THERMODYNAMIC ANALYSIS AND MULTI-OBJECTIVE OPTIMIZATIONS OF A COMBINED RECOMPRESSION SCO2 BRAYTON CYCLE–TCO2 RANKINE CYCLES FOR WASTE HEAT RECOVERY

ABSTRACT:

A thermodynamic (Energy and Exergy) analysis and optimization of a newly-conceived combined power cycle were conducted in this chapter 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 S-CO2 Brayton cycle and a bottoming S-CO2 Rankine cycle. The bottoming cycle configurations included a simple tCO2 Rankine cycle and a split tCO2 Rankine cycle. The topping supercritical CO2 recompression Brayton cycle used a combustion chamber as a heat source, and waste heat from a topping cycle was recovered by the tCO2 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 tCO2 Rankine cycle cannot fully recover the waste heat due to the high exhaust temperature from the top cycle, and therefore an advance split tCO2 Rankine cycle was employed in order to recover most of the waste heat. Results show that the highest thermal efficiency was obtained with recompression S-CO2 Brayton cycle – split tCO2 Brayton cycle. Also, the results show that the combined CO2 cycles is a promising technology compared to conventional cycles.

1. 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.

Researchers have theoretically demonstrated that low exhuast temperature can power supercritical and transcritical CO2 Rankine cycles [1-2]. Sarker [2] provides an organized review of supercritical CO2 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 supercritical CO2 Rankine cycle has clear advantages to steam and organic Rankine cycles, and he discusses pathways to developing aspects of this cycle (parameter optimization, hardware components, control strategies, etc.). Wang and Dai [3] compared the exergoeconomic performance for two bottoming cycles (transcritical CO2 and ORC) designed to optimize waste heat recovery from a supercritical CO2 recompression Brayton topping cycle. Parametric optimization indicates that the tCO2 bottoming cycle has superior performance at lower PRc (off-design conditions), and that higher turbine inlet temperatures improve tCO2 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 [4] developed a tCO2 cycle for recovering waste heat from the pre-cooler of a sCO2 Brayton cycle, and they modeled the performance improvement for this new combined cycle relative to that of a simple sCO2 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. [5] compared the performance of two cycles act as a bottoming cycle to extract useful work from low-grade waste heat. The organic Rankine cycle (ORC) is most commonly used, but the authors found that the CO2 transcritical 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, most research in supercritical CO2 cycles used two ways to represent heat exchanger performance: using fixed heat exchanger effectiveness or pinch point temperature differences. However, due to sCO2 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.

1. SYSTEM ANALYSIS:

In this study, two combined CO2 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 sCO2 recompression Brayton cycle with a bottom tCO2 split flow Rankine cycle -called cycle I- (Fig.3a) and 2) a top sCO2 recompression Brayton cycle with a bottom tCO2 simple Rankine cycle –called cycle II- (fig.3b).

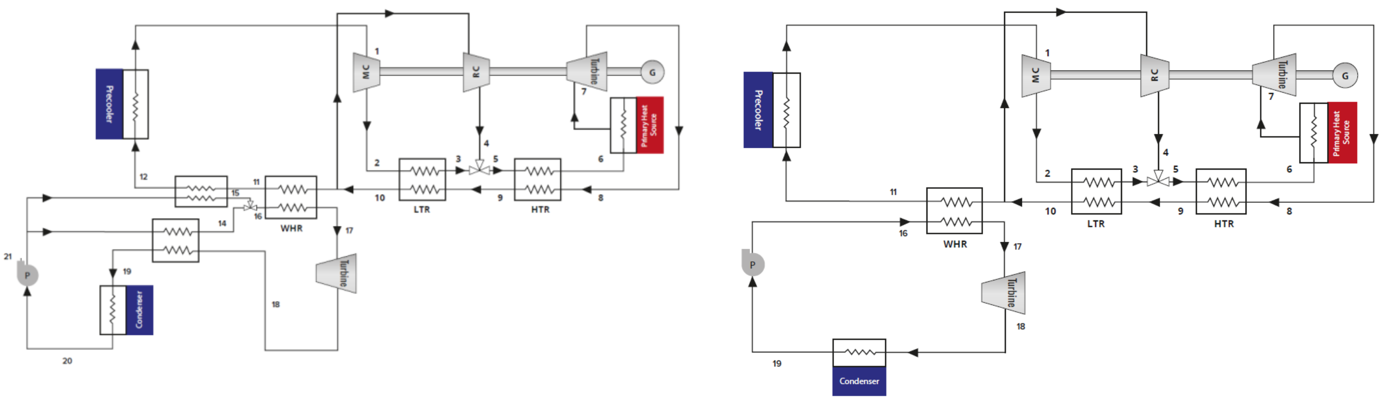

Figure 1 Combined CO2 power cycles a) recompression sCO2 Bryton cycle – split flow tCO2 Rankine cycle b) recompression sCO2 Bryton cycle – simple tCO2 Rankine cycle

2.1 Turbomachinery

The turbomachinery analysis is modeled on the energy balance of individual components to study the performance of turbines and compressors. During the turbomachinery modeling, some basic assumptions are considered: (i) The cycle is assumed to function in a steady state (ii) the turbine expansion, the compressors and the pump are considered adiabatic with given isentropic efficiencies (iii) kinetic and potential energy effects are negligible (iv) each component of the cycle is sufficiently insulated. Compressors and turbine isentropic efficiencies are defined in the equations (1) and (2)

$$\eta_c = \frac{h_{out} - h_{in}}{h_{out_{ise}} - h_{in}} \quad (1)$$

$$\eta_t = \frac{h_{in} - h_{out}}{h_{in} - h_{out_{ise}}} \quad (2)$$

Where h_{in} and h_{out} represent the inlet and outlet actual enthalpy respectively, and $h_{out_{ise}}$ the isentropic outlet enthalpy.

The specific actual work can be calculated using equation (3)

$$w = h_{in} - h_{out} \quad (3)$$

2.2 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 capacity. These techniques are not valid for recuperators operating under inconstant capacitances, such as CO2 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 PCHEs are divided into sub-heat exchangers (nodalization) as it is shown in Figure 2. Nodalization is a heat exchanger modeling strategy that is necessary when a CO2 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.

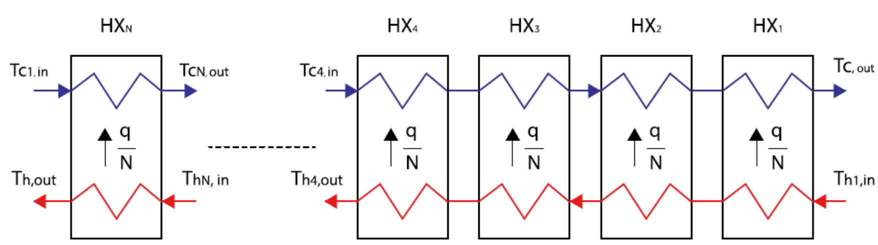

Figure 2 HX nodes

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 3 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 seems to be enough for analysis.

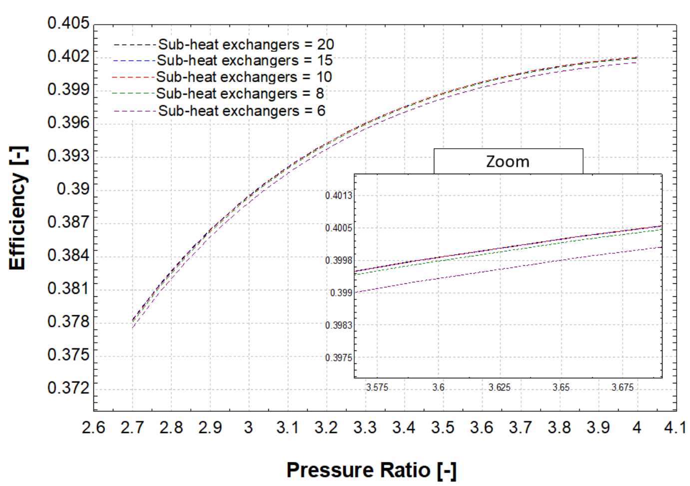

Figure 3. Efficiency at different number of Sub-Heat Exchangers

The counter-flow effectiveness and number of transfer units (NTU) is shown in equations (4) and (5) respectively.

$$\varepsilon = \frac{1 - \exp[-NTU * (1 - C_R)]}{1 - \mathrm{C_R} * \exp[-NTU * (1 - C_R)]} \tag{4}$$

$$NTU = \frac{\ln[\frac{1 - C_R}{1 - \varepsilon}]}{1 - C_R} \tag{5}$$

Where $\mathrm{C_R}$ 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 (6) and (7) using an energy balance, then it is equally divided between the sub-heat exchangers by using equation (9).

$$\dot{q_H} = \dot{C}_H * \left(T_{h_{in}} - T_{h_{out}}\right) = \dot{m}_H * (h_{h_{in}} - h_{h_{out}}) \tag{6}$$

$$\dot{q_C} = \dot{C}_C * (T_{C_{in}} - T_{C_{out}}) = \dot{m}_C * (h_{C_{in}} - h_{C_{out}}) \tag{7}$$

$$\dot{q_H} = \dot{q_C} \tag{8}$$

$$q_i = \frac{\dot{q}_{total}}{N} \tag{9}$$

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, $T_{h_{in}}, T_{C_{in}}$ and $h_{h_{in}}, h_{C_{in}}$ 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 (10) and (11)

$$h_{h_{out}} = h_{h_{in}} - \frac{\dot{q}_i}{\dot{m}_h} \tag{10}$$

$$h_{c_{out}} = h_{c_{in}} - \frac{\dot{q}_i}{\dot{m}_c} \tag{11}$$

Where $\dot{m}_h$ and $\dot{m}_c$ 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 ($\dot{\mathrm{C}}$) of each side for the sub-heat exchanger, is done through equations (12) – (15):

$$C_{P_h} = \frac{\left(h_{h_{in}} - h_{h_{out}}\right)}{\left(T_{h_{in}} - T_{h_{out}}\right)} \tag{12}$$

$$C_{P_C} = \frac{\left(h_{C_{out}} - h_{C_{in}}\right)}{\left(T_{C_{out}} - T_{C_{in}}\right)} \tag{13}$$

$$C_h = \dot{m}_H * C_{P_h} \tag{14}$$

$$C_c = \dot{m}_c * C_{P_c} \tag{15}$$

To calculate the sub-heat exchanger performance, the dimensionless effectiveness (ε) is defined in equation (16):

$$\varepsilon = \frac{\dot{q}_\imath}{\dot{q}_{\imath max}} = \frac{\dot{q}_\imath}{C_{min} * \left(T_{h_{in}} - T_{c_{out}}\right)} \tag{16}$$

Calculating the conductance for each sub-heat exchanger as it shown in equation (17):

$$UA_i = C_{min} * NTU_i \tag{17}$$

Where NTU is the dimensionless number of transfer units that are defined in equation (5).

3. RESULT

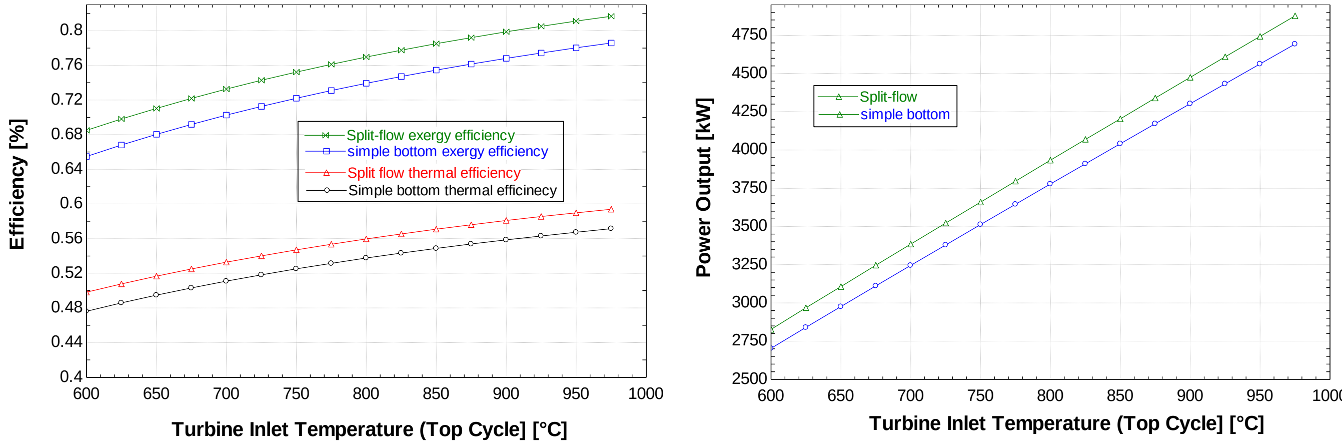

Figure 4. a) Thermal and Exergy efficiency b) Power output comparison as a function of maximum operating temperature for the simple and new combined cycles

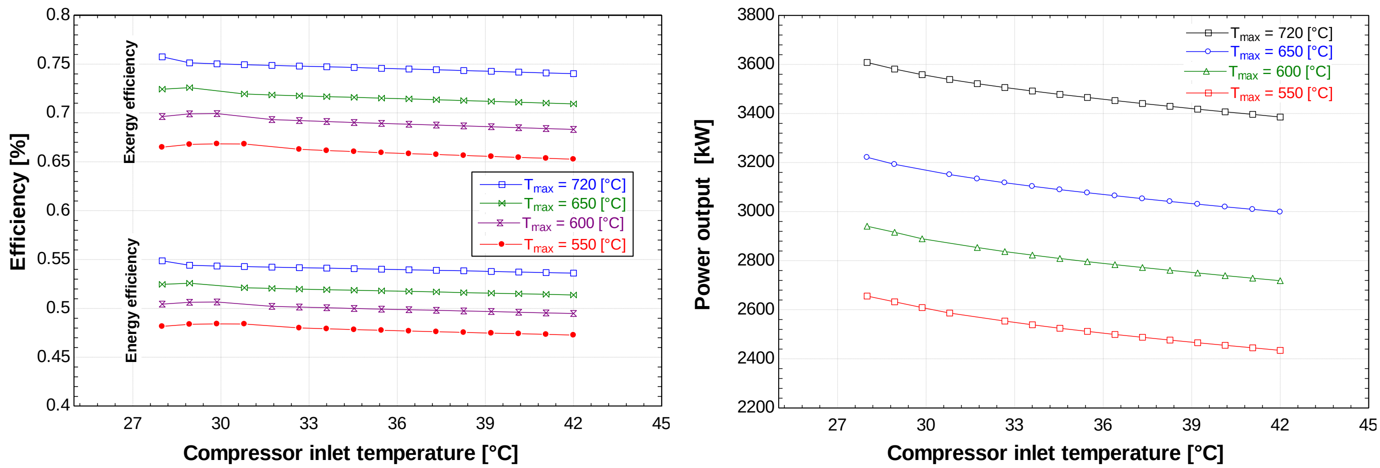

Figure 5. a) Thermal and Exergy efficiency b) Power output comparison as a function of minimum operating temperature for the simple and new combined cycles

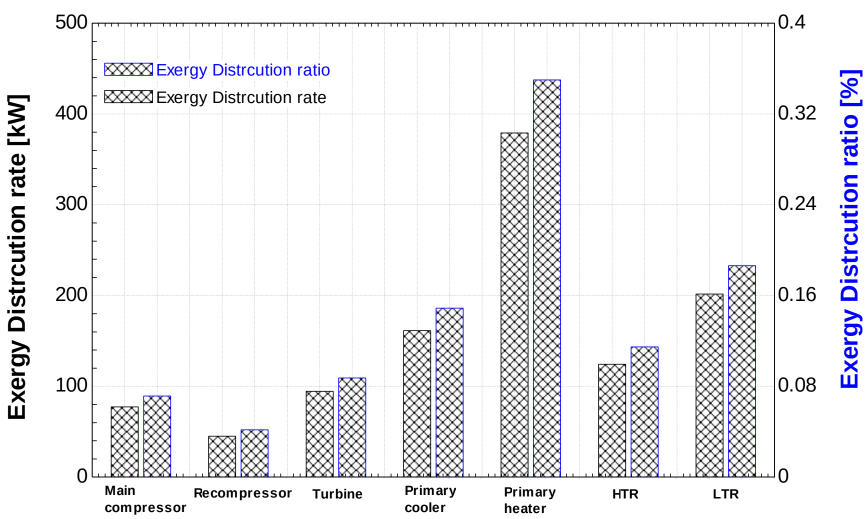

Figure 6. Exergy destructions rate and ratio in the S-CO2 components

4. CONCLUSION

The energy and exergy analysis of the two advanced combined cycles were conducted in this chapter. 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 sCO2 Rankine cycle for the purpose of improving both efficiency and power output. The two cycles comparison is based on the parametric analysis of the maximum cycle operating temperature. The result demonstrate that the new-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.

. REFREANCE:

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