VOL. 10, NO 21, NOVEMBER, 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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PARAMETRIC ANALYSIS OF CARBON DIOXIDE TRANSCRITICAL AND SUPERCRITICAL POWER CYCLES USING LOW GRADE HEAT SOURCE

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ABSTRACT

This paper considers carbon dioxide transcritical and supercritical power cycles driven by low temperature flue gases exhaust from a gas turbine. Transcritical CO2 Rankine and supercritical CO2 Brayton cycles were studied at steady state conditions and their performance were compared. Furthermore, the study carried out parametric analysis to investigate the cycles' performance in terms of thermal efficiency and the network output at different turbine inlet temperatures. A mathematical model was developed to carry out the analysis based on the first law of thermodynamics. In order to simulate cycle performance and generate parametric tables a simulation model was developed using Engineering Equations Solver (EES). The efficiencies of the cycles were compared and it was found that transcritical Rankine cycle generates higher efficiency and net power output compared to supercritical Braytoncycle for the same turbine inlet conditions. Parametric analysis showed that as the turbine inlet temperature increases, the gas heater pressure that gives optimum efficiency increases.

Keywords: Power cycles, carbon dioxide trans critical, engineering equation solver, parametric analysis.

INTRODUCTION

The emerging environmental problems associated with power generation by burning the fossil fuel has encouraged the researches towards green energy concept [1]. That introduces to us the likes of heat recovery, cogeneration and green energy that contribute less damage to the environment and increase the thermal efficiency of a power generation unit by burning less fossil fuel.

Carbon dioxide physical properties have been studied and found to have low critical point of 7.38MPa and 31 ºC. This lower critical point allows the CO2 to change states at low pressure and temperature and favors to convert low quality heat energy into useful power. There are many other working fluids that can be alternative for steam as a working fluid like R245fa, R123, R134a and zeotropic mixtures [2]. Among all these working fluids carbon dioxide has favorable thermal properties and low environmental impact. It is also non-explosive and non-toxic[3]. Moreover, carbon dioxide allows better temperature match with the heat source temperature profile during the isobaric heat transfer with a low grade temperature heat source [4, 5].

Recently, many studies were devoted to investigate the transcritical and supercritical carbon dioxide power cycles driven by various temperature ranges along with different configurations [6]. Chen *et al.* investigated the carbon dioxide transcritical performance utilizing low-grade waste heat energy and it was compared to the Organic Rankine cycle (ORC) and R123 as its working fluid. It was found that carbon dioxide transcritical power cycle had a slightly higher power output than the ORC under the same conditions due to the irreversibilities involved in the ORC power cycle [7]. Cayer *et al.* [2] analyzed and optimized the thermodynamic parameters of a transcritical CO2 power cycle using genetic algorithm and artificial neural network to recover low-grade heat source. Zhang *et al.* [5] investigated a solar energy heat source Rankine cycle using supercritical cycle and carbon dioxide as the working fluid for combined power and heat production.

Nowadays, the use of supercritical carbon dioxide power cycles for extraction of nuclear energy is increasing due to its enhanced safety, sustainability, simplicity, superior economy and compactness[8]. It's highly expected that the supercritical CO2 would benefit the renewable, nuclear and fossil power plants for its remarkable properties as a working fluid in its supercritical state.

In this paper T-CO2 Rankine and S-CO2 Braytoncycles performance were compared in terms of the cycles' thermal efficiencies. Wider ranges of heat source temperatures are studied by developing parametric analysis. For the considered temperature range gas turbine exhaust gas can be the heat source.

System description and assumptions

As mentioned earlier this study investigates two different CO2 cycles under the following assumptions. The system is at steady state, the kinetic and potential energies as well as the heat loss are negligible and no heat regeneration. The turbine and pump/compressor isentropic efficiencies are assumed to be 90%.

VOL. 10, NO 21, NOVEMBER, 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

Transcritical CO2 rankine cycle

The cycle is made up of four components: a pump, an evaporator, a turbine and a condenser. Figure 1 illustrates a T-CO2 Rankine cycle composed of the following processes:

- [1-2] a non-isentropic expansion in the turbine.
- [2-3] an isobaric heat rejection in the condenser.
- [3-4] a non-isentropic compression in the pump.
- [4-1] an isobaric heat addition in the evaporator.

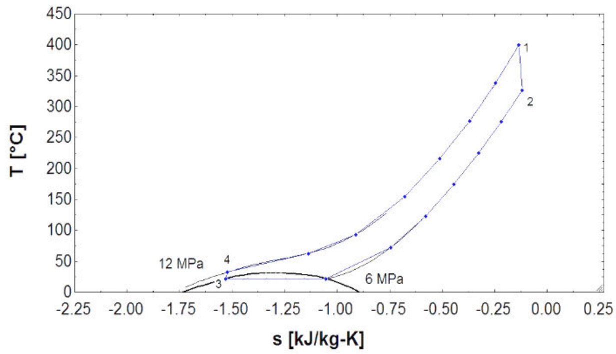

Figure-1. T-CO2 rankine cycle T-s diagram.

In the evaporator the supercritical CO2 is converted into a supercritical vapor by exchanging heat with the heat source. The high pressure and temperature CO2 vapor is then directed to expand in the turbine to generate work. After that the condensation process takes place to dissipate the heat energy content after the expansion in order to convert the CO2 into 100% liquid state at the pump inlet.

Supercritical CO2 brayton cycle

The S-CO2 Brayton cycle differs from the T-CO2 Rankinecycle in that the cycle operates fully in the supercritical region or at a pressure higher than the critical pressure of CO2 (7.38MPa). Figure-2 illustrates a S-CO2 Brayton cycle.

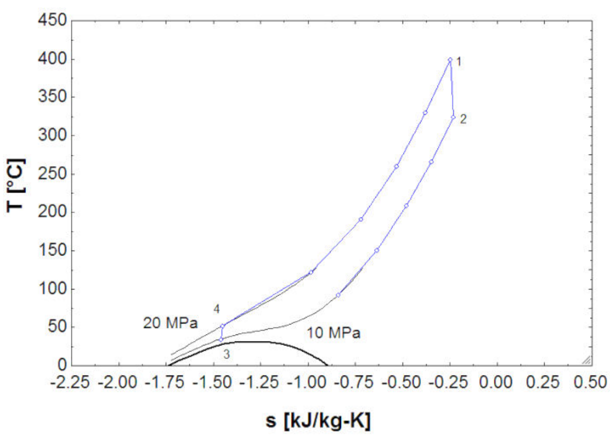

Figure-2. S-CO2 brayton cycle T-s diagram.

Mathematical models and performance criteria

Figure-3 represents the configuration and the state points of CO2 cycle. The mathematical model for each component was developed based on conservation of energy.

- Evaporator/Gas heater energy balance for heat addition process at constant pressure.

$$q_{in} = h_1 - h_4 \quad (1)$$

- The amount of turbine power output is estimated as (non-isentropic expansion)

$$h_2 = h_1 - \eta_t * (h_1 - h_{2s}) \quad (2)$$

$$w_{t,out} = h_1 - h_2 \quad (3)$$

- Condenser/gas cooler energy balance for heat rejection at constant pressure

$$q_{out} = h_2 - h_3 \quad (4)$$

- Pump/compressor energy balance for power in (non-isentropic compression)

$$h_4 = h_3 - \frac{h_{3s} - h_4}{\eta_{p_c}} \quad (5)$$

$$w_{p/c,in} = h_4 - h_3 \quad (6)$$

- Net power output

$$w_{net} = w_{out} - w_{in} \quad (7)$$

- Cycle thermal efficiency

$$\eta_{th}(\%) = \frac{w_{net}}{q_{in}} * 100 \quad (8)$$

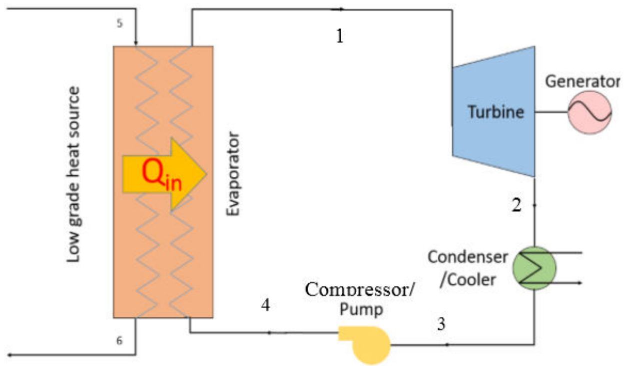

Figure-3. Schematic representation of CO2 cycle configuration and state points.

The numerical analysis has been carried out based on the mathematical model, the pre-defined conditions and assumptions and finally solved by EES. A subroutine was written in a way to estimate thermal efficiency and network output for different turbine inlet conditions. Furthermore, Microsoft Excel has been used as well to plot the parametric tables obtained from EES. The T-CO2 Rankine cycle and the S-CO2 Brayton cycle both operate at the same turbine inlet temperature and pressure. Fluid properties at each state point are automatically evaluated with two known properties as the EES database

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has CO2 properties. The EES was also used to generate the parametric tables.

RESULTS AND DISCUSSIONS

Parametric cycle performance

The thermal efficiency of each cycle was evaluated and considered as the key overall performance indicator. The efficiencies of the cycles were simulated for the temperature range 100-160 °C and gas heater pressure range 12 to 36MPa. Figures 4 and 5 show for every specific turbine inlet temperature as the gas heater pressure increases, first the thermal efficiency increases, reaches maximum and then drops. Hence, every turbine inlet temperature has distinct pressure that gives optimum efficiency. As the heat source/turbine inlet temperature increases, the pressure that gives optimum efficiency moves towards the right. Moreover, transcritical Rankine

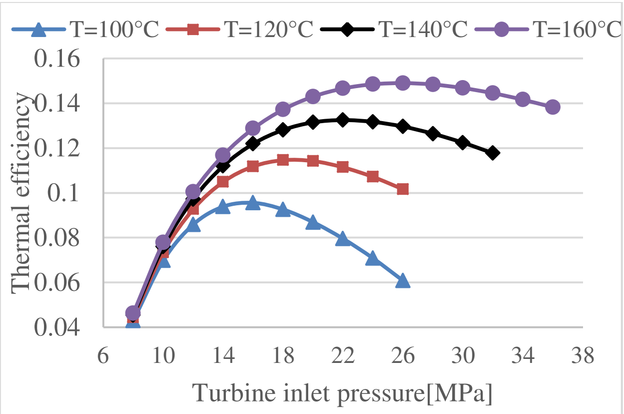

Figure-4. T-CO2 rankine cycle thermal efficiency versus turbine inlet pressure for different turbine in let temperatures.

cycle provides better performance than supercritical Brayton cycle as shown in Figures 4 and 5.

The main reason that creates the difference is the compression work of the cycle is significantly reduced in the T-CO2 Rankine cycle as compared to the S-CO2 Brayton cycle because pumping liquid needs less work compared to gas. Therefore, the thermal efficiency of the T-CO2 Rankine cycle is higher than that of the S-CO2 Brayton cycle.

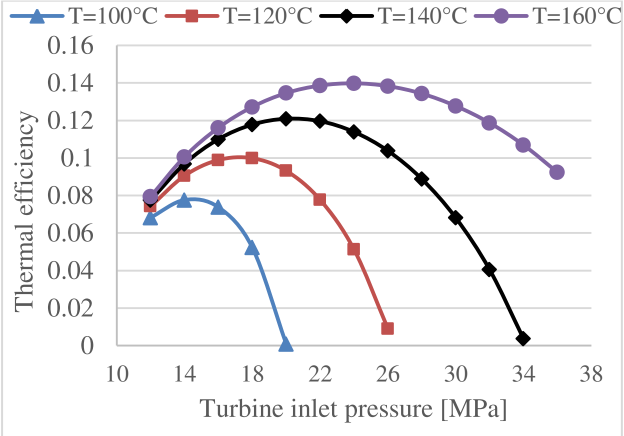

Figure-5. S-CO2 brayton cycle thermal efficiency versus turbine inlet pressure for different turbine inlet temperatures.

The Influence of the turbine inlet pressure and temperature on the cycle net power output

Net power output may is one way to measure performance of a cycle but it may be misleading, since the cycle fuel consumption may be higher than another cycle for the same power output. However, it is a good additional performance index with known cycle efficiency.

Figures 6 and 7 show the trend of network output of the two cycles. As can be seen for a range of gas heater pressure, the profile of the network output is the same as that of their thermal efficiency. At a given pressure and temperature, for the same reasoning, the T-CO2 cycle produces higher network output compared to the corresponding S-CO2 cycle at the same conditions.

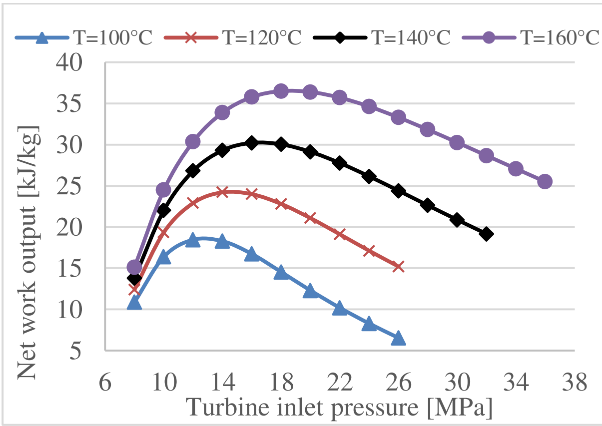

Figure-6. T-CO2 rankine cycle net work out vs. turbine inlet pressure for different turbine in let temperatures.

It is also important to notice that the pressure that gives the maximum network output is less than the pressure that gives optimum efficiency while the remaining operating conditions are the same. Hence, there is no a pressure that gives maximum efficiency and net work output at the same time. In this case, running the cycle for maximum efficiency or for maximum work output depends on the availability of the heat source. If the heat sources are free, say gas turbine exhaust gas or solar energy, then it should be run for maximum power output.

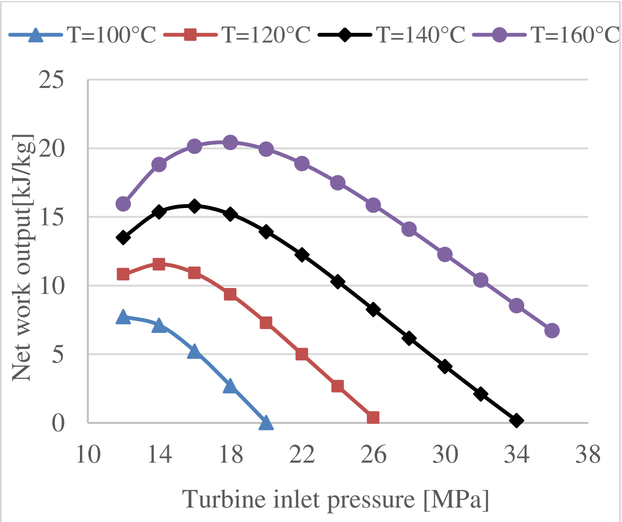

Figure-7. S-CO2 brayton cycle net work out versus turbine inlet pressure for different turbine inlet temperatures.

Nomenclature

T	Temperature (°C)
p	Pressure (MPa)
h	Enthalpy (kJ/kg)
x	Saturated mixture quality
Cp	Isobaric specific heat capacity (kJ/kg.K)
w	Work (kJ/kg)
Q	Heat energy (kJ/kg)
T-CO2	Transcritical Carbone dioxide
S-CO2	Supercritical Carbon dioxide
EES	Engineering equations solver

Subscripts

in	inlet
out	outlet
th	thermal
p	pump
c	compressor

CONCLUSIONS

In the present study, investigation of carbon dioxide transcritical and supercritical power cycles performance driven by low temperature flue gases waste as heat source was carried out based on parametric analysis.

The cycles utilize flue gases exhaust temperature in the range of 100 to 160 °C as a heat source which can be obtained from a gas turbine exhaust gas. Based on the first law analysis at given conditions T-CO2cycle produces higher thermal efficiency and network output compared to S-CO2 cycle. The optimum turbine inlet pressure that gives the highest thermal efficiency is found to increase as the turbine inlet temperature increases. This study generates clearer overview on the use of carbon dioxide cycles for the conversion of medium temperature heat source into power.

ACKNOWLEDGMENTS

This work is supported by Ministry of Higher Education, FRGS grant number 0153AB-I70. The authors thank Universiti Teknologi PETRONAS for providing supports to conduct the project.

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