(1)=XMIN
      AX(2)=XMAX
      AX(3)=YMIN
      AX(4)=YMAX
      DO 265 I=1,4
      IF (AX(I)) 262,261,262
  261  IUSED(I)=0
       GO TO 265
  262 XA=ABS(AX(I))
      IF((10000.01-XA)*XA-100.)264,261,261
  264  IUSED(I)=1
  265 CONTINUE
       IXFLG=IUSED(1)+IUSED(2)
       IYFLG=IUSED(3)+IUSED(4)
      XINC=(XMAX-XMIN)/100.
      YINC=(YMAX-YMIN)/50.
      J=0
      M=0
      DO 286 I=1,NX
      XMXA = (X(I) - XMIN)/XINC + 1.5
      IF (XMXA .GT. 110.0 .OR. XMXA .LT. -10.0) GO TO 283
```

```
      MXA = XMXA
      IF (MXA .GT. 101 .OR. MXA .LT. 1) GO TO 283
  282 XMYA = (Y(I) - YMIN)/YINC + 0.5
      IF (XMYA .GT. 60.0 .OR. XMYA .LT. -10.0) GO TO 283
      MYA = XMYA
      IF (MYA .LT. 51 .AND. MYA .GT. -1) GO TO 281
  283 IF(J)284,284,285
284   WRITE (IOOUT,8) N
    8 FORMAT(25HINUMBER OF INPUT POINTS = I4/)
      WRITE (IOOUT,9) XMIN, XMAX, YMIN, YMAX
    9 FORMAT(8H XMIN = 1PE10.3/8H XMAX = 1PE10.3/
     1  8H YMIN = 1PE10.3/ 8H YMAX = 1PE10.3/)
      WRITE (IOOUT,1) XINC, YINC
    1 FORMAT(/17H INCREMENT IN X =  1PE10.3/
     1 17H INCREMENT IN Y = 1PE10.3/)
      WRITE (IOOUT,287)
  287 FORMAT(21H VALUES OUTSIDE RANGE  //
     124H POINT         X            Y    /5H  NO.)
      J=1
  285 WRITE (IOOUT,288) I, X(I), Y(I)
  288 FORMAT(1X,I3,3X,2(1PE12.3))
      GO TO 286
  281 M=M+1
      IX(M) = MXA
      IY(M) = MYA
  286 CONTINUE
      NX=M
    7 IF(NX)98,98,44
   98 WRITE (IOOUT,97)
   97 FORMAT(/23H NO VALUES INSIDE RANGE /)
      GO TO 100
   44 IF(J .GT. 0) WRITE (IOOUT,21) NX
   21 FORMAT(/26H NO. OF POINTS IN RANGE =  I3/)
      CALL SSSORT (-NX, IY, IX)
      IZRO=101
      IFLG=0
      IF(XMIN*XMAX)604,602,602
  604 IZRO=(-XMIN/XINC)+1.
      IFLG=1
  602 DO 88 I=1,101
   88 IUSED(I) = IBLANK
      IX(NX+1)=1
      IY(NX+1)=-1
      I=1
      MAXX=0
      J=1
      LC=50
      XA=ABS(YINC/2.)
      WRITE (IOOUT,103)
  103 FORMAT(1H1,13X,10(10HI++++.++++),1HI/)
  111 IF(IFLG)114,114,161
```

```
161 IUSED(IZRO) = IIIIII
114 IF(NX-I)113,117,117
117 IF(LC-IY(I))199,115,113
199 I=I+1
    GO TO 114
115 IXS=IX(I)
    IUSED(IXS) = IISTAR
    I=I+1
    GO TO 114
101 FORMAT(1X,F10.3,2X,1H+,50A1,51A1)
102 FORMAT(13X,1H+,50A1,51A1)
159 FORMAT(1X,E10.3,2X,1H+,50A1,51A1)
113 ALC=50-LC
    YVAL=YMAX-ALC*YINC
    NM=(LC/KX)*KX
    IF(LC-NM)120,121,120
120 WRITE (IOOUT,102) (IUSED(IXS), IXS = 1,101)
    GO TO 125
121 IF(IYFLG)151,151,152
151 WRITE (IOOUT,101) YVAL, (IUSED(IXS), IXS = 1,101)
    GO TO 125
152 WRITE (IOOUT,159) YVAL, (IUSED(IXS), IXS = 1,101)
125 IF(MAXX)451,452,451
451 DO 453 IXS=1,101,2
453 IUSED(IXS) = IBLANK
    MAXX=0
452 DO 456 IYS=J,I
    IXS=IX(IYS)
456 IUSED(IXS) = IBLANK
457 J=I
454 LC=LC-1
    IF(LC)150,111,201
201 IF(ABS(YVAL-YINC)-XA)203,203,111
203 DO 204 IXS=1,101,2
204 IUSED(IXS) = IMINUS
    MAXX=1
    GO TO 114
150 WRITE (IOOUT,106)
106 FORMAT(/14X,10(10HI++++.++++),1HI/5X,11(9X,1HI))
    AX(1)=XMIN
    DO 130 M=2,6
130 AX(M)=AX(M-1)+20.*XINC
    DO 131 M=7,11
131 AX(M)=(AX(M-6)+AX(M-5))/2.
    IF(IXFLG)153,153,143
153 WRITE (IOOUT,105) (AX(M), M = 1,11)
105 FORMAT(/  8X,6(F10.3,10X)/18X,5(F10.3,10X))
    GO TO 100
143 WRITE (IOOUT,104) (AX(M), M = 1,11)
104 FORMAT(/10X,6(E10.3,10X)/20X,5(E10.3,10X))
100 WRITE (IOOUT,99)
 99 FORMAT (1H )
    RETURN
    END
```

```
      SUBROUTINE SSSORT(NX,NA,IX)
C    THIS ROUTINE IS CALLED BY SUBROUTINE PLOTT.
      DIMENSION NA(1),IX(1),IU(16),IL(16)
      M=1
      I = 1
      J = IABS(NX)
      IF (J .LE. 1) RETURN
   10 K=I
      IJ=(J+I)/2
      NT=NA(IJ)
      MT = IX(IJ)
      IF(NA(I) .GE. NT) GO TO 20
      NA(IJ)=NA(I)
      IX(IJ) = IX(I)
      NA(I)=NT
      IX(I) = MT
      NT=NA(IJ)
      MT = IX(IJ)
   20 L=J
      IF(NA(J) .LE. NT) GO TO 40
      NA(IJ)=NA(J)
      IX(IJ) = IX(J)
      NA(J)=NT
      IX(J) = MT
      NT=NA(IJ)
      MT = IX(IJ)
      IF(NA(I) .GE. NT) GO TO 40
      NA(IJ)=NA(I)
      IX(IJ) = IX(I)
      NA(I)=NT
      IX(I) = MT
      NT=NA(IJ)
      MT = IX(IJ)
      GO TO 40
   30 NA(L)=NA(K)
      IX(L) = IX(K)
      NA(K)=NTT
      IX(K) = MTT
   40 L=L-1
      IF(NA(L) .LT. NT) GO TO 40
      NTT=NA(L)
      MTT = IX(L)
   50 K=K+1
      IF(NA(K) .GT. NT) GO TO 50
      IF(K .LE. L) GO TO 30
      IF( L-I .LE.  J-K ) GO TO 60
      IL(M)=I
      IU(M)=L
      I=K
      M=M+1
      GO TO 80
   60 IL(M)=K
      IU(M)=J
      J=L
```

```
      M=M+1
      GO TO 80
   70 M=M-1
      IF(M .EQ. 0) RETURN
      I=IL(M)
      J=IU(M)
   80 IF(J-I .GE. 11) GO TO 10
      I=I-1
   90 I=I+1
      IF(I .EQ. J) GO TO 70
      NT=NA(I+1)
      IF(NA(I) .GE. NT) GO TO 90
      MT = IX(I+1)
      K=I
  100 NA(K+1)=NA(K)
      IX(K+1) = IX(K)
      K=K-1
      IF(K .EQ. 0) GO TO 110
      IF(NT .GT. NA(K)) GO TO 100
  110 NA(K+1)=NT
      IX(K+1) = MT
      GO TO 90
      END
```

Subroutine BLUBOX and Function F

Subroutine BLUBOX uses the performance map generated by subroutine BLKBOX to obtain a black box model to be used in long term simulations. It is referenced by subroutine TYPE23 and has inputs from subroutine BLKBOX via the labeled common blocks ANUM and COORD. The operation of this routine is discussed in Section 3.7. Function F is used to interpolate in the performance map.

```
      SUBROUTINE BLUBOX(TBOIL,TCON,WOUT,OUT,DPAR,PAR)
C    THIS SUBROUTINE USES THE BLACKBOX REPRESENTATION (PERFORMANCE
C    MAP) PRODUCED BY BLKBOX TO SIMULATE THE RANKINE ENGINE. IT
C    INTERPOLATES IN THE BLACKBOX TO OBTAIN THE VALUES OF THERMAL
C    EFFICIENCY AND THE RATIOS OF PUMP AND EXPANDER POWER TO THEIR
C    DESIGN VALUES.
C    PAR IS AN ARRAY OF USER SUPPLIED INPUT PARAMETERS, DPAR IS
C    AN ARRAY OF INPUT PARAMETERS AS DEFINED BY SIZER, OUT IS THE
C    ARRAY OF OUTPUT VARIABLES.
      DIMENSION OUT(15),DPAR(20),PAR(40),ENTH33(3,3),ENTH3(3),WEXP33(3,3
     1),WEXP3(3),WPMP33(3,3),WPMP3(3)
C    IF MORE THAN 40*40 POINTS ARE DESIRED IN THE BLACKBOX THE COMMON
C    BLOCK ARRAYS ENTH,WEXP, AND WPUMP SHOULD BE REDIMENSIONED.
C    THE COMMON BLOCKS ANUM AND COORD CARRY THE PERFORMANCE MAP
C    AND INFORMATION NEEDED TO INTERPOLATE IN IT FROM BLKBOX.
      COMMON /ANUM/ ENTH(40,40),WEXP(40,40),WPUMP(40,40)
      COMMON /COORD/ DLTH,DLTC,THIH,THIC,N
      REAL MBOIL,MCON
      T1=(THIC-TCON)/DLTC
      T2=(THIH-TBOIL)/DLTH
      IT1=IFIX(T1)+1
      IF(T1-IT1+0.5) 15,10,10
   10 CONTINUE
      IT1=IT1+1
   15 CONTINUE
      IT2=IFIX(T2)+1
      IF(T2-IT2+0.5) 25,20,20
   20 CONTINUE
      IT2=IT2+1
   25 CONTINUE
      X1=T1+2-IT1
      X2=T2+2-IT2
      DO 30 L=1,3
      DO 30 K=1,3
      IK=IT1+K-2
      IL=IT2+L-2
      ENTH33(K,L)=ENTH(IK,IL)
   30 CONTINUE
      DO 40 K=1,3
      ENTH3(K)=F(ENTH33(K,1),ENTH33(K,2),ENTH33(K,3),X2)
   40 CONTINUE
      ENT=F(ENTH3(1),ENTH3(2),ENTH3(3),X1)
      DO 130 L=1,3
      DO 130 K=1,3
      IK=IT1+K-2
      IL=IT2+L-2
      WEXP33(K,L)=WEXP(IK,IL)
  130 CONTINUE
      DO 140 K=1,3
      WEXP3(K)=F(WEXP33(K,1),WEXP33(K,2),WEXP33(K,3),X2)
  140 CONTINUE
      WEXPR=F(WEXP3(1),WEXP3(2),WEXP3(3),X1)
```

```
      DO 230 L=1,3
      DO 230 K=1,3
      IK=IT1+K-2
      IL=IT2+L-2
      WPMP33(K,L)=WPUMP(IK,IL)
  230 CONTINUE
      DO 240 K=1,3
      WPMP3(K)=F(WPMP33(K,1),WPMP33(K,2),WPMP33(K,3),X2)
  240 CONTINUE
      WPMPR=F(WPMP3(1),WPMP3(2),WPMP3(3),X1)
C     NEGATIVE THERMAL EFFICIENCIES CAUSED BY A POOR FIT OF THE
C     DATA ARE NOT ALLOWED.
      IF(ENT.LE.0.0) GO TO 300
      WDOUT=PAR(1)
      WPD=DPAR(14)
      MBOIL=DPAR(6)
      CPBOIL=PAR(3)
      MCON=DPAR(11)
      CPCON=PAR(9)
      WACT=WEXPR*WDOUT
      WAUX=WOUT-WACT
      WP=WPMPR*WPD
      QA=(WACT+WP)/ENT
      QR=(WACT+WP)-QA
      TBOILO=TBOIL-QA/(MBOIL*CPBOIL)
      TCONO=TCON-QR/(MCON*CPCON)
      OUT(1)=QA
      OUT(2)=QR
      OUT(3)=TBOILO
      OUT(4)=TCONO
      OUT(5)=WP
      OUT(6)=WACT
      OUT(7)=WAUX
      OUT(8)=MBOIL
      OUT(9)=MCON
      OUT(10)=WEXPR
      OUT(11)=WPMPR
      OUT(12)=ENT
      RETURN
  300 CONTINUE
      OUT(1)=0.0
      OUT(2)=0.0
      OUT(3)=TBOIL
      OUT(4)=TCON
      OUT(5)=0.0
      OUT(6)=0.0
      OUT(7)=WOUT
      OUT(8)=0.0
      OUT(9)=0.0
      OUT(10)=0.0
      OUT(11)=0.0
      OUT(12)=0.0
      RETURN
      END
```

Program MAIN1

This is the main program used to generate performance maps and do design studies with the Rankine engine model. Although the model is capable of producing performance maps during a TRNSYS execution, it is much simpler to do this function in a program which is distinct of TRNSYS.

```
      DIMENSION XIN(10),OUT(15),PAR(40),INFO(8)
C     THIS ROUTINE IS USED ONLY WHEN USING THE RANKINE ENGINE
C     FOR A DESIGN STUDY OR FOR PRODUCING A PERFORMANCE MAP.
C     IT SHOULD NOT BE INCLUDED IN A TRNSYS SIMULATION.
C     READ THE INPUT PARAMETERS AND WRITE THEIR VALUES AND NAMES.
      READ(-,-)(PAR(J),J=1,32)
      WRITE(-,20)
   20 FORMAT(1X,' WDOUT TBOIL CPBOIL ULL DELTB DELTOB IBHX TCON CPCON
     1ULV DELTC DELTOC ICHX WFRIC REVE ENE REVC ENC CPSL PLOSS ERK ERV
     2IOSCL IPRT ')
      WRITE(-,21)
   21 FORMAT(1X,' IMAP N TMAXH TMINH TMAXC TMINC ISAVE IPLOT ')
      WRITE(-,-)(PAR(J),J=1,32)
C     READ THE NUMBER OF STATES TO BE CONSIDERED IN THIS SIMULATION.
      READ(-,-) N
      DO 2 IT=1,N
C     READ THE INPUT VARIABLES AND WRITE THEIR NAMES AND VALUES.
      READ(-,-)(XIN(L),L=1,4)
      WRITE(-,3)
    3 FORMAT(1X,' TBOIL  TCON  WOUT  IRUN ')
      WRITE(-,-)(XIN(K),K=1,4)
      TIME=1
      T=0
      DTDT=0
      CALL TYPE23(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C     WRITE THE OUTPUT VARIABLE VALUES AND NAMES.
      WRITE(-,1)
    1 FORMAT(1X,' QA  QR  THOUT  TCOUT  WC  WACT  WAUX  MBOIL  MCON  ')
      WRITE(-,-)(OUT(J),J=1,9)
      ENTH=OUT(12)
      WRITE(-,4)ENTH
    4 FORMAT(1X,' THE THERMAL EFFICIENCY = ',F12.6)
      WRITE(-,7)OUT(10),OUT(11)
    7 FORMAT(1X,' WACT/WDOUT = ',F12.6,' WC/WPUMP DESIGN = ',F12.6)
      WRITE(-,8)
    8 FORMAT(1X,//)
    2 CONTINUE
      STOP
      END
```

Appendix B. Air Conditioner - Heat Pump Model and Program Listing

The Air conditioner-heat pump model is a modified version of the curve fit of performance data offered in reference 20. The unit models air-to-air type heat pumps with capacities between 8.79 kw (2.5 tons) and 35.2 kw (10 tons). The model assumes that the system COP is not a function of unit size for the given range. The model also modulates its capacity to match heat pump output to cooling load if the required amount of cooling is less than the amount the unit can supply at those conditions. If the unit could not supply the load while operating in the cooling mode at some time a warning is printed. In heating mode if the unit cannot meet the desired load, additional energy is added in the form of electric auxiliary. This increased energy is outputed in the term WAUX.

The call to this unit is in the standard TRNSYS input format (23).

```
      SUBROUTINE TYPE20(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C   THIS IS A ROUTINE WHICH MODELS A HEAT PUMP IN THE COOLING MODE
C   OR THE HEATING MODE VIA A CURVE FIT OF GE WEATHERTRON DATA.
C   IT IS MODIFIED FROM THE GE PHASE O NSF REPORT.
C   XIN IS AN INPUT VARIABLE ARRAY, PAR IS AN INPUT PARAMETER ARRAY,
C   OUT IS AN OUTPUT ARRAY. INPUTS MUST BE IN THE UNITS KJ, M, HOUR,
C   AND DEG C. THE INTERNAL CURVE FITS ARE FOR THE UNITS BTU, FT,
C   HOUR, AND DEG F.
      DIMENSION XIN(10),OUT(10),PAR(10)
      TODB=XIN(1)*1.8+32.
      RH=XIN(2)/100.
      QLOAD=XIN(3)/1.055
      IMODE=XIN(4)
C  IF IMODE=1 THE HEATPUMP IS IN THE HEATING MODE
C  IF IMODE=2 THE HEATPUMP IS IN THE COOLING MODE OF OPERATION.
      IRUN=XIN(5)
      QCAPYD=PAR(1)/1.055
      COPD=PAR(2)
      TIDB=PAR(3)*1.8+32.
      TIWB=PAR(4)*1.8+32.
      WFAN=PAR(5)/1.055
      WAUX=WFAN
C   IRUN IS AN EXTERNAL CONTROLLER.
      IF(IRUN) 50,50,5
    5 CONTINUE
      GO TO (1,2),IMODE
    1 CONTINUE
C  THIS IS THE HEATING MODE.
C   THIS DETERMINES THE DESIGN COMPRESSOR INPUT POWER IN THE HEATING
C   MODE OF OPERATION.
      PDH=QCAPYD/COPD
C   CAPY IS THE FRACTION OF DESIGN CAPACITY BASED ON THE A.R.I.
C   240 STANDARDS FOR HEATING .
      CAPY=0.6512+0.016*TODB-0.00233*TIDB*(TODB+20.)**0.197
      QCAPY=CAPY*QCAPYD
      QSENS=QCAPY
C   THIS TEST DETERMINES IF FROST CAN FORM ON THE EVAPORATOR COIL.
      IF(TODB.LT.20.) GO TO 3
C   P IS THE FRACTION OF DESIGN COMPRESSOR INPUT POWER IN THE HEATING
C   MODE OF OPERATION.
      P=0.444+0.004*TIDB+3.8E-05*TIDB*TODB+9.7E-04*TODB+5.76E-05*TODB**2
      GO TO 4
    3 CONTINUE
C   THE COMPRESSOR INPUT POWER CHANGES WHEN LESS THAN 20 DEG F
C   BECAUSE OF DEFROSTING.
      P=0.484+0.0034*TIDB+6.4E-05*TIDB*TODB
    4 CONTINUE
      WCOMP=P*PDH
C   LESS THAN 0 POWER INPUT IS NOT ALLOWED.
      IF(WCOMP.LT.0.0) GO TO 50
C   TESTS TO DETERMINE IF FROST IS PRESENT ON THE EVAPORATOR COIL.
      IF(TODB.GT.39..OR.TODB.LT.28.) GO TO 100
      IF((TODB+40.).GT.RH) GO TO 100
C   IF FROST IS PRESENT ON THE COIL ELECTRIC AUXILIARY IS USED
```

```
C    TO MELT THE ICE. THE ENERGY IS ACCOUNTED TO WAUX.
      WAUX=WFAN+0.1*QCAPY/COPD
      GO TO 100
    2 CONTINUE
C   THIS IS THE COOLING MODE
C    THIS DETERMINES THE INPUT COMPRESSOR POWER IN THE COOLING
C    MODE OF OPERATION AT DESIGN CAPACITY.
      PDC=QCAPYD/COPD
C    CAPY IS THE FRACTION OF DESIGN CAPACITY BASED ON THE A.R.I.
C    240 DESIGN STANDARDS FOR COOLING.
      CAPY=0.252+0.0144*TIWB-6.96E-04*TODB-1.67E-05*TODB**2
      QCAPY=CAPY*QCAPYD
C    P IS THE FRACTION OF DESIGN COMPRESSOR INPUT POWER IN THE COOLING
C    MODE OF OPERATION.
      P=0.0074+0.0073*TIWB+0.0053*TODB
      WCOMP=P*PDC
C    LESS THAN 0 POWER INPUT IS NOT ALLOWED.
      IF(WCOMP.LT.0.0) GO TO 50
      QSENS=QCAPY*(1.18+0.0278*TIDB-0.04*TIWB)
C    QSENS IS THE AMOUNT OF SENSIBLE COOLING OBTAINED, IT CAN NOT
C    BE LARGER THAN THE AMOUNT OF TOTAL COOLING.
      IF(QSENS.GT.QCAPY) QSENS=QCAPY
      GO TO 100
   50 CONTINUE
      QCAPY=0.0
      QSENS=0.0
      WCOMP=0.0
      WAUX=0.0
      GO TO 40
  100 CONTINUE
C    THIS DETERMINES THE FRACTION OF LOAD TO CAPACITY.
      PARTLD=QLOAD/QCAPY
C    TEST FOR LOAD EXCEEDING CAPACITY.
      IF(PARTLD.LE.1.0) GO TO 30
      GO TO (20,25),IMODE
   20 CONTINUE
C    IN THE HEATING MODE LOADS GREATER THAN CAPACITY ARE MEET BY
C    ELECTRIC AUXILIARY.
      QELEC=QLOAD-QCAPY
      WAUX=QELEC+WAUX
      GO TO 40
   25 CONTINUE
      IF(IRUN) 40,40,32
   32 CONTINUE
C    IN THE COOLING MODE A WARNING IS OUTPUTED WHEN CAPACITY IS
C    EXCEDED BY LOAD.
      WRITE(*,31) TIME
   31 FORMAT(5X,' THE UNIT COULD NOT MEET THE LOAD AT TIME= ',F10.4)
      GO TO 40
   30 CONTINUE
      IF(PARTLD.LE.0.0) GO TO 50
      QCAPY=QLOAD
C    THE UNIT IS MODULATED FOR OPERATION AT PART LOAD.
      QSENS=PARTLD*QSENS
```

```
      WCOMP=PARTLD*WCOMP
      WAUX=PARTLD*WAUX
   40 CONTINUE
      OUT(1)=QCAPY*1.055
      OUT(2)=QSENS*1.055
      OUT(3)=WCOMP*1.055
      OUT(4)=WAUX*1.055
      RETURN
      END
```

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