STEAM RANKINE CYCLE COOLING SYSTEM: 
ANALYSIS AND POSSIBLE REFINEMENTS

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Abstract—In the present communication, thermal modelling and energy conservation studies on the steam Rankine cycle cooling system [Rankine engine (RE) cycle coupled with vapour compression (VC) cycle] have been undertaken using steam + NH3, steam + R-22 and steam + R-12 as the working fluids, and their comparative studies are presented having parametric variations of various operating conditions. The overall performance is dependent on the thermal efficiency of the RE-cycle and the COP of the VC-cycle subsystems.

Subsequently, various methods of improvement of the efficiency of the Rankine engine cycle, viz. regeneration, superheating, recompression, etc. have been analysed quantitatively and their influences on Rankine engine efficiency performance are presented herewith.

It has been found that the steam Rankine cycle cooling system performance is higher than a freon based cooling system [1] and the Rankine cycle performance can also be further improved by using external superheating and/or recompression of the turbine exhaust. Numerical results are presented for the sake of illustration.

Steam Rankine cycle Regeneration Recompression Superheating and desuperheating Thermodynamic cycle analysis Thermal efficiency and COP Refinements

NOMENCLATURE

\[ \text{COP} = \text{Coefficient of performance} \]
\[ C_p = \text{Specific heat of refrigerant (kJ/kg °C)} \]
\[ E = \text{Effectiveness of regenerative heat exchanger} \]
\[ h = \text{Enthalpy of fluid (kJ/kg)} \]
\[ p = \text{Pressure (bar)} \]
\[ q = \text{Specific heat transfer rate (kJ/kg)} \]
\[ RCC = \text{Rankine cycle cooling} \]
\[ RE = \text{Rankine engine} \]
\[ RHE = \text{Regenerative heat exchanger} \]
\[ S = \text{Entropy (kJ/kg °C)} \]
\[ T = \text{Temperature (°C)} \]
\[ V-C = \text{Vapour compression} \]
\[ w = \text{Work (kJ/s)} \]

Subscripts
\[ ^b = \text{Boiler} \]
\[ ^f,l = \text{Fluid/liquid} \]
\[ ^g,v = \text{Gas/vapour} \]
\[ ^o = \text{Overall} \]
\[ V-C = \text{Vapour compression} \]
\[ R-C = \text{Rankine cycle} \]
\[ RCC = \text{Rankine cycle condenser} \]
\[ T = \text{Turbine} \]
\[ P = \text{Pump} \]
\[ VCC = \text{Vapour compression condenser} \]
\[ EVC = \text{Vapour compression evaporator} \]

1. INTRODUCTION

The Rankine heat engine cycle is commonly used in power plants as it can utilize waste heat and can also be utilized efficiently for cooling purposes [2-6]. However, as regards the efficiency of the
Rankine cycle, there is a considerable scope for improvement. A Rankine cycle heat engine coupled with a conventional vapour compression refrigeration cycle has also been considered as a promising solar cooling option [1]. The thermodynamic and technical feasibility of the organic Rankine cycle for electric power generation has already been shown by several investigations [2-9]. Generation of mechanical power by solar heat input has not yet been shown to be economical even in large scale systems, and hence, it is difficult to envision a small Rankine cycle cooling system. The problems associated with a solar Rankine cycle cooling system are many but one of the important facts is that the solar collection efficiency decreases as the operating temperature increases, while the Rankine engine efficiency increases with the source temperature.

The efficiency limitations and economic considerations of the Rankine cycle cooling system make the Rankine cycle suitable only for large scaling cooling capacity. The application of a steam Rankine cycle heat engine to the realm of solar air conditioning and power production is limited due to poor overall performance and high capital cost associated with it. The use of organic working fluids in the Rankine cycle cooling system is restricted by the physical and thermodynamic properties of the fluid and, to, from the environmental point of view. A relevant literature review [10-15] indicates that relatively little attention has been given to modelling of Rankine cycle heat engines driving a vapour compression refrigeration system from the point of view of energy conservation, especially using steam in the Rankine cycle and a refrigerant in the vapour compression cycle.

As the exhaust vapour temperature of the turbine is significantly higher than the temperature of condensation, a regenerator can be used to utilize the desuperheating of the exhaust vapour to preheat the fluid leaving the condenser before it enters the boiler.

Ways and means for the improvement of efficiency have also been proposed by various authors, but no concerted effort has been made by the authors towards a quantitative comparison between these refinements. In this communication, the authors have studied steam Rankine cycle cooling system using steam + NH₃, steam + R-22 and steam + R-12 as working fluids. Possible refinements and improvements in COP are also investigated from the point of view of energy conservation. Detailed parametric studies on Rankine cycle cooling systems have been presented, and comparative studies on possible refinements for process improvements in the Rankine engine efficiency have been undertaken. The results presented are useful for thermal design of the Rankine cycle cooling system.

2. RANKINE CYCLE SYSTEM OPTIONS

A simple Rankine cycle system schematic is shown in Fig. 1(a). Such a system uses a working fluid vapour taken from a boiler at elevated temperature and pressure and expanded through a

![Fig. 1. Schematic diagram of the Rankine engine cycle system and it's temperature-entropy (T-S) diagram.](image-url)
prime-mover/turbine to produce work and/or electricity. The expanded low pressure vapour is then condensed to a liquid and pumped back to the boiler where it is re-vaporized in a continuous process.

With the proper choice of components, particularly the working fluids, the Rankine cycle can be made to operate at efficiencies of 50% or greater of the Carnot efficiency over a broad range of temperatures. The thermodynamic properties of working fluids inherently contribute to the maximum possible theoretical efficiency of the cycle, and these thermodynamic properties, along with other physical properties of the fluid, determine the type and efficiency of the component hardware required.

Steam is, by far, the most important and common Rankine cycle working fluid and is currently being used in all major electrical power plants. Steam has many advantages: low cost, non-flammable, nontoxic, etc.

One of the most useful tools for visualizing and analysing the Rankine cycle is a temperature entropy (T–S) chart, Fig. 1(b). The rejected heat is represented by the labelled area on the chart. The overall efficiency is given by

$$\eta_0 = \frac{\text{net work}}{\text{input heat}} = \frac{\text{net work}}{\text{net work} + \text{rejected heat}}.$$  

It should be noted that the efficiency would be a maximum for any given high and low temperature combination if the net work area is a rectangle on the T–S chart, i.e. for a Carnot heat engine.

In order to analyse the cycle, the steady flow energy equation is applied to each component separately. If changes in kinetic and potential energies are neglected, the steady flow energy equation reduces to

$$q = w + \Delta h.$$
The pump work per unit mass flow of the working fluid is given by
\[ W_{\text{pump}} = (h_4 - h_3). \]

The pump work can also be obtained by the following equation
\[ w = -\int v \, dp. \]

Considering the fluid to be incompressible, the specific volume of saturated liquid corresponding to the condenser pressure can be taken from steam tables and is designated by \( v_f \) and
\[ W_{\text{pump}} = v_f (p_4 - p_3) / \eta_p. \]

The heat transferred to the working fluid in the boiler is given by \( q_{in, \text{boiler}} = (h_4 - h_3) \) and also \( p_4 = p_1 \).

Thus, the work output per unit mass flow from the turbine is given by
\[ W_{\text{out, turbine}} = (h_1 - h_2). \]

The heat rejection in the condenser is given by
\[ q_{out} = (h_2 - h_3) \quad \text{and} \quad p_2 = p_3. \]

Therefore,
\[ \text{Thermal efficiency} = \frac{W_{\text{turbine}} - W_{\text{pump}}}{q_{in}}. \]

When the pump work is neglected, the above equation reduces to
\[ \text{Thermal efficiency} = \frac{(h_1 - h_2)}{(h_1 - h_3)}. \]

The description and schematics of the various Rankine cycle system options are given in Figs 2-7 along with their \((T-S)\) diagrams. The individual descriptions are as follows:

(a) Figures 2(a) and (b) show the saturated steam Rankine cycle cooling system along with it's \(T-S\) diagram. The various thermodynamic processes that are taking place are as follows:

![Fig. 3. Schematic diagram of the Rankine cycle with a non-mixing type of regenerative heat exchanger and it's \((T-S)\) diagram.](image-url)
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Fig. 4. Schematic diagram of the Rankine cycle with internal superheating and it's (T–S) diagram.

In the Rankine engine cycle:
1–2 Isentropic expansion through the turbine
2–3 Constant temperature and constant pressure heat rejection in the condenser
3–4 Isenthalpic pumping process
4–1 Heat addition in the boiler.

In the V-C cycle:
5–6 Constant temperature and constant pressure heat rejection in the condenser
6–7 Isenthalpic expansion
7–8 Isothermal heat absorption (evaporation) in the evaporator
8–5 Actual compression in the compressor.

(b) Figures 3(a) and (b) show the presence of a non-mixing type of regenerative heat exchanger after the turbine outlet.
1–2 Isentropic expansion in the turbine
2–2' Heat loss of the condenser exit stream in the RHE
2'–3 Heat rejection in the condenser
3–4 Isenthalpic pumping process
4–4' Heat addition to the exit stream of the condenser
4'–1 Heat input to the boiler.

(c) Figures 4(a) and (b) show the internal superheating of the steam in the boiler. The various processes taking place in the cycle are the same as those taking place in the Rankine cycle part in the Rankine cycle cooling system.

(d) Figures 5(a) and (b) show the presence of a mixing type of regenerative heat exchanger (RHE) after the turbine.
1–2 Isentropic expansion in the turbine
2–3 Heat rejection in the condenser
3–4 Isenthalpic pumping
4–6 Mixing in the RHE
1–5 Bleeding in the turbine
5–6 Mixing in the RHE
6–7 Isenthalpic pumping
7–1 Heat addition in the boiler.

(e) Figures 6(a) and (b) show a Rankine cycle with regeneration and recompression where the turbine exhaust is compressed to a fixed compression ratio (30–40%).
1–2 Isentropic expansion in the turbine
2–2’ Isentropic compression
2’-5 Mixing in the RHE
2-3 Heat rejection in the condenser
3–4 Isenthalpic pumping
4–5 Mixing in the RHE
5–6 Isenthalpic pumping
6–1 Heat addition in the boiler.

(f) Figures 7(a) and (b) show the presence of a regenerator and an external superheater after the boiler in the Rankine cycle.

1–2 Isentropic expansion in the turbine
2–2’ Heat loss to the boiler exit stream
2’–3 Heat rejection in the condenser
3–3’ Isenthalpic pumping
3’–4 Heat addition in the boiler

Fig. 6. Schematic of the Rankine cycle using recompression after the turbine exhaust and its (T–S) diagram.
3. MODELLING OF THE RANKINE CYCLE COOLING SYSTEM AND REFINEMENTS

The general modelling procedure for the components of the Rankine cycle cooling system is well established [1] and described here as follows:

Turbine. The flow characteristics of a turbine used in the Rankine cycle can be represented by a single converging nozzle with frictional effects included through departure from isentropic flow equations. The actual expansion work done by the turbine may be given by

$$W_T = \eta_T \cdot M_{RC} \cdot (h_i - h_{in}) = M_{RC} \cdot (h_i - h_2)$$

where $M_{RC}$ is the mass flow rate of the Rankine cycle fluid, $h_i$ is the enthalpy of the fluid at the inlet to the turbine, $h_{in}$ is the isentropic enthalpy at the exit of the turbine, $h_2$ is the actual enthalpy at the exit of the turbine and $\eta_T$ is the isentropic efficiency of the turbine.

Pump. The pump work required to pump the Rankine cycle fluid from the lower condenser pressure to the boiler pressure can be given by

$$W_p = M_{RC} \cdot v_3 \cdot \Delta P / \eta_p$$

where $\eta_p$ is the pump efficiency.

Compressor. It is assumed that a reciprocating compressor is employed, and again, a non-isentropic compression process is considered through the isentropic efficiency of the compressor. Thus, the required compression work is given by

$$W_c = M_{VC} \cdot (h_{in} - h_i) \cdot \frac{1}{\eta_c} = M_{VC} \cdot (h_3 - h_6)$$

and the refrigerating effect is given by:

$$q_g = M_{VC} \cdot (h_3 - h_7).$$

Modelling of the various Rankine cycle options is given as follows.

(a) The diagrams 2(a) and (b) show the Rankine cycle cooling system without any modifications in it. In this system, the Rankine cycle turbine is coupled with the compressor of the vapour compression cycle and, thus, provides shaft power to it. The boiler conditions and the turbine
Fig. 8(a,b). Caption opposite.
efficiency are also known. The condenser temperature is assumed to be constant. Thus, $h_1$ is calculated at the boiler temperature, and $h_2$ is calculated as follows:

$$S_1 = S_2 = S_{26} + x_2 S_{26g}$$

or

$$x_2 = \frac{S_1 - S_{26}}{S_{26g}}$$

$$h_2 = h_{26} + x_2 h_{26g}.$$ 

Hence, using

$$\eta_1 = \frac{h_1 - h_2}{h_1 - h_1}.$$ 

$h_3$ can be calculated. $h_1$ can be computed from the condenser temperature and pressure and, taking the pump work to be negligible, $h_3 = h_4$

$$\eta_{R-C} = \frac{h_1 - h_2}{h_1 - h_4}.$$ 

For the V-C cycle:

Taking $h_1 = h_5$ at the condenser temperature and $h_6$ being calculated at the evaporator temperature and pressure, isentropic compression gives

$$S_e = S_5 + c_p \ln \frac{T_5}{T_6}.$$
From the above equation, \( T_5 \) can be calculated and the vapour enthalpy at that temperature can also be calculated. Let it be equal to \( h_\gamma \), then

\[
h_5 = h_\gamma + c_v(T_5 - T_\gamma).
\]

The V-C cooling cycle COP is given by:

\[
\text{COP} = \frac{h_\gamma - h_\delta}{h_\gamma - h_\eta}(\eta_c)
\]

where \( \eta_c \) is the efficiency of the compressor. So, the overall system COP of is given by

\[
\text{COP}_o = \eta_{R-C} \times \text{COP}_{V-C}.
\]

To improve the efficiency of the Rankine cycle cooling system, the efficiency of the Rankine cycle has to be improved, which means that there have to be some modifications in the Rankine cycle to enhance its efficiency.

(b) In Figs 3(a) and (b), a Rankine cycle with the presence of a non-mixing type of regenerative heat exchanger is shown. The regenerative heat exchanger has been put at the exhaust of the turbine and before the condenser. For this RHE, various effectiveness values were assumed, like \( E = 0.0, 0.6, 0.8 \) and 1.0. The effectiveness value of 0.0 signifies that no RHE was used. The enthalpy at point 1 is calculated using the boiler temperature and pressure.

The dryness fraction at point 2 is given by the isentropic expansion condition

\[
S_1 = S_2 = (1 - x_2)S_2 + x_2S_2,
\]

and the enthalpy \( h_2 \) is given by

\[
x_2 = \frac{h_3 - h_2}{h_{1g} - h_2}.
\]

So, the actual enthalpy is given by the relation

\[
\text{efficiency of the turbine} = \eta_T = \frac{h_1 - h_\gamma}{h_{1g} - h_2}.
\]

For the same dryness fraction, the enthalpy at state 4 is given by \( h_{4,0} = x_1h_4 + (1 - x_2)h_{4f} \). Now, the effectiveness of the regenerative heat exchanger is defined as

\[
E = \frac{h_2 - h_\gamma}{h_2 - h_{4,0}}.
\]

From the above equation, \( h_2 \) can be calculated. Heat exchange through the RHE is given by

\[
h_{4L} - h_{4L} = h_{3} - h_{2s}.
\]

from which \( h_{4L} \) can be calculated.

The efficiency of the Rankine cycle is given by

\[
\eta_{R-C} = \frac{h_1 - h_\gamma}{h_1 - h_\delta}.
\]

(c) Figures 4(a) and (b) show yet another way of increasing the efficiency of the Rankine cycle, i.e. to superheat the steam entering the turbine. The following calculation procedure was adopted. Knowing \( h_1 \) from the tables, \( h_\gamma \) is calculated from the isentropic condition:

\[
S_1 = S_2 = S_2 + x_2S_{2f}.
\]

\( x_2 \) can be calculated from the above equation and can be used in the following equation

\[
h_\alpha = h_\gamma + x_2h_{2f}.
\]

The turbine efficiency \( \eta_T = h_1 - h_\gamma/h_1 - h_\delta \).
Now, inserting the value of \( h_3 \), we can calculate \( h_2 \). The efficiency of the Rankine cycle is given by
\[
\eta_{R-C} = \frac{h_1 - h_2}{h_1 - h_4}.
\]

(d) In the effort to increase the efficiency of the Rankine cycle, the calculations were done for the Rankine cycle having a mixing type of RHE incorporated in it [refer to Figs 5(a) and (b)]. In the diagram, it can be seen that a part of the steam is bled off from the turbine which goes to the RHE. The bled off steam is then mixed with the steam coming from the condenser and, thus, raises the enthalpy of the condenser leaving fluid. Assuming a fixed pressure ratio (i.e. \( p_3/p_4 \)), the calculation is as follows:

Referring to the \( T-S \) diagram shown in Fig. 5(b)
\[
S_1 = S_2 = S_M + x_3 S_{fg}.
\]

The efficiency of the turbine is given by
\[
\eta_T = \frac{h_1 - h_3}{h_1 - h_M}.
\]

From the above equation, \( h_3 \) can be calculated. Also, we have
\[
S_1 = S_3 = S_M + S_{fg} x_2
\]
from which \( x_2 \) can be calculated, and then,
\[
h_3 = h_M + h_{fg} x_2.
\]

Taking \( h_3 = h_4 \) and \( h_6 = h_7 \) for the isenthalpic pumping process, the efficiency of the turbine is given by
\[
\eta_T = \frac{h_1 - h_2}{h_1 - h_M}
\]
from which \( h_2 \) can be calculated.

Assuming unit mass flow of the fluid in the cycle and \( m \) units of steam are bled off from the turbine, the energy balance for the mixing type RHE gives
\[
m = \frac{h_4 - h_1}{h_3 - h_4}.
\]

The efficiency of the Rankine cycle is given by
\[
\eta_{R-C} = \frac{h_1 - h_5 + (1 - m)(h_5 - h_2)}{(h_1 - h_4)}.
\]

(e) The other option being studied here is recompression of part of the turbine outlet to a particular pressure and mixing the outlet of the compressor with the condenser outlet as shown in Figs 6(a) and (b). The conditions of the boiler are known and, thus, \( h_1 \) can be calculated. Assuming a pressure ratio \( p_1/p_2 \), \( p_2 \) can be calculated because \( p_2 \) is known once \( h_2 \) is known.
\[
S_1 = S_{1'} = S_M + x_2 S_{fg}.
\]

From the above equation, \( x_2 \) can be calculated and substituted in the following equation.
\[
h_1' = h_M + x_2 h_{fg}.
\]

The efficiency of the turbine is given by
\[
\eta_T = \frac{h_1 - h_2}{h_1 - h_1'}
\]
from which, \( h_2 \) and \( p_2 \) can be calculated.
\[
S_2 = S_{1'} = S_M + x_3 S_{fg}.
\]
Substituting the value of \( x_{21} \) in the following equation,
\[ h_{21} = h_{21t} + x_{21} h_{2t} \]
\( h_{21} \) can be calculated and substituted in the following equation,
\[ \eta_c = \frac{h_{21} - h_2}{h_{21} - h_2} \]
from which \( h_2 \) can be calculated.
Assuming unit mass flow rate of the fluid in the cycle and if \( m \) is the mass of the fluid being recompressed, then the energy balance applied to the RHE is given by
\[ mh_2 + (1 - m)h_4 = h_5 \quad \text{or} \quad m = \frac{h_5 - h_4}{h_2 - h_4} \]
The efficiency of the modified Rankine cycle is calculated by:
\[ \eta_{R-C} = \frac{(h_1 - h_2) - m(h_2 - h_2)}{(h_1 - h_5)} \]
(f) The Rankine cycle efficiency can also be improved by incorporating a regenerator and a superheater as shown in diagrams 7(a) and (b). In the diagrams, it can be seen that the regenerator is used to heat the stream coming from the boiler by the turbine outlet exhaust stream. The superheater does external superheating on the exit stream of the regenerator. Assuming a degree of superheat and adding it to the boiler temperature, the conditions at state point \( t \) are known.
\( h_1 \) is known. The dryness fraction at state point \( 2(x_{21}) \) is given by the isentropic flow condition, viz.
\[ S_1 = S_{2t} + x_1 S_{2t} \]
and the enthalpy \( h_{21} \) is given by the following equation:
\[ h_{21} = h_{21t} + x_1 h_{21t} \]
The efficiency of the turbine is given by
\[ \eta_T = \frac{h_1 - h_2}{h_1 - h_{21}} \]
from which \( h_2 \) can be calculated.
\( h_4 \) is known from the conditions of the boiler and \( h_3 \) is known from the conditions of the condenser. The effectiveness of the regenerator is defined as:
\[ E = \frac{(h_5 - h_4)}{h_2 - h_4} = \frac{(h_5 - h_2)}{h_1 - h_4} \]
from which \( h_4 \) can be calculated and also the temperature at that corresponding state point. The temperature rise in the regenerator in the stream coming from the boiler is subtracted from the assumed degree of superheat in the superheater to give the effective and consistent degree of superheat. The efficiency of the Rankine cycle is given by
\[ \eta_{R-C} = \frac{(h_1 - h_2)}{(h_4 - h_3) + (h_1 - h_3)} \]

4. MODELLING RESULTS AND DISCUSSION

Performance evaluation of the Rankine cycle cooling system and the evaluation of the methods of improvement of the Rankine cycle efficiency can be carried out from the modelling studies. In the Rankine cycle cooling system using steam in the Rankine cycle with NH\(_3\), R-22 and R-12 in the V-C cycle, thermal modelling has been carried out, and the operating parameters are as follows:

- Boiler or inlet turbine temperature \((T_b) = 80-100^\circ\text{C}\)
- Rankine cycle condenser temperature \((T_{REC}) = 45-55^\circ\text{C}\)
- V-C cycle condenser temperature \((T_{VCC}) = 45-55^\circ\text{C}\)
V-C cycle evaporator temperature \( (T_{\text{evc}}) = 0-10^\circ \text{C} \)

Turbine efficiency \( \eta_T = 0.7-0.85 \)

Compressor efficiency in the V-C cycle \( \eta_C = 0.7-0.85 \).

The specific heats of the various fluids used are given below:

<table>
<thead>
<tr>
<th>Fluid</th>
<th>( C_v ) (kJ/kg °C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>NH(_3)</td>
<td>3.62</td>
</tr>
<tr>
<td>R-22</td>
<td>0.77</td>
</tr>
<tr>
<td>R-12</td>
<td>0.76</td>
</tr>
</tbody>
</table>

The effectiveness of the regenerative heat exchanger used is taken to be 0.0-1.0.

The effect of varying each of the above parameters, while keeping the other parameters constant at optimum value, have been studied. The operating range of variation of each of the parameters is assumed to be in the practical range. The parametric variations have been studied from the graphs, which show the trends of the overall system efficiency (efficiency of the Rankine cycle multiplied by the COP of the V-C cycle) with the changing parameters. It must be stated that \( E = 1.0 \) corresponds to the ideal regenerative heat exchanger while \( E = 0.0 \) corresponds to the absence of the RHE.

Figure 8(a) gives the variation of the overall efficiency with the evaporator temperature and the compressor efficiency of the vapour compression cycle. It can be invariably seen that, with the increase of the evaporator temperature, the overall efficiency of the system increases. The increase in the overall efficiency is the maximum in the case of steam + NH\(_3\) and when the regenerative heat exchanger is present in the Rankine cycle. With the increase in the compressor efficiency, the overall efficiency increase is the maximum as far as the steam + NH\(_3\) combination is concerned.

Figure 8(b) shows the change of overall efficiency with a change of the condenser temperature of the Rankine cycle and the vapour compression cycle. It can be seen that the decrement in the overall efficiency with a change in the Rankine cycle condenser temperature is slowed down when the regenerative heat exchanger is present in the Rankine cycle. The overall efficiency in the case of the cooling system without the RHE is reduced by nearly half of its value. The same trend can be seen in the case of the change in the condenser temperature of the V-C cycle. It can be seen from these graphs that the steam + NH\(_3\) combination gives a better performance.

Figure 8(c) shows the change of overall efficiency with a change in turbine efficiency and the turbine inlet temperature of the Rankine cycle. It can be seen that, with the RHE, there is an improvement of around 20% in the overall efficiency, keeping the other parameters the same. The other two curves show the change in the overall efficiency with the change in the turbine inlet

Table 1. Comparative assessment of three processes for improving the efficiency of the Rankine cycle under saturated conditions: effect of boiler temperature, turbine efficiency and condenser temperature on Rankine cycle efficiency \((\eta_{\text{R,C}})\)

<table>
<thead>
<tr>
<th>Process</th>
<th>Boiler temp (°C)</th>
<th>Turbine efficiency</th>
<th>Condenser temp. (°C)</th>
<th>( \eta_{\text{R,C}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Presence of a mixing</td>
<td>80</td>
<td>0.080</td>
<td>0.75</td>
<td>0.080</td>
</tr>
<tr>
<td>type of RHE with bleeding from the</td>
<td>85</td>
<td>0.087</td>
<td>0.78</td>
<td>0.083</td>
</tr>
<tr>
<td>turbine, ( p_1/p_0 = 0.8 )</td>
<td>90</td>
<td>0.090</td>
<td>0.81</td>
<td>0.086</td>
</tr>
<tr>
<td>Recompression of the</td>
<td>80</td>
<td>0.084</td>
<td>0.75</td>
<td>0.084</td>
</tr>
<tr>
<td>turbine outlet using</td>
<td>85</td>
<td>0.089</td>
<td>0.78</td>
<td>0.099</td>
</tr>
<tr>
<td>a compressor of a fixed</td>
<td>90</td>
<td>0.095</td>
<td>0.81</td>
<td>0.106</td>
</tr>
<tr>
<td>pressure ratio, ( p_1/p_2 = 0.3 )</td>
<td>95</td>
<td>0.120</td>
<td>0.84</td>
<td>0.110</td>
</tr>
<tr>
<td>Presence of a non-mixing</td>
<td>80</td>
<td>0.079</td>
<td>0.75</td>
<td>0.079</td>
</tr>
<tr>
<td>type of RHE after the</td>
<td>85</td>
<td>0.085</td>
<td>0.78</td>
<td>0.082</td>
</tr>
<tr>
<td>turbine</td>
<td>90</td>
<td>0.089</td>
<td>0.81</td>
<td>0.085</td>
</tr>
<tr>
<td></td>
<td>95</td>
<td>0.098</td>
<td>0.84</td>
<td>0.088</td>
</tr>
</tbody>
</table>

\( \eta_{\text{R,C}} = \text{Rankine cycle thermal efficiency.} \)
Table 2. Comparison between various refined processes for improving the efficiency of the Rankine cycle using superheated steam

<table>
<thead>
<tr>
<th>Process</th>
<th>Boiler temp. (°C)</th>
<th>Turbine efficiency</th>
<th>Condenser temp. (°C)</th>
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<td>( \eta_{r} )</td>
<td>( \eta_{r,c} )</td>
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</tr>
<tr>
<td></td>
<td>375</td>
<td>0.185</td>
<td>0.84</td>
</tr>
<tr>
<td>External superheating</td>
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<td>0.185</td>
<td>0.75</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.188</td>
<td>0.78</td>
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<td>0.75</td>
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<td></td>
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Pressure = 6 bar

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Pressure = 8 bar

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<td>( \eta_{r} )</td>
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<tr>
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The temperature of the Rankine cycle. It can be found from the graphs that the overall efficiency of the Rankine cooling system is improved by (10-15%) with the installation of the regenerative heat exchanger in the Rankine engine cycle. It can also be seen that the steam + NH₃ combination is best suited for cooling purposes in the parameter ranges considered in the present calculations. It can be mentioned that, for a Rankine cycle cooling system, the performance can be further improved by incorporating some refinements, viz. regeneration, superheating, etc. in the Rankine engine cycle while the cooling temperature desired and the surrounding sink temperatures are fixed. Thus, the cooling performance can be improved by improving the performance of the Rankine heat engine using and/or incorporating a regenerative heat exchanger, mixed regeneration and recompression, etc. Thermodynamic evaluation of these options has been carried out for the Rankine engine.

Table 1 gives the comparative assessment of various processes for improving the efficiency of the Rankine cycle. The three processes considered are:

1. Presence of a mixing type regenerative heat exchanger with bleeding from the turbine.
2. Recompression of the turbine outlet, using a compressor, to a fixed pressure ratio.
3. Presence of a non-mixing type of RHE after the turbine exhaust.
It can be seen that, with the change in the turbine inlet temperature the change in the efficiency of the Rankine cycle is from 10 to 20% in process 2 and, with the same range of turbine efficiency, the increase in the Rankine cycle efficiency in process 2 is from 10 to 25%. The reduction in efficiency is less in the case of process 2 in the case of increase in the condenser temperature of the Rankine cycle. As regards the non-mixing type of RHE, the effectiveness of the RHE was taken to be 0.6 but the performance of this process (i.e. process 3) was not comparable to the other two processes. It can be seen that the recompression of the turbine outlet gives a better Rankine cycle efficiency as compared to the efficiencies achieved with the presence of a mixing type and a non-mixing type of RHE.

Referring to Table 2 which gives the comparative assessment of the three processes for the improvement of the efficiency of the Rankine cycle at different saturation pressure levels, i.e. 4, 6 and 8 bar, respectively, the three processes are:
(1) Internal superheating in the boiler.
(2) External superheating with a superheater and a regenerator before the superheater.
(3) Recompression in the turbine exhaust.

In process 3, the recompression ratio of \( p_3/p_2 \) was taken to be 0.3. It can be seen that, over the same temperature range, the increment in the efficiency of process 3 was from 5 to 20%. With the change in the turbine efficiency, the increase in the efficiency of the Rankine cycle in process 3 is of the order of 10% in the case when the pressure is 4 bar and higher, around 20%, in the case of higher pressures, i.e. 6 and 8 bar. In the case of other processes, also, there is an increase in efficiency but the quantum of increase in process 3 is more. The increase in efficiency is due to the increase in the enthalpy after recompression in the compressor, and thus, the increase in enthalpy of the steam in the mixing type RHE is more. Another notable feature which can be seen from the tables is that, with the increase in the pressure levels, the efficiency in all three processes increases but the increase is the maximum (11%) in the case of process 3 as compared to 6% in process 1.

It can also be seen that external superheating is a better option as compared to internal superheating because it gives higher efficiencies in the Rankine cycle. The above results hold good for other parametric changes, i.e. the change in the turbine efficiency and the change in the condenser temperature. It can be seen that, of all the three processes that have been studied, the process of recompression is the best to improve the efficiency of the Rankine cycle cooling system.

Table 3(a) gives the quantitative assessment of the change in the Rankine cycle efficiency with a change in the recompression ratio of the process of recompression of the turbine outlet. It can be seen that the higher recompression ratio gives a lesser Rankine engine cycle efficiency. The enthalpy increase in the case of the lesser compression ratio is more as compared to higher compression ratios, which accounts for the higher efficiency.

Table 3(b) shows the effect of \( p_1/p_2 \) on the ratio of the pressure at the inlet of the turbine to the pressure at which the steam is bled off from the turbine. It can be seen that, with the increase of

<table>
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<th>p_2/p_2</th>
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<th>( \eta_{R,C} )</th>
<th>( T_2 ) (°C)</th>
<th>( \eta_{R,C} )</th>
<th>( T_3 ) (°C)</th>
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<th>( T_4 ) (°C)</th>
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</table>

<table>
<thead>
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<th>p_3/p_1</th>
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<th>( \eta_{R,C} )</th>
<th>( T_3 ) (°C)</th>
<th>( \eta_{R,C} )</th>
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<td>0.082</td>
<td>85</td>
<td>0.088</td>
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<td>0.100</td>
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</table>
this ratio, the efficiency of the cycle increases because, in that case the enthalpy of the bled off steam is tending towards the enthalpy of the turbine inlet steam, which signifies the increase in the enthalpy which shows an increase in the heat input without much affecting the work obtained from the turbine.

5. CONCLUSIONS

Thermodynamic analysis and thermal modelling of the Rankine cycle cooling system with and without a RHE have been carried out for a set of three working fluid combinations, i.e. steam + NH₃, steam + R-22 and steam + R-12. It can be seen that the combination (steam + NH₃) is the best amongst these fluids with the overall efficiency greater than the other fluids by nearly 10–20%. This efficiency increase is further enhanced with the installation of the regenerative heat exchanger in the Rankine cycle.

Further improvement in the overall efficiency can be made by incorporating a regenerative heat exchanger in the vapour compression cycle of the system. It can also be seen that the effect of the vapour compression cycle condenser is more pronounced as compared to the Rankine cycle condenser. Again, the effect of the evaporator temperature on the overall efficiency is more pronounced as compared to the effect of the boiler temperature. The various processes for the improvement of the efficiency of the modified Rankin cycle were studied, and it was found that the efficiency of the Rankine cycle was improved over a simple Rankine cycle (10–20%). It can be seen that superheating the turbine inlet and then recompressing the exhaust of the turbine would be the best option for the improvement of the efficiency of the Rankine cycle. Solar heat could be used for taking the boiler temperature to the saturated level.

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REFERENCES