Computer Modeling of a Regenerative Solar-Assisted Rankine Power Cycle

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Advanced solar-driven thermodynamic cycles for electric power generation are under study for application of the DSN Energy Conservation Project at Goldstone, California. This article presents a detailed interpretation of the computer program that describes the performance of one of these cycles; namely, a regenerative Rankine power cycle. Water is used as the working medium throughout the cycle. The solar energy collected at relatively low temperature level presents 75 to 80 % of the total heat demand and provides mainly the latent heat of vaporization. Another energy source at high temperature level superheats the steam and supplements the solar energy share. A program summary and a numerical example showing the sequence of computations are included. The outcome from the model comprises line temperatures, pressures, component heat rates, specific steam consumption, percentage of solar energy contribution, and the overall thermal efficiency.

I. Introduction

One of the problems associated with the effort to convert solar thermal energy to electric power via power cycles, is the low conversion efficiency. Solar energy collected by low cost flat plate collectors can inefficiently reach a temperature level of about 100°C. Such a temperature makes the thermal-to-mechanical energy conversion efficiency on the order of 10 %. On the other hand, expensive solar concentrators (ranging from 2 to 10 times the cost of flat plate collectors) can achieve much higher temperatures with a conversion efficiency in the order of 20 %.

As a candidate solution to the efficiency versus cost problem, several investigators (Refs. 1, 2 and 3) have recently studied an advanced solar-assisted Rankine cycle with two regenerators. In this advanced cycle, solar energy is used as the primary source of heat for supplying the latent heat of vaporization for the working fluid. Evaporation temperatures consistent with the capability of low-cost solar collectors are sufficient. This means that the evaporation temperature will range from 90°C for flat plate collectors to 160°C for low concentration ratio[1] types.

A thermal superheater utilizing a secondary source of energy at a high temperature level is added. The superheater will supplement the solar energy share and will elevate the maximum cycle temperature to that attained in commercial power plants.

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[1] There are several definitions in the literature for this ratio. The concentration ratio in this work refers to the ratio of the projected area of the opening receiving the Sun rays to the projected area of the absorbing tube(s).

The choice of the working fluid in this cycle is restricted to water only. The chemical instability of other candidates of organic working fluids prohibits their application at temperatures above 300°C. Water is also a nonhazard material; it is nontoxic, easily detectable and nonflammable. Moreover, the latent heat of water is the largest quantity of energy compared with liquid sensible heat or vapor superheat. This advantage makes the superheat share supplied by the secondary energy source small compared to the latent heat share contributed by the Sun. The overall arrangement of the advanced cycle results in a significantly higher thermal efficiency (about double) and a reduction of collector size (about one half) compared with a simple Rankine cycle without superheat that is 100 % powered by solar energy.

Definitions of terms used appear at the end of this article.

II. Thermodynamic Cycle

The system components and the states of the working fluid are shown in the flow diagram (Fig. 1) on the temperature entropy diagram (Fig. 2) and the enthalpy-entropy (Mollier) diagram (Fig. 3). The advanced Rankine power cycle is composed of the same simple Rankine cycle components; namely, a boiler, a superheater, a turbine, a condenser and a pump, in addition to two extra heat exchangers; a vapor-vapor regenerator and a vapor-liquid regenerator.

The cycle of events is described as follows: starting from the condenser end as shown in Fig. 1, the condensate (state 1) as saturated water is pumped to state 2 from the condenser pressure, P_c, to the evaporation pressure, P_e, in the boiler. Prior to the boiler entrance section (state 3), the water passes through the vapor-liquid heat exchanger and picks up some regenerative heat. In the solar powered boiler, the accumulated solar energy provides the latent heat of vaporization at the corresponding boiler pressure. A set of low-cost high-temperature flat-plate collectors or low-to-medium concentration ratio types can be used as the boiler. The evaporation temperature, t_e, can vary from 90°C for flat-plate types to 160°C for medium concentration types. The evaporation pressure will vary from 0.7 to 6 atm depending on the above collection temperature. It is important to note that the cycle can also be operated as a "waste-heat assisted" Rankine cycle by replacing the low-level-temperature solar heat by any other source of waste heat such as engine exhaust gases if available.

The dry and saturated water vapor (state 4), leaves the boiler for the vapor-vapor heat exchanger where it is superheated with regenerative heat to state 5. The high temperature superheater then follows to raise the maximum cycle temperature to around 600°C (state 6) where the heat is added by the secondary source of energy. The fuel used in the superheater can be of any phase (solid, liquid, or gaseous) and it presents a small fraction (20 to 25 %) of the total external heat added to the cycle.

Adiabatic expansion of the superheated steam in the turbine will produce the mechanical work required. The turbine exhaust steam (state 7), still at high temperature, is used to supply the two regenerative heat quantities; in the vapor-vapor heat exchanger to state 8 and in the vapor-liquid heat exchanger to state 9. The only heat rejection process in the cycle takes place in the condenser from state 9 to state 1 and the cycle is repeated.

The operational function of the two extra regenerators in the cycle is to reduce the heat input without affecting the network output. This function results in a large increase in the thermal efficiency.

III. Thermodynamic Properties

A. Enthalpy Scale

The enthalpy of water is calculated based on a reference temperature of 0°C, i.e., the enthalpy of saturated water at 0°C is 0 kcal/kg. The following are expressions used to construct the enthalpy scale.

(1) By approximating the specific heat of water liquid in the range of operating temperatures (up to 160°C) to be 1 kcal/kg°C, the enthalpy of saturated water liquid at t°C is equal to

$$H_f = t, \text{ kcal/kg} \tag{1}$$

(2) The latent heat of evaporation expression for water in kcal/kg is

$$H_{fg} = 597.8 - 0.583\, t \tag{2}$$

where t is the saturation temperature in °C, or

$$H_{fg} = 757.05 - 0.583\, T \tag{3}$$

where T is the saturation temperature in K. The error in both expressions, Eqs. (2) and (3), is found to be ± 0.4 kcal/kg

(3) From Eqs. (1) and (2), the enthalpy of saturated water vapor at temperature t°C is given by

$$H_g = 597.8 + 0.417\ t, \text{ kcal/kg} \tag{4}$$

(4) The specific heat at constant pressure of superheated steam can be expressed approximately by a linear relationship with the absolute temperature T,

$$C = a + bT \tag{5a}$$

The mean specific heat value between temperatures T_1 and T_2 is determined from

$$C_{1,2} = \frac{1}{(T_2 - T_1)} \int_{T_1}^{T_2} C\,dT$$

or

$$C_{1,2} = a + \frac{b}{2}\,(T_1 + T_2) \tag{5b}$$

The values of a and b for steam at atmospheric pressure and in the temperature range from 400 to 1000 K, are calculated using Ref. 4, thus

$$\begin{aligned} a &= 0.3924, \text{ kcal/kg K} \\ b &= 15 \times 10^{-5}, \text{ kcal/kg K}^2 \end{aligned} \tag{6}$$

(5) The enthalpy of superheated steam at temperature t°C and at a pressure equal to the saturation pressure of steam at temperature t_s°C is

$$H_{t,ts} = (597.8 + 0.417\ t_s) + C_{t,ts}\,(t - t_s), \text{ kcal/kg} \tag{7}$$

where $C_{t,ts}$ is the mean value of specific heat of the superheated steam between temperatures t and t_s as determined by Eq. (5b).

B. Entropy Scale

Having established an enthalpy scale, the entropy scale can be constructed by taking an entropy datum ($s = 0$) for saturated water liquid at 0°C (273.15 K). The following entropy expressions are then made:

(1) With a liquid specific heat of 1 kcal/kg°C, the entropy of saturated water liquid at T K is

$$S_f = \ln \frac{T}{273.15}\ , \text{ kcal/kg°C} \tag{8}$$

(2) Entropy of saturated steam at temperature t_s°C (T_sK) is calculated using Eqs. (3) and (8) as

$$S_g = \ln \frac{T_s}{273.15} + \frac{(757.05 - 0.583\ T_s)}{T_s} \tag{9}$$

(3) Entropy of superheated steam at temperature TK and at a pressure equal to the saturation pressure of steam at temperature T_s K is

$$S_{t,ts} = S_g\ (\text{at } T_s) + \int_{T_s}^{T} C\,\frac{dT}{T}$$

or by using Eq. (5a)

$$S_{t,ts} = \ln \frac{T_s}{273.15} + \frac{(757.05 - 0.583\ T_s)}{T_s} + a \ln \frac{T}{T_s} + b\,(T - T_s) \tag{10}$$

where a, b are given by Eq. (6).

C. Saturation Pressure

The saturated water vapor pressure P in a millimeter of mercury (mm Hg) corresponding to a saturation temperature T K can be expressed by the formula

$$P = \text{antilog}\left[7.8553 - \frac{1555}{T} - \frac{11.2414 \times 10^{-4}}{T^2}\right] \tag{11}$$

IV. Performance Calculations

To determine the thermodynamic properties of each state and the rates of heat and work exchanged for each process in the cycle, the following set of parameters should be given:

Temperatures

1. Evaporation temperature in the boiler, t_e °C
2. Condensation temperature, t_c °C
3. Maximum cycle temperature leaving the superheater, T_6 °C

Component Efficiency

4. Isentropic efficiency of the feed water pump, E_p
5. Isentropic efficiency of the turbine, E_T
6. Effectiveness of the vapor-vapor heat exchanger, E_{VV}
7. Effectiveness of the vapor-liquid heat exchanger, E_{VL}

As mentioned before, the evaporation temperature, t_e, may vary from 90°C to 160°C, according to the collector type. The condensation temperature, t_c, depends entirely on the type of cooling medium used. It varies from 25°C (77°F) for water cooled condensers to 40°C (104°F) for air-cooled ones. The maximum cycle temperature, t_6 in Fig. 2 has a ceiling value limited by the phenomenon of creep and other metallurgical effects on materials subjected to high temperature. A design value for the temperature t_6 should not exceed 600°C (1112°F) for a continuous safe operation.

Regarding the load versus the steam flow-rate control mechanism, the present study considers a "nozzle cut-off" in the turbine that is actuated by the turbine speed. The "throttling" control mechanism, on the other hand, is not recommended for its irreversibility and the associated changes in cycle configuration. Therefore, the above set of input parameters will be kept unchanged during load variations, and the only expected variation is the steam flow rate.

Considering a unit mass (1 kg) of steam flowing in the cycle, the following is an ordered sequence of calculation steps, which also presents the computer program sequence.

A. Cycle Limiting Pressures

This is determined by using Eq. (11) and the temperatures t_e and t_c for the pressures P_e and P_c, respectively.

B. Turbine Isentropic End Temperature

The turbine exit temperature in case of isentropic expansion (state 7′ on Figs. 2 and 3) has to be determined first before applying the turbine isentropic efficiency E_T. There exist several approaches to solve for state 7′, and all of them allow for a trial-and-error procedure. Two of these approaches are:

Approach 1: This assumes that the superheated steam follows the ideal gas relations. Accordingly, in the isentropic process from state 6 to state 7′, the temperature $T_{7'}$ is determined from

$$\frac{T_{7'}}{T_6} = \left(\frac{P_c}{P_e}\right)^{\gamma - 1/\gamma} \tag{12}$$

where γ is the mean specific heat ratio between temperatures T_6 and $T_{7'}$. A first estimate of γ can be taken as $1 \cdot 3$ for superheated steam. The second corrected estimate of γ will be calculated after calculating the temperature $T_{7'}$ using Eq. (12), and the mean specific heat at constant pressure $C_{6,7'}$ using Eq. (5b). Usually, this approach converges very rapidly to the true temperature required, and either one or two trials is enough for engineering purposes.

Approach 2: In this approach, the entropy of states 6 and 7′ are set equal using Eq. (10). The temperature $T_{7'}$ is then found by trial and error as follows:

$$S_6 = \ln \frac{T_e}{273.15} + \left(\frac{757.05}{T_e} - 0.583\right) + a \ln \frac{T_6}{T_e} + b\,(T_6 - T_e)$$

$$S'_7 = \ln \frac{T_c}{273.15} + \left(\frac{757.05}{T_c} - 0.583\right) + a \ln \frac{T_{7'}}{T_c} + b\,(T_{7'} - T_c)$$

with the equality

$$S_6 = S_{7'}$$

and dividing by a for both sides, then

$$\frac{1}{a} \ln\left(\frac{T_e}{T_c}\right) + \ln\left(\frac{T_6 T_c}{T_e}\right) + \frac{757.05}{a}\left(\frac{1}{T_e} - \frac{1}{T_c}\right) + \frac{b}{a}\,(T_6 - T_e + T_c)$$

$$- \frac{b}{a}\, T_{7'} = \ln T_{7'} \tag{13}$$

By raising both sides of Eq. (13) to the base e, and rearranging the terms, then

$$T_7' = \frac{\left(\frac{T_c T_6}{T_e}\right)\left(\frac{T_e}{T_c}\right)^{1/a}}{\exp\left[\frac{757.05}{a}\left(\frac{1}{T_c} - \frac{1}{T_e}\right) - \frac{b}{a}\left(T_6 + T_c - T_e\right)\right]} \exp\left(-\frac{bT_7'}{a}\right) \tag{14}$$

The values of a and b from Eq. (6) are substituted in Eq. (14) to yield

$$T_7' = \frac{T_6 \cdot \left(\frac{T_e}{T_c}\right)^{1.5484}}{\exp\left[1929.3\left(\frac{1}{T_c} - \frac{1}{T_e}\right) - \frac{(T_6 + T_c - T_e)}{2616}\right]} \exp\left(-\frac{T_7'}{2616}\right) \tag{15}$$

The temperature T_7' is found by substituting an initial estimate of T_7' in the right-hand side of Eq. (15) and getting the corrected value from the left-hand side of Eq. (15), and repeating the procedure. The convergence of this approach is also rapid but needs more than three trials, as will be shown later in an example.

C. Actual Turbine End-Temperature

The turbine isentropic efficiency E_T is defined as the ratio

$$E_p = \frac{\text{actual turbine work}}{\text{isentropic turbine work}} = \frac{H_6 - H_7}{H_6 - H_7'} \approx \frac{T_6 - T_7}{T_6 - T_7} \tag{16}$$

since the expansion process lies entirely in the superheat region, the above approximation simply means that the mean value of specific heat for the isentropic and actual expansion processes $C_{6,7}'$ and $C_{6,7}$ are equal.

The actual turbine end temperature T_7 is then found from Eq. (16) as

$$T_7 = T_6 - E_T(T_6 - T_7') \tag{17}$$

D. Turbine Work

This is calculated from

$$W_T = C_{6,7}\,(T_6 - T_7) \tag{18}$$

where $C_{6,7}$ is the mean specific heat at constant pressure between temperatures T_6 and T_7.

E. Pump Work

The pump work is usually very small compared with the turbine work and is sometimes neglected in most Rankine cycle analyses. However, it will be calculated in this program to size the pump and the driving motor. The actual pump work is given the enthalpy difference of states 1 and 2 as

$$W_p = (H_2 - H_1) = (P_e - P_c) \times \left(\frac{1\ \text{atm}}{760\ \text{mm Hg}}\right)\left(\frac{1.033\ \text{kg}_f/\text{cm}^2}{1\ \text{atm}}\right)\cdot\left(\frac{10^4\text{cm}^2}{\text{m}^2}\right) \times \left(\frac{\text{kcal}}{427\ \text{kg}_f\text{m}}\right)\cdot\frac{V_1}{E_P},\ \text{kcal/kg} \tag{19}$$

where V_1 is the specific volume of saturated water in m^3/kg at temperature t_c, E_P is the pump isentropic efficiency, and P_e and P_c are the evaporation and condenser pressures in mm Hg, respectively. The specific volume V_1 is considered uniform at 0.001 m^3/kg and Eq. (19) is reduced to

$$W_P = \frac{(P_e - P_c)\ \text{in mm Hg}}{31415\,E_P},\ \text{kcal/kg} \tag{20}$$

Also, the temperature of compressed liquid water leaving the pump is taken as the condenser temperature for heat exchange calculations

$$T_2 \approx T_1 = T_c \tag{21}$$

F. Vapor-Vapor Heat Exchanger

This is the device that transfers heat to the superheated steam from state 4 to state 5 by cooling the turbine exhaust from state 7 to state 8. It is considered a "balanced" counterflow heat exchanger, since the two heat transfer fluids possess the same heat capacity.

The temperature pattern of each heat transfer fluid is represented by two straight lines as shown in Fig. 4. The effectiveness expression E_{VV} is written as (Ref. 5):

$$E_{VV} = \frac{T_7 - T_8}{T_7 - T_e} = \frac{T_5 - T_e}{T_7 - T_e} \qquad (22)$$

from which the temperatures T_5 and T_8 are calculated:

$$T_5 = T_e + E_{VV}\,(T_7 - T_e) \qquad (23)$$

$$T_8 = T_7 - E_{VV}\,(T_7 - T_e) \qquad (24)$$

G. Vapor-liquid Heat Exchanger

This heat exchanger acts as a "feed water heater" from state 2 to state 3 by utilizing the exhaust steam leaving the vapor-vapor heat exchanger. The temperature pattern is sketched in Fig. 5. The heat exchanger is considered an "unbalanced" counterflow type since the heat capacities of the two heat transfer fluids are unequal. The effectiveness expression E_{VL} is based on the minimum heat capacity fluid (the vapor in this case) as

$$E_{VL} = \frac{(T_8 - T_9)}{(T_8 - T_c)} = \frac{C_{2,3}(T_3 - T_c)}{C_{8,9}(T_8 - T_c)} \qquad (25)$$

where $C_{2,3}$ is the mean specific heat of liquid water in the range of temperatures from T_2 to T_3 and $C_{8,9}$ is that for superheated steam in the range of temperatures T_8 to T_9. By taking $C_{2,3}$ as 1 kcal/kg°C, the temperatures T_3 and T_9 are then written as

$$T_9 = T_8 - E_{VL}\,(T_8 - T_c) \qquad (26)$$

$$T_3 = T_c + C_{8,9} \cdot (T_8 - T_c) \cdot E_{VL} \qquad (27)$$

In the computer model, the ceiling value of the temperature T_3 is assigned the evaporation temperature T_e. This means that if the temperature T_3, when computed from Eq. (27), exceeds the temperature T_e, then T_3 will be set equal to T_e as a maximum value. The enthalpy H_3 is calculated using Eq. (1).

H. Cycle Performance

The cycle performance characteristics and heat rates are determined as follows:

(1) The heat supplied by solar energy

$$Q_s = (H_4 - H_3)$$

$$Q_s = (597.8 + 0.417\, t_e) - t_3$$

or

$$Q_s = (757.05 + 0.417\, T_e - T_3) \qquad (28)$$

(2) The heat supplied by the external fuel for superheat

$$Q_{ex} = (H_6 - H_5)$$

$$Q_{ex} = C_{5,6}(T_6 - T_5) \qquad (29)$$

where $C_{5,6}$ is the mean specific heat between states 5 and 6 as determined from Eq. (5).

(3) Total heat added to the cycle

$$Q_{\text{add}} = Q_s + Q_{ex}$$

(4) The percentage of fuel heat added to total heat added

$$\lambda_s = Q_s/Q_{\text{add}}$$

(5) Net work done per unit mass of steam

$$W_{\text{net}} = (W_T - W_P)$$

(6) Rankine thermal efficiency

$$W_{\text{rankine}} = \frac{W_{\text{net}}}{Q_{\text{add}}}$$

(7) Specific Stream Consumption (SSC)

This is the amount of steam rate in kg/hr necessary to produce a net one horsepower output.

$$\text{SSC} = \frac{632}{W_{\text{net}}}, \quad \text{kg/hp hr}$$

(8) Carnot efficiency based on the cycle minimum and maximum temperatures T_c and T_6

$$E_{\text{Carnot}} = 1 - \frac{T_c}{T_6}$$

(9) Relative efficiency which measures the deviation from the ideal cycle

$$E_{\text{relative}} = E_{\text{rankine}}/E_{\text{carnot}}$$

V. Summary of Program Steps

The following is an ordered list of the program sequence with the necessary repetitive or conditional commands.

A. Input Data

A.1. Input temperatures

A.1.1 Evaporation temperature, t_e °C

A.1.2 Condenser temperature, t_c °C

A.1.3 Maximum superheat temperature, t_6 °C

A.2. Input component efficiency

A.2.1 Pump isentropic efficiency, E_p

A.2.2 Turbine isentropic efficiency, E_T

A.2.3 Vapor-vapor heat exchanger effectiveness, E_{VV}

A.2.4 Vapor-liquid heat exchanger effectiveness, E_{VL}

B. Pressure Calculations

B.1 $T_e = t_e + 273.15$

$T_c = t_c + 273.15$

$T_6 = t_6 + 273.15$

B.2 $$P_e = \text{antilog}\left[7.8553 - \frac{1555}{T_e} - \frac{11.2414}{(T_e/100)^2}\right] \text{ mm Hg}$$

B.3 $$P_c = \text{antilog}\left[7.8553 - \frac{1555}{T_c} - \frac{11.2414}{(T_c/100)^2}\right] \text{ mm Hg}$$

B.4 $$W_P = \frac{(P_e - P_c)}{31415\, E_P} \text{ , kcal/kg}$$

Next, either one of the following two approaches should be used to determine the isentropic turbine-end temperature.

C. Approach 1, by Ideal Gas Relations

C.1 Initiate a DO LOOP COMMAND

C.2 Set an initial value for $\gamma^{(0)} = 1.30$ or an initial value for the parameter $X = \gamma - 1/\gamma$ i.e., $X^{(0)} = 0.2308$ (based on $\gamma^{(0)} = 1.3$)

C.3 $$T_{7'} = T_6\left(\frac{P_c}{P_e}\right)^X \text{ , °K}$$

C.4 $C_{6,7'} = 0.3924 + 7.5 \times 10^{-5}\,(T_6 + T_{7'})$, kcal/kg K

C.5 $$X = \frac{0.11024}{C_{6,7}}$$ [This is an updated value for X]

[The gas constant for superheated steam = 0.11024 kcal/kg K]

C.6 GO TO STEP C.3. Either repeat the procedure three or four times only since the convergence is rapid, or repeat until two consecutive values of X differ by a preassigned error.

C.7 Exit from the loop with the final value of $T_{7'}$.

D. Approach 2, by Equating the Entropy

D.1 Initiate a DO LOOP COMMAND

D.2 Set an initial value of $T_{7'}$. This could be taken as T_6, i.e., $T_{7'}^{(0)} = T_6$

D.3 $$T_{7'} = \frac{T_6\left(\dfrac{T_e}{T_c}\right)^{1.54842}}{\exp\left[1929.3\left(\dfrac{1}{T_c} - \dfrac{1}{T_e}\right) - \left(\dfrac{T_6 + T_c - T_e}{2616}\right)\right]}$$

$$\times \exp\left(\frac{-T_{7'}}{2616}\right)$$

This is an updated value of $T_{7'}$.

D.4 GO TO STEP D.3. Either repeat the procedure six or seven times or repeat until two consecutive values of T_7' differ by a preassigned error.

D.5 Exit from the loop with the final value of T_7'.

E. Heat Exchangers and Cycle Performance

Having calculated T_7' by either approach (1) or (2), then

E.1 $T_7 = T_6 - E_T (T_6 - T_7')$, K

E.2 Conditional command: IF $T_7 < T_e$ STOP ELSE proceed.

E.3 $C_{6,7} = 0.3924 + 7.5 \times 10^{-5} (T_6 + T_7)$, kcal/kg K

E.4 $W_T = C_{6,7} (T_6 - T_7)$, kcal/kg

E.5 $T_5 = T_e + E_{VV} (T_7 - T_e)$, K

E.6 $T_8 = T_7 - E_{VV} (T_7 - T_e)$, K

E.7 $T_9 = T_8 - E_{VL} (T_8 - T_c)$, K

E.8 $C_{8,9} = 0.3924 + 7.5 \times 10^{-5} (T_8 + T_9)$, kcal/kg K

E.9 $T_3 = T_c + C_{8,9} \cdot E_{VL} (T_8 - T_c)$, K

E.10 Conditional Command. IF $T_3 > T_e$ THEN assign $T_3 = T_e$ ELSE proceed

E.11 $Q_s = 757.05 + 0.417\ T_e - T_3$, kcal/kg

E.12 $$Q_{ex} = 0.3924 (T_6 - T_5) + 7.5 \times 10^{-5} (T_6^2 - T_5^2), \text{ kcal/kg}$$

E.13 $Q_{\text{add}} = Q_s + Q_{ex}$, kcal/kg

E.14 $$\lambda_s = \frac{Q_s}{Q_{\text{add}}}$$

E.15 $W_{\text{net}} = W_T - W_P$, kcal/kg

E.16 $E_{\text{rankine}} = W_{\text{net}}/Q_{\text{add}}$

E.17 $E_{\text{carnot}} = 1 - (T_c/T_6)$

E.18 $E_{\text{relative}} = E_{\text{rankine}}/E_{\text{carnot}}$

E.19 $$\text{SSC} = \frac{632}{W_{\text{net}}} \text{ , kg/hp hr}$$

VI. Example of Calculations

The following numerical example is given to show the relative weight of each property involved in the calculations and to identify the order of execution steps.

A. Input Temperatures

$t_e = 90°\text{C}$, evaporation temperature

$t_c = 40°\text{C}$, condenser temperature

$t_6 = 600°\text{C}$, maximum cycle temperature

B. Input Component Efficiencies

$E_P = 0.60$, for the pump

$E_T = 0.75$, for the turbine

$E_{VV} = 0.80$, for the vapor-vapor heat exchanger

$E_{VL} = 0.80$, for the vapor-liquid heat exchanger

C. Program Steps

$T_e = 363.15$ K

$T_c = 313.15$ K

$T_6 = 873.15$ K

$P_e = 525.91$, mm Hg

$P_c = 55.37$, mm Hg

$W_P = 0.025$, kcal/kg

D. If Approach 1 Is Selected, Then

$X^{(0)} = 0.2308$

$T_7'^{(1)} = 519.34$ K

$C_{6,7}'^{(1)} = 0.4968$ kcal/kg K

$X^{(1)} = 0.2219$

$T_7'^{(2)} = 529.87$ K

$C_{6,7}'^{(2)} = 0.4976$

$X^{(2)} = 0.2215$

$T_7'^{(3)} = 530.29$ K

$C_{6,7}'^{(3)} = 0.4976$

$X^{(3)} = 0.2215$

Final temperature $\underline{T_7' = 530.29 \text{ K}}$

E. If Approach 2 Is Selected, Then

$$T_7' = 644.1140 \quad \exp\left(\frac{-T_7'}{2616}\right)$$

$T_7'^{(0)} = 873.15$

$T_7'^{(1)} = 461.32$

$T_7'^{(2)} = 539.98$

$T_7'^{(3)} = 523.98$

$T_7'^{(4)} = 527.20$

$T_7'^{(5)} = 526.55$

$T_7'^{(6)} = 526.68$

$T_7'^{(7)} = 526.65$

Final temperature $T_7'^{(8)} = 526.66$ K

The error between the two approaches is negligible in the final answers. Let us complete the program steps as if Approach 1 was chosen.

$T_7 = 616.00$ K, $\quad T_7 > T_e$

$C_{6,7} = 0.5041$, kcal/kg K

$W_T = 129.63$, kcal/kg

$T_5 = 565.43$ K

$T_8 = 413.72$ K

$T_9 = 333.26$ K

$C_{8,9} = 0.4484$, kcal/kg°C

$T_3 = 349.21$ K

Note $T_3 < T_e$

$Q_s = 559.27$, kcal/kg

$Q_{ex} = 153.95$, kcal/kg

$Q_{add} = 713.22$, kcal/kg

$\lambda_s = 0.7841$ $\left\{\begin{array}{l}\text{i.e., 78.41\% solar}\\ \text{21.59\% fuel}\end{array}\right.$

$W_{net} = 129.605$, kcal/kg

$E_{rankine} = 0.1817$ i.e., 18.17%

$E_{carnot} = 0.6414$ 64.14%

$E_{relative} = 0.2833$

SSC = 4.8764, kg/hp hr

A short computer program was written based on the above sequence of cycle calculations. Several runs were made with the condenser temperature and the evaporator temperature as parameters, keeping the component efficiencies and the maximum cycle temperature fixed. The results are plotted in Figs. 6 and 7 to form a performance map to help the system designer. Figure 6 shows that the thermal efficiency increases rapidly by increasing the evaporation temperature in this low range of working temperatures and results in a little decrease in the solar share λ_s. A variation of λ_s from 80 to 75 % occurs when the evaporation temperature changes from 80°C to 130°C at a condenser temperature of 40°C and the efficiency rises from 16 to about 24%.

The component efficiencies were arbitrarily selected to be as close as possible to practical values. The specific steam consumption plot in Fig. 7 was added in the analysis for sizing the components, steam and water lines for a given power output. A small value of the specific steam consumption, will result in lower equipment, operation, and maintenance cost. On the other hand, small values of specific steam consumption lie in the region of high thermal efficiency and small solar share resulting in more dependence on external fuel heat. A tradeoff would then have to be made for the selection of the optimum design point that solves the thermal efficiency versus operation costs problem.

Definition of Terms

a,b	specific heat constants
C	specific heat at constant pressure, kcal/kg°C
E_P	pump isentropic efficiency
E_T	turbine isentropic efficiency
E_{VL}	vapor-liquid heat exchanger effectiveness
E_{VV}	vapor-vapor heat exchanger effectiveness
H	specific enthalpy, kcal/kg
P	pressure, mm Hg
Q	heat rate
S	entropy, kcal/kg K
SSC	specific steam consumption, kg/hp hr
t	temperature, °C
T	absolute temperature, K
W	mechanical work, kcal/kg
λ_s	percentage of solar heat to total heat added

Subscripts

c	condenser
e	evaporation
ex	extra for superheat
f	liquid phase
g	gas phase
s	solar share

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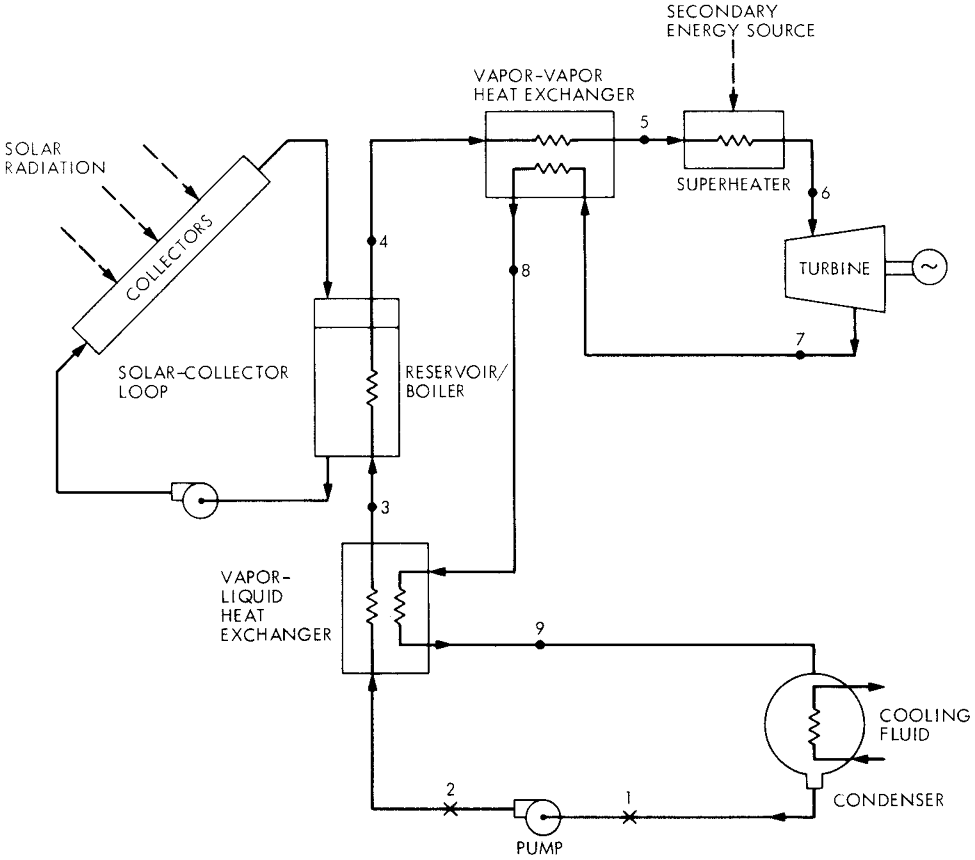

Fig. 1. Flow diagram of the solar-assisted regenerative Rankine cycle

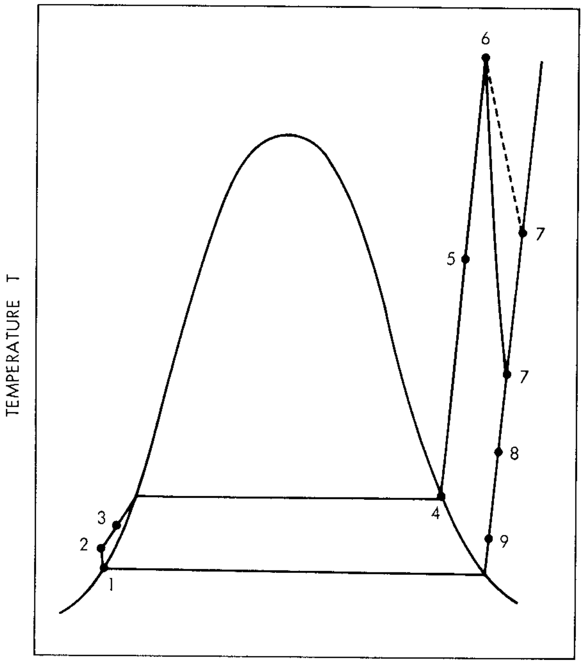

Fig. 2. Temperature entropy diagram for steam showing the states of the regenerative Rankine cycle (not to scale)

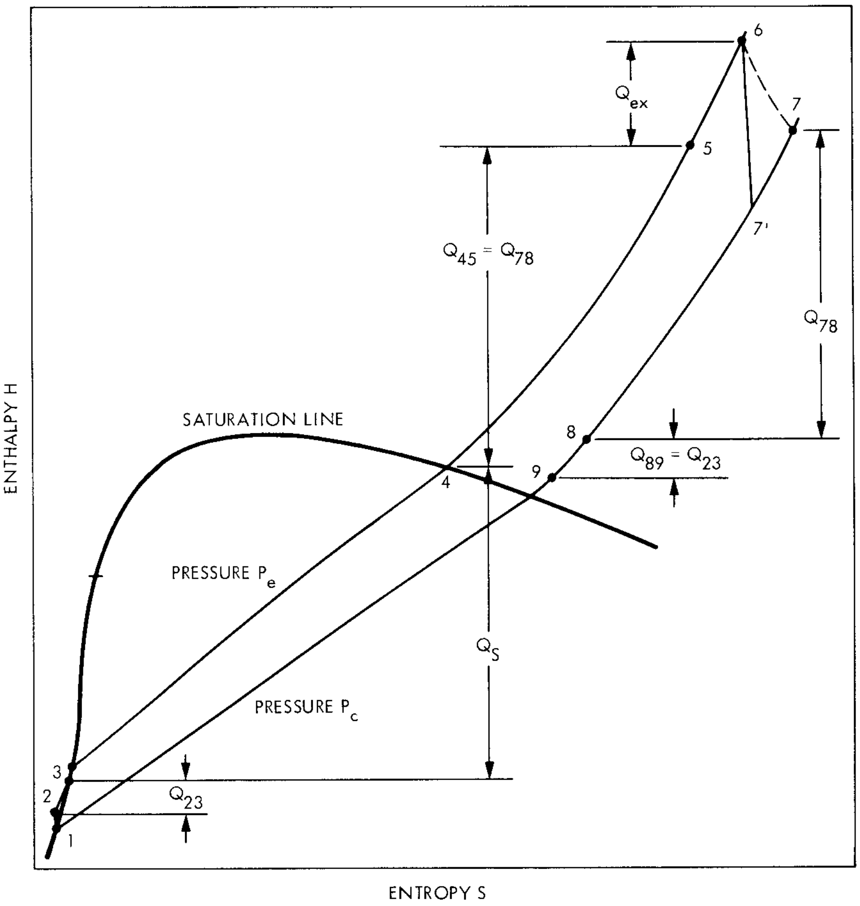

Fig. 3. Enphalpy-entropy (Mollier) diagram for steam showing the states of the regenerative Rankine cycle (not to scale)

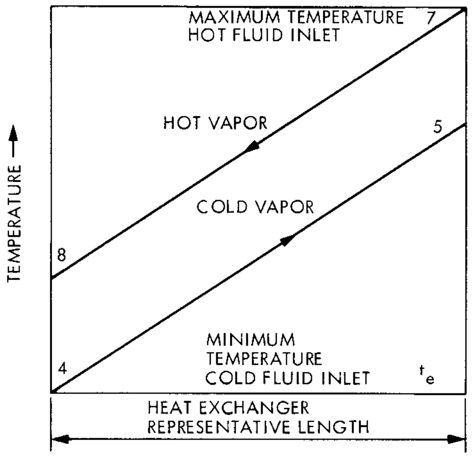

Fig. 4. Vapor-vapor heat-exchanger temperature pattern

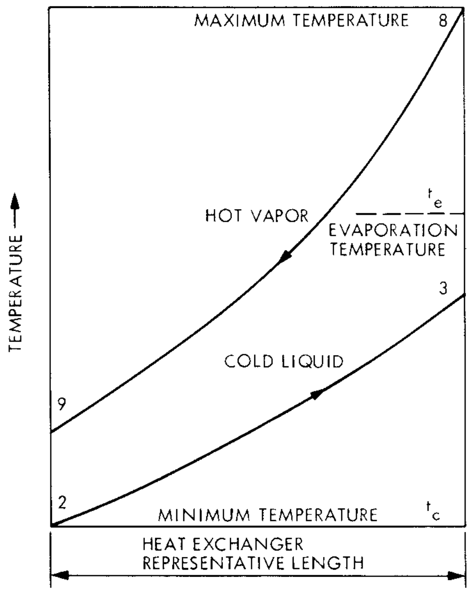

Fig. 5. Vapor-liquid heat-exchanger temperature pattern

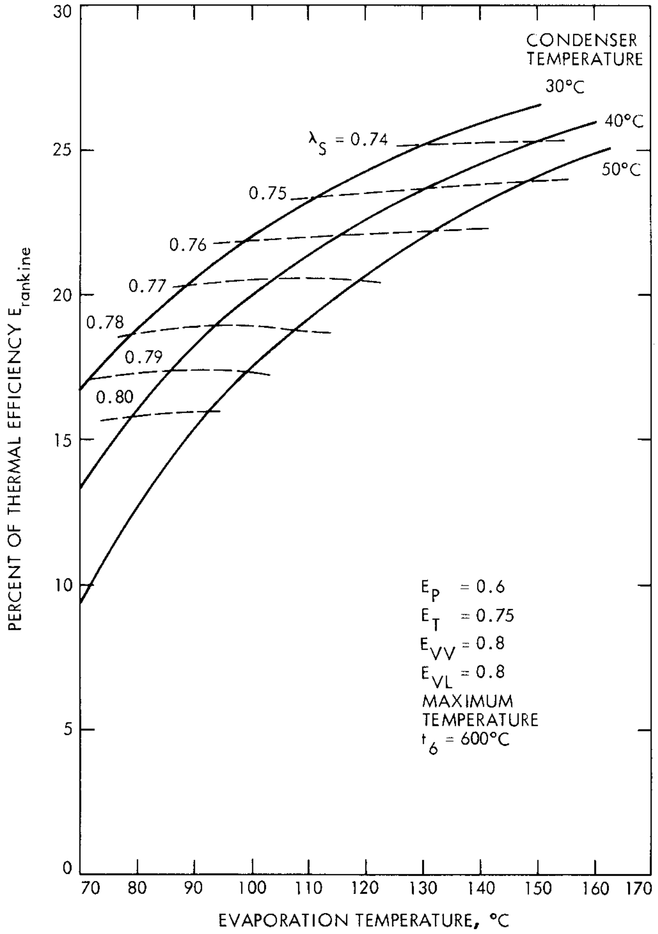

Fig. 6. Rankine efficiency and percentage of solar share versus the evaporation temperature

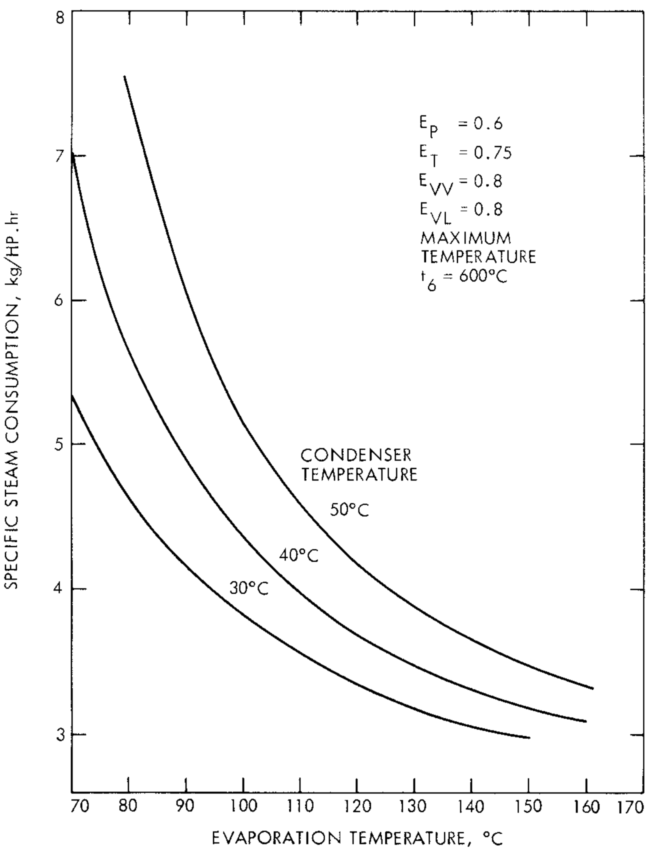

Fig. 7. Specific steam consumption versus the evaporation temperature for variable condenser temperatures