Thermodynamic Analysis and Multi-Objective Optimizations of a Combined Recompression \( \text{SCO}_2 \) Brayton Cycle-TCO\(_2\) Rankine Cycles for Waste Heat Recovery

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Received 05 March 2018, Accepted 08 May 2018, Available online 12 May 2018, Vol.8, No.3 (May/June 2018)

Abstract

A thermodynamic analysis and optimization of a newly-conceived combined power cycle were conducted in this paper for the purpose of improving overall thermal efficiency of power cycles by attempting to minimize thermodynamic irreversibilities and waste heat as a consequence of the Second Law. The power cycle concept comprises a topping advanced recompression supercritical carbon dioxide (sCO\(_2\)) Brayton cycle and a bottoming transcritical carbon dioxide (tCO\(_2\)) Rankine cycle. The bottoming cycle configurations included a simple tCO\(_2\) Rankine cycle and a split tCO\(_2\) Rankine cycle. The topping sCO\(_2\) recompression Brayton cycle used a combustion chamber as a heat source, and waste heat from a topping cycle was recovered by the tCO\(_2\) Rankine cycle due to an added high efficiency recuperator for generating electricity. The combined cycle configurations were thermodynamically modeled and optimized using an Engineering Equation Solver (EES) software. Simple bottoming tCO\(_2\) Rankine cycle cannot fully recover the waste heat due to the high exhaust temperature from the top cycle, and therefore an advanced split tCO\(_2\) Rankine cycle was employed in order to recover most of the waste heat. Results show that the highest thermal efficiency was obtained with recompression sCO\(_2\) Brayton cycle – split flow tCO\(_2\) Rankine cycle. Also, the results show that the combined CO\(_2\) cycles is a promising technology compared to conventional cycles.

Keywords: Rankine cycle, Brayton cycle, sCO\(_2\), tCO\(_2\), Engineering Equation Solver etc.

Introduction

The unprecedented growth in the world population and economic activity, along with rising concerns about environmental issues, mean that energy efficiency will play a vital role in the development of future energy systems. Motivated by limited energy resources, the accelerating growth of energy demand, cost, and growing environmental concerns, there has been a focus on improving such poor energy production efficiency.


Sarker provides an organized review of tCO\(_2\) Rankine cycle configurations from the literature, focusing on low-grade heat supplies, and he provides a performance comparison with other working fluids. He finds that the tCO\(_2\) Rankine cycle has clear advantages to steam and organic Rankine cycles (ORC), and he discusses pathways to developing aspects of this cycle (parameter optimization, hardware components, control strategies, etc.). Wang and Dai compared the exergoeconomic performance for two bottoming cycles (tCO\(_2\) and ORC) designed to optimize waste heat recovery from a sCO\(_2\) recompression Brayton topping cycle. Parametric optimization indicates that the tCO\(_2\) bottoming cycle has superior performance at lower pressure ratio (PRc) (off-design conditions), and that higher turbine inlet temperatures improve tCO\(_2\) exergoeconomic performance, unlike the ORC. Both combined cycles have similar second-law efficiency, and the ORC was shown to have a slightly lower total product unit cost. Yari and Sirousazar developed a tCO\(_2\) cycle for recovering waste heat from the pre-cooler of a sCO\(_2\) Brayton cycle, and they modeled the performance improvement for this new combined cycle relative to that of a simple sCO\(_2\) cycle. The authors reported that their new system improved the first and second law
efficiencies by 5.5%, to 26%, and that it reduced exergy destruction by 6.7%, to 28.8%. Chen et al. compared the performance of two cycles act as a bottoming cycle to extract useful work from low-grade waste heat. The ORC is most commonly used, but the authors found that the tCO₂ power cycle showed better performance. Specifically, this cycle had a slightly higher power output than ORC, and it did not have a pinch limitation in the heat exchanger.

According to the literature that shown in Table 1, most research in sCO₂ cycles used two ways to represent heat exchanger performance: using fixed heat exchanger effectiveness or pinch point temperature. However, due to CO₂ properties, assuming a constant recuperator effectiveness - minimum temperature approach leads to markedly different conductance values in heat exchanger size and consequently cost. The first contribution in this study is developing a computationally efficient technique to design heat exchangers by using constant conductance (UA) to represent heat exchanger performance and thereby deliver improved accuracy in calculations. The second contribution in this research is the newly-conceived combined power cycle is proposed.

### Nomenclature

- **sCO₂**: Supercritical Carbon Dioxide
- **tCO₂**: Transcritical Carbon Dioxide
- **ORC**: Organic Rankine Cycle
- **HTF**: Heat Transfer Fluid
- **PR**: Pressure Ratio
- **EES**: Engineering Equation Solver
- **UA**: Heat Exchanger Conductance
- **Cp**: Specific Heat
- **HTR**: High Temperature Recuperator
- **LTR**: Low Temperature Recuperator
- **LMTD**: Log-Mean Temperature Difference
- **PCHE**: Printer Circuit Heat Exchangers
- **TIT**: Turbine Inlet Temperature
- **CIT**: Compressor Inlet Temperature
- **WHR**: Waste Heat Recovery
- **GA**: Genetic Algorithm

### Mathematical Modeling Approach

Carbon dioxide (CO₂) properties sharply vary at and near critical points (T=30.98 °C & P=7.38 MPa). Illustrates the instability of CO₂’s specific heat (Cp) (the amount of thermal energy needed to raise the temperature of a system by 1 °C per unit of mass) near the critical point, presenting a variety of temperatures and pressures. Despite the sCO₂ advantages of reducing the compressor work due to high density near critical points, the thermodynamic characteristics fluctuate wildly, thereby influencing the CO₂ properties. The sharp alteration in pressure and temperature near critical point makes specific heat an impractical measure. As CO₂ properties fluctuate near the critical point, design difficulties with turbomachinery and heat exchangers arise [20]. So, following the conventional method that assume a constant capacity to deal with a heat exchanger whose properties change arbitrarily near the critical point is invalid (K. Gregory et al, 2009). An adjustment needs to be made in order to reuse the conventional equations, which are described in detail in the heat exchanger model in the next section.

### Table 1 Literature input parameters and combined power cycle efficiency

<table>
<thead>
<tr>
<th></th>
<th>Wang</th>
<th>Akbari</th>
<th>Yari</th>
<th>Wang</th>
<th>Besarati</th>
<th>Pichel</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Top cycle</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>sCO₂</td>
<td>32</td>
<td>35</td>
<td>35</td>
<td>32</td>
<td>55</td>
<td>30</td>
</tr>
<tr>
<td>tCO₂ ORC Isopentane</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Bottom cycle</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>sCO₂</td>
<td>7.4</td>
<td>7.4</td>
<td>7.4</td>
<td>8</td>
<td>-</td>
<td>7.4</td>
</tr>
<tr>
<td>tCO₂ ORC Isopentane</td>
<td>20.72</td>
<td>22.2</td>
<td>24.3386</td>
<td>20</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td><strong>Maximum pressure [MPa]</strong></td>
<td>550</td>
<td>650</td>
<td>550</td>
<td>800</td>
<td>500</td>
<td></td>
</tr>
<tr>
<td><strong>Main compressor inlet temperature [°C]</strong></td>
<td>86%</td>
<td>86%</td>
<td>86%</td>
<td>95%</td>
<td>95%</td>
<td>95%</td>
</tr>
<tr>
<td>HTR effectiveness [-]</td>
<td>90/70</td>
<td>90/87</td>
<td>90/80</td>
<td>90/85</td>
<td>90/87</td>
<td>93/85</td>
</tr>
<tr>
<td>LTR effectiveness [-]</td>
<td>85/80</td>
<td>85/80</td>
<td>85/80</td>
<td>89/85</td>
<td>89/85</td>
<td>89/85</td>
</tr>
<tr>
<td>Compressor/Pump efficiency [%]</td>
<td>negligible</td>
<td>negligible</td>
<td>negligible</td>
<td>negligible</td>
<td>50 kPA/HX</td>
<td></td>
</tr>
<tr>
<td>Pressure drop [-]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Combined Cycle efficiency</strong></td>
<td>0.449</td>
<td>0.4523</td>
<td>0.4422</td>
<td>0.49</td>
<td>0.4672</td>
<td>0.5433</td>
</tr>
<tr>
<td></td>
<td>0.435</td>
<td></td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

![Figure 1 Specific heat variation at different temperatures and pressures](image-url)
In this study, two combined CO\textsubscript{2} power cycles are subjected to thermodynamic analysis and optimization in order to improve calculation accuracy and to improve the cycle efficiency and power output. With respect to improving the accuracy of the analytical model, a computationally efficient technique using constant conductance (UA) to represent heat exchanger performances is executed. The cycles involved will be 1) a top sCO\textsubscript{2} recompression Brayton cycle with a bottom tCO\textsubscript{2} split flow Rankine cycle - called cycle I (Figure 2a) and 2) a top sCO\textsubscript{2} recompression Brayton cycle with a bottom tCO\textsubscript{2} simple Rankine cycle – called cycle II (figure 2b).

\begin{equation}
\eta_c = \frac{h_{\text{out}} - h_{\text{in}}}{h_{\text{out, isentropic}} - h_{\text{in}}}
\end{equation}

where \( h_{\text{in}} \) and \( h_{\text{out}} \) are the actual inlet and outlet enthalpies, respectively, and \( h_{\text{out, isentropic}} \) is the isentropic outlet enthalpy. Two properties at any state are sufficient to calculate the others properties in the same state. The inlet turbine and compressor temperature and pressure are assumed, while taking into consideration the pressure drop in the cycle. With known two-inlet turbomachinery properties and one-outlet turbomachinery properties, the model obtains the turbomachinery outlet properties using equations (3) and (4):

\begin{equation}
s_{\text{in}} = s_{\text{out, isentropic}}
\end{equation}

\begin{equation}
h_{\text{out, isentropic}} = f(P_{\text{out}}, s_{\text{out, isentropic}})
\end{equation}

Where \( s_{\text{in}} \) and \( s_{\text{out, isentropic}} \) are the inlet actual specific entropy and outlet isentropic specific entropy respectively. After specific isentropic outlet enthalpy is calculated in equation the actual enthalpy can be obtained using the isentropic turbomachinery efficiencies.

The specific actual work can be calculated using equation (5)

\begin{equation}
w = h_{\text{in}} - h_{\text{out}}
\end{equation}

**Heat exchanger**

The conventional techniques for the analysis of heat exchangers (log-mean temperature difference (LMTD) and effectiveness-NTU) rely upon assumptions to set up the equations, such as constant specific heat. These techniques are not valid for recuperators operating under inconstant capacitances, such as CO\textsubscript{2} near the critical point. To overcome this impediment, two approaches will be explored: Develop a numerical complex model or divide the heat exchanger into numerous small sub heat exchangers (Nodalization).

In the model presented below, the printed-circuit heat exchangers (PCHEs) are divided into sub-heat exchangers (nodalization) as it is shown in figure 3. Nodalization is a heat exchanger modeling strategy that is necessary when a CO\textsubscript{2} working fluid is used due to its significant properties changing at or near the critical point. Each sub heat exchanger is then modeled independently (each component is evaluated as a separate control volume). At each sub-heat exchanger, the capacitance is almost the same and therefore the conventional techniques (LMTD and effectiveness-NTU) can be used after the adjusting of heat exchanger.
The appropriate number of sub-heat exchangers were studied to characterize the high variation of properties near the critical point. Too many nodes slow down the computational analysis, while too few nodes reduce the calculation accuracy. The system is first modeled with 20 sub-heat exchangers for each heat exchanger in the cycle, then dropped to 15, where there was not a big difference in the system efficiency. Then it reduces to 10 sub-heat exchangers, the efficiency still looks identical. Then, when the system is modeled with 8 sub-heat exchangers, a slight difference occurs. Finally, the system is tested with 6 sub-heat exchangers, there is a noticeable difference.

**Figure 2** Figure 2 shows different number of sub-heat exchangers versus cycle efficiency. Starting with ten sub-heat exchangers, the efficiency starts to converge. From 10 to 20 nodes, the efficiency seems identical, and therefore, 10 sub-heat exchangers seem to be enough for analysis.

The counter-flow effectiveness and number of transfer units (NTU) is shown in equations (6) and (7) respectively.

$$\varepsilon = \frac{1 - \exp[-NTU(1-C_R)]}{1-C_R \cdot \exp[-NTU(1-C_R)]}$$

(6)

$$NTU = \frac{\ln[1-C_R]}{1-C_R}$$

(7)

Where $C_R$ represents the dimensionless capacity ratio describing the heat exchanger balanced.

With the nodalization method, the total heat transfer rate is calculated first in either one of equations (8) and (9) using an energy balance, then it is equally divided between the sub-heat exchangers by using equation (10) and (11)

$$q_i = \frac{\dot{q}_{total}}{N}$$

(11)

Where $\dot{C}_H$, $\dot{C}_C$ and $\dot{m}_H$, $\dot{m}_C$ are the capacitance rate and mass flow rate of the hot and cold streams respectively, $T_{h_{out}}, T_{c_{out}}$ and $h_{h_{out}}, h_{c_{out}}$ are the inlet temperature and enthalpy of the hot and cold streams, $T_{h_{out}}, T_{c_{out}}$ and $h_{h_{out}}, h_{c_{out}}$are the out temperature and enthalpy of the hot and cold streams respectively, and $N$ is the number of sub-heat exchangers.

Then enthalpies for each sub-heat exchangers is calculated using equation (12) and (13)

$$h_{h_{out}} = h_{h_{in}} - \frac{\dot{q}_i}{m_H}$$

(12)

$$h_{c_{out}} = h_{c_{in}} - \frac{\dot{q}_i}{m_C}$$

(13)

Where $m_H$ and $m_C$ are the mass flow rate of hot and cold streams, $\dot{q}$ is the heat transfer rate of the sub-heat exchanger.

Calculation of the average specific heat $C_p$ heat and heat capacity rate ($\dot{C}$) of each side for the sub-heat exchanger, is done through equations (14), (15) and (16)

$$C_{P_h} = \frac{h_{h_{in}}-h_{h_{out}}}{T_{h_{in}}-T_{h_{out}}}$$

(14)

$$C_{P_c} = \frac{h_{c_{in}}-h_{c_{out}}}{T_{c_{out}}-T_{c_{in}}}$$

(15)

$$C_h = \dot{m}_H \cdot C_{P_h}$$

(16)

$$C_c = \dot{m}_C \cdot C_{P_c}$$

(17)

To calculate the sub-heat exchanger performance, the dimensionless effectiveness ($\varepsilon$) is defined in equation (18)

$$\varepsilon = \frac{\dot{q}_i}{\dot{q}_{imax}} = \frac{\dot{q}_i}{c_{min}(T_{h_{in}}-T_{c_{out}})}$$

(18)

Calculating the conductance for each sub-heat exchanger as it shown in equation (19)

$$UA_i = C_{min} \cdot NTU_i$$

(19)

Where NTU is the dimensionless number of transfer units that are defined in equation (19)

**Optimization**

The Multi-objective optimization technique conducted in this work is based on a genetic algorithm (GA) using Engineering Equation Solver (EES). Using the GA method allows the model to handle non-linear and non-differentiable optimization tasks. The two objective targets in the optimization processes are: 1) the overall thermal efficiency and 2) the power output.
Both objective targets need to be maximized simultaneously, while at some points they are conflicted to each other. Thus, the weighted sum method is employed in this study for the purpose of properly solving the conflicted of the multi-objective functions.

**Optimization Domain**

For an appropriate optimization, the variables bounds have to be predetermined to govern the optimization process and provide more reliable solutions. Based on the literature review at Table 1, the upper and lower bounds are specified. The lower and upper bounds set at the acceptable values to allow the model to test as mush variables as it could be. The optimization domain of the lower and upper variables is shown in Table 2.

**Results and Discussion**

The parametric analysis of the maximum cycle operating temperature show that the simple bottom Rankine cycle cannot fully recover the waste heat from the top recompression sCO₂ Brayton cycle. Thus, a newly-conceived bottom cycle is proposed to utilize the remaining waste heat from the top cycle. Figure 3 and Figure 4 shows thermal efficiency, exergy efficiency, and power output of the two combined cycles at different maximum operating temperature.

### Design Values

As noted above, several cycle configurations will be thermodynamically modeled and optimized for thermal efficiency and power output using Engineering Equation Solver Software (EES).

**Table 3 Decision and design variables**

<table>
<thead>
<tr>
<th>Decision variables</th>
<th>Cycle I</th>
<th>Cycle II</th>
</tr>
</thead>
<tbody>
<tr>
<td>compressor inlet temperature (°C)</td>
<td>32</td>
<td>32</td>
</tr>
<tr>
<td>compressor inlet pressure [MPa]</td>
<td>6.8</td>
<td>6.8</td>
</tr>
<tr>
<td>Pump inlet temperature</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>Turbine Isentropic efficiency [%]</td>
<td>93</td>
<td>93</td>
</tr>
<tr>
<td>Compressor Isentropic efficiency [%]</td>
<td>89</td>
<td>89</td>
</tr>
<tr>
<td>Inlet cooling air temperature [°C]</td>
<td>22</td>
<td>22</td>
</tr>
<tr>
<td>Inlet heating air temperature [°C]</td>
<td>900</td>
<td>900</td>
</tr>
<tr>
<td>Recompressor efficiency [%]</td>
<td>89</td>
<td>89</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Design variables</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Main compressor inlet pressure</td>
<td>vary</td>
<td>vary</td>
</tr>
<tr>
<td>Mass Flow rate</td>
<td>vary</td>
<td>vary</td>
</tr>
<tr>
<td>Pressure Ratio</td>
<td>vary</td>
<td>vary</td>
</tr>
<tr>
<td>Total fixed UA</td>
<td>vary</td>
<td>vary</td>
</tr>
<tr>
<td>Recompression fraction [-]</td>
<td>vary</td>
<td>vary</td>
</tr>
<tr>
<td>second WHR</td>
<td>Vary</td>
<td>Vary</td>
</tr>
<tr>
<td>Turbine inlet temperature (°C)</td>
<td>Vary</td>
<td>Vary</td>
</tr>
<tr>
<td>Compressor inlet temperature (°C)</td>
<td>Vary</td>
<td>Vary</td>
</tr>
</tbody>
</table>

The primary heater inlet temperatures source is assumed to come from a concentrated solar power tower plant. While air-cooled heat rejection is used instead of water-cooled heat rejection to reduce water consumption. Despite the air-cooled shortcomings such as overall thermal efficiency penalty and larger air-cooled heat exchanger than wet-cooled heat exchanger, the scarce of water makes air-cooled heat rejection attractive.

The input variables are divided into two categories, decision variables and design variables. The main difference is that the decision variables are assumed to be constant for all runs, while the design variables are the optimizer which is varied for each run. The initial parameter assumptions of the heat exchangers and turbomachinery are presented in Table 3.
Figure 3 indicates that the energetic and exergetic efficiency increase with the increase of the maximum cycle operating temperature of both combined cycles. While increasing ambient temperature increases the external irreversibilities (exergy losses) and therefore reducing the system performance. Another interesting finding is that, exergy and energy efficiency follow the same pattern, which they are linearly increasing as the turbine inlet temperature increases. Figure 3 shows how higher energetic and exergetic of the newly-conceived cycle at all different maximum cycle operating temperatures can be achieved by adding two recuperators to utilize the remaining waste heat that the simple bottom Rankine cycle cannot recover. The newly-conceived cycle, using two more recuperators, increases the cycle efficiency by about 2% to 2.5% compared to the simple bottom cycle.

Figure 4 demonstrates the higher power output of the newly-conceived cycle compared to the simple bottom cycle when they operate at same maximum turbine inlet temperature. An interesting finding from the simple bottom tCO2 Rankine cycle results is that, increasing the maximum operating temperature, above 390 °C, leads to two conflicts results. The first result is increasing the cycle efficiency, and the second result is lowering the waste heat recovery effectiveness, and thereby having a lower system efficiency. So, system thermal efficiency is optimized by balancing the cycle efficacy against the waste heat recovery effectiveness to have a higher system efficiency. Thus, the simple bottom cycle turbine inlet temperature has to be less than 390 °C in order to maintain a high waste heat recovery effectiveness. Otherwise, a drop in the waste heat recovery effectiveness occur which lower the system efficiency. To overcome this issue, a newly-conceived cycle is proposed. It allows higher bottom cycle turbine inlet temperature without adverse effecting the waste heat recovery effectiveness by adding two recuperators to the system. Increasing the maximum cycle operating temperature lead to an increase of enthalpy difference across the turbine and therefore an increase of power output.

According to Figure 5 the combined cycles improve the overall cycle thermal efficiency by about 2% - 2.5% (simple bottom cycle), and 4% - 4.5% (split-flow bottom cycle) compared to the stand-alone recompression sCO2 Brayton cycle. The optimum turbine inlet temperature is determined based on the concentrated solar power (CSP) heat source availability, which will be studied in a future work. Also, economic analysis are important to be combined with the power cycle analysis to find out whether that higher efficiency, due to the higher operating temperature, is associated with the higher cost, due to larger heat exchanger and higher temperature material, worth or not.

The impact of the main compressor inlet temperature on the first law efficiency, second law efficiency, power output has been studied, as it shown in Figure 8 and Figure 9.

![Figure 5](image5.png)  
**Figure 5** Thermal efficiency improvement as a function of maximum operating temperature for simple and new combined cycles

![Figure 6](image6.png)  
**Figure 6** Impact of compressor inlet temperature and turbine inlet temperature on the newly-conceived cycle efficiency

![Figure 7](image7.png)  
**Figure 7** Impact of compressor inlet temperature and turbine inlet temperature on the newly-conceived cycle power output

Figure 8 and Figure 9 show how the minimum operating inlet temperature and maximum operating inlet temperature affect the cycle efficiency and power output. Increasing the turbine inlet temperature has a direct positive effect to the power output and efficacy. Also, it is important to note that, a high-temperature-
resistant material may be needed with the increasing of maximum operating temperature, thereby increasing the system cost.

Exergy analysis of the newly-conceived combined power cycle is conducted in order to identify thermodynamic losses in each cycle component for the purpose of improving overall thermal efficiency by attempting to minimize thermodynamic irreversibilities. External irreversibilities (exergy loss) and internal irreversibilities (exergy destruction) have been determined through the exergy analysis, as they shown in Figure 8

![Figure 8 Exergy destruction rate and ratio in the SCO2 components](image)

Figure 8 shows the exergy destruction in the top recompression sCO2 cycle components. Primary heater, which transfer heat from the primary source to the cycle, accounts for 35% of the thermodynamic losses, follow by the low temperature recuperator, which has an internal cycle heat transfer, accounts for 18% of the total thermodynamic losses. One way to minimize the thermodynamic losses on the primary heat exchanger and low temperature recuperator is to minimize heat exchanger temperature difference between the two streams (cold – hot).

Conclusion

The energy and exergy analysis of the two advanced combined cycles were conducted in this paper. The internal irreversibilities (exergy destruction) and external irreversibilities (exergy losses) for each component were investigated in order to provide appropriate guiding improvements. The top sCO2 recompression Brayton cycle’s waste heat is utilized by a bottom tCO2 Rankine cycle for the purpose of improving both efficiency and power output. The result demonstrate that the newly-conceived cycle, sCO2 recompression Brayton coupled with a tCO2 split-flow Rankine cycle, surpasses the simple combined cycle, sCO2 recompression Brayton coupled with a tCO2 simple Rankine cycle, in respect to energy and exergy efficiencies and power output.

Based on the exergy analysis, primary heater has the highest thermodynamic losses, follow by the low temperature recuperator (LTR). On the other hand, the turbine and compressors have the lowest thermodynamic losses. The high potential improvements of the cycle should be focused on the heat exchangers and especially primary heater and low temperature recuperator.

Acknowledgments

The authors acknowledge that the work presented here was funded by Qassim University, Saudi Arabia.


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