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The S-CO2 Brayton cycle code development and the thermal dynamic analysis of a small scaled facility

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ABSTRACT

The S-CO2 Brayton cycle is a transform power conversion technology with high thermal efficiency, compactness and low cost. It could be applied into various areas, such as ship power, waste heat reuse, solar energy, etc. The cycle was first proposed in last middle century and saw profound progress in these few years as the development of small turbo machinery technology and high compact heat exchangers. The 100 kW class power conversion loop has been demonstrated in USA. The Mega Watts class facility is ongoing. Other countries including Europe, China and Korea have also supported relative projects. To better understand the performance of S-CO2 Brayton cycle, the thermal dynamic conversion code was developed which has the abilities of analyzing the performances of simple basic cycle, regeneration cycle and recompression cycle. The code was written in MATLAB language. The NIST REFPROP software was built in so not only the supercritical carbon dioxide but also other substances could be analyzed as the work medium. The optimize method was coupled with the code to find the local best parameter combinations under the specified constrains. A small scaled S-CO2 power conversion loop was analyzed with the code under steady normal conditions. The various factors which would influence the facility performance were discussed. This analysis could provide theoretical guidance for the facility design.

INTRODUCTION

The S-CO2 Brayton cycle is a transforming power conversion technology with high thermal efficiency, compactness and low cost. It could be applied into various areas, such as ship power, waste heat reuse, solar energy, etc. The cycle was first proposed in last middle century and saw profound progress in these few years as the development of small turbo machinery technology and high compact heat exchangers.

The thermal balance code is used for steady analysis and thermal design of the whole cycle. For example, the code could be used to get the cycle efficiency for specified parameters, to analyze the parameter effects and cycle optimization that meets a set of design requirements. Up to now, the thermal dynamic code for supercritical carbon dioxide cycles are usually self-developed.

Dostal et al. (2004)[1] developed the thermal dynamic code CYCLES to investigate the cycle performance of recompression cycle. The compression cycle was found to yield the highest efficiency and simplicity.

Dyreby(2014)[2] developed a code capable of predicting the design-point, off-design, and part-load performance of S-CO2 power.

Chang (2004[3]，2006[4]) evaluated the suitability of commercial software ASPEN PLUS and HYSYS in S-CO2 cycle design. The Lee-Kesler-Plocker correlation is used to calculate the carbon dioxide properties.

Besides the steady analysis code, the transient analysis code have also been developed or improved based on the traditional code.

Kemp（2006）[8] analyzed the PBMR with Flownex software.

Hexemer et al.(2009)[5] analyzed the Integrated Systems Test S-CO2 loop with TRACE Plant analysis code. The source code modifications were performed to model the turbo machinery components.

Pope (2009)[6] analyzed the system safety performance of GFR coupled with S-CO2 cycles with RELAP5-3D software.

Colonna(2011)[7]analyzed the 10MWe recompression cycle with Open Modelica to demonstrate the code capabilities.

Very limited test facilities have been setup around the world to demonstrate the S-CO2 cycle technical feasibility, such as 100 kWe class recompression loop from the Sandia National Laboratories (SNL) [9]and the Naval Nuclear Laboratory Integrated System Test (IST) [10-12]. The Mega Watts class facility is under construction to test turbine performance.

NPIC is now [13, 14] building a Mega Watts Class closed S-CO2 facility. The design parameters have been determined. The real test facility may be different from the design one because of equipment performance, heat losses and pressure losses. Also the test facility could be improved for future applications.

To better understand the performance of S-CO2 Brayton cycle, the thermal dynamic conversion code was developed which has the abilities of analyzing the performances of simple basic cycle, regeneration cycle and recompression cycle. The code was used to analyze a conceptual design of Mega Watts loop. This analysis could provide theoretical guidance for the future facility design and operation.

FLUID PROPERTIES

The carbon dioxide basic parameters are listed in Table 1. The critical pressure and temperature are relatively medium (7.38MPa, 304.13 K). When the cycle pressure operates above the critical point, which means supercritical state, the fluid is only single phase. At the fixed pressure, the point at which the heat capacity reaches its maxima is usually called pseudocritical point. Before this point, the carbon dioxide is liquid-like with high density, high viscosity and high thermal conductivity; after this point, the carbon dioxide is gas-like, with low density, low viscosity and low thermal conductivity. The S-CO2 cycle makes use of the special characteristics of carbon dioxide, which puts the compressor inlet near the critical point to suppress the compress work. At the same time, the turbine works at high temperature, high inlet pressure region. The different states for the equipment suggest the potential of high thermal efficiency. Figure 1 shows some pictures of typical fluid properties variations for carbon dioxide.

Table 1 fluid information for carbon dioxide

Parameter	**Description**
Mole Mass	44.01 kg/kmol
Triple Temperature	216.59 K
Critical temperature	304.13 K
Critical Pressure	7.3773 MPa
Critical Density	467.6 kg/m^3

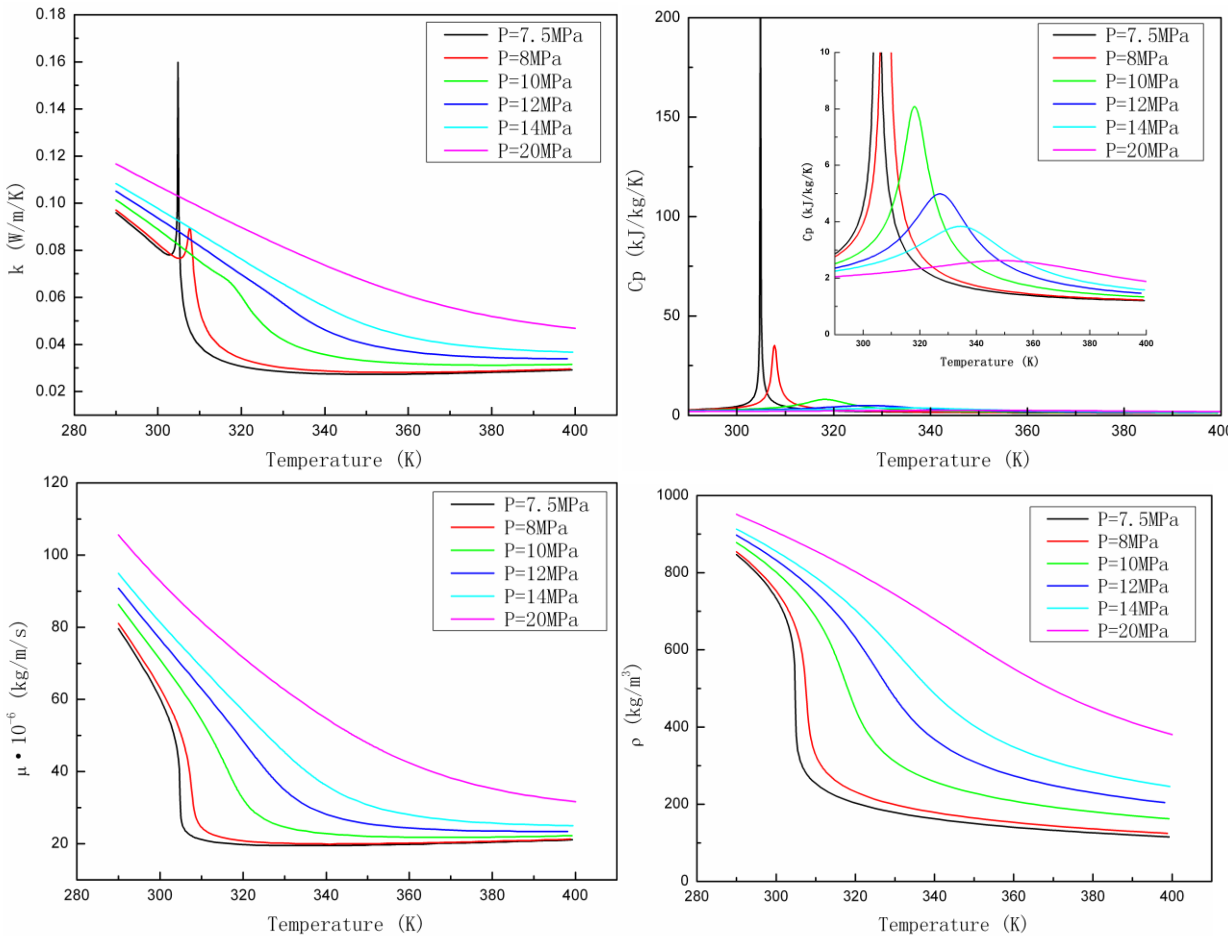

Figure 1 typical fluid properties profile for carbon dioxide

CODE DEVELOPMENT

The thermal dynamic code was developed with MATLAB. The code consists of different modules, such as compressor, turbine, heat exchanger, etc. Each module was described in a separate function file. The modules could be combined together for different cycle layouts. The information exchange was through the function interface. The cycle calculation was performed through an iteration method.

The NIST software was embedded to calculate the fluid properties for carbon dioxide. The software could also be used to other substances, such as helium, nitrogen and so on. The software applies accurate method and complex formulas to describe the equations of states. However, the computational time costs a lot. If large number of cases are to be calculated, most of the time are wasted on the properties calculation. For this question, the MATLAB parallel computing technique was applied as well as the high performance computer. The total computing time could be saved a lot. Near the critical point, there exists the possibility of numerical instability which would cause the computational failure sometimes. If the optimized point is searched by the program, the near-critical region is likely to be searched. For this question, the instability point was dealt with to avoid this problem.

The compressor and turbine were both rotational equipment, with the first one transforming energy from motor to fluid and the second one transforming the fluid energy into electricity. The compressor ratio and expansion ratio are two important parameters to describe the equipment characteristics. The equipment efficiency was accounted for in two aspects. First, the efficiency could influence the power transferred from or to the motor/generator. Second, the efficiency could influence the outlet temperature. Take the compressor as an example to show the calculation procedure. First, with the inlet pressure and the pressure ratio, the outlet pressure could be obtained. Second, based on the iso-entropy assumption, the ideal outlet entropy is obtained. The compressor work is obtained by the enthalpy difference divided by the efficiency. At last, the outlet enthalpy is obtained by the summation of the inlet enthalpy and compressor work.

The heat exchanger model was based on the PCHE geometry which is widely used in S-CO2 cycles. Each channel has the shape of semi-circle. Each plate was echoed with numerous semi-circle channels. One block was bounded with numerous plates.

The PCHE could be simplified into plate heat transfer question between one hot channel and one cold channel based on the following assumptions: The hot channel and cold channel was arranged alternately, one layer of hot channel and one layer of cold channel. The mass flux was equally distributed into each channel. The non-uniform wall temperature along the circumferential direction was neglected.

The supercritical carbon dioxide was not ideal gas, so the traditional logarithmic temperature difference method was no long suitable. The one dimensional finite volume method was used here. The heat transfer was divided into several sections. For each section, the heat transfer was calculated. The fluid states for the hot side and cold side were then updated. Based on this method, the next section was then calculated in similar method until the whole heat exchanger was solved. For the co-current heat exchanger, both hot and cold inlets are at the same end. The solution process could be completed from one end to the other end. For the counter-current heat exchanger, both hot and cold inlets are at opposite ends. At this situation, one outlet is first assumed. Then the inlet condition for this assumed outlet is calculated. If the error between this value and the given value is small enough, the solution is obtained. The nonlinear equation solver method was used in MATLAB to solve this kind of equations.

There are two kinds of heat exchangers in the cycle, the recuperator and the cooler. For the recuperator, both sides are supercritical carbon dioxide. Both inlet conditions are known and the outlet conditions should be calculated through the program. For the cooler, the hot side is S-CO2 and the cold side is water. The hot side inlet and outlet temperatures are known, the cold side mass flow rate should be calculated to meet the specified cooling requirements. These two kinds of heat exchangers are dealt separately in the program.

The heater was assumed as an ideal heat point source. The heater power was added to the fluid as long as it passed away. Meanwhile, the pressure loss was abstracted from the inlet pressure to get the outlet pressure.

For one cycle calculations, the compressor inlet pressure and temperature is known. The heater outlet temperature is known. The geometry parameters for the recuperator and cooler are known. With the compressor model, the compressor outlet pressure and temperature are calculated. The inlet temperature and outlet pressure for the low pressure side of recuperator are assumed first. Then the other information could be calculated, such as the outlet pressure and temperature for the high pressure side, the inlet pressure and outlet temperature for the low pressure side. For the heater model, the heating power and outlet temperature is known, so the mass flow rate is calculated. For the turbine, the inlet pressure and outlet pressure are known, so the outlet temperature is calculated. For the cooler, the inlet temperature and outlet pressure and temperature for carbon dioxide are known, as well as the inlet pressure and temperature for water, so the mass flow rate could be known. Till now, all the information in the cycle has been updated one time. Then it will check the convergence based on the mass flow rate. The basic program procedure and structure are depicted in Figure 2.

The cycle optimization process is a typical multi-variable optimization problem. The MATLAB software supports many optimization solver techniques, such as Interior-reflective Newton Method, Subspace Trust-region，SQP and so on. At default, the local optimization method is used. MATLAB also supports global optimization method, such as Global Search, MultiStart, Pattern Search, GA and Simmulated Annealing. The optimization process needs lots of computation time for one case, especially for global optimization method. The optimization process also needs the program to be highly robust, or else the optimization process fails easily. Figure 3 shows the optimization process for one case.

If the initial point is near the optimized point, it will need less iteration numbers to get the target. Actually, for each step, it needs lots of calculations.

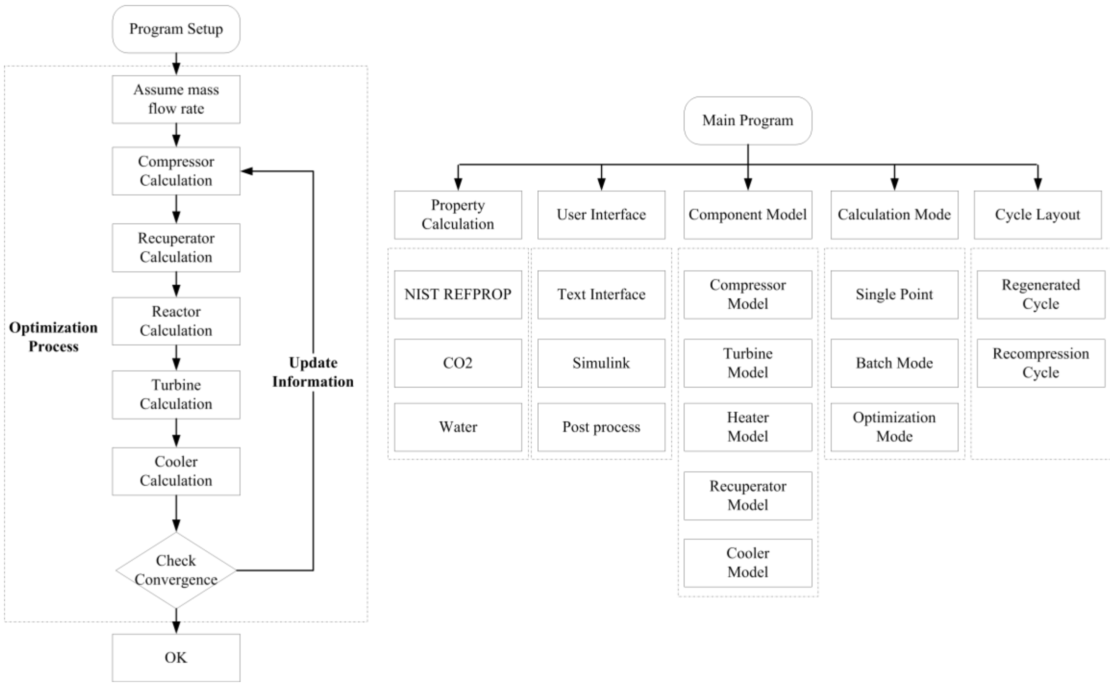

Figure 2 Program procedure and structure

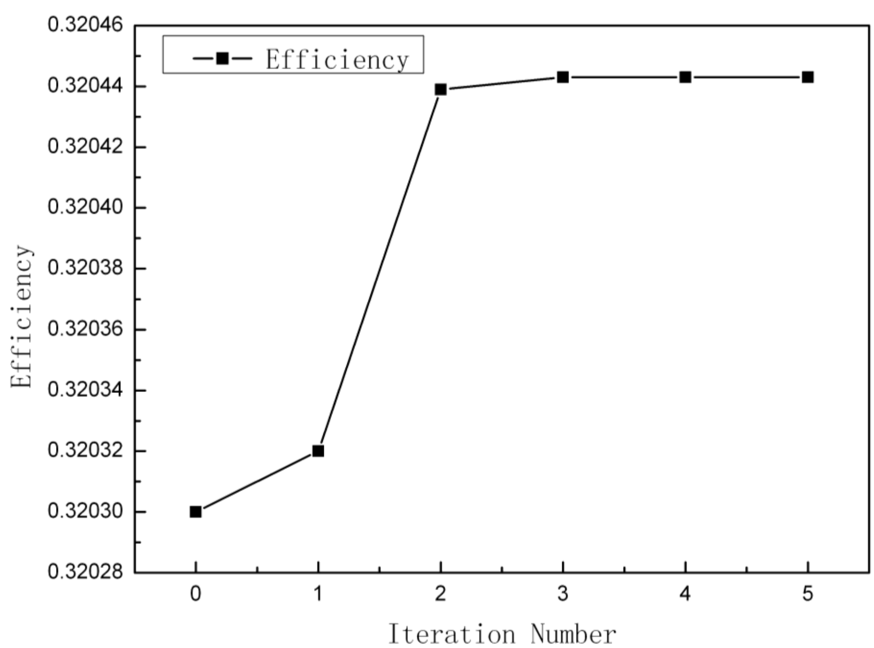

Figure 3 Optimization iteration process

RESULTS AND DISCUSSION

The SCOPE program was used to support the design of a Mega Watts-Class test facility. The loop flow chart is depicted in Figure 10. The carbon dioxide is compressed in the compressor and goes through the cold side of the recuperator where it gains heat from the exhaust gas from the turbine outlet at the hot side of the recuperator. Then it goes through the heater with the outlet temperature as high as 500 ℃. The high temperature, high pressure fluid goes into the turbine with pressure and temperature decreased at the turbine outlet. The enthalpy difference between the outlet and inlet is the part of energy transferred into electricity. The exhausted gas goes through the hot side of the recuperator and the cooler to be cooled down to the specified point of compressor inlet.

The compressor outlet pressure and inlet temperature are 15.0 MPa and 32℃. The compressor ratio is 1.8. The compressor efficiency is assumed 0.8. The turbine efficiency is assumed 0.85. The pressure drop and power for the heater are 100 kPa and 6MW, respectively. The heater outlet temperature is 500 ℃. The cooling water temperature is 25℃.

Pressure Ratio

For the pressure ratio sensitive analysis, the compressor outlet pressure is fixed at 15.0MPa and inlet temperature is fixed at 32℃. Figure 4 shows the dependency of cycle efficiency on pressure ratio. As the increase of the pressure ratio, the cycle efficiency is increased first and then decreases after reaches the maxima point. The maxima point is the place where the compressor inlet pressure is near the critical point. After this point, the compressor inlet is below the critical point and resides in the subcritical region. The medium density is greatly decreased along with the increased consumed work. At some cases, the increasing rate of consumed work by compressor is greater than that of generated work by turbine. So the cycle efficiency is decreased, accordingly.

There exists discrepancy between the thermal efficiency and net efficiency. The net efficiency accounts for the pumped cooling water work. Near the critical point, the heat capacity of carbon dioxide is very large. At the same time, the temperature difference between the water and co2 is relatively small. If the cooler geometry is fixed, the cooling water mass flux should be increased in order to cool the co2 down to the specified compressor inlet temperature. So the cooling water pumped work is increased (Figure 5). From this point of view, the cooler design parameters are also very important to the net efficiency. If the cooler is too small, there will be no enough capacity to eliminate the extra heat in the cycle.

From the point of getting large net cycle efficiency, the compressor inlet point should be close to and just above the critical point and the inlet temperature is below the pseudocritical temperature. In actual applications, the compressor inlet should leave a distance from the critical point, because the large property variations near the critical point may lead to unstable operation of compressor. In following calculations, the compressor inlet pressure is fixed at 8.3MPa.

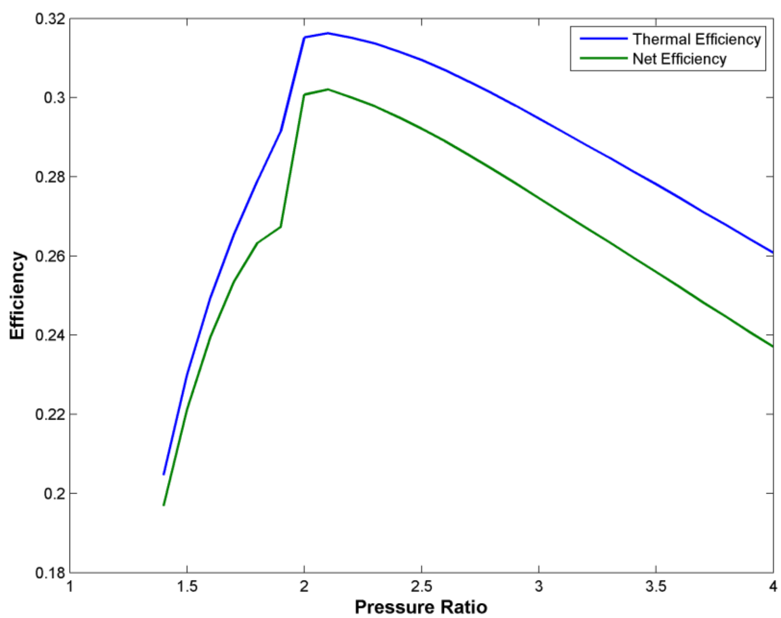

Figure 4 The pressure ratio effect on cycle efficiency

Recuperator Volume fraction

If the total heat exchanger volume is fixed, the recuperator volume fraction could be optimized. If the recuperator volume fraction is too small, the recuperated heat is small, which leads to large irreversible heat loss and low net efficiency. If the recuperator volume fraction is too large and the cooler volume is decreased, the cooling water mass flux has to be increased to fulfill the cooling demand. This will also decrease the net efficiency, as a result. So from the point of view, there will exist an optima recuperator volume fraction for a fixed total volume.

Figure 6 presents the efficiency trend with the recuperator volume fraction. The total heat exchanger volume is 0.6m^3. From the figure, it is suggested that the optimized recuperator volume fraction is about 0.75.

Recuperator Length

After the recuperator volume is fixed, the geometry parameter could also be optimized. The recuperator volume is dependent on the length and section area. If the length is determined, the other parameter is determined too. Figure 7 presents the efficiency dependency on recuperator length. As the increase of recuperator length, the thermal efficiency increases first and then decreases after it reaches the maxima point. The reason behind this is the competing relationship between the heat transfer and pressure drop.

When the recuperator length is small, the recuperated heat is small. The irreversible heat loss is large because of large temperature difference at the same end of the heat exchanger. As the increase of the recuperator length, the recuperated heat is increased. At the same time, the pressure drop in the recuperator is increased, too. The increase of pressure drop will decrease the net efficiency. Under the influence of the two factors, the net efficiency will have an optima recuperator length for a fixed volume.

The trend is similar for that of cooler length. The relationship is shown in Figure 8.

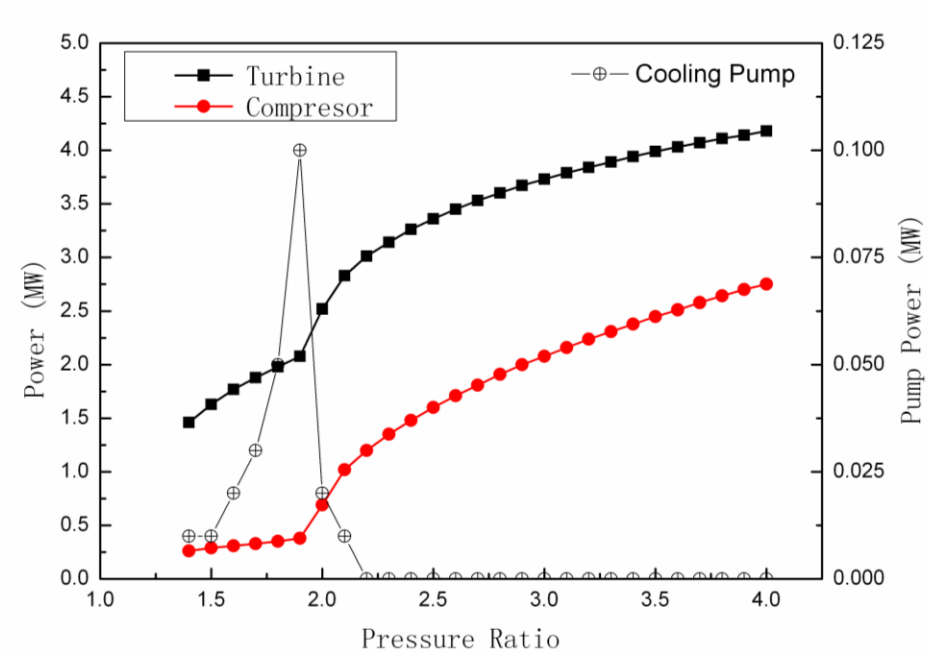

Figure 5 The pressure ratio effect on turbine, compressor and pump power

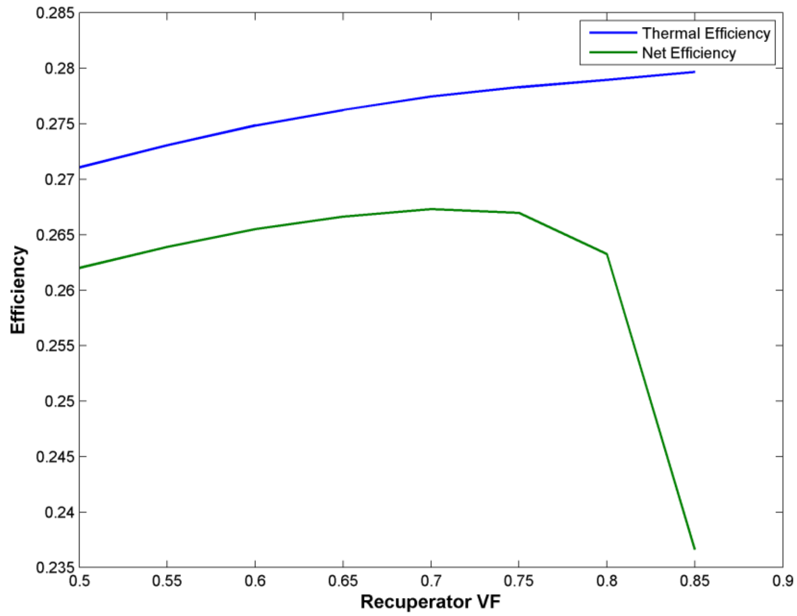

Figure 6 The recuperator volume fraction (VF) effect on cycle efficiency

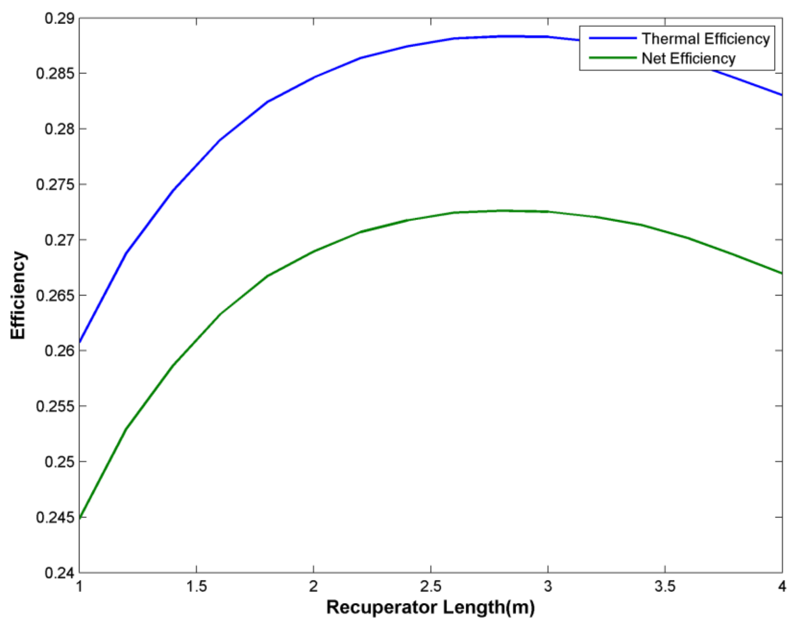

Figure 7 The recuperator length effect on cycle efficiency

The total heat exchanger(HX) volume effect

As the increase of total volume, the net efficiency is increased. There are two factors behind this. First, as the increase of total heat exchanger volume, the recuperated heat is increased which will decrease the irreversible heat loss. Second, as the increase of total heat exchanger volume, the flow area is increased for the fixed length heat exchanger along with the decrease of pressure drop. So the increase of heat exchanger volume leads to the increase of recuperated heat and decrease of pressure drop and the increase of net efficiency. When the heat exchanger volume is increased to some extent, the increasing rate of net efficiency is getter smaller. Thus there will be a balance between the heat exchanger volume and net efficiency. Figure 9 presents the relationship of heat exchanger volume with cycle efficiency. In this calculation, when the heat exchanger volume is larger than $0.5m^3$, the cycle efficiency increasing rate is getting small.

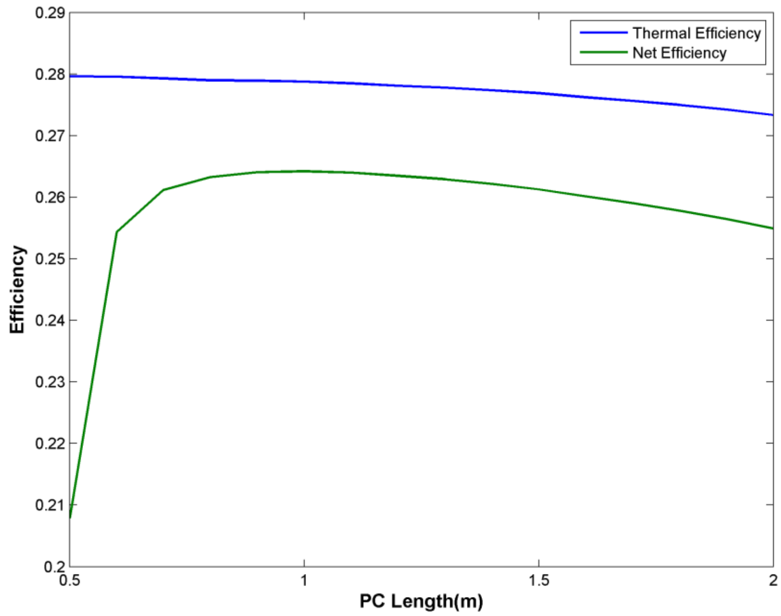

Figure 8 The pre-cooler length effect on cycle efficiency

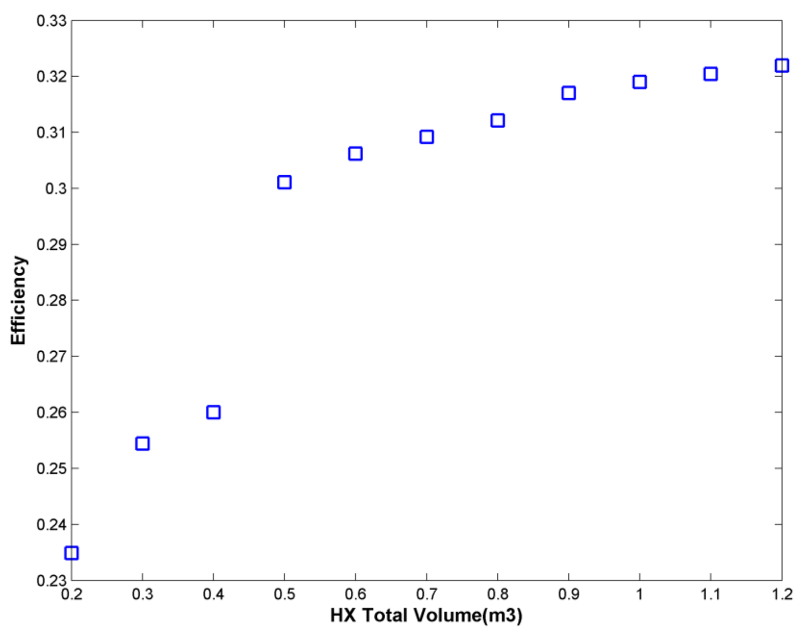

Figure 9 The heat exchanger(HX) total volume effect on cycle efficiency

The thermal balance diagram

After the discussions above, the thermal balance diagram for the MW-class test facility is shown in Figure 10. For this design, the thermal power is 6.0 MW. The thermal efficiency is 28.9% and net efficiency is 27.7%. The total heat exchanger volume (only the core plates) is 0.6 m^3. The recuperator volume fraction is 0.8. The turbine generated power is 2.05 MW. The compressor wasted power is 0.37MW. The cooling pump power is 0.02 MW. The net electricity for this loop is 1.66 MW.

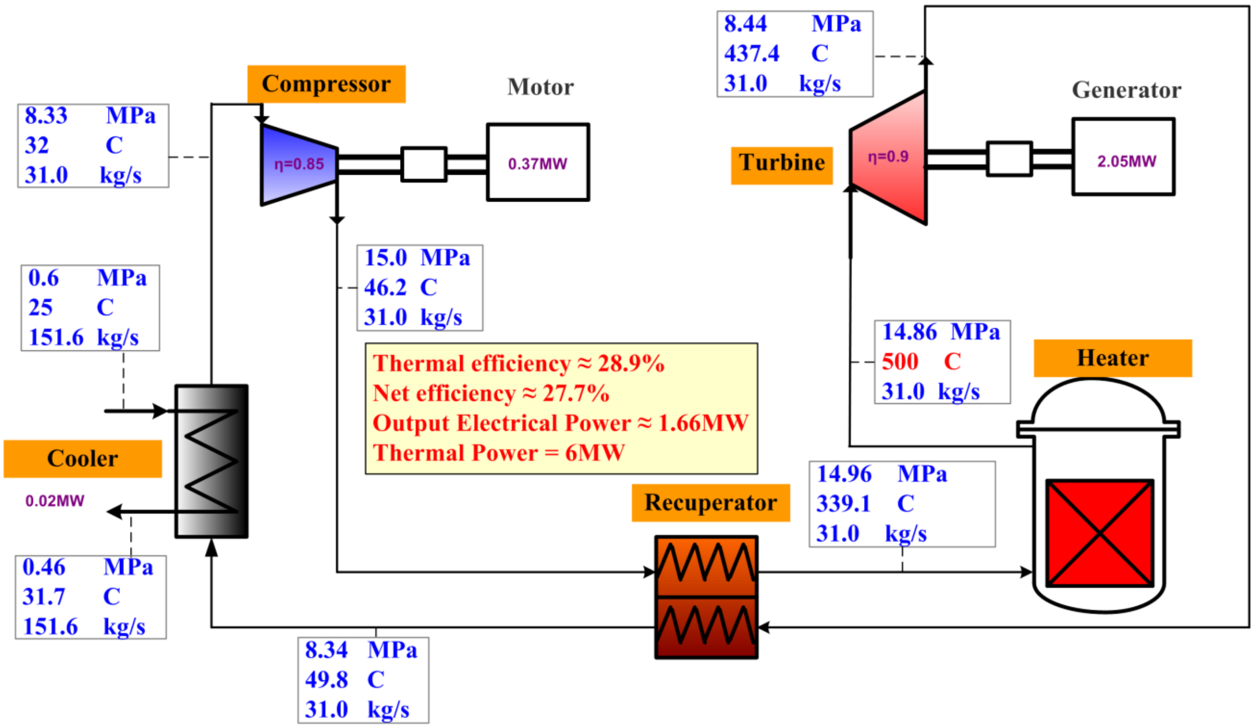

Figure 10 The thermal diagram for MW class loop (conceptual)

Conclusions

To better understand the performance of S-CO2 Brayton cycle, the thermal dynamic conversion code was developed which has the abilities of analyzing the performances of simple basic cycle, regeneration cycle and recompression cycle. The code was written in MATLAB language. The NIST REFPROP software was built in to calculate the properties.

The code was used to analyze a conceptual design of Mega Watts loop. There exists an optimal point for the pressure ratio, recuperator volume fraction, recuperator length and cooler length. For the pressure ratio sensitive analysis, the compressor inlet point should be close to and just above the critical point. In actual applications, the compressor inlet should leave a distance from the critical point. For the recuperator volume fraction, the optimal value is around 0.75. For the reuperator length, the optimal value is around 2.5 meters. For the cooler length, the optimal value is around 0.8 meter.

Increasing the total heat exchanger volume will increase the thermal efficiency. When the volume is bigger than 0.5m^3, the increasing rate is small.

Finally, a thermal balance diagram was proposed. This analysis could provide theoretical guidance for the future facility design and operation.

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